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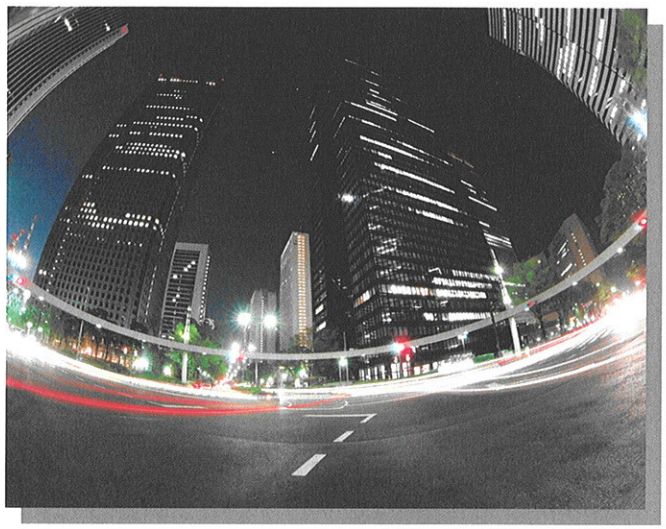
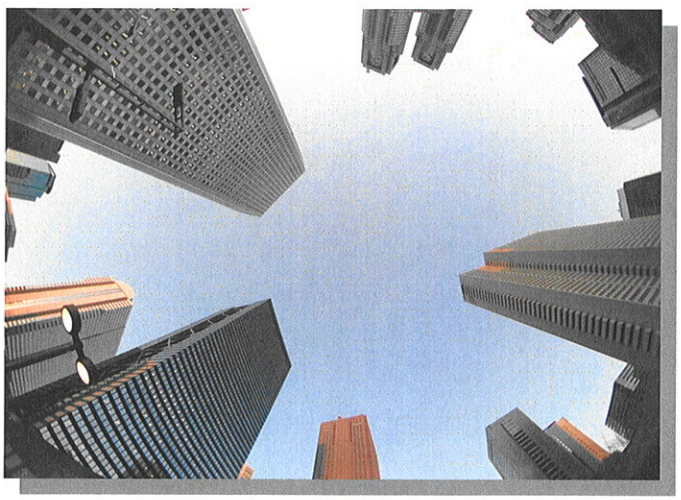
COMMERCIAL SYSTEM DESIGN COURSE (MODULE 1)





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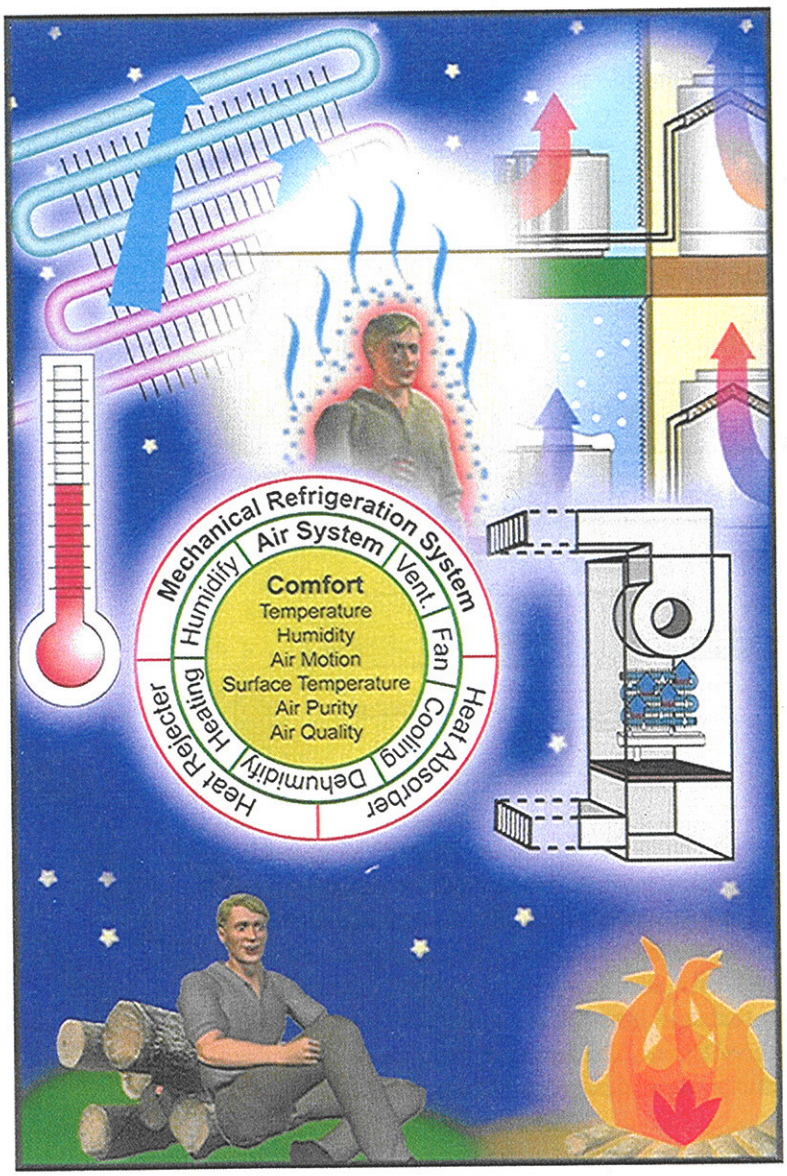


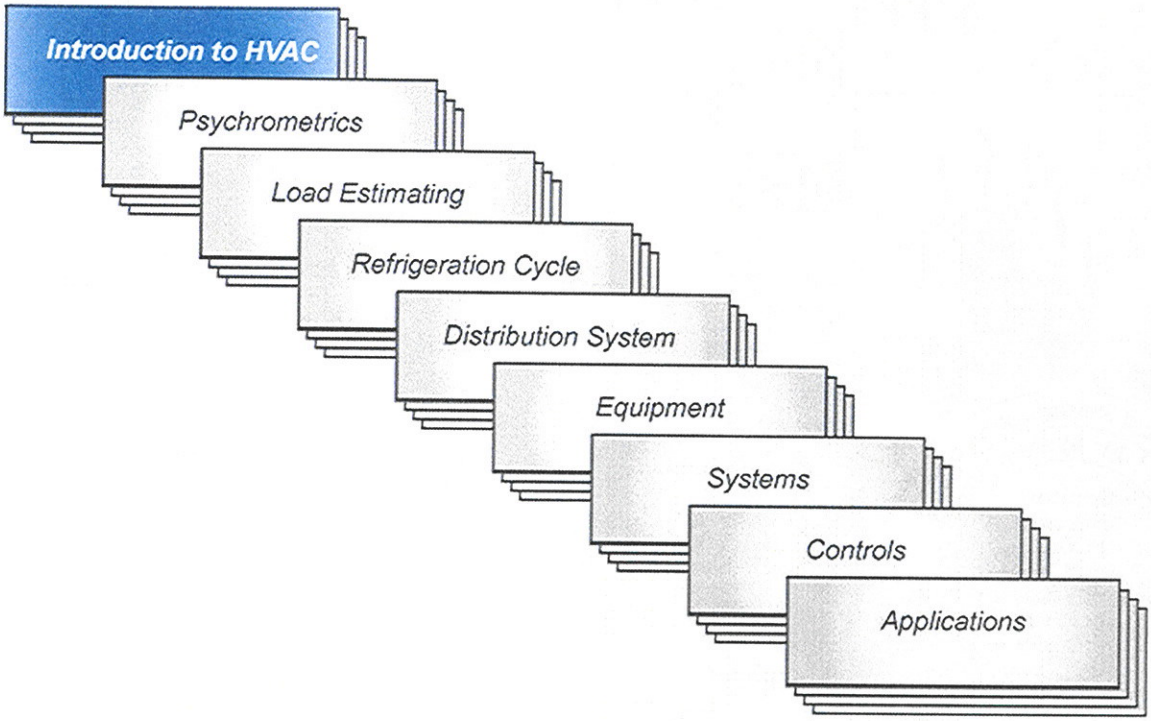
INTRODUCTION TO HVAC

ABCs of Comfort

Technical Development Program

Technical Development Programs (TDP) are modules of technical training on HVAC theory, system design, equipment selection and application topics. They are targeted at engineers and designers who wish to develop their knowledge in this field to effectively design, specify, sell or apply HVAC equipment in commercial applications.

Although TDP topics have been developed as stand-alone modules, there are logical groupings of topics. The modules within each group begin at an introductory level and progress to advanced levels. The breadth of this offering allows for customization into a complete HVAC curriculum – from a complete HVAC design course at an introductory-level or to an advanced-level design course. Advanced-level modules assume prerequisite knowledge and do not review basic concepts.



The Carrier TDP modules deal primarily with the design and operation of comfort air conditioning. To design these comfort air-conditioning systems, it is first necessary to understand what comfort is, and how a system designer can influence the human perception of comfort. The "ABCs of Comfort" is a module of the introductory series and is intended to introduce system designers to the parameters that influence human comfort, and how the air system and mechanical refrigeration system work together to control these conditions. The material presented helps the designers determine one of the first objectives of the system design, which is to establish the comfort standards for the project.

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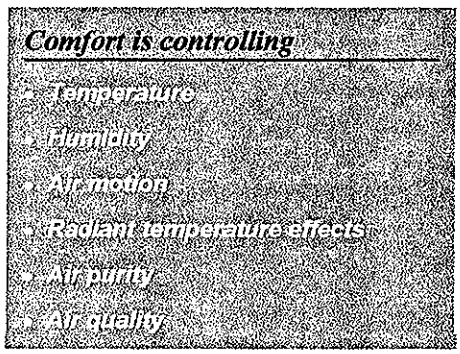
Introduction

HVAC is the business of providing comfort. In order to design a ***comfort*** system, one must first understand what comfort is. This training module will introduce the principles of ***human comfort***, the environmental factors that influence comfort, and how air conditioning works to maintain comfort. It will also explain how to determine the comfort specifications for a project.

We will begin by defining human comfort and the six environmental factors that influence our perception of comfort. These factors are: temperature, ***humidity***, air motion, radiant temperature, ***ventilation*** (air quality) and contaminant control (air purity). Each of us perceive comfortable levels of these factors in different ways, and each of these factors has limits that an HVAC system must maintain. However, there is no absolute mix of conditions that will assure that every person is comfortable in every situation. It is the designer's responsibility to understand the requirements, then to identify an HVAC system that will maintain these conditions within acceptable limits. This design often requires a trade-off in design, cost, complexity, and comfort.



In order to define comfort specifications, the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (***ASHRAE***) has developed a comfort chart. This chart will be explained later on in this module.

We will also explore how total air-conditioning systems are actually two interdependent systems: the air system and the ***mechanical refrigeration system***. Finally, the primary function of the components of these systems will be explained.

Why Do We Need Air Conditioning?

There are two purposes for conditioning the air: comfort air conditioning and ***process air conditioning***.



Figure 1

Comfort depends on how the space is used.

Process Air Conditioning

The purpose of process air conditioning is to improve an industrial process so that the quality or quantity of the product is improved. For example, the printing process requires certain air conditions to maintain high quality standards. This also applies to pharmaceutical and electronic equipment assembly areas or in refrigeration (food preservation) applications. In these applications, when air conditions are out of the desired range, product quality and reliability suffer. The conditions to be maintained are fixed by the nature of the process or material being handled. The number of variables controlled and the acceptable range of those variables vary greatly from process to process.

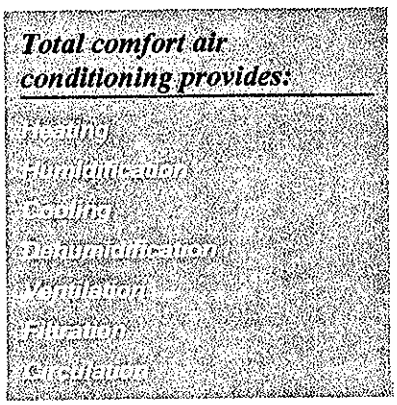
Comfort Air Conditioning

The second, basic reason for conditioning air is to maintain indoor conditions that are considered comfortable to the occupants. This process is called comfort air conditioning.

While comfort air conditioning focuses on people, process air conditioning focuses on a product or process, if the people are comfortable as well, so much the better. This module will focus on comfort air conditioning.

Comfort

Most people think of cooling when air conditioning is mentioned, but it means much more than that. Total comfort air conditioning is provided by a system that will ***heat***, ***humidify***, ***cool***, ***dehumidify***, ventilate, ***filter***, and circulate indoor air in a controlled manner to achieve a desired set of conditions. Most air-conditioning systems do only a partial job of controlling these comfort variables. But, at the very least, all systems must maintain temperature, humidity, and air motion within an acceptable range to be perceived as acceptable. Comfortable temperature ranges from 68 to 78 degrees Fahrenheit (°F), with 75° F being a typical design goal. Humidity is comfortable from 30 to 60 percent relative humidity, with 50 percent being a typical design goal in the cooling season.



An explanation of how the human body regulates temperature and how space conditions affect human comfort will help in understanding why these conditions represent good design goals.

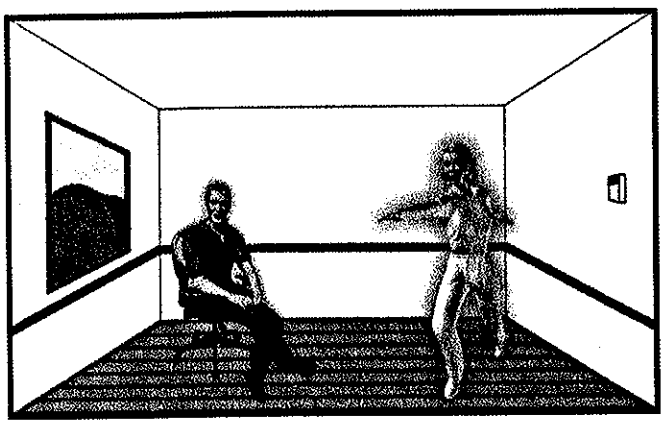


Figure 2

People generate heat depending on activity.

Body Temperature Control

The food we eat acts as fuel, which our body converts to useful energy. In addition to energy, heat is also produced. Whether you are digesting food, raising your arm, or using your eyes, energy is used and heat is generated. The more vigorous the activity, the greater the heat generated.

Our bodies are designed to function in a healthy manner at an inner body temperature of about 98.6° F. In order to maintain that temperature, any excess heat must be rejected. In a conditioned space, the body always generates more heat than needed to keep the inner body temperature at 98.6° F. Therefore, heat is rejected from our bodies constantly. Four ways our bodies transfer heat to the air around us are by conduction, convection, radiation, and evaporation.

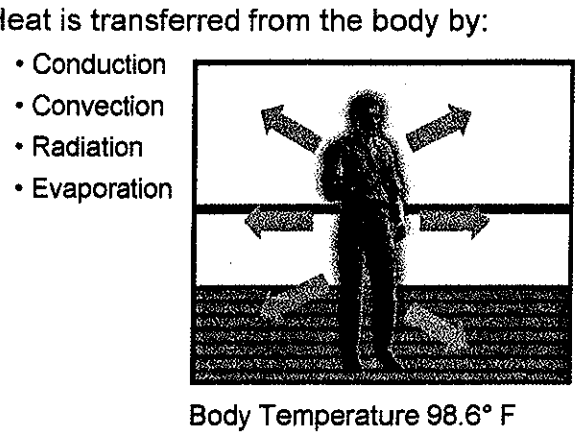


Figure 3

Heat is transferred from the body by four processes.

Conduction

Conduction is the transfer of heat directly through a substance or from one substance to another as heat flows from a higher temperature area to a lower one. The flame of the candle heats the end of the metal bar. The heat moves along the bar, from the higher temperature area to the lower temperature area, by conduction.

The heat is conducted into the fingers of the person holding the bar if the process is allowed to continue. Also, the heat from the bar is conducted to the cooler air surrounding it. The heat from the flame is conducted to the bar, through the bar, from the bar into the fingers, as well as into the air.

Likewise, in a conditioned space, heat generated by a person is conducted to the objects the person touches and to the air surrounding them.

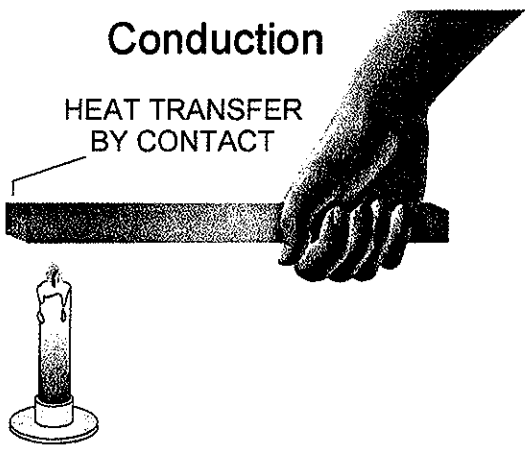
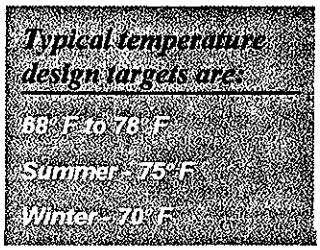


Figure 4

Conduction – Heat Transfer by Contact



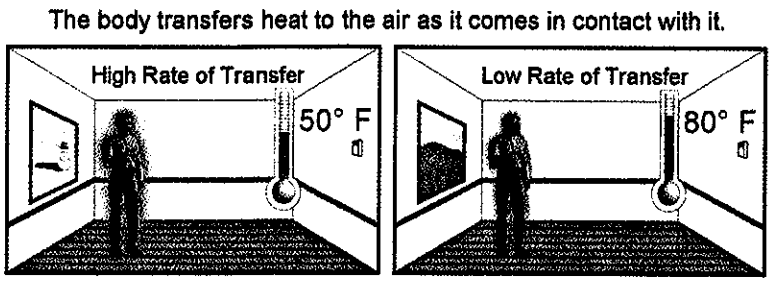


Figure 5

Air temperature controls conduction.

When the temperature is too low, occupants will complain about feeling cold. When the air temperature is too warm, they will complain about feeling hot. Consequently, the basic device used to control the amount of heating or cooling capacity delivered by a comfort air-conditioning system is a temperature controller, called a ***thermostat***. It is located in the conditioned space.

Temperatures considered comfortable vary with the level of activity, clothing, and personal preference. The design range for comfortable temperature for normally clothed people who are involved in sedentary, or near-sedentary activities that occur in an office, school, theater, restaurant, retail shop, hospital or residence is about 68 - 78° F. A typical design target for heating is about 70° F and for cooling it is about 75° F. This will be explored further later in this module.

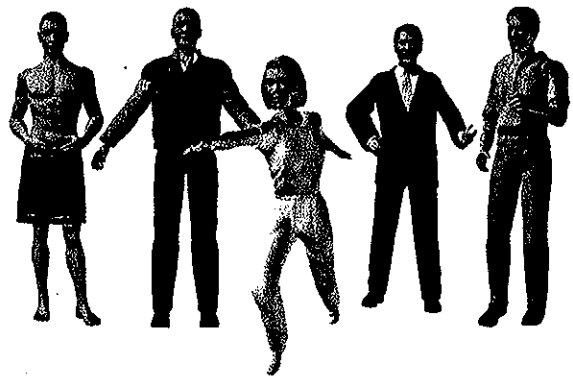


Figure 6

Clothing insulates our body, slowing conduction.

Clothing slows down the conduction process by adding insulation to our bodies. In the cooling season, adding insulation is not advisable because it slows down the conduction process necessary to reject excess heat. In the heating season, clothing helps to reduce body heat loss, adding to comfort.

However, there is a practical limit to how much clothing can be added before people feel uncomfortable because their body movement is restricted. In sedentary occupations, most efforts to create comfort with a space temperature below 70° F have been unsuccessful.

Convection

Our bodies also transfer heat by ***convection***. Convection is heat transfer by the movement of a fluid (gas or liquid) from one place to another. Heated molecules within a flowing medium, such as air, carry their heat directly to a new area. Thus, convection air currents carry heat to our body or away from it. Convection currents are easy to see in a pan of water being heated on a stove, especially just before boiling begins. Natural convection currents like these are also called density currents because their movement is caused by differences in fluid density that are a result of temperature differences. Warmer fluids are less dense and tend to rise, while cooler fluids are denser and tend to fall. Even though we cannot see them, convection currents operate in the air around us, very much like they do in a pan of water.

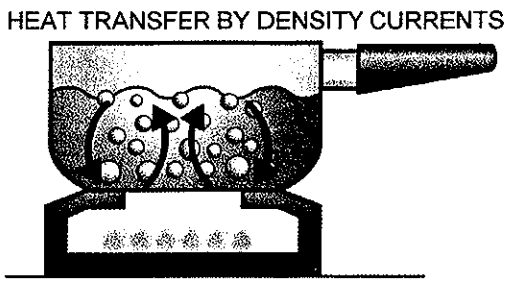


Figure 7

Convection – Heat Transfer by Density Currents

Natural air convection currents form around our bodies in a conditioned space. As body heat is conducted into the air close to us, it becomes warmer than the air farther away. Since warm air is lighter, or less dense, than cool air, the warm air floats upward. The warm air is replaced by cooler air, and the process continues.

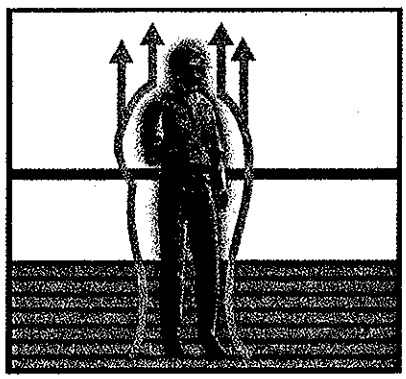
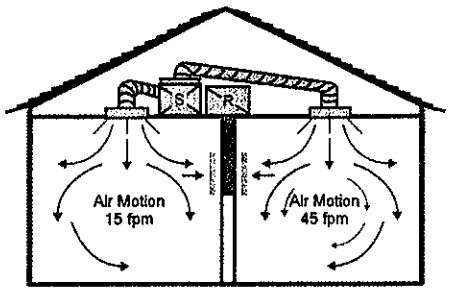


Figure 8

Natural Convection – Warm Air Rises Upward

When fans are used, the convection process is forced rather than natural. Using fans allows the convection process to move along more rapidly than it would by natural means.

Forced air systems send air from a fan through a supply ***ductwork*** system and out the supply ***diffusers*** located in the ceiling, wall or floor of the air-conditioned space. Return ***grilles*** pull the room air into a return ductwork system that takes the air back to the central equipment for conditioning. Then the fan sends the air back to the space again.



Forced-convection: convective motion is caused by a fan. Air motion influences comfort.

Figure 9

Forced Convection – Fans Provide Air Motion

The way the supply air diffusers and return air grilles are selected and laid out in the conditioned space is called ***room air distribution***. Room air distribution uses forced convection currents to assure comfort. If air moves too slowly, temperatures within the space will be uneven and the occupants may feel stuffy. If the air moves too rapidly, complaints about draftiness and feeling cold might be expressed.

Air Motion Comfort:

Comfortable air motion depends on the clothing of the occupants and their activity. For normally clothed people who are involved in sedentary, or near-sedentary activities, a reasonable target for room air motion is 15 to 45 feet per minute (fpm). When supply diffusers are selected, located, and ducted according to manufacturer's recommendations, this design goal is easily achieved.

Radiation

Radiation is the third way heat is transferred from our bodies to conditioned spaces. Radiant heat travels from a warmer object to a colder object without heating up the area in between. For example, a camper sitting in front of a fire on a cold night can get warmed even though the air between him and the fire remains cold. Some of the fire's radiant heat travels to the camper, striking him, and warming the side of his body facing the fire.

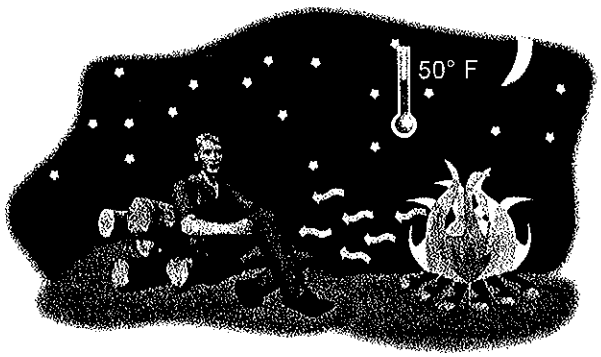


Figure 10

Radiation – Heat Transfer by Electromagnetic Waves

Just as the radiant heat of the campfire flowed to the cooler camper, radiant heat can also flow from our bodies to cooler surfaces surrounding us in conditioned spaces. For instance, body heat will flow from the man standing in the room to the colder outside window behind him, by means of ***radiation***. The colder the window is, the more rapidly the heat leaves his body.

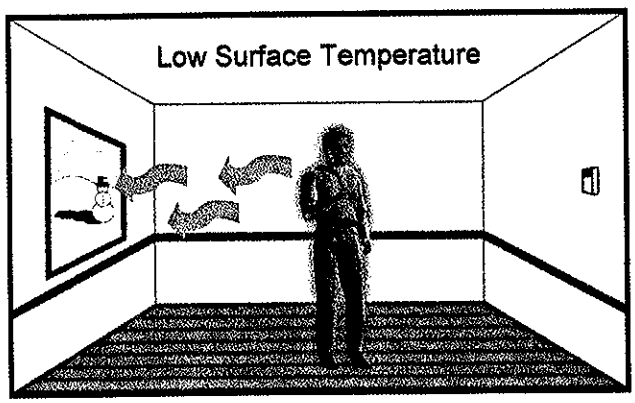


Figure 11

Radiation – Warmth radiates from the body toward cooler surfaces.

Conversely, radiant heat can flow from a surface to our bodies when the surface is warmer than our body. Even when the air temperature surrounding us is acceptable, the temperature of the surfaces surrounding us can cause discomfort because of radiant heat flow if those surfaces are very cold or hot.

The surface temperature of the ceiling, walls, floor, and glass that surrounds us is determined largely by how well insulated they are. The better the insulation, the closer the inside surface temperature will be to the inside air temperature. The closer the surface temperatures are to the room air temperature, the more comfort is enhanced.

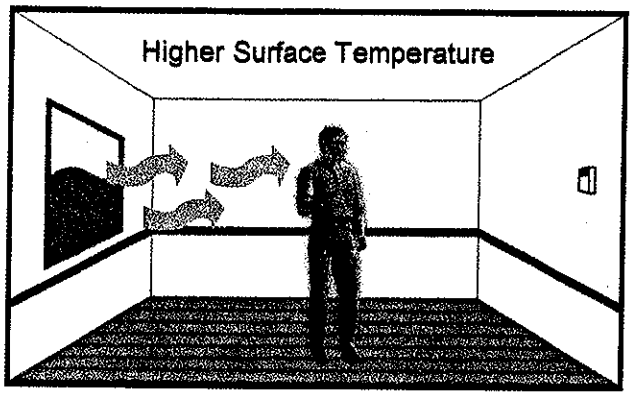


Figure 12

Radiation – Higher surface temperatures radiate toward the body.

Surface Temperature Comfort:

Ideal surface temperatures to minimize radiant heat loss are 70° to 80° F.

Therefore, quality construction reduces discomfort caused by radiant heat loss from our bodies in the winter. It also reduces discomfort caused by radiant heat gain to our bodies in the summer.

The ideal comfort situation exists when the average temperature of all the surfaces surrounding the occupant is between 70 and 80° F. Adequate wall and ceiling insulation, multiple layers of glass, and internal or external solar shading devices all help to achieve this ideal situation. Substantial space temperature adjustment is required to maintain comfort when surface temperatures are much different than the space air temperature. For instance, a building with poorly insulated outside walls and single glass may have to be set for 78° F space temperature in the winter to give the same feeling of comfort as a well insulated space with triple glass set at 70° F.

Likewise, a poorly insulated building without adequate solar shading may have to be set for 68° F in the summer to provide the same feeling of comfort as a well insulated building with adequate solar shading set at 75° F.

Besides the comfort incentive, energy costs for air conditioning add another important motivation for providing adequate building insulation and glass quality. In most cases, the energy impact of high quality construction is well known, but the comfort aspect is often forgotten. The energy penalty of inadequate insulation and poor glazing is compounded when the thermostat is reset in an attempt to restore comfort.

Here are some reasonable minimum guidelines for insulation and glazing in residential construction with outside temperatures at -10° F and 95° F and a space temperature of 75° F in both summer and winter.

Barrier	Minimum	Surface Temp.* Summer/Winter	Example Construction
Walls	R-11 U-0.099	76.4° F / 69.3° F	Wood frame with sheathing and siding, ½-in. sheetrock inside, 3½-in. fiberglass insulation
Roof	R-19 U-0.036	75.5° F / 72.9° F	Sheetrock ceiling on bottom of wooden trusses, 6-in. fiberglass insulation above ceiling
Floor Slab	R-5 U-0.231	78.1° F / 61.6° F	Closed-cell foam board slab between foundation and floor slab runs vertically down below frost lines
Doors	U-0.39	76.4° F / 60.5° F	All closures weather-stripped from factory
Single Glass	U-1.13	90.4° F / 10.0° F	Single glass (pane) window
Single Glass w/Storm	U-0.49	81.6° F / 46.7° F	Single glass (pane) window with storm

* Outside Summer 95° F with 75° F inside
Outside Winter -10° F with 75° F inside

Figure 13

Typical Residential Surface Temperatures

Commercial construction guidelines are similar. Here is a table with the same type information but for common commercial construction, again at -10° F and 95° F with a 75° F space in both summer and winter.

Solar heat absorbed by the surface can make these temperatures much higher. This will be dealt with in more detail later.

Barrier	Minimum	Surface Temp.* Summer/Winter	Example Construction
Walls	U-0.124 U-0.580	76.7° F / 67.8° F 82.9° F / 41.6° F	Steel frame masonry
Roof	R-16 U-0.095	76.2° F / 70.1° F	Steel frame insulation over deck
Floor Slab	U-0.360 U-0.322	79.8° F / 70.2° F 79.3° F / 66.4° F	Steel joist masonry
Doors	U-0.70	80.4° F / 51.9° F	Swinging type
Single Glass	U-1.22	89.6° F / 5.3° F	Single pane window less than 40% glass
Double Pane Glass	U-0.64	83.7° F / 38° F	Double pane glass

* Outside Summer 95° F with 75° F inside
Outside Winter -10° F with 75° F inside

Figure 14

Typical Commercial Surface Temperatures

Evaporation

Evaporation is the fourth way heat is transferred from our bodies to the conditioned spaces we occupy. Moisture, in the form of perspiration, is given off through the pores of our skin. As this moisture evaporates, it turns into invisible water vapor. The body provides the heat necessary to accomplish this change of state, from water to vapor. Each pound of water evaporated absorbs about 1000 ***British thermal units (Btu)*** of heat from the body. Evaporation from our bodies goes on constantly, whether it is visible or not. When you see drops of perspiration on your skin, your body is producing more heat than it can reject at the normal rate.

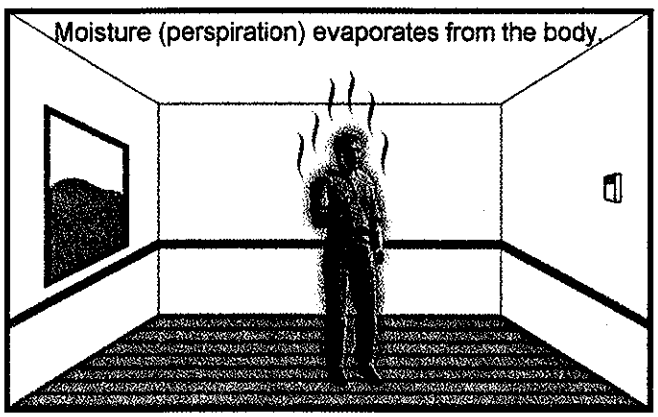


Figure 15

Evaporation helps control body heat.

Convection currents affect evaporation. If there were no air motion, the layer of air closest to your body would soon absorb most of the water vapor it could hold. If this happened, the evaporation process would come to a halt. Moving air prevents the build-up of a blanket of moist air around you.

Convection currents move the air that has absorbed evaporated moisture away from your body. Behind that air comes drier air that allows the evaporation process to continue. The faster the air flows over your body, the more rapidly evaporation takes place and, therefore, the greater the sensation of coolness you feel. While convection currents affect evaporation in an important way, relative humidity affects it most of all.

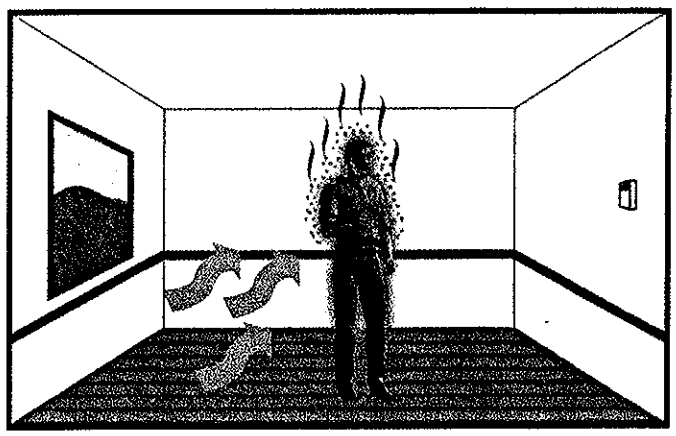
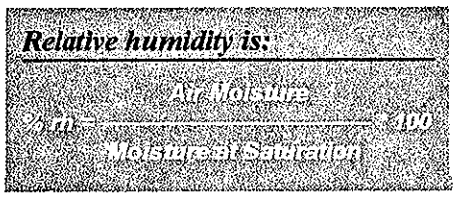


Figure 16

Convection currents increase the evaporation rate.

Relative Humidity and Comfort

Relative humidity is a measure of how much moisture is in the air. In reality, it is a ratio that indicates the ability of the air to absorb more moisture, expressed as a percent.



The lower the percent, the lower the moisture content in the air and the greater the air's ability to absorb more moisture. Since relative humidity influences comfort in an important way, let's take a closer look at it.

Since each pound of water evaporated from our skin removes about 1000 Btu from our bodies, evaporation is an important means for keeping us cool. The relative humidity of the air surrounding us sets a limit on how fast evaporation can take place. The more humid the air, the slower evaporation will progress. The less humid, the more rapidly evaporation will progress.

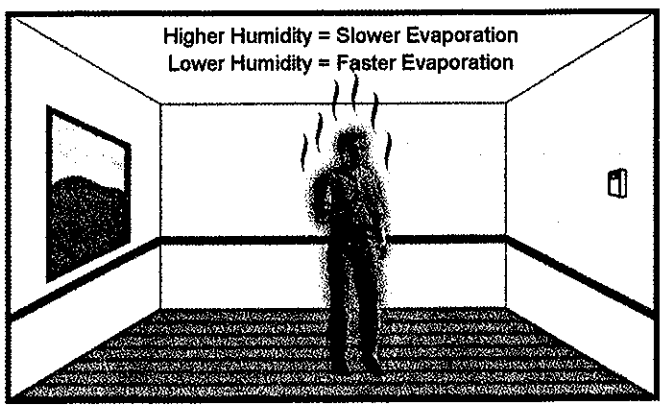


Figure 17

Humidity affects the rate of evaporation.

There is an upper limit to the water vapor any specific volume of air can hold. Air is said to be saturated or at saturation when it reaches that level. Air is saturated with all the water vapor it can hold without having the water change state from a vapor (a gas) into a liquid or a solid. When that limit is exceeded, the vapor turns to liquid, forming droplets of condensation. Outdoors it shows up as fog, rain, sleet, or snow.

Consider the air in the room illustrated. Obviously, there is no rain or fog in the room, so the air is at a condition, which is at or less than saturation. When the temperature of the air is measured, it is found to be 70° F. We will need some more information to determine how close to saturation the air is.

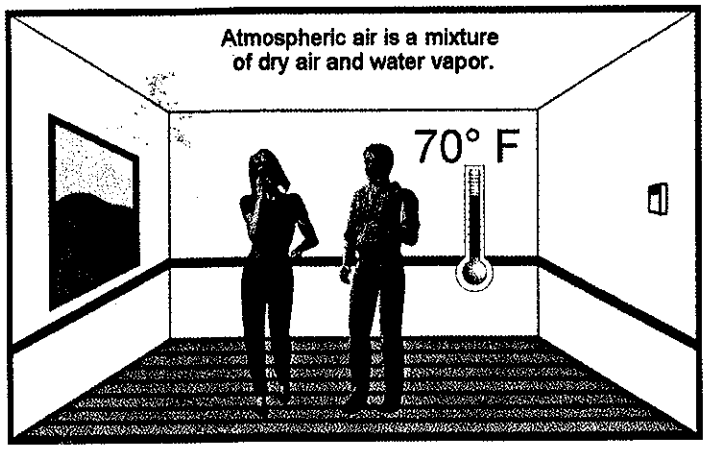


Figure 18

Air is a mixture of dry gases and water vapor.

Air has weight. The dry gases in air, as well as its water vapor, contribute to that weight. The amount it weighs per unit of volume (space) is its density, which is expressed as pounds per cubic foot of air (lb/ft^3). Mostly it is temperature that influences air density. Warmer air is less dense. Colder air is denser.

The amount of volume, or space, one pound of air takes up is its specific volume. Density and specific volume are like flip sides of the same coin. Mathematically speaking, specific volume is the reciprocal of density (specific volume = 1/density). Likewise, density is the reciprocal of specific volume (density = 1/specific volume). As density increases, specific volume decreases, and vice versa.

Figure 19 gives you an appreciation of the specific volume of a typical sample of air and water vapor mixture in an air-conditioned room. The sample shown has a specific volume of 13.3 cubic feet per pound. In other words, one pound of air at the conditions in this room takes up a space the size of a cube that is about 28½ inches on each side. The density of the same air sample is 0.075 pounds per cubic foot (1/13.3). This moist air sample is about the same weight as four packs of chewing gum.

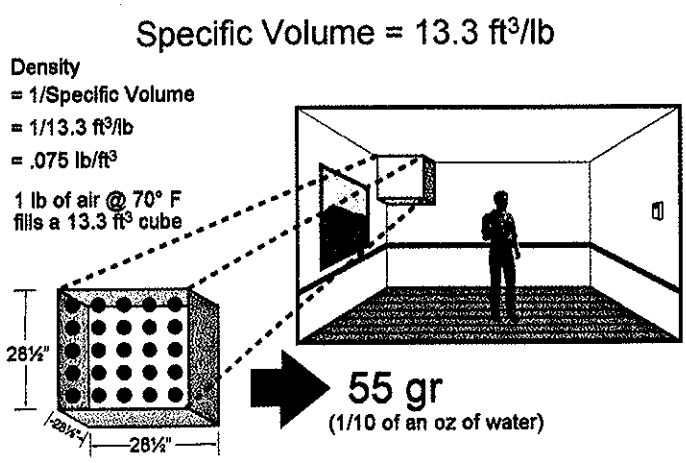


Figure 19

One pound of air and water mixture containing 55 grains of moisture.

When the moisture of the air is measured, it is determined to be about 55 grains of water vapor per pound of dry air. This is a very small amount of water vapor, since there are 7000 grains in a pound of water. So in this example, each pound of dry air contains about 1/10 of an ounce in the form of water vapor. So in this example, each pound of dry air contains about 1/10 of an ounce in the form of water vapor. If the air were at 50° F and the same 55 grains of moisture, the air would be saturated.

1 lb water = 7000 gr

Air at 50° F and 100% relative humidity has 55 grains, or 1/10 ounce about the weight of two sheets of paper.

If this same pound of 70° F air was saturated with water vapor, it could hold a maximum of 110 grains of water vapor. Psychrometrics is required to determine the maximum amount of water vapor the air sample can hold. Psychrometrics is the study of dry air and water vapor mixtures, the kind we find in any room. The psychrometric chart shows how the ***specific humidity*** of our air sample is determined to be 110 grains per pound at saturation. Following the 70-degree dry bulb line vertically up to the saturation line (100 percent rh), and over to the right gives the value of 110 grains per pound of dry air.

The air sample is not saturated since it only contains 55 grains of water vapor per pound of dry air, rather than 110 grains.

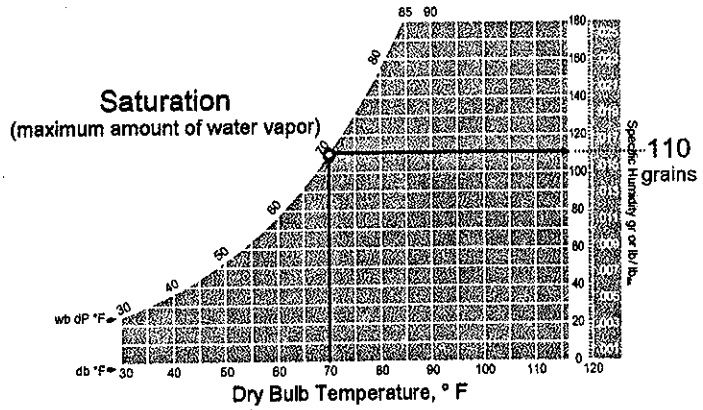


Figure 20

A psychrometric chart is used to determine saturation.

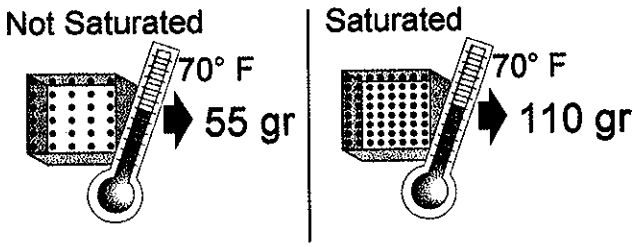


Figure 21

Air Sample – Not Saturated and Saturated

To find the relative humidity of the air, divide the amount of moisture the sample contains (55 grains per pound) by the maximum amount it could hold at saturation (110 grains per pound), and multiply by 100 to get the percent. The answer is 50 percent for the illustrated room. Therefore, the room conditions are 70° F and 50 percent relative humidity (rh).

Because the air has a 50 percent rh, you know that it contains half as much water vapor as it could hold if it were saturated. Since the moisture holding ability of air is influenced by temperature, the relative humidity of the air in the room changes if its temperature changes.

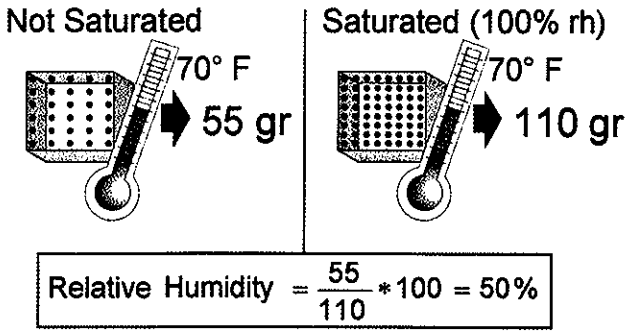


Figure 22

Relative Humidity of 70° F Moist Air

Let's raise the temperature to 92° F. Since no more water vapor has been added, the actual moisture content, also called specific humidity, is still 55 grains per pound of dry air (gr/lb). However, warm air has a greater ability to hold moisture than cool air. At 92° F, a pound of dry air holds approximately 220 grains of water vapor when saturated.

The relative humidity of the air sample is now only 25 percent, as shown. One pound of 70° F air is 50 percent of the way to being saturated when containing 55 grains of water vapor, but when heated to 92° F, it is only 25 percent of the way to saturation. Air at 92° F can hold twice the moisture as 70° F air.

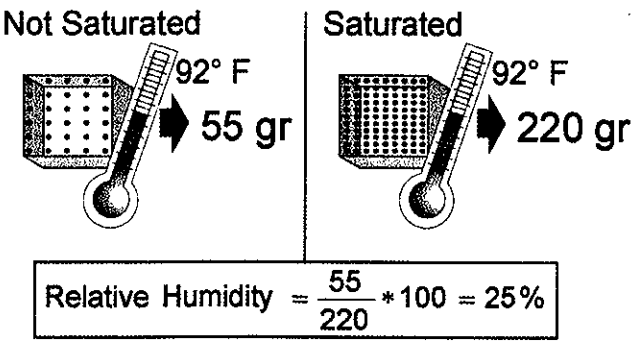


Figure 23

Relative Humidity of the Same Air at 92 F.

As previously illustrated, cold air can hold less moisture than warm air. A practical application of this fact can be seen in heated spaces situated in cold climates. The air within a building always starts out being outdoor air. The moisture within the building is a combination of what comes in from outside plus the vapor added by people, cooking, washing, and water exposed to the indoor air. If a building has good vapor barriers, some of the moisture added internally can be retained; but if the barriers are poor, most of the moisture leaks out rather rapidly.

If the outdoor air is 20° F and 50 percent rh, as shown here, its specific humidity is only about 7.5 grains per pound. Its specific humidity is very low at 50 percent rh because cold air can hold very little water vapor before becoming saturated. When this same air comes inside by leakage (***infiltration***), or is brought in through ventilation, it still contains 7.5 grains per pound, but drops way down to 7 percent rh at 70° F. It will not take much air this dry to drop the space relative humidity below the lower limit for comfort, which is about 30 percent.

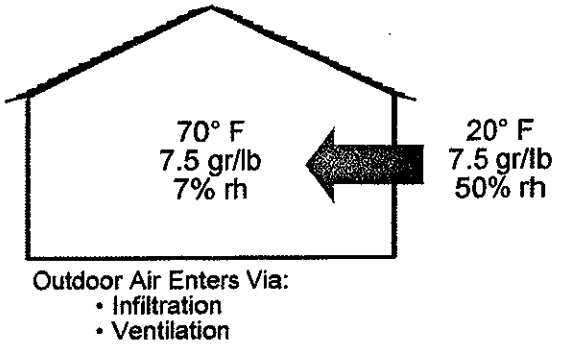


Figure 24

Infiltration of outside air lowers space relative humidity.

During the heating season, evaporation works against us. The heating system is used to limit the loss of heat from our bodies by heating the air, but evaporation is accelerating body heat loss as the air becomes drier. Consequently, a ***humidifier*** becomes an important part of a total comfort air-conditioning system installed in a cold climate.

If the air around your body has a low relative humidity, your body will be able to give off more heat through evaporation than if there is a high relative humidity. Lowering the relative humidity speeds up evaporation. Increasing relative humidity slows it down.

Comfortable relative humidity ranges from about 30 percent to 60 percent, with an ideal goal in the cooling season being 50 percent. Relative humidity of 55 percent is a more reasonable goal for climates with a moderately humid climate. Relative humidity of 60 percent is the limit for high humidity climates.

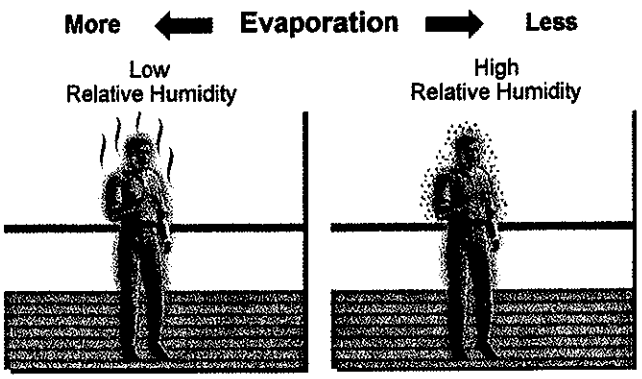


Figure 25

Relative humidity influences the rate of evaporation.

In the cooling season, when relative humidity is above 60 percent, people often complain of being uncomfortable, regardless of the space temperature. It is often true that a space with a slightly warmer temperature but lower relative humidity is perceived to be more comfortable than one with a lower temperature but higher relative humidity. Since cooling systems dehumidify the air only when the ***compressor*** is running, it is important to avoid oversizing equipment more than about 20 percent to avoid excessive humidity at partial ***cooling load***. Oversized systems spend too much time with the compressor cycled off to do a good job of dehumidifying the indoor air. This is especially important in humid, summer climates. Humidity levels above 60 percent have the potential to create indoor air quality issues in addition to comfort complaints.

Comfortable humidity levels

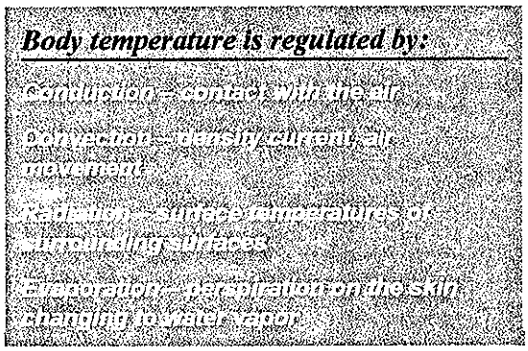
are in the range of 30 % to 60 %, with a target of 50 % for summer design. Levels above 60 % can become an IAQ concern because of mold growth.

Heating season goals for relative humidity are limited by the severity of the winter climate and the insulation and glazing quality of the building. When the outdoor temperature drops, cold outside walls and windows can cause moisture to condense as liquid water, damaging walls, draperies, moldings, and floor coverings. It is wisest to reset the humidistat, which controls the humidifier, according to outside temperature. If the humidity level is not reset, a practical design goal for locations with a winter design temperature not less than 10° F is 30 percent. Milder climates may tolerate somewhat higher humidities, as shown.

Outdoor Temperature	Outdoor Relative Humidity	Maximum Indoor Humidity
-10° F	30 to 70	20%
0° F	30 to 70	25%
10° F	30 to 70	30%
20° F	30 to 70	35%
30° F	30 to 70	40%

Figure 26

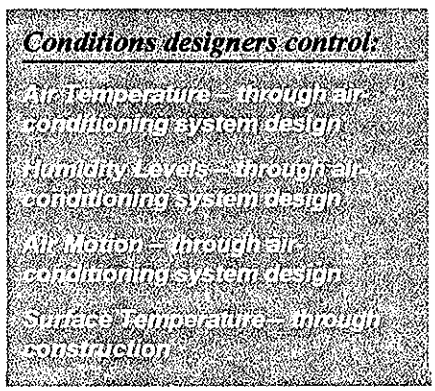
Recommended Maximum Winter Humidity Levels



Body Regulation and Space Control

Now we will summarize human comfort as it pertains to air conditioning by relating the four ways our bodies transfer heat to four building conditions the air-conditioning designer can control.

Conduction, convection, radiation, and evaporation, the four ways our bodies transfer heat to the environment, occur simultaneously. These methods work together attempting to maintain our bodies in a comfortable state. As we have seen, there are several conditions within the space that help our bodies achieve the comfort they are trying to maintain.



These conditions include air temperature, space relative humidity, air motion, and surface temperature of outside barriers. The designer of a comfort air-conditioning system can influence air temperature, space relative humidity, and air motion so that comfort is assured for most people in most circumstances. The goal of the comfort air-conditioning designer is to engineer systems that maintain these conditions within a reasonable range at a reasonable cost and energy usage. Proper attention must be given to the ***air system*** and ***mechanical refrigeration system*** to achieve this goal. First we will examine the air system and then the mechanical refrigeration system. Both are equally important.

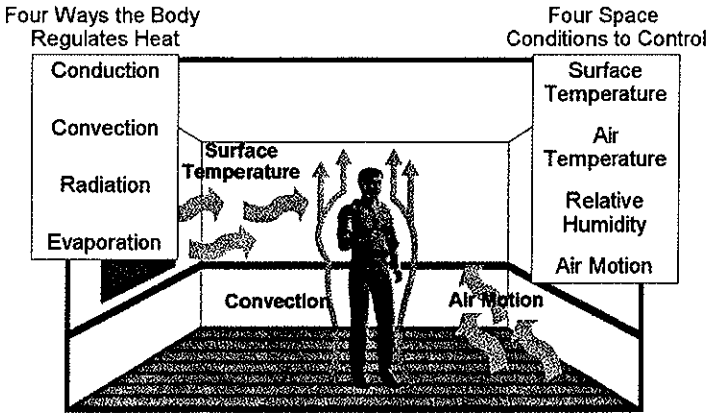


Figure 27

Body Temperature Regulation and Space Control

It can be noted from the discussion that comfort parameters have acceptable control ranges. Some of these are the result of personal preferences, and some are the result of seasonal variations.

People dress differently in summer than they do in winter, which influences the insulating value provided by clothing. It is typical for people to dress in lighter clothes with less insulating value in the summer, making higher temperatures acceptable. This variation can result in temperatures perceived as comfortable in one season as not being acceptable in another.

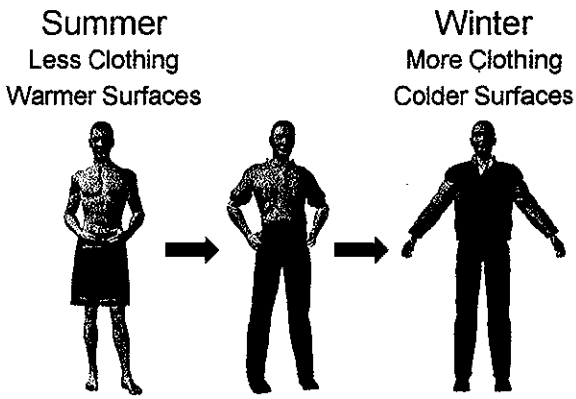


Figure 28

Clothing levels vary by season.

Similar results are observed with transmission loss and gains. It is not uncommon to have greater temperature difference inside to outside in winter than summer. This can result in conditions like colder glass temperatures that require higher space temperatures to compensate for the additional radiant heat loss.

For the designer this means it is important to evaluate conditions for all seasons of the year. This is true even when a space requires cooling all year round in order to be sure the selected comfort criteria is maintained.

Comfort Chart

ASHRAE has conducted considerable research on what constitutes comfort. The results are a comfort chart that can be used by the designer to select the appropriate comfort conditions for any occupancy, activity level, or season.

Since the conditions that define optimum comfort depend on individual preferences, dress, activity, and body function, it is impossible to have one specific number that satisfies everyone. To account for human differences, studies have been done using a large number of people to define what people perceive as comfortable.

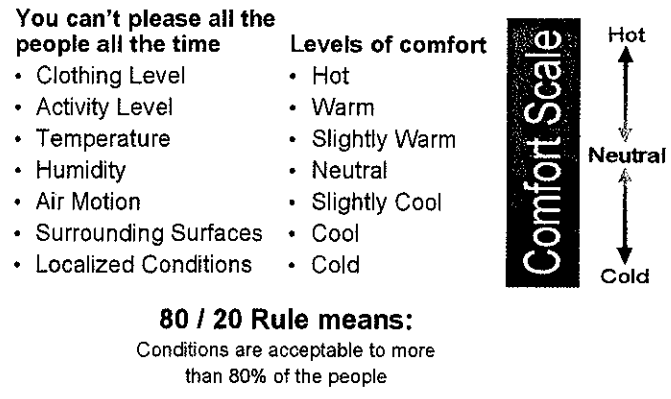


Figure 29

Comfort is subjective and depends on many variables.

Each of four comfort variables has been tested along with dress, activity, and localized conditions to determine what range of each is considered acceptable. The results of these tests have resulted in the development of the ASHRAE Comfort Standard, ASHRAE 55. This standard can serve as the designer's guide in determining the indoor comfort requirements for a project.

As is done with many studies that involve human judgment, the 80/20 rule is used. This rule is, when greater than 80 percent of the people are satisfied with the conditions, then the conditions are considered acceptable. In reality, since comfort is so subjective and a function of numerous variables, a 10 percent dissatisfaction level is used for acceptability with each individual comfort condition. This should result, when all conditions are considered, in meeting the 80/20 rule.

Comfort perception varies considerably, so a seven-step scale is used to evaluate the level of comfort. The subjects of the studies rate comfort as, hot, warm, slightly warm, neutral, slightly cool, cool and cold. Results then are based on a 90 percent rating of those conditions.

As discussed, temperature and humidity have the greatest influence on comfort and are the primary controllable functions of air-conditioning systems. The comfort envelope defines a range of acceptable temperature and humidity. Other conditions that influence comfort are assumptions and corrections to the values in the chart. Air motion, surrounding surface temperatures, clothing levels, activity level, and local conditions, such as floor-to-ceiling temperature difference, cycles per hour, and temperature drift, are assumptions on which the chart is based.

Designers can then use the chart and the modifications, which account for the other variables, to select the design conditions appropriate for the project.

Clothing Level

Clothing acts as an insulator inhibiting the ability of the body to dissipate heat. This can both help and hinder in the ability to maintain comfort. The more layers and the heavier clothing people have on, the less heat the body will be able to dissipate. The insulating value of clothing is measured in ***CLO*** units, or clothing insulating value. One CLO unit is equivalent to an R-value (resistance to heat flow) of 0.88 Btu/(h * ft^2 * F), or about equal to wrapping the body in a ¼ inch of cellular foam insulation.

- Clothing is an insulator
- Insulating value is measured in CLO – Clothing Insulation Value
- Varies from summer to winter with lower value in areas with less annual variation

Typical Values
Summer 0.35 – 0.6
Winter 0.8 – 1.2

Business Suit – 1.0 CLO



Figure 30

The insulating value of clothing is measured in CLO units.

Tables have been established for typical clothing assemblies based on men and women and typical dressing situations. These tables are found in ASHRAE Standard 55 and The ASHRAE Fundamentals Handbook. When the assemblies for the project are not listed, a table of CLO values for different clothing pieces is provided. The CLO value of the assembly can be estimated by adding individual component CLOs using a formula.

As discussed earlier, people have different dressing habits in summer and winter. The comfort chart assumes a CLO value of 0.5 for summer dress and 0.9 for winter dress. These values are typical for most office and retail applications. As a guideline, typical values of 0.35 to 0.60 in summer and 0.8 to 1.2 in winter are appropriate.

Activity Level

The activity level of the people in a space also affects the perceived comfort level for temperature and humidity. Activity level is measured in metabolic rate units (***MET***), and is used to calculate the heat energy produced by the body. One MET is equivalent to a heat generation rate of 18.4 Btuh/ft^2. The average body is 19 ft^2, therefore, the body generates approximately 350 Btuh per MET or about 420 Btuh per person for performing standard office tasks.

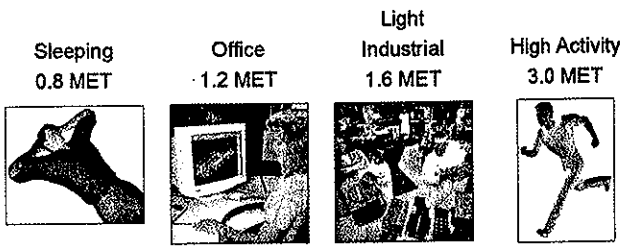


Figure 31

Activity level is measured in MET (Metabolic Rate Temperatures) and can be less for higher activity levels.

Typical levels vary from 0.8 MET when sleeping, to 3.0 MET doing heavy activity. The values in the comfort chart are based on an activity level representative of people seated at rest and doing typical office work. The metabolic rate for this activity level is 1.2 MET.

The average human body

The higher the activity level the lower the temperature generally found acceptable. Designers may be able to adjust design conditions to account for this. However, people often compensate for the activity in their dress. When designing, you must anticipate the impacts of both the activity and compensation of clothing to activity level.

Effective Temperature

Humidity has a significant impact on comfort. People commonly feel comfortable at a lower temperature in winter if the humidity is higher. Conversely, in the summer people feel more comfortable at a higher temperature if the humidity is lower.

To account for this difference, the concept of ***effect temperature*** is used. The effective temperature is dry bulb and relative humidity that feel the same as that dry bulb temperature at 50 percent relative humidity. Notice the lines on the comfort chart below reflect this concept.

- Humidity affects how we perceive comfort
- Effective temperature (ET) is the temperature and humidity that feels the same as that dry bulb temperature at 50% relative humidity
- In winter we feel more comfortable at a lower temperature if the dew point is higher
- In summer we feel more comfortable at a higher temperature if the dew point is lower

Figure 32

Effective Temperature

The Comfort Chart

ASHRAE has plotted, on the psychrometric chart, an envelope of the acceptable range of temperature and humidity based on clothing and activity levels in a typical office environment. This data allows the designer to select operating conditions that should achieve the 80/20 rule for comfort. The effective temperature results in the sloping angle to the sides of both the upper and lower limits of the operative temperature. The impact of clothing can be seen in the difference in the limit values being different for summer and winter.

- Do not exceed 60% rh
- Lower limit rh is not required but it affects comfort
- Energy codes require 5 degrees between cooling and heating set points

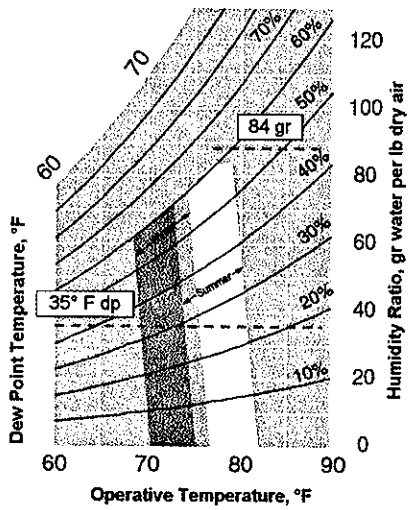


Figure 33

ASHRAE Comfort Chart

Notice that the line does not extend above 60 percent relative humidity or 84 grains. Above 60 percent, comfort is usually compromised and the potential for mold growth and indoor air quality issues are likely. A dotted line at 35° F dew point represents a design goal for a lower limit of space humidity. However, the current ASHRAE comfort envelope has removed the lower limit requirement due to concerns over condensation on cold walls and glass. Several studies have shown that maintaining the space humidity at the line represented by the 35° F dew point will help avoid complaints often associated with very dry air.

Also notice that the summer and winter conditions overlap in the 73° F to 75° F range. One might assume that this temperature range then would be a good compromise for both winter and summer with one set point for both. However, energy codes require a 5-degree temperature difference between summer and winter control points. Designers need to consider both the summer and winter control set points.

Conditions that Impact Comfort

A number of conditions could exist that will impact comfort if the settings from the comfort chart are applied. The first is the impact of air motion. When the air motion is increased the evaporation is also increased. As a result, in areas where the higher air motion can be tolerated, the temperature may be up to 7° F higher than indicated in the chart. This has the most impact in high activity level applications. For lower activity levels, like the office environment, this should be less than 5.5° F.

While air motion helps with the cooling effect, airflow that exceeds the velocity limits shown may result in drafts that could be objectionable. This is the case if air motion is such that papers move on a desk. Air velocity in excess of 100 fpm is not recommended.

- Air Motion
  - Control temperature can be compensated by up to 7° F
  - Drafts over 100 fpm maintain between 25 and 40 fpm, depending on temperature, turbulence and activity level
- Head-to-Toe difference of 5° F
- Temperature Cycling > 4° F/hour
- Temperature Drifts > 1° F/hour

Figure 34

Non-uniform Impacts

Another phenomenon that can result in discomfort is too great a thermal gradient from head to toe. Warm air rises and cold air settles based on the thermal density currents. With a great difference in this temperature, discomfort is sensed even when the temperatures at both extremes are within the comfort zone. A head-to-toe temperature difference should not exceed 5° F.

People are also sensitive to the time rate of change in temperature. Many systems control capacity by cycling stages of cooling or heating capacity. It has been determined that this rate should not exceed a change of greater than 4° F based on a maximum of one cycle every 15 minutes. The peak-to-peak differences in cycles per hour should also not exceed 2° F. These issues are associated with the system's ability to control temperature at part load conditions.

A similar issue is related to the amount of intentional drift or ***swing*** a system is allowed to exhibit. Drift is intentionally letting the control temperature not meet the thermostat set point. This can be done by offset in the thermostat or by control strategies that reduce cooling capability during the occupied hours. To maintain comfort, the temperature drift should not exceed 1° F per hour.

The final impact on how comfort is perceived is the impact of radiant surface temperatures. When the temperature of one surface is warmer than another surface it radiates heat energy to the cooler surface. If we consider a room and a person in the room, if their body is at a higher temperature than all the surfaces in the room, the body will radiate heat energy to all the surfaces. In most actual applications each of the room surfaces is at different temperatures.

Radiant heat exchange is governed by three key principles. First, the heat transfer is based on the temperature difference between the body and the surface. The greater the temperature difference the greater the heat transfer. The second principle is that the heat transfer is a function of the distance squared so the further away the surface is from the person the less the impact. Third, the heat transfer is a function of the absorption (ability to absorb heat) of the surface and the angle between surfaces. It should be obvious that the calculation of radiant effects is very complicated. While the equations are available to calculate the radiant impacts, a few simple rules will help designers maintain comfort under most conditions.

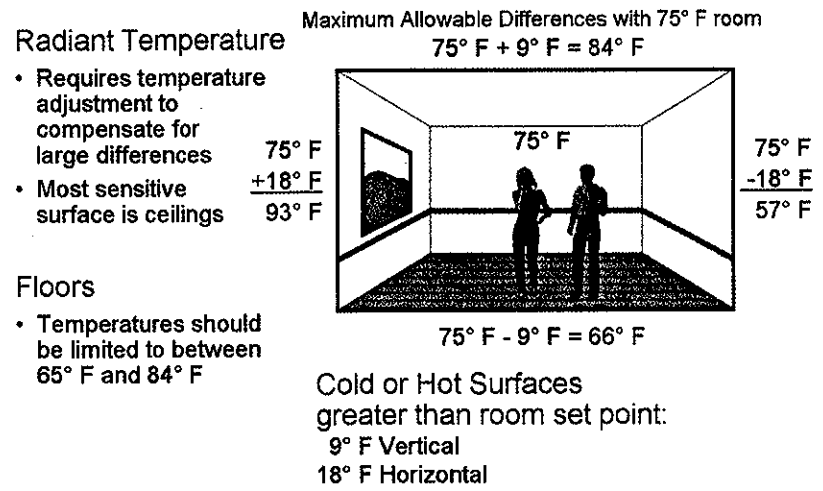


Figure 35

Surface Temperatures Influence Comfort

People tend to be most sensitive to ceiling temperatures. If the ceiling and floor temperatures are within 9° F of the room control point, and wall, glass, and other horizontal surfaces are within 18° F, comfort conditions should be maintained.

Floor temperatures present a similar problem, and with normal footwear if the floor temperature is kept between 65° F and 84° F, comfort conditions should not be compromised.

Comfort System Requirements

Now that comfort standards have been defined, how does a designer go about determining what the actual requirements should be for a project?

Figure 36 shows a form with the data a designer should consider in preparing the comfort requirements for a project. One of the first steps in design is to define the comfort requirements and the ranges acceptable to the owner. Ranges that are tighter than actually required can result in excess cost, both initially and for operation. Likewise, understated requirements result in occupants who are uncomfortable, unhappy, and often less productive.

The first step is to identify the type of space by usage and the activity to be performed. Buildings often have several spaces with different activity and usages; requirements need to be established for each.

Comfort Design Standards

Space / Activity						
Temperature	Occupied		Unoccupied		Throttling Range	Swing
	Design	Min	Design	Min		
Summer						
Winter						
Humidity	Design	Max	Min			
Summer						
Winter						
Air Motion	Max	Min	Notes			
fpm						
Ventilation	Occupied		Unoccupied		Notes:	
	Design	Min	Min			
cfm						
Direct Exhaust						
Filtration	Condition		MERV	Type	Other	
Radiant Surfaces	Glass	Wall	Floor	Roof	Other	
Summer						
Winter						
Acoustics	Room RC		Other Notes:			
Notes:						

Figure 36

Defining Comfort Requirements

Next, determine temperatures that should be maintained in the space. Temperature can be determined from the comfort chart with the appropriate adjustments for activity level, clothing and the like. It is necessary to consider a temperature requirement at four operational points, both for summer and winter. First, occupied temperature both an upper (cooling) and lower (heating) control point during summer operation must be determined. Remember that a 5° F offset is normally required between heating and cooling. The second set is the control point for unoccupied conditions for both heating and cooling. Most commercial buildings require that the set point of the thermostat be ***setup*** for cooling and ***setback*** for heating. In some buildings, not all these heating and cooling conditions are controlled during certain seasons of the year. For example, heating may not be available during summer seasons, or during summer unoccupied periods, cooling may be shutoff and the space temperature allowed to drift to whatever temperature it will. The strategies used are often dependent on the location of the project.

The same set of four conditions also needs to be evaluated in winter operation. Remember, because of the conditions reflected on the comfort chart, the conditions may be different for each season. When making the temperature decisions think about changes in activity level and clothing during each of the seasons.

Finally for temperature, consider how much ***throttling range***, or thermostat drift, will be allowed. Also consider the amount of temperature swing allowed. Swing is the intentional offset in space temperature above the thermostat setting. These two items are how much above or below the selected control setting a temperature can be and still be considered in range. Typically, no more than 3 degrees should be allowed for either of these two items.

In a similar manner the humidity requirements for the space need to be determined. Consider here special requirements that may exist because of materials in the space or activity. In some cases, even though the prime function of the design is comfort, materials and processes in the space require certain temperature and humidity ranges. In selecting humidity requirements, use caution that a maximum of 60 percent or less can be maintained to help avoid indoor air quality problems. On the lower end of the humidity scale, consider if humidification might be required. In typical office spaces, conditions within the comfort chart are normally acceptable. Humidity levels can be specified in terms of relative humidity or a dew point. Dew point is more appropriate when the absolute level of humidity in the space must be held to a set limit.

Next, consider the air motion. If it is traditional office space, then the range of 15 to 45 fpm is acceptable. But, if greater air motion can be used to compensate for the temperature level this is a time to consider it.

Ventilation requirements are often dictated by building codes. This phase of the design is a good time to check the requirements and determine the required airflows. ASHRAE requirements allow for airflows to meet both the building requirements and the requirements for people. The building requirement is air to purge the gases that outgas from building materials and maintenance activities. The people requirement is for oxygen replacement and odor dilution. This is addressed in greater detail in other TDP modules. These are minimums and greater rates may be required in some instances. Also, be sure to note any air that is being exhausted directly from the space. Lab and kitchen hoods, bath and kitchen fans and exhaust fans from processes may remove air from the space. Ventilation air, by definition, must at least replace this amount of air.

Any special requirements in terms of filtration should be addressed in the selection of comfort criteria. An example is activities occurring in the space that generate some type of particulate or gaseous contaminant that needs to be filtered. This is a good time to define these issues and what levels of filtration may be required. Filters can be used for particulate, gaseous, and biological contaminants.

In this phase of the design, it is a good time to evaluate if some of the surfaces may present radiant problems in either heating or cooling. Calculate surface temperatures for the surfaces that may present a problem. Since the HVAC systems can only be designed to compensate for this problem, it may be early enough in design that the building materials or design can be modified to help overcome potential problems.

Finally, consider any acoustical requirements of the space. This issue is covered more completely in TDP-901, Vibration and Acoustics. All six of the conditions related to comfort have now been considered for the project.

Defining Comfort Conditions

First, let's look at defining the comfort conditions in a typical office environment. Temperatures are selected based on the ASHRAE comfort chart. Notice a 5-degree difference is maintained between heating and cooling, and a lower winter temperature is selected to account for typical clothing difference in the two seasons. It has been determined that this project does not need to provide unoccupied cooling in the winter and does not require heating in the summer.

Comfort Design Standards

Space / Activity	Office					
Temperature	Occupied		Unoccupied		Throttling Range	Swing
	Design	Min	Design	Min		
Summer	75	70	85	0	1 1/2	0
Winter	70	68	0	60		
Humidity	Design	Max	Min			
Summer	50	60				
Winter	None		None			
Air Motion	Max	Min	Notes			
fpm	30	15				
Ventilation	Occupied		Unoccupied		Notes:	
	Design	Min	Min			
cfm	20 / P	0.25 / ft²	0			
Direct Exhaust	0.25 / ft²					
Filtration	Condition		MERV	Type	Other	
	Standard		6	Pleat		
Radiant Surfaces	Glass	Wall	Floor	Roof	Other	
Summer	84F @ 95F					
Winter	38F @-10F				35 % glass	
Acoustics	Room RC		Other Notes:			
	RC 35 - N					
Notes:						

Figure 37

Example of Comfort Requirements for an Office

Humidity requirements are again from the ASHRAE comfort chart. Typical of many office environments the design team has decided that humidification will not be provided.

Air motion is the typical design range of 15 to 45 fpm.

Ventilation air provides 20 cfm/person in the occupied period and 0.25 cfm/ft^2 during the unoccupied period. The bathroom fans exhaust the same amount as the building ventilation requirement.

The building has no special filtration requirements.

The glass selected has a summer design surface temperature of 84° F or 9 degrees above the room, about half the maximum allowed. However, the winter temperature has a 30-degree difference and this is a potential problem. Maybe better glass could be specified. If not, the winter heating set point may need to be increased.

In a different application we have a light manufacturing area. In this case the manufacturing area is working with cardboard boxes. Again the primary goal of air conditioning is the comfort, and productivity of the workers. However, because of the paper boxes and the process, a few special requirements are necessary.

Higher temperatures are allowed because the workers are more active than the office workers in the last example. Humidity, however, has a lower limit because of handling paper; in this case winter humidification may be required.

Other special requirements are that an exhaust hood on the curing oven requires 300 cfm of exhaust air, and because of the curing process of the glue, continuous ventilation is required. The boxes also generate a lot of paper particle dust. To control this, a better filtration system is required. With this information, a comfort design chart similar to the one for the office can be generated.

After going through this process for all the building spaces the designer has a good picture of what the total air-conditioning design must do to maintain comfort. In the next section the two interdependent elements of the air-conditioning system and the components of each will be examined.

This completes our coverage of basic human comfort. Before you continue, please complete the Work Session 1. Work session answers can be found in the back of the book.

Comfort Design Standards

Space / Activity	Box Manufacturing Area					
Temperature	Occupied		Unoccupied		Throttling	Swing
	Design	Min	Design	Min	Range	
Summer	78	70	0	0	2	3
Winter	68	65	0	55		
Humidity	Design	Max	Min	Storage Requires Min of 25%		
Summer	45	55				
Winter	30		25			
Air Motion	Max	Min	Notes			
fpm	30	15				
Ventilation	Occupied		Unoccupied		Notes:	
	Design	Min	Min			
cfm	20 / P	0.5 / ft^2	.025 / ft^2		Oven Exhaust	
Direct Exhaust	300 cfm					
Filtration	Condition		MERV	Type	Other	
	High Particle		10	Bags		
Radiant Surfaces	Glass	Wall	Floor	Roof	Other	
Summer	84F @ 95F			83F @ 95F		
Winter	38F @-10F				35 % glass	
Acoustics	Room RC		Other Notes:			
	RC 50 - N					
Notes:						

Figure 38

Example of Comfort Requirements for Light Manufacturing

Air-Conditioning Systems

Now that comfort has been addressed, as well as the controllable space conditions that determine comfort, we will turn our attention to the air-conditioning system. First we will present the systems approach to air-conditioning design. Then we will deal with the air system to show how load is absorbed by conditioned air as well as how air is treated to provide air motion, ventilation, filtration, heating, cooling and dehumidification. Finally, the mechanical refrigeration system, which provides the capacity necessary to cool and dehumidify the air supplied to the conditioned space, will be discussed.

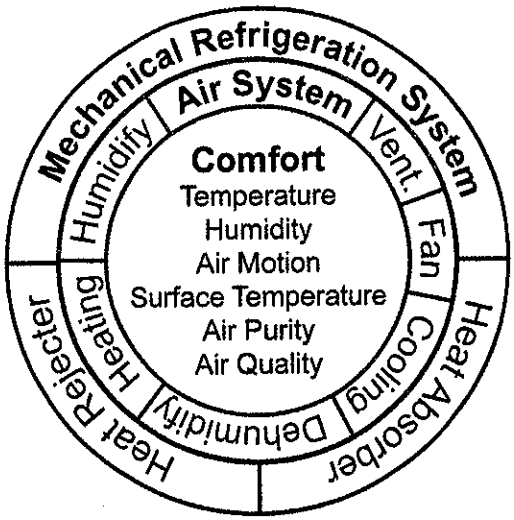
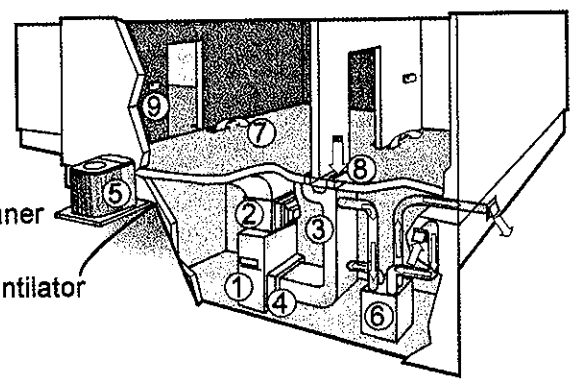


Figure 39

Interaction of Comfort to Air and Refrigeration Systems

Many air-conditioning system designers think in terms of components. However, the systems that provide the maximum comfort for the money invested are those designed by people who think in terms of systems and subsystems rather than components.

Not only does a systems approach benefit the customer, but it also makes air-conditioning design easier to understand. Once the basic systems, subsystems, and components are understood, the large array of options offered in our industry can be easily recognized as variations on a few basic themes.



1. Furnace
2. Evaporator
3. Humidifier
4. Electronic Air Cleaner
5. Condensing Unit
6. Heat Recovery Ventilator
7. Supply Diffusers
8. Return Grilles
9. Comfort Controls
   - Thermostat
   - Humidistat
   - HRV Controller

Figure 40

Typical Residential System Components

The residential air-conditioning system shown here illustrates these concepts quite well. Many types of ***air handlers*** and condensing units exist. But even though their size, shape and location may change, these components provide the same function and relate to other components in their respective systems the same way whether a 3-ton residential system or a 60-ton commercial system is considered.

Since several components are interconnected as a system, the action of one component affects many or all of the others in some way. Thus, decisions made in selecting or installing one component affect the other components. Trade-offs are involved when making system decisions. What is beneficial for one component may be disadvantageous for another. The designer who uses a systems approach makes choices which produce the greatest benefit for the system overall.

Some components belong to more than one system. For example, the ***evaporator*** (cooling coil) is part of three systems: the air system, the mechanical refrigeration system, and the cooling system. Since such components have multiple purposes, they must perform acceptably in all systems in which they occur and in all modes of operation.

As a part of the air system, the evaporator must be selected and installed in a way that permits an even flow of supply air through it. Also, its air pressure drop, when wet, must be included in the summary of resistances to airflow that the fan must overcome.

As part of the mechanical refrigeration system, the evaporator must be piped correctly to the ***refrigerant metering device*** and condensing unit to ensure reasonable internal operating temperatures for the evaporator at both full and partial cooling loads.

System	Components	Subsystems
Air System	Furnace Evaporator Electronic Air Cleaner Humidifier Heat Recovery Ventilator Supply Diffusers Return Grilles	Ductwork Electrical Power Wiring Electrical Control Wiring
Mechanical Refrigeration System	Evaporator Compressor Condenser Metering Device	Refrigerant Piping Condensate Piping Electrical Power Wiring Electrical Control Wiring
Heating System	Furnace or Fan Coil Humidifier Thermostat Humidistat Heat Recovery Ventilator HRV Controller	Ductwork Gas/Oil Piping Flue Gas Venting Electrical Power Wiring Electrical Control Wiring Condensate Piping
Cooling System	Furnace or Fan Coil Evaporator Metering Device Condensing Unit Thermostat Heat Recovery Ventilator HRV Controller	Ductwork Condensate Piping Refrigerant Piping Electrical Power Wiring Electrical Control Wiring

Figure 41

Comfort systems consist of systems, components and subsystems.

As a part of the cooling system, the evaporator must combine with the metering device and condensing unit to produce adequate, but not excessive, capacity. Both sensible and latent cooling loads must be met. The condensate, which accumulates as the evaporator cools the air, must be removed from the coil and properly disposed.

Air System

Let's start our study of the air-conditioning system with the air system.

When we are outside, we depend on clothing and weather patterns for our comfort. If the combination of temperature, relative humidity, air motion, and clothing is within a reasonable range, we feel comfortable. Our bodies reject just the right amount of heat to maintain our inner temperature in a stable way. But if the combination of these conditions is not right, discomfort results. If clothing cannot solve the comfort problem, there is nothing we can do to change the outside conditions. Comfort is not achieved.

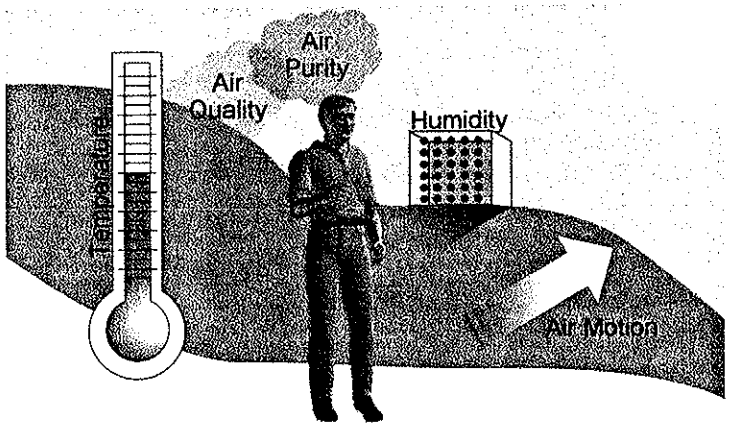


Figure 42

The air system controls five conditions of space comfort.

Building Loads

When we seek shelter in an enclosed space, we can control the air conditions in a way that assures comfort over a very broad range of outside conditions. If the building was perfectly isolated from outside effects, and only people were present, we could maintain comfort by simply removing the heat given off by the people. But, in reality, the situation is not that simple. No matter how diligently the building designer strives to insulate the building from outside effects, it is still influenced.

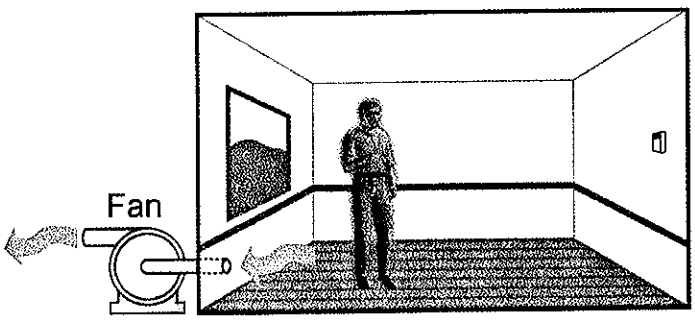


Figure 43

In a perfect world, only heat generated by people would need to be removed.

In addition to body heat, other heat gains enter the space from within the building. Electric lights, appliances, computers, copy machines, heated food and drinks, and any equipment that is motor-driven adds heat to the air within the building. All these items tend to increase the temperature and some of them increase the humidity of the indoor air, slowing down the body's ability to reject heat, and eventually causing discomfort. The problem does not end with internal effects.

- People
- Lights
- Motors
- Appliances
- Cooking
- Computers
- Process

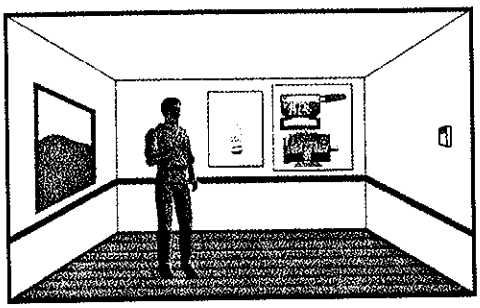


Figure 44

Internal Heat Gains

No building is perfectly isolated from outside effects. Sunlight enters the space in the form of radiant heat energy through windows or glass walls, and glass doors. Once sunlight strikes an object inside, it turns into heat that moves into the air by conduction and convection. Sunlight also strikes some parts of the exterior of the building. Outside walls and roofs that are sunlit warm up and conduct heat into the building. All these effects are called solar loads.

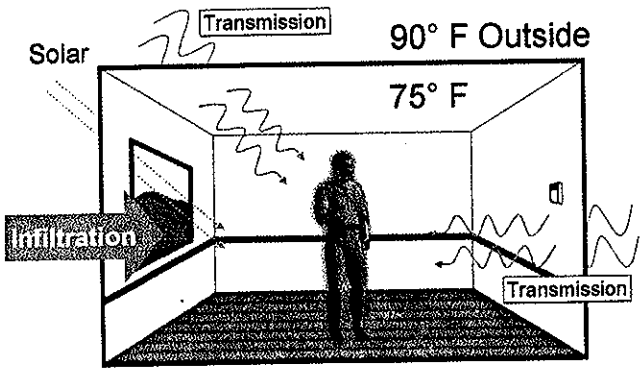


Figure 45

External Heat Gains

Infiltration vs. Ventilation

Likewise, when the outside air temperature is higher than the inside temperature, heat is conducted inward, through all exterior building barriers, adding further to the heat entering the building. Heat transferred in this way is called ***transmission load***.

Outdoor air infiltrates (leaks) into the building through cracks around windows and doors. Fans that mechanically ventilate the space also bring in outdoor air. Since the temperature and humidity of the outdoor air is often quite different than the indoor air, infiltration and ventilation can change indoor temperature and humidity in a significant way. The air-conditioning system must offset the outdoor air load in order to maintain comfortable inside conditions.

Together, the outside effects of solar, transmission, infiltration and ventilation, along with the inside effects of people, lights and equipment determine the amount of heat entering or leaving the building per unit of time. This rate of heat transfer is called the load of the building. It is expressed in British thermal units per hour (Btuh). Just as the total of all these effects is called load, each of the outside and inside components that make up the total load also is called a load (e.g. solar load, transmission load, lighting load, equipment load, people load, and outdoor air load).

Any effect that increases space temperature or humidity is called a cooling load. Solar load, people, equipment, and lighting are always cooling loads, as shown in the example illustrated in Figure 46.

Any effect that decreases space temperature is called a ***heating load***. Transmission, infiltration and ventilation are heating loads when the outside temperature is below the inside temperature. Transmission, infiltration, and ventilation are cooling loads when the outside temperature is above the inside temperature

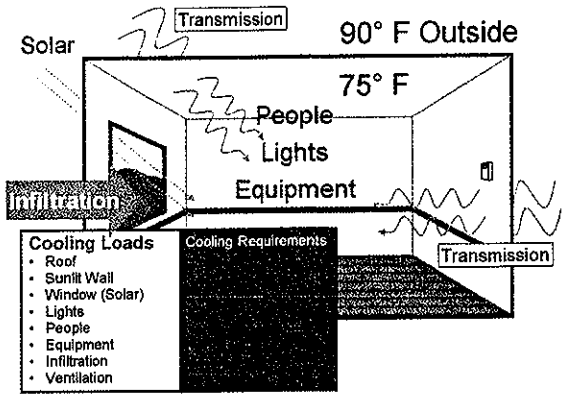


Figure 46

Building Loads – Internal Gains Plus External Gain

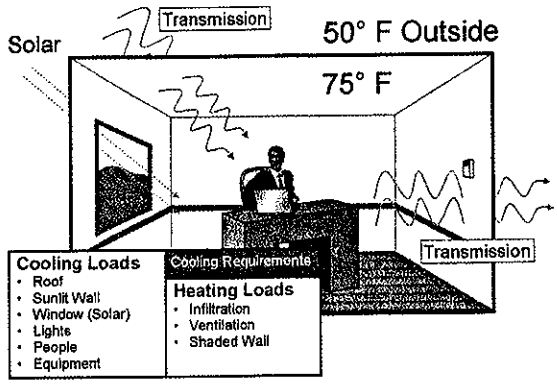


Figure 47

Cooling Loads – If Heat Gains are Greater than Heat Losses

Cooling load vs. heating load

The total load of the building is the combined result of all the individual load components working together. Often, inside and outside conditions combine in such a way that cooling load components exist at the same time as heating load components as shown in Figure 47.

If the combined result causes heat to enter the building more rapidly than it leaves, the entire building is said to have a cooling load. The space temperature and humidity will climb unless something is done to reject heat and moisture from the indoor air. This is the case in the example illustrated.

Since it is 50° F outside, the outside wall that is not sunlit has a heating load. Likewise, outdoor air that comes into the building represents a heating load since the outside air temperature is below the inside air temperature. At the same time, the sunlit glass, wall, and roof present a cooling load, as do the computer (equipment), lighting, and the person in the room. In this case, the cooling loads outweigh the heating loads.

Likewise, if the combination of all load components causes heat to leave the building more rapidly than it enters it, the entire building has a heating load. The space temperature and humidity will fall unless something is done to add heat and humidity to the indoor air.

In the example, the lights, people and any equipment provide cooling loads. However, the heating load components outweigh the cooling load components. The roof, walls, window, and any outdoor air that comes into the building are all heating loads. Since it is nighttime, there is no solar load. The building has a heating load under these conditions.

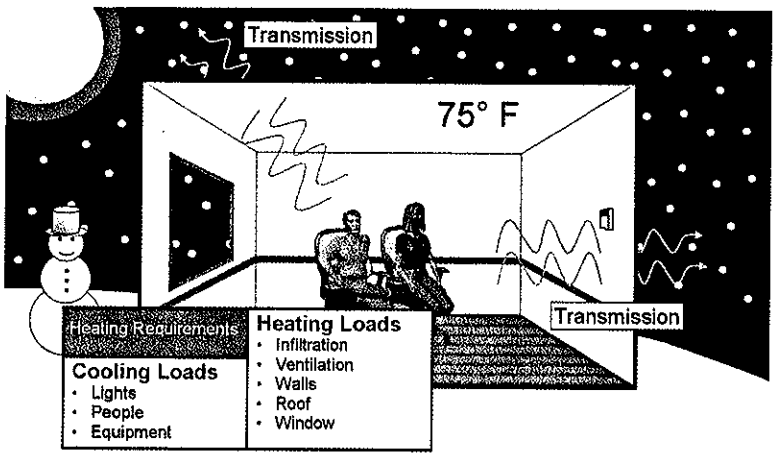


Figure 48

Heating Loads – If Heat Losses Exceed Heat Gains

Air System – Components

The job of the air-conditioning system is to match the heating or cooling load with heating or cooling capacity necessary to maintain desirable inside conditions. Many heating-only systems accomplish this by using a boiler with ***convective baseboard*** heaters. No fan is required. The same thing is accomplished with baseboard electric resistance heat.

When cooling, dehumidifying, filtering and direct control of air motion are required, however, an air system becomes essential. Only with the air system can the space temperature, humidity, and air motion be simultaneously controlled, year round.

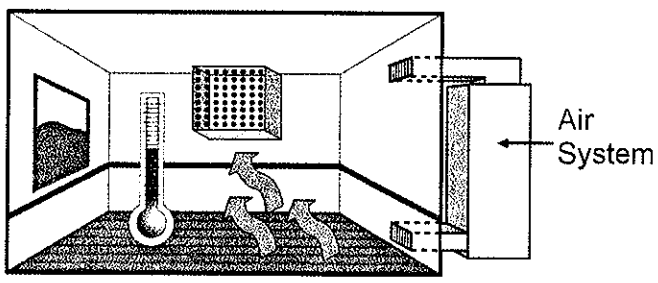


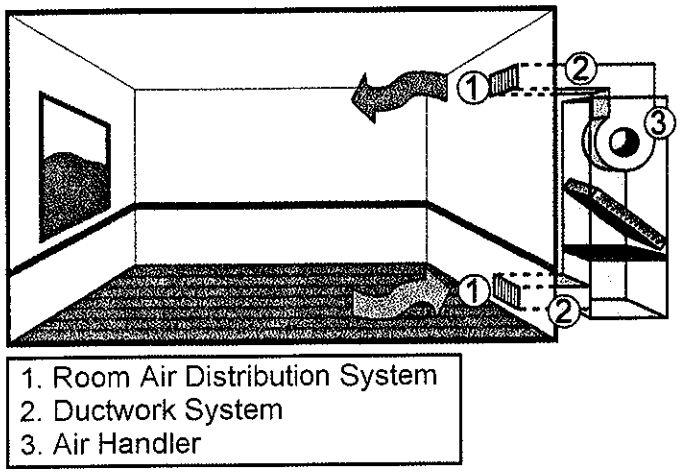
Figure 49

Air systems maintain air motion, temperature, and humidity.

Actually, a comfort air-conditioning system must simultaneously maintain, within a reasonable range, all three air conditions affecting the rate of heat loss from our bodies: temperature, humidity, and air motion. The system also must provide adequate air quality by properly ventilating and filtering the indoor air.

A comfort air-conditioning system controls these parameters by moving air through the building. The air system accomplishes this task. The air system is usually made up of three subsystems: the room air distribution system, the ductwork system and the air handler. The air handler ***circulates*** the proper quantity of air, at the right temperature, humidity, and purity to maintain the desired conditions.

The following section shows how the air system maintains comfort conditions in the air-conditioned spaces within the building.



1. Room Air Distribution System
2. Ductwork System
3. Air Handler

Figure 50

An air system is made up of the room air distribution system, the ductwork system, and the air handler.

Room Air Distribution

The room air distribution system is made up of the supply air outlets and return air inlets in the conditioned spaces of the building. The air motion in a room is affected by the openings through which conditioned air is supplied. These supply air outlets are usually called diffusers or ***registers***.

The number, size, shape, and location of the supply air diffusers all affect room air motion significantly. The air distribution aspect of comfort air-conditioning design is very important, yet it is often neglected. The very best equipment will perform only as well as the air distribution system that supports it. Proper design of ductwork and return openings are also vitally important but will not offset errors made in the selection and layout of the supply diffusers.

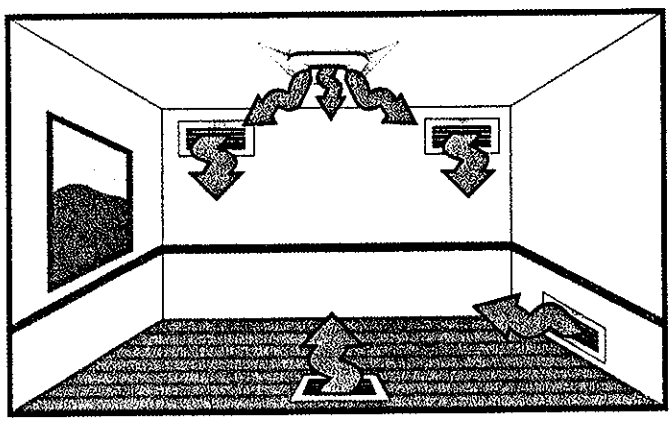


Figure 51

Supply Diffusers and Registers

Return air openings are usually called grilles. The total airflow capacity, measured in cubic feet per minute (cfm) of the return grilles must be at least equal to the total airflow capacity of the supply diffusers because the air returned becomes supply air after being conditioned by the air handler. The performance of the return grilles is less sensitive to location and number than supply diffusers.

At the air handler, air returned from the conditioned space is mixed with outdoor air (ventilated), filtered and heated or cooled, as the building load indicates. The first treatment the return air receives when it enters the air handler is an improvement of its quality by mixing it with outdoor air.

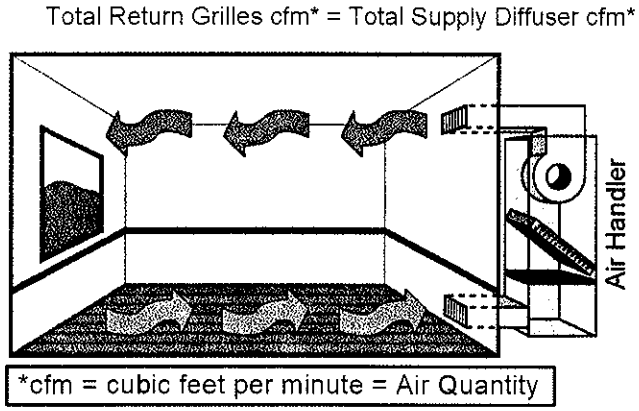


Figure 52

Return Grilles – air provided must be equal to air removed.

Ventilation

The process of mixing outdoor air with return air at an air handler and supplying it to the building spaces is called mechanical ventilation. It dilutes odors and pollutants in the conditioned spaces of the building. It also replaces oxygen used by people as they breathe the air in the space.

Many residential systems lack mechanical ventilation. They rely on infiltration to do the job instead. Ventilation is considered so important that some commercial systems are supplemented by a separate system that provides only ventilation air. Also, ventilation is being included on an increasing number of tightly constructed residences.

While mechanical ventilation is not the norm on residential systems or some small-zone commercial equipment, it is required on commercial systems.

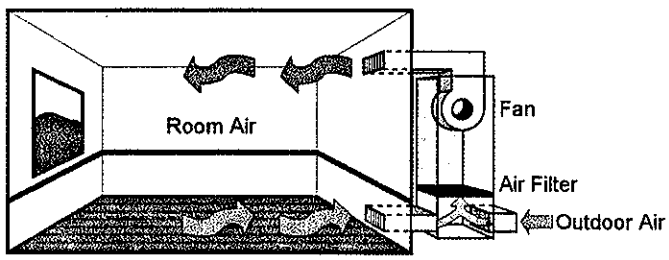


Figure 53

Mechanical ventilation replaces oxygen and dilutes odors.

Filtration

After return and outdoor air are mixed to provide ventilation, it is drawn through a filter, which purifies the air. Many types of filters are available. At the least, they remove dust and dirt from the air before conditioning and supplying it to the space. More efficient filters that remove pollen, bacteria, smoke and even harmful and irritating gases, are available. After mixing and filtration, the air is ready to be heated or cooled, and humidified or dehumidified.

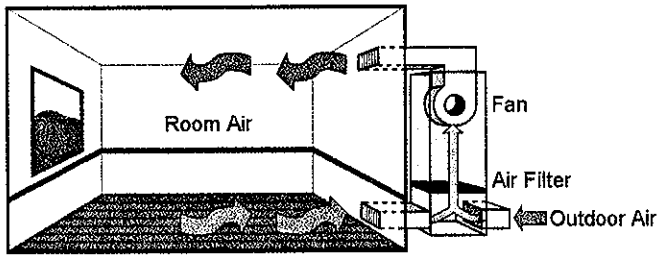


Figure 54

Filtration purifies the mixed return and outdoor air.

Heating

If there is a heating load, heat must be added to the air to make up for the heat lost from the room. A heating coil within the air handler adds this heat, raising the temperature of the supply air above the room temperature. Air typically leaves the supply diffusers at a temperature of 90 - 140° F. As shown here, the heat exchanger that adds heat to the air can be located on the discharge side of the fan in a ***blow-thru*** configuration. It also can be located on the inlet side of the fan in a ***draw-thru*** configuration. Residential air handlers usually put the heat exchanger in the blow-thru arrangement. Commercial products vary the heat exchanger's location based on size and style of equipment.

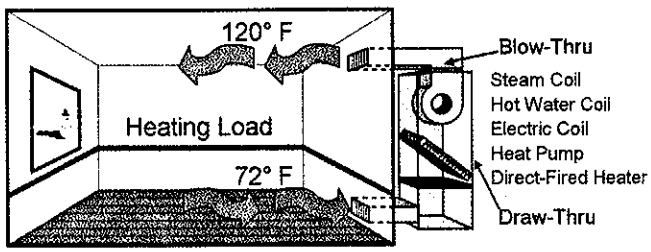
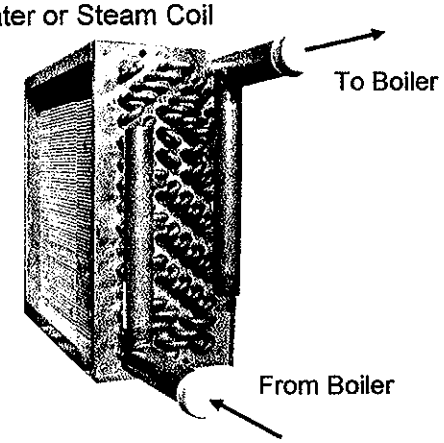


Figure 55

Heating Locations – before (draw-thru) or after (blow-thru) the fan

1. Hot Water or Steam Coil



There are four common methods of heating air within an air handler. One way is to pass air through a hot water or steam coil that is connected to a boiler. As the air passes over the surface of the coil, it absorbs heat from the coil. The heat in the air will be used to offset heat loss as it circulates through rooms with a heating load.

Figure 56

Hot Water or Steam Heating Coils

Another way to heat the air is with an electric resistance heating coil. The wire elements of the heater heat up due to resistance when electricity passes through them. As the air passes over them, it absorbs the heat and carries it to the conditioned space.

2. Electric Resistance Heating Coil

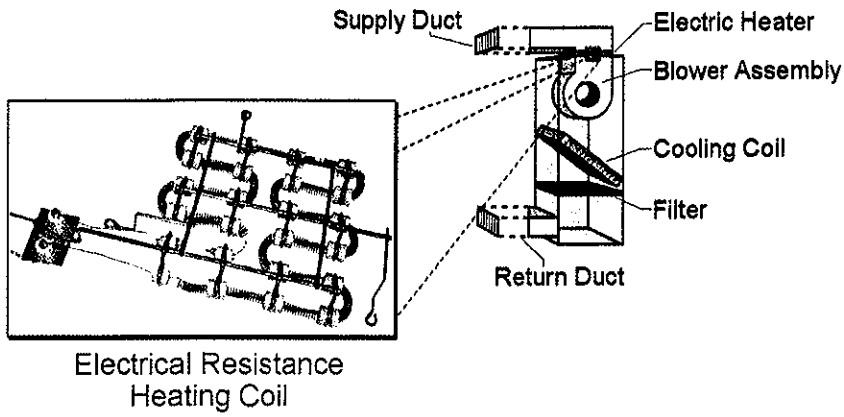


Figure 57

Electric Resistance Heating Coil

Another popular way to heat electrically is with a ***heat pump*** coil. It uses much less electricity than electric resistance heaters. The heat pump indoor coil looks just like a cooling coil. In fact, it does double duty, serving as a cooling coil in the cooling mode and as a heating coil in the heating mode. It accomplishes this by a reversal of the ***refrigerant*** flow in the mechanical refrigeration system. An electric resistance heater usually accompanies the heat pump indoor coil for the times when the heat pump coil cannot supply the needed heating capacity.

3. Heat Pump Indoor Coil (Heat/Cool)

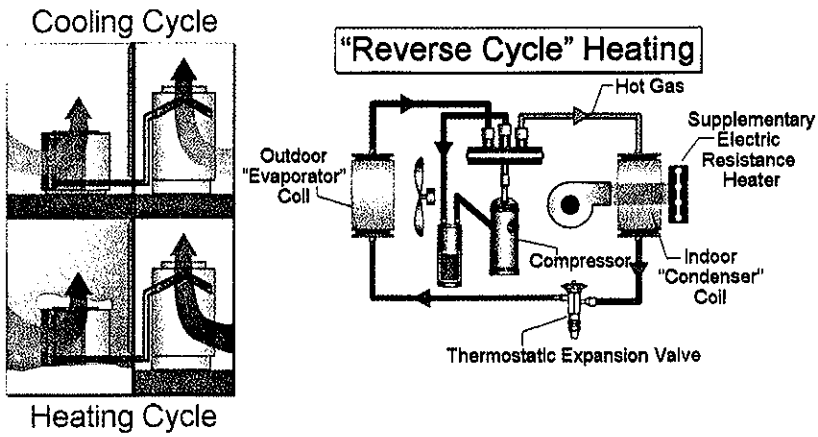


Figure 58

Reverse Cycle Heat Pump

Finally, air may be heated by a flame within a heat exchanger over which the air flows. This approach is called direct-fired or fossil fuel-fired heating since fossil fuels are burned in the heat exchanger over which the air handler circulates air. Oil, natural gas, and liquefied propane (LP) gas are the fossil fuels used. Direct-fired heat exchangers are usually located on the discharge side of the air handler fan. This approach is popular for residences and small commercial applications using upflow, downflow or horizontal furnaces. It is also popular for commercial jobs using roof-mounted packaged equipment.

4. **Direct-Fired Heat Exchanger** (Natural Gas, Oil, LP Gas)

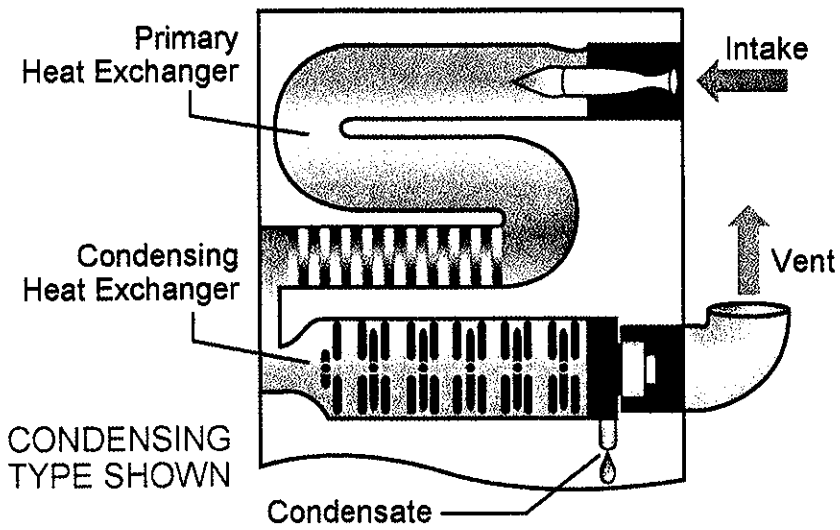


Figure 59

Gas Heaters – Direct-Fired, Gas, Oil, LP

Cooling

When there is a cooling load, air must be supplied at a lower temperature and humidity than the air in the conditioned space. This allows the supply air to absorb heat and humidity while maintaining acceptable space conditions. The air supplied to the space absorbs heat added by solar, transmission and infiltration loads from outside as well as from people, equipment, and lights from inside. It also absorbs humidity added by the infiltration of outdoor air, plus humidity added from the inside by people, hot foods, and liquids. The air handler must remove this heat and humidity so that the air can be used to condition the space again.

The air handler removes heat and humidity by passing the air through a cooling coil. The coil is made up of metal tubes with thin, metal sheets, called fins, attached to them. Cool refrigerant or water flows through the cooling coil, absorbing heat from the air passing over its tubes and fins, thus cooling the air. This module covers the internal workings of the refrigerant evaporator, not the chilled water type of cooling coil.

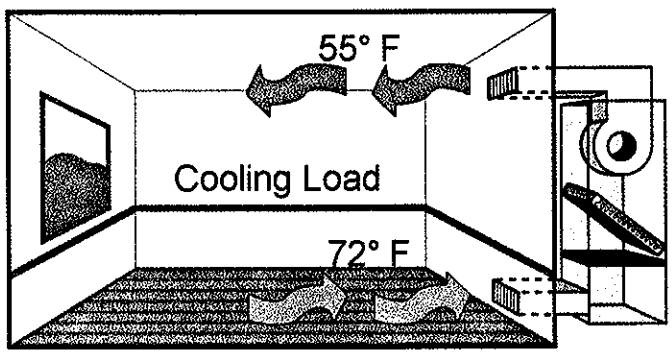


Figure 60

Cooling Air Handler

Moisture is removed from the air that flows over the cooling coil because some of the water vapor in the air turns into liquid as it hits the cold coil. As the water vapor condenses, the air is dehumidified and the condensate (liquid water) is collected and drained away from the air handler. The following example helps to demonstrate why this dehumidifying process takes place.

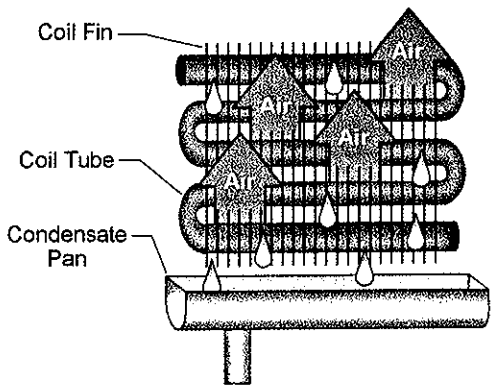


Figure 61

Cooling and Dehumidifying Coil

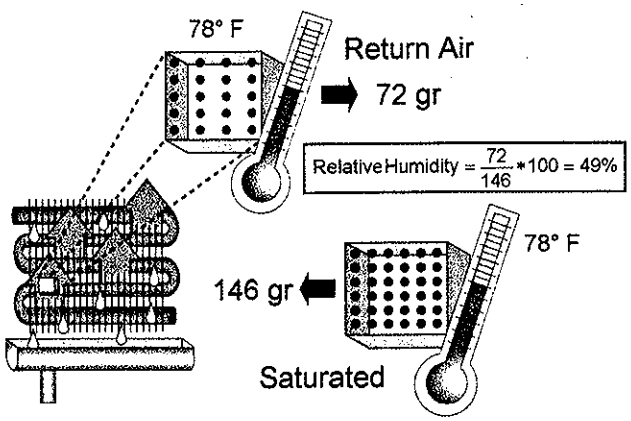


Figure 62

Relative Humidity of Return Air

If a residential system uses no ventilation air, the air enters the cooling coil approximately at room conditions. Assume that the return air conditions are 78° F and 72 grains per pound. Air at 78° F can hold 146 grains per pound of moisture when saturated with water vapor. Therefore, the ***relative humidity*** (rh) of the air entering the cooling coil is approximately 49 percent (72/146 * 100). As it enters the cooling coil, the return air is not even close to being saturated.

However, as the return air begins to pass through the cooling coil, the air temperature drops. Colder air can hold less water vapor than warmer air, so the relative humidity of the air climbs as it passes through the cooling coil. Air at 72 grains per pound (gr/lb) will have a relative humidity of 100 percent at about 58° F. At that temperature, the air will be saturated. So, if the air leaves our coil at 55° F, it will be saturated, but its specific humidity at 55° F will be 64 gr/lb at 100 percent rh. That's about 8 gr/lb below the entering air condition of 72 gr/lb. Therefore, moisture was removed from the air by the coil. Now the supply air can enter the room and absorb more moisture. Let's see how the coil removed the moisture.

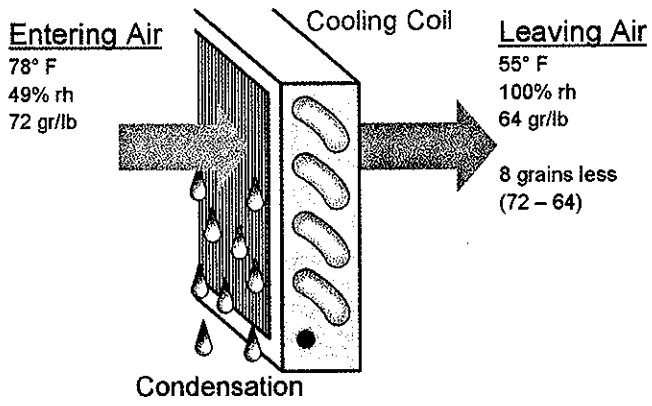


Figure 63

Cooling and Dehumidifying – Relative humidity increases, but actual moisture (grains) decreases.

At peak cooling load, most cooling coils run at a surface temperature of about 52-58° F. The air leaves the cooling coil at about 53-60° F. A typical leaving air temperature is 55° F, which will be used for this example.

The saturation temperature of the air entering the cooling coil is 58° F, as noted. At 58° F, all of the water is held within the air as vapor. If the air temperature drops below the saturation temperature of the entering air (58° F), some of the moisture within it will change from vapor into liquid.

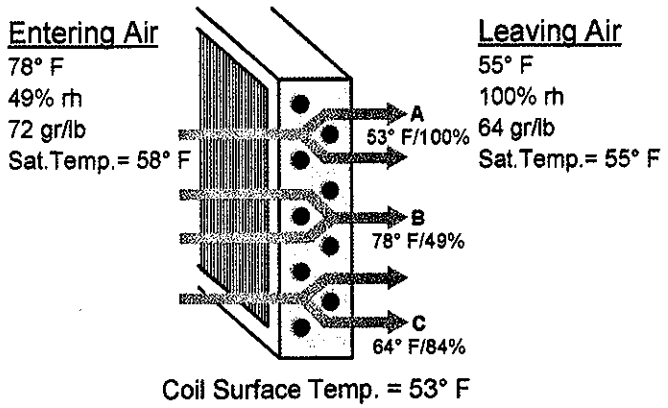


Figure 64

Actual coil performance is a mixture of several airstreams.

The air leaving the coil is really a mixture of several air streams at various conditions. Air that comes in closer contact with the coil tubes and fins mixes with air that slips by farther away. The air that is farther away from the tubes and fins is less affected by the coil surface than the air that contacts it more directly.

In reality, none of the air is brought all the way down to the surface temperature of the coil and none of it slips through totally unaltered. However, the illustration helps to show that the leaving air condition is a mixture of several airstreams treated to various degrees. The illustration also shows why various air-cooling coils have different efficiencies.

Evaporator coil terms:

Airstream "A" comes in direct contact with a coil tube. It drops to the coil surface temperature, 53° F. This is well below the saturation temperature of the entering air (58° F). Therefore, moisture is condensed on the coil tube and fin(s) connected to it. This air leaves the coil at 53° F, saturated. Airstream "B" goes through the coil without touching a fin or tube. It may come out just about the way it went in, at 78° F/49% rh with a 58° F saturation temperature. Airstream "C" leaves the coil at a condition somewhere between "A" and "B."

Air cooling coils are designed so that the vast majority of the air passing through them comes in direct contact with the coil tubes and fins. As demonstrated in the example, the leaving air temperature (55° F) is much closer to the coil surface temperature (53° F) than it is to the entering air temperature (78° F).

The air that passes through without contacting the coil is called bypassed air. It can be expressed as a decimal fraction or percent of the total air, in which case it is called ***bypass factor***.

In our example, about 92 percent of the air is directly contacted by the coil surface: ([78° F - 55° F] ÷ [78° F - 53° F]). Only 8 percent passes through without contacting the coil surface: ([55° F -53° F] ÷ [78° F - 53° F]), making its bypass factor 0.08. A lower bypass factor means that more of the air passing through the coil comes in direct contact with the coil surface, improving cooling and dehumidification efficiency. Higher bypass factors decrease cooling and dehumidification efficiency.

$$\text{Contact Factor} = \frac{(78° F - 55° F)}{(78° F - 53° F)} = 0.92 = 92\%$$

$$\text{Bypass Factor} = \frac{(55° F - 53° F)}{(78° F - 53° F)} = 0.08 = 8\%$$

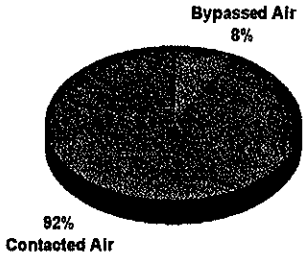


Figure 65

Bypass and Contact Factors

To lower bypass factor:

Bypass factors for comfort air-conditioning equipment vary from about 0.01 to about 0.20. A typical figure for most applications is about 0.08. Anything that increases the amount of air in contact with the coil surface will reduce the bypass factor and improve the heat absorbing efficiency of the coil, especially its ability to dehumidify.

The speed, or velocity, in feet per minute (fpm), at which air travels through the coil affects bypass factor. The slower the air moves through the coil, the better chance the air has to come in contact with the coil surface. Therefore, lower air velocity means lower bypass factor. Air velocity through the cooling coil can be reduced by selecting a larger coil or by selecting a system that can provide the necessary cooling capacity at lower airflow quantity (cfm). Bypass factor is seldom considered directly when selecting cooling equipment. Instead, the manufacturer's capacity ratings include the effect of bypass factor, without directly mentioning it. Nevertheless, the bypass factor concept helps you to understand why systems with oversized evaporators perform most efficiently.

For all cooling equipment, reduced airflow (cfm) means reduced coil velocity and total cooling capacity. However, the coil's ability to remove moisture (its latent cooling capacity) increases at lower airflow and velocity while its ability to reduce the supply air temperature (its sensible cooling capacity) decreases.

The closer together the coil tubes and fins are, the less air passes through untreated. This reduces bypass factor. When selecting applied commercial air handlers, the air-conditioning designer can select various types of mixing boxes, filters, heating coils, cooling coils, fans, and accessories for any given unit size. This allows the designer the ability to select various types of cooling coils, with a differing number of coil tube rows and fin spacing, which directly affects bypass factor. So, to a limited extent, the engineer using an applied air handler can directly select bypass factor. When using other types of air handlers, there is only one coil type offered for each capacity size. There is no choice of bypass factor.

Now compare the supply air conditions entering the room with the conditions in the example room. This shows how the supply air absorbs heat and humidity from the room in the cooling mode. For the sake of simplicity, we will assume that the conditions of the air supplied to the room are the same as the conditions leaving the cooling coil.

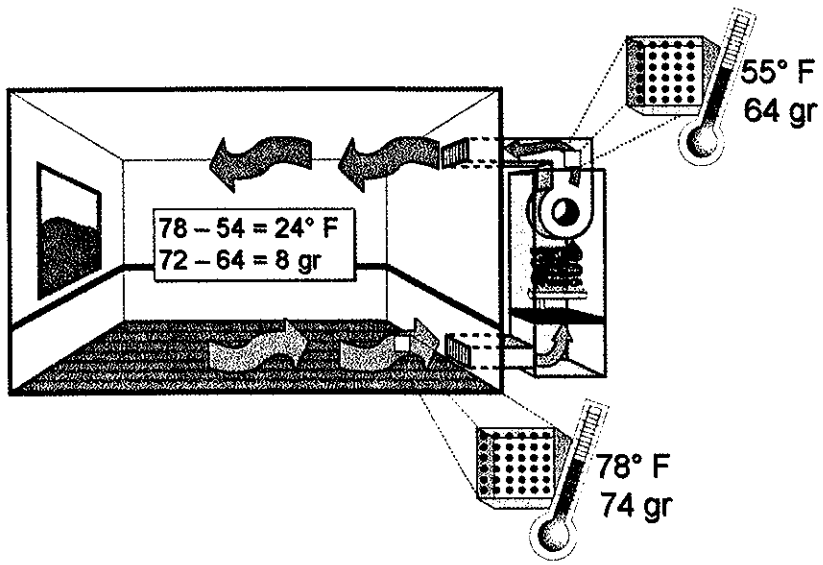


Figure 66

Supply Air Absorbing Heat and Humidity

The 55° F supply air is cooler than the 78° F room air. Therefore, the supply air absorbs heat in the room as it rises to 78° F. The supply air is also drier than the room air. It enters at 64 gr/lb, allowing each pound of air circulated to absorb 8 grains of water vapor from the room before arriving at the 72 gr/lb room condition.

In summary, when there is a cooling load in the space, cool, dry, clean air is circulated through the room where it absorbs heat and moisture. The same quantity of air is returned to the air handler that is supplied to the room. The air entering the air handler is mixed with outdoor ventilation air, when ventilation is provided, then filtered and sent to the cooling coil (evaporator), which removes heat and moisture. The condensate water goes down the drain, and

the cooling coil absorbs the heat. As long as there is a cooling load on the conditioned space and cooling capacity is provided to match that load, the cycle repeats itself.

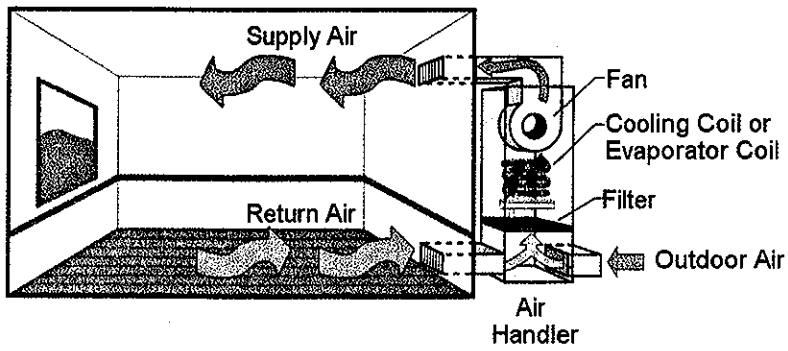


Figure 67

Air System Components

This completes our review of the air system, which is part of the air-conditioning system. As you read the next section, keep in mind that the cooling coil in the air system is the same as the evaporator coil described in the next section, the mechanical refrigeration system. This completes our coverage of the air system. Before you continue, please complete the Work Session 2 found at the back of the book.

Mechanical Refrigeration System

Mechanical refrigeration

When studying comfort air conditioning, a question that often arises is, "What makes the cooling coil cold?" The answer is, the mechanical refrigeration system. This section explains the process going on within the coil that makes the coil cold.

The mechanical refrigeration system is the portion of the air-conditioning system that provides cooling capacity. It can also provide heating capacity if a reverse cycle heat pump is used.

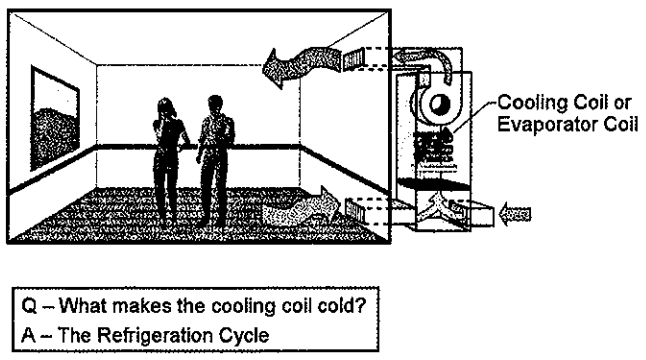


Figure 68

The refrigeration cycle provides cooling capacity and dehumidification.

Change of State Provides the Capacity

The cooling coil is cold because refrigerant is boiling at a low temperature inside the coil.

The basic physical principle behind the heat absorption by the cooling coil in a mechanical refrigeration system can be illustrated by heating a pot of water. As the water absorbs heat, its temperature rises until it reaches the boiling point. At sea level, water boils at 212° F. The heat absorbed by the water, while its temperature is rising, is called ***sensible heat***. We can measure sensible heat processes with a thermometer.

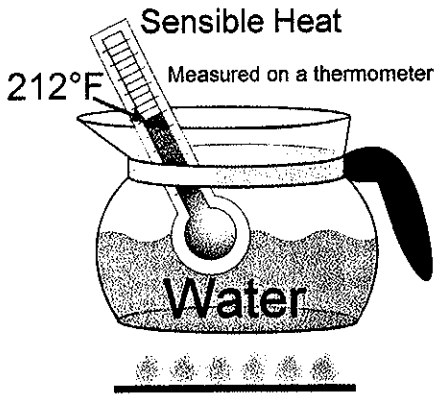


Figure 69

Sensible heat can be directly measured.

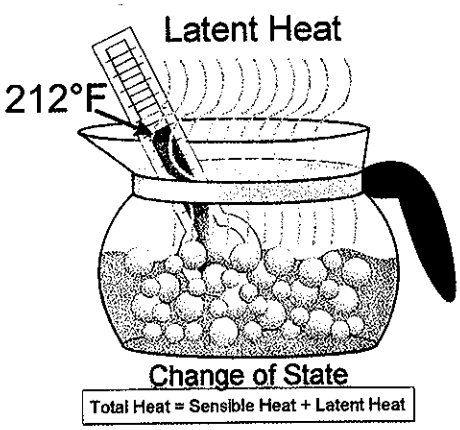


Figure 70

Latent heat changes the state and cannot be directly measured.

As the water boils, it absorbs still more heat, but the temperature does not change. The heat absorbed when a substance changes state at a constant temperature is called latent heat. You cannot measure latent heat with a thermometer because it happens at a constant temperature. Yet, it is a much bigger heat addition than the sensible heat addition required to raise its temperature to the boiling point.

Instead of raising the temperature of the water, latent heat causes the water to change state from a liquid to a vapor. This process can be observed by watching bubbles form within the body of the liquid. The vapor rising up from the pot contains both sensible and latent heat. The total heat of any fluid (liquid or gas) is its sensible heat plus its latent heat.

The first principle that makes mechanical refrigeration possible is the ability of a liquid to absorb a large amount of latent heat as it changes state at a constant temperature to become a vapor. In simple, everyday terms, this is the boiling process, but it is called evaporation in mechanical refrigeration terminology.

Conversely, a vapor must reject a large amount of heat when changing state at a constant temperature to become a liquid. This process is called condensing.

Instead of water, as used in the illustration, a chemical compound, called a refrigerant is the fluid used in the mechanical refrigeration system to accomplish this latent heat transfer. Also, instead of using an unsealed pot as a container, the mechanical refrigeration system uses a sealed cooling coil, called the evaporator, where the boiling process takes place. Instead of a burner providing the heat necessary to boil the water in the pot, air or water flowing over the evaporator provides the heat necessary to cause the refrigerant to boil. As a result, the air or water gives up heat to the refrigerant within the evaporator. The air or water leaves the evaporator at a cooler temperature than when it began to flow over it.

The second principle that makes mechanical refrigeration possible is that changing its pressure can change the boiling temperature of a liquid. The pressure scale most commonly used to measure refrigerants is pounds per square inch gauge (psig). It is the pressure read on the service technician's gauge manifold set.

Water can be used to illustrate how boiling point is determined by pressure.

Water, at a pressure of 20 pounds per square inch above atmospheric pressure (psig) boils at 258° F, instead of 212° F. If the pressure is equal to standard atmospheric pressure (0 psig), it boils at less than 212° F.

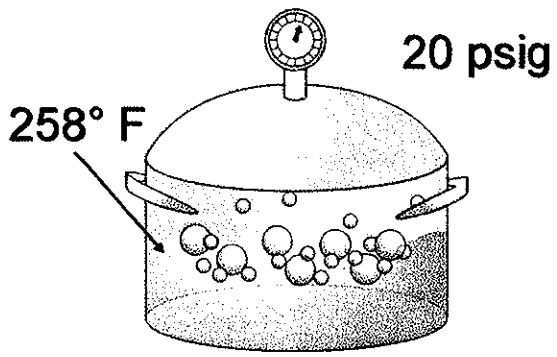


Figure 71

Pressure affects the boiling point.

Water has the same characteristics of a refrigerant, but the boiling point of water is much too high for conventional mechanical refrigeration. However, absorption refrigeration equipment uses an absorbing chemical and heat to create a refrigeration cycle that allows water to be used as a refrigerant (R-718) at very low pressures.

Four principles of mechanical refrigeration:

Heat will only flow from a higher temperature to a lower temperature.

The greatest amount of heat energy occurs during a change of state.

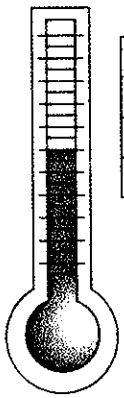
The boiling point of a substance is dependent on the pressure.

Flow only occurs if there is a pressure difference.

Raising the pressure of a fluid increases its boiling point and lowering it decreases its boiling point. The boiling point of a fluid used as a refrigerant is called its saturation temperature. Whether it is evaporating from a liquid into a gas or condensing from a gas into a liquid, it does so at the saturation temperature that matches the pressure for the refrigerant used.

Refrigerants used in the mechanical refrigeration cycle are chemical compounds that have much lower boiling points at standard atmospheric pressure than water.

R-134a (HFC-134a) boils at -15° F. HFC-134a is designed for automotive air conditioning, household refrigerators and freezers, comfort air conditioners, and transport and commercial refrigeration. R-22 (HCFC-22), presently the most commonly used refrigerant for comfort air conditioning, boils at -41° F. Refrigerant HFC-410A, a replacement for HCFC-22, boils at -62° F. Because HCFC-22 is still a commonly used refrigerant for both commercial and residential comfort work, it is used in the following sections to illustrate the mechanical refrigeration system. Comparisons are made to HFC-410A where appropriate. The mechanical refrigeration systems will function the same, however, pressures are much higher.



Water	212° F
HFC-134a	-15° F
HCFC-22	-41° F
HFC-410A	-62° F

Figure 72

Refrigerants and Boiling Points at Atmospheric Pressure

Components of the System

In most systems, the cooling coil is also called the evaporator because the refrigerant boils, or evaporates, inside the coil as it absorbs heat from the air passing over it. The refrigerant absorbs a great deal of heat as it changes state from a liquid into a vapor. The refrigerant continues to boil as it travels through the coil, until it is completely vaporized. Thus, it changes from a cold liquid to a cold vapor as it absorbs heat. The cooling action of the refrigeration cycle results from this latent heating of the refrigerant by the air, called evaporating.

The desired boiling point, or saturated evaporator temperature, is maintained by the proper combination of refrigerant type and pressure maintained in the evaporator. The manufacturer provides capacity rating data for the appropriate refrigerant at various saturation temperatures that match certain pressures. The combined action of the compressor and the metering device maintain the desired pressure in the evaporator.

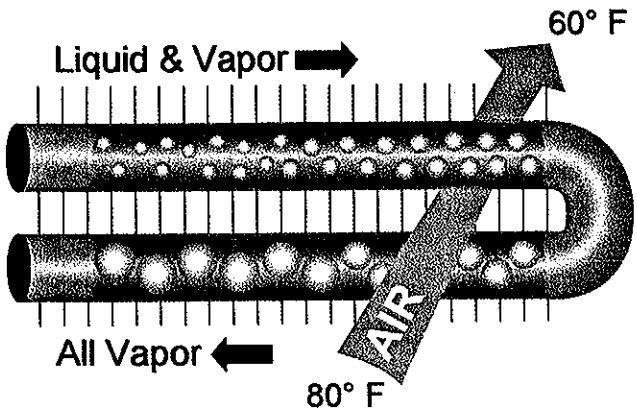


Figure 73

An evaporator absorbs the heat from the air.

If the environment or cost was not a concern, we could keep adding refrigerant to the evaporator in proportion to the cooling load and no other parts would be needed for the refrigeration cycle. However, our environment is such a major concern that such a strategy is illegal. Additionally, refrigerant prices prohibit such a wasteful action.

The rest of the mechanical refrigeration system makes it possible to use the same refrigerant over and over again. In addition to the evaporator, the major components of any mechanical refrigeration system include the compressor, a ***condenser*** and a metering device. In this case, the metering device is a ***thermostatic expansion valve*** (TXV), but many other types exist.

The same refrigerant that changes state from a vapor to a liquid in the ***condenser***, condensing at about 120° F, also changes state from a liquid into a vapor in the evaporator, evaporating at about 40° F. In order for the refrigerant to have such different saturation temperatures in two areas of the same system, there must be two different pressure zones.

The compressor and metering device work together to create a high-pressure zone, called the ***high side*** of the system, and a low-pressure zone, called the ***low side*** of the system. The high side includes the compressor discharge, hot gas line, condenser, liquid line, and the inlet side of the metering device. The low side includes the outlet side of the metering device, evaporator, suction line, and compressor inlet.

For an HCFC-22 system operating at an outside temperature of 95° F and supplying 55° F air to the conditioned space, a typical low side saturation temperature is about 40° F, which matches a pressure of about 68 psig. A typical high side saturation temperature is about 120° F, which matches a pressure of about 260 psig.

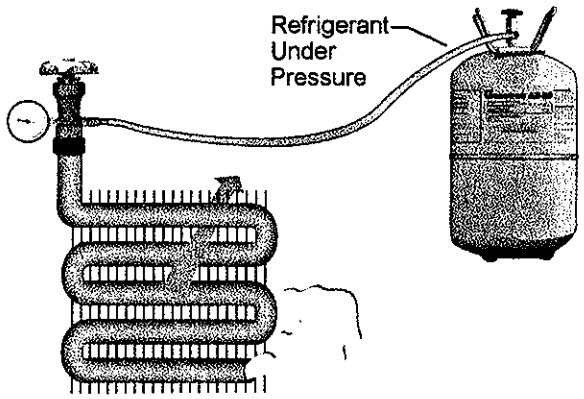


Figure 74

An open cycle would require only an evaporator.

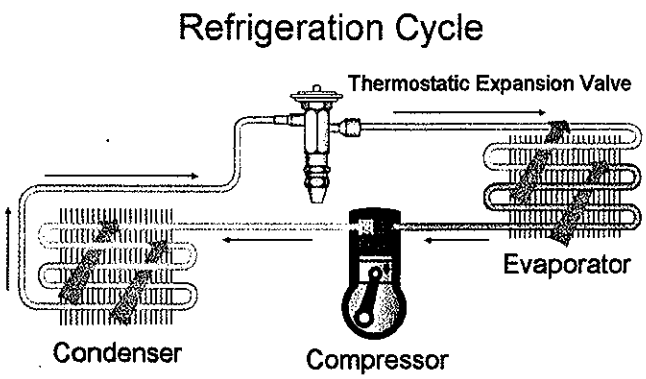


Figure 75

A closed cycle requires four basic components.

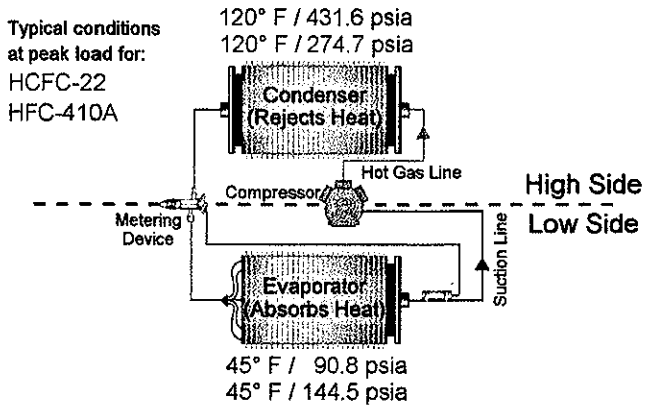


Figure 76

High Side/Low Side – Two Pressure Zones

For comparison, if HFC-410A is used, it will have the same condensing and evaporating temperatures but higher pressures for both.

Now we will follow the refrigerant flow around the system. Once it has changed into a vapor in the evaporator, it goes to the compressor.

Many types of compressors are used for comfort air conditioning: ***reciprocating compressor***, ***scroll compressor***, ***screw compressor***, ***rotary compressor*** and ***centrifugal compressor***. The reciprocating type will be used in this discussion to demonstrate the function of the compressor. The first function of the compressor is to draw refrigerant vapor away from the evaporator as the vapor is formed. The vapor is carried to the compressor through a refrigerant pipe called the suction line. By removing vapor as it is formed, the compressor maintains the evaporator at the pressure necessary to boil refrigerant at the proper temperature, in this case 40° F, at 68 psig (120 psig for HFC- 410A).

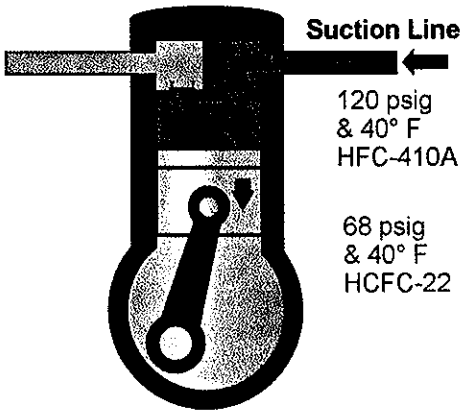


Figure 77

A compressor draws vapor from the evaporator – suction.

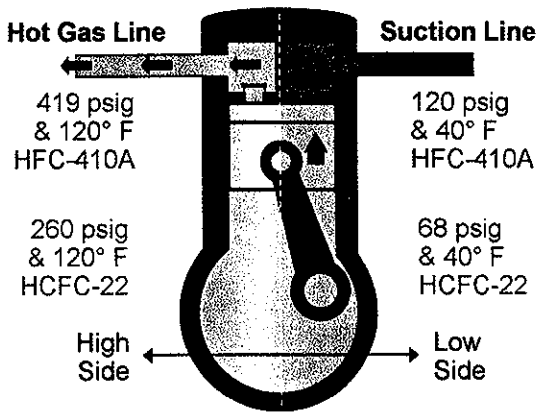


Figure 78

The compressor then raises the pressure – discharge

The other function of the compressor is to raise the pressure, and thus the saturation temperature of the refrigerant, high enough to reject the heat contained in the refrigerant at the condenser. The compressor is a dividing line in the mechanical refrigeration cycle for two pressure zones. It separates the low side that is the low-pressure, low-temperature zone to which the evaporator belongs, from the high side, which is the high-pressure, high-temperature zone to which the condenser belongs.

The high-pressure, high-temperature gas leaving the compressor travels through a refrigerant pipe called the hot gas line into the condenser. The purpose of the condenser is to do just the opposite of the evaporator. The condenser rejects heat from the refrigerant that was absorbed in the evaporator. It also rejects the heat added by compressing the gas at the compressor. In order to have the heat flow from the refrigerant within the condenser to the air or water flowing over it, the refrigerant must be hotter than the air or water.

For example, with an air-cooled condenser, if the outdoor air temperature is 95° F, the refrigerant temperature has to be about 25° F higher, or 120° F. HCFC-22 boils at 120° F at a pressure of about 260 psig (419 psig for HFC-410A). At this pressure, the refrigerant gas can reject heat to the cooler air (95° F) flowing over the condenser coil, condensing back into a liquid so that it can be reused in the evaporator the next time around the cycle. Like the evaporator, the condenser is designed primarily to do a latent heat transfer process that changes the state of the refrigerant.

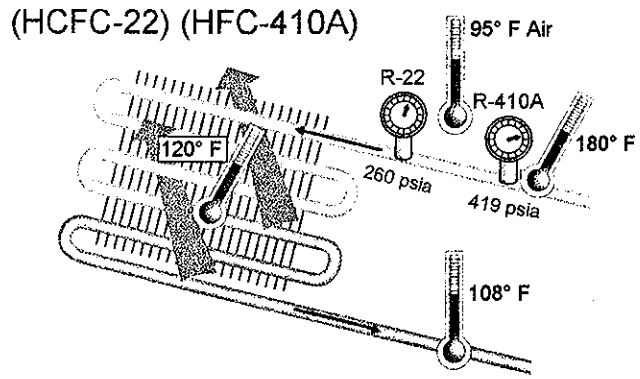


Figure 79

The condenser removes the heat and condenses the refrigerant to a high temperature liquid.

Water-cooled condensers come in various forms, but they work on the same principle as the air-cooled variety. Since water-cooled condensers are more efficient, the condensing temperature of the refrigerant only needs to be about 15° F higher than the average condenser water temperature. Since the example shows cooling tower condenser water entering at 85° F and leaving at 95° F, the average condenser water temperature is 90° F. Therefore, the required condenser refrigerant (saturation) temperature is 105° F (90 + 15° F). The condenser water rises about 10° F, from 85° F to 95° F, as it absorbs the heat rejected by the condenser. The leaving condenser water is re-cooled for reuse by a device called the cooling tower.

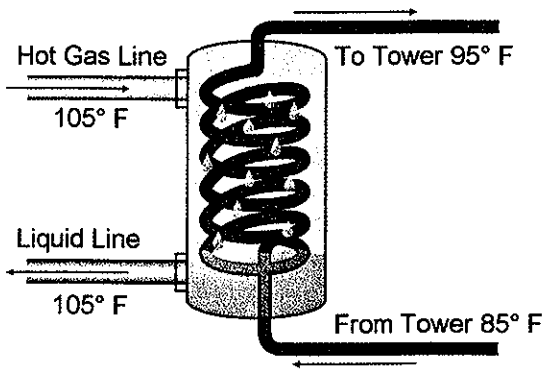


Figure 80

Water-Cooled Condenser Example

At this point in its journey through the mechanical refrigeration system, the refrigerant has been changed back into a liquid in the condenser, but it is still too hot (120° F in the air-cooled example) to be used in the evaporator (40° F).

The liquid refrigerant flows from the condenser to the metering device, in this case a thermostatic expansion valve, by way of a refrigerant pipe called the liquid line. At the metering device the refrigerant undergoes a rapid pressure reduction, from about 260 psig in the condenser to 68 psig in the evaporator (120 psig and 419 psig for HFC-410A, respectively).

As a result, the saturation temperature of the refrigerant drops from 120° F to 40° F. Now the refrigerant is ready to absorb heat, once again, in the evaporator. The refrigerant leaves the metering device as a low-pressure, low-temperature mixture of mostly liquid with a bit of vapor mixed in. Just as the compressor divided the high side from the low side of the system, the metering device also separates the two pressure zones.

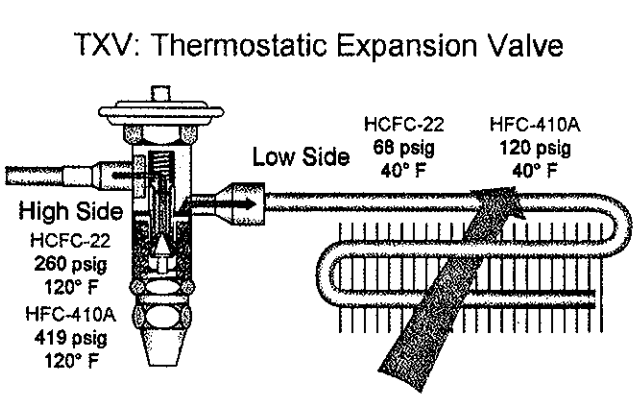
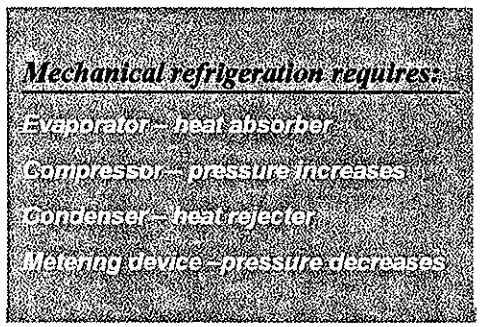


Figure 81

Metering Device

In summary, the refrigerant boils into vapor in the evaporator as it absorbs heat from the air passing over its tubes and fins. The compressor takes that vapor and raises its pressure and temperature. The refrigerant changes back into a liquid in the condenser as its heat is rejected to the air or water flowing over the condenser tubes and fins. The metering device reduces the pressure and temperature of the refrigerant to the level found in the evaporator. The refrigerant cycle repeats as long as the mechanical refrigeration system is operating to provide capacity for the air-conditioned space.



You can see how air is cooled and dehumidified by the mechanical refrigeration system within an air-conditioning system. The air and mechanical refrigeration systems work together to provide the sensible and latent heat removal capacities necessary to maintain acceptable temperature and humidity levels in the occupied area.

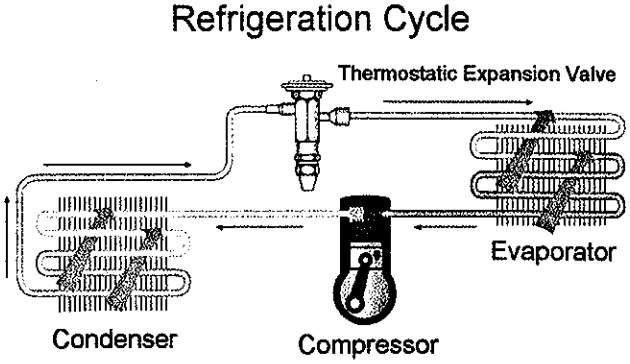


Figure 82

The refrigeration cycle works to cool and dehumidify air.

Summary

This module has covered foundational information about heating, ventilating, and air-conditioning design and service work. These same principles hold true for air-conditioning systems that vary in size from a room air conditioner to the largest commercial system. The style, size, and design of each component or system will vary from system to system, but the basic principles still hold true for each.

When a comfort air-conditioning system is designed, the engineer should decide what the goals are for maintaining the variables in the conditioned area that most directly affect comfort. Temperature, humidity, and air motion are always important. The surface temperature of the conditioned space is also very important but usually beyond the influence of the air-conditioning engineer. Each of these variables influences the rate at which heat is rejected from our bodies by conduction, convection, radiation, and evaporation, thereby affecting human comfort.

The design of the air system will determine whether the capacities of the heating and humidifying equipment and the mechanical refrigeration system are properly delivered to the areas being conditioned. If the capacity is not properly distributed, the temperature, humidity, and air motion goals the designer determined would not be achieved. Systems thinking during the system selection, load estimating, equipment selection, room air distribution design, duct design, and piping design steps of the design process will help to ensure good system performance at a reasonable cost.

This completes our coverage of the mechanical refrigeration system. Please complete Work Session 3.

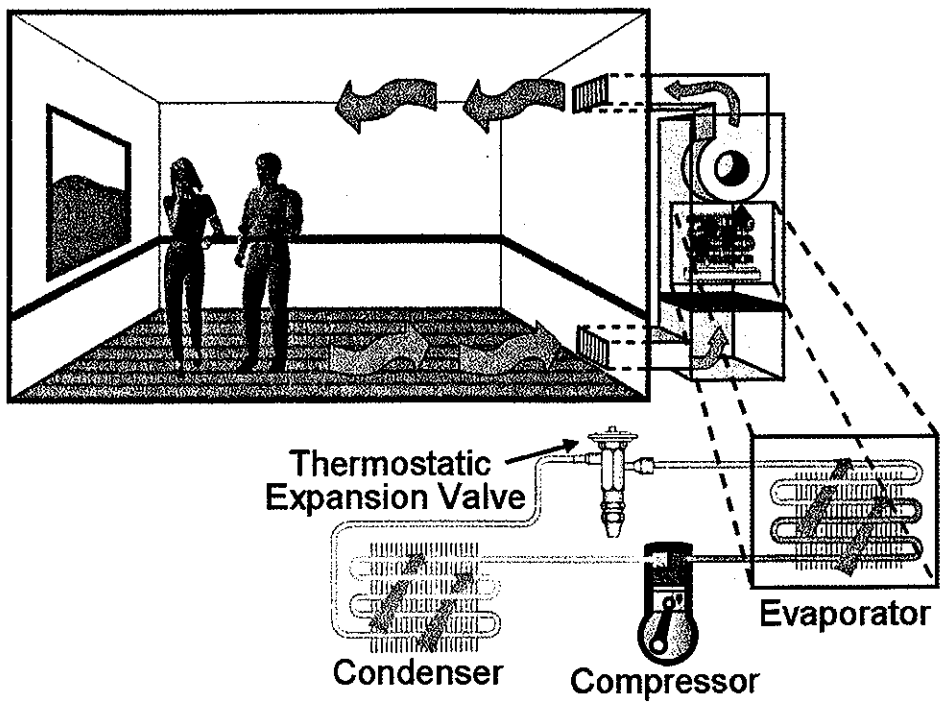


Figure 83

Systems work together.

Work Session 1

Multiple-choice questions may have more than one correct answer; identify all correct selections.

1. True or False? The two reasons for conditioning air are for people or things. Comfort air conditioning is for people. Process air conditioning is for things. _____

2. A total comfort air-conditioning system maintains which of the following air conditions within a comfortable range? _____

   a) Temperature
   b) Humidity
   c) Air motion
   d) Air purity

3. A typical comfort design goal for indoor temperature is _____° F. (Give one number or range.)

4. A typical comfort design goal for indoor relative humidity in the summer is _____ percent (Give one number or range.)

5. Which of the following methods of heat transfer is not used by our bodies to maintain comfort? _____

   a) Evaporation
   b) Insulation
   c) Convection
   d) Conduction

6. Which of the following space conditions affect(s) heat loss from our body by conduction? _____

   a) None, our bodies do not lose heat by conduction.
   b) Relative humidity
   c) Surface temperature of walls, windows, and ceiling
   d) Air temperature

7. Which of the following space conditions affect(s) heat loss from our bodies by evaporation? _____

   a) None, our bodies do not lose heat by evaporation.
   b) Air motion
   c) Relative humidity
   d) Surface temperature of walls, ceiling and windows

8. True or False? The only significant disadvantage of a poorly insulated building is high-energy usage. _____

9. Relative humidity tells you: _____

   a) the exact amount of moisture in each pound of dry air in grains/pound.
   b) how close the air is to being saturated with water vapor.
   c) how much more moisture the air is able to hold, expressed as a percent.
   d) the amount of moisture in the air compared to what it could hold at its maximum.

10. Air with a water content of 55 grains/pound has a relative humidity of 50 percent at 70° F but only 25 percent at 92° F because: _____

   a) the grains change as temperature changes.
   b) as temperature increases, air becomes denser.
   c) cooler air can hold more water vapor than warmer air.
   d) warmer air can hold more water vapor than cooler air.

11. True or False? Increasing relative humidity makes one feel warmer, all other conditions being equal. _____

12. Which combination will make a person feel warmest? _____

   a) Decrease temperature, increase humidity, decrease air motion.
   b) Increase temperature, increase humidity, increase air motion.
   c) Increase temperature, decrease humidity, decrease air motion.
   d) Increase temperature, increase humidity, decrease air motion.

Work Session 2

Multiple-choice questions may have more than one correct answer; identify all correct selections.

1. True or False? With proper air conditioning, comfort can be assured over a broad range of outside conditions for people located in an enclosed shelter. _____

2. True or False? Internal loads always add to the heat in a building. They are always heating loads. _____

3. Which of the following internal effects produce load in a building? _____

   a) Furniture
   b) Ventilation
   c) People
   d) Lights

4. Which of the following external effects can create either a cooling or heating load in a building depending on the weather? _____

   a) Transmission
   b) Equipment
   c) Infiltration
   d) Solar load

5. A building has a cooling load when the: _____

   a) Outside temperature is above 40° F.
   b) Combined load components cause heat to enter the building more rapidly than it leaves.
   c) Thermostat is turned down.
   d) Combined load components cause heat to leave the building more rapidly than it enters.

6. Air system design requires decisions about which of the following building conditions? _____

   a) Building insulation
   b) Humidity
   c) Temperature
   d) Air quality

7. Room air distribution design is the process of: _____

   a) selecting and make the air ductwork.
   b) predicting how the fan will operate.
   c) locating the thermostat(s).
   d) selecting and locating supply and return air openings.

8. True or False? Moisture is removed from the air that flows over the cooling coil because some of the water vapor in the air turns into liquid as it hits the cold coil. _____

9. The difference between infiltration and ventilation is that infiltration is: _____

a) mechanically controlled while ventilation just happens.

b) something that just happens, but ventilation is mechanically controlled.

c) the outward leakage of air from the building while ventilation is the inward leakage from the outside.

d) a way to dilute indoor odors and pollutants and to replace oxygen, but ventilation is not.

10. In heating mode, at about what temperature should the air leave the supply diffusers? _____

a) 80° F

b) 110° F

c) 160° F

d) 200° F

11. Which of the following methods are commonly used to heat air in the air handler? _____

a) Heat wheel

b) Hot water coil

c) Absorber

d) Gas-fired heat exchanger

e) Solar grid

12. To maintain the desired temperature and relative humidity in an air-conditioned space with a cooling load, the supply air must have its temperature: _____

a) equal to the space temperature, but its humidity below it.

b) and humidity equal to the space temperature and humidity.

c) below the space temperature, but its humidity equal to it.

d) and humidity below the space temperature and humidity.

13. To dehumidify the air returning from a building, the cooling coil must have its surface temperature: _____

a) below the temperature of the return air.

b) below the saturation temperature of the return air.

c) equal to the saturation temperature of the room air.

d) below the temperature of the fluid running through it.

14. Which of the following actions can a system designer take to change the efficiency with which a cooling coil cools and dehumidifies the air returned from a building?

_____

a) Change the fan speed.

b) Choose a different coil.

c) Turn up the thermostat.

d) Insulate the ductwork.

Work Session 3

Multiple-choice questions may have more than one correct answer; identify all correct selections.

1. Latent heat is: _____

   a) measurable with a thermometer.
   b) what happens when you heat water from 55° F to 210° F.
   c) what happens when water boils.
   d) more important in the refrigeration cycle than sensible heat.

2. As the pressure on a fluid (liquid or gas) increases, its boiling point: _____

   a) increases.
   b) decreases.
   c) remains unchanged.
   d) may or may not change.

3. The four basic components of the mechanical refrigeration cycle include the: _____, _____, _____, and _____.

4. The evaporator: _____

   a) rejects heat as refrigerant boils from a liquid into a vapor within it.
   b) absorbs heat as refrigerant boils from a liquid into a vapor within it.
   c) rejects heat as refrigerant changes from a vapor into a liquid within it.
   d) absorbs heat as refrigerant changes from a vapor into a liquid within it.

5. Which of the following components divides the high side from the low side of the system? _____

   a) Receiver
   b) Compressor
   c) Suction line accumulator
   d) Metering device

6. To provide cooling, why isn't refrigerant just pumped from a tank through an evaporator and out into the air? _____

   a) It is too damaging to our environment.
   b) It would not work for cooling. Without being enclosed, the evaporator would not cool the air flowing over it.
   c) It is too complicated.
   d) It is too expensive.

7. The refrigerant line entering the compressor is the: _____

   a) suction line.
   b) liquid line.
   c) hot gas line.
   d) condensate line.

8. The refrigerant line leaving the condenser is the: _____

   a) suction line.
   b) liquid line.
   c) hot gas line.
   d) condensate line.

9. The compressor: _____

   a) pumps liquid refrigerant.
   b) lowers the pressure and temperature of the refrigerant within the cycle.
   c) draws refrigerant out of the condenser.
   d) raises refrigerant pressure and temperature.

10. A typical high side saturation temperature at peak cooling load for a system using a water-cooled condenser is about: _____

    a) 50° F.
    b) 70° F.
    c) 105° F.
    d) 120° F.

11. A typical low side saturation temperature at peak cooling load is about: _____

    a) 50° F.
    b) 40° F.
    c) 30° F.
    d) 20° F.

Appendix

Comfort Conditions Form

Table of Design Conditions

Glossary

Work Session Answers

Comfort Conditions Form

Space/Activity						
Temperature	Occupied		Unoccupied		Throttling Range	Swing
	Design	Min	Design	Min		
Summer						
Winter						
Humidity	Design	Max	Min			
Summer						
Winter						
Air Motion	Max	Min	Notes			
fpm						
Ventilation	Occupied		Unoccupied		Notes:	
	Design	Min	Min			
cfm						
Direct Exhaust						
Filtration	Condition		MERV	Type	Other	
Radiant Surfaces	Glass	Wall	Floor	Roof	Other	
Summer						
Winter						
Acoustics	Room RC		Other Notes			
Notes:						

Table of Design Conditions

Category	Temperature		Relative Humidity		Supply Air		Sound	Ventilation			Occupancy	Filters	Exhaust
	Summer °F	Winter °F	Summer %	Winter %	Speed fpm	Changes / hour	NC or RC(N)	cfm / Person	cfm / ft^2	Default / Person	People / 1000 ft^2	Eff. % MERV	cfm or cfm/ft^2
Lodging (Hotels, Motels, Resorts, Dormitories)													
Bedroom / living room	74 - 78	74 - 76	50 - 60	30 - 35	25 - 45	4 - 10	25 - 35	5	0.06	11	10	10-15%	20 - 50/rm
Barracks	74 - 78	74 - 76	50 - 60	NR	25 - 45	4 - 10	25 - 35	5	0.06	8	20	10-15%	20-50/Fixture
Conference room	74 - 78	70 - 74	50 - 60	20 - 30	30 - 50	12 - 15	40 - 50	5	0.06	6	50	MERV 6	NS
Kitchen general	82 - 88	70 - 74	65 max	NR	30 - 50	12 - 15	40 - 50	5	0.06	6	10	MERV 6	0.70 / ft^2
Kitchen pastry	76	70 – 74	65 max	NR	30 – 50	12 – 15	40 – 50	5	0.06	6	10	MERV 6	0.70 / ft^2
Lobbies	80	72	50 – 60	20 – 30	30 – 45	NS	35 – 45	5	0.06	11	10	MERV 8	NS
Kitchen private	74 – 78	70 – 74	65 max	NR	30 – 50	12 – 15	40 – 50	5	0.06	6	2	MERV 6	0.30 / ft^2
Hallways	80	72	50 – 60	20 – 30	30 – 45	NS	35 – 45	5	0.06	11	10	MERV 8	NS
Guest rooms	74 – 78	74 – 76	50 – 60	30 – 35	25 – 45	4 – 10	25 – 35	5	0.06	11	10	10 – 15%	20 – 50/rm
Multi-purpose	74 – 78	70 – 74	50 – 60	20 – 30	30 – 50	12 – 15	40 – 50	5	0.06	6	120	MERV 6	NS
Office areas	74 – 78	70-74	50 – 60	20 – 30	25 – 45 0.7-2 cfm/ft^2	4 – 10	30 – 45	5	0.06	17	5	30 – 60	NS
Laundry	85	65	60	NR	50 +	10 – 12	NR	7.5	0.06	11	20	8	As Req.
Office Buildings													
Office	74 – 78	70 – 74	50 – 60	20 – 30	25 – 45 0.7-2 cfm/ft^2	4 – 10	30 – 45	5	0.06	17	5	30 – 60	NR
Reception	74 – 78	70 – 74	50 – 60	20 – 30	25 – 45 0.7-2 cfm/ft^2	4 – 10	30 – 45	5	0.06	7	30	30 – 60	NR
Data entry	74 – 78	70 – 74	50 – 60	20 – 30				5	0.06	6	60	30 – 60	NR
Lobbies	80	72	50 – 60	20 – 30	30 – 45	NS	35 – 45	5	0.06	11	10	30 – 60	NR
Computer rooms	74 – 78	70 – 74	50 – 60	20 – 30				5	0.06	20	4	30 – 60	NR
Toilets	74 – 78	70 – 74	50 – 60	20 – 30	30 – 45	NS	NR	NS	0.12	NS	NS		50-70/Fixture

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Category	Temperature		Relative Humidity		Supply Air		Sound	Ventilation			Occupancy	Filters	Exhaust
	Summer °F	Winter °F	Summer %	Winter %	Speed fpm	Changes / hour	NC or RC(N)	cfm / Person	cfm / ft^2	Default / Person	People / 1000 ft^2	Eff. % MERV	cfm or cfm/ft^2
Conf. Room	74 – 78	70 – 74	50 – 60	20 – 30	30 – 50	12 – 15	40 – 50	5	0.06	6	50	30 – 60	NR
Storage	80	72	50 – 60	20 – 30	30 – 45	NS	NR	NS	0.12	NS	1	10 – 15	1.00
Parking Garage	80 – 100	40 – 55	NR	NR	30 – 75	4 – 6	35 – 50	NR	NR	NR	NS	10 – 15	0.75
Break room	74 – 78	70 – 74	55 – 60	20 – 30	25 – 30	8 – 12	35 – 40	7.5	0.18	9	100	35 +	0.30
Corridors	80	72	50 – 60	20 – 30	30 – 45	NS	35 – 45	5	0.06	11	10	MERV 8	NS
Residence – Single Family, Multi-Family, Apartment, Manufactured Housing													
Living areas	74 – 78	74 – 76	50 – 60	30 – 35	25 – 45	4 – 10	25 – 35	5	0.06	11	10	10-15%	20-50/rm
Kitchens	74 – 78	74 – 76	50 – 60	30 – 35	25 – 45	4 – 10	25 – 35	5	0.06	11	10	10-15%	0.30
Baths	74 –78	74 – 76	50 – 60	30 – 35	25 – 45	4 – 10	25 – 35	5	0.06	11	10	10-15%	50 cfm int. 20 cfm cont.
Retail													
Sales-main	74 – 78	70 – 74	50 – 60	20 – 30	25 – 30	8 – 12	35 – 50	7.5	0.60	16	15	MERV 6	NR
Sales-upper	74 – 78	70 – 74	50 – 60	20 – 30	25 – 30	8 – 12	35 – 50	7.5	0.60	16	50 – 100	MERV 6	NR
Mall common spaces	74 – 78	70 – 74	50 – 60	20 – 30	25 – 30	8 – 12	35 – 50	7.5	0.06	9	40	MERV 6	NR
Barber shop	74 – 78	70 – 74	50 – 60	20 – 30	25 – 30	8 – 12	35 – 50	7.5	0.06	10	25	MERV 6	0.50
Salon	74 – 78	70 – 74	50 – 60	20 – 30	25 – 30	8 – 12	35 – 50	20	0.12	25	25	MERV 6	0.60
Clothing stores	74 – 78	70 – 74	50 – 60	20 – 30	25 – 30	8 – 12	35 – 50	7.5	0.06	11	20	MERV 6	NR
Drug store	74 – 78	70 – 74	50 – 60	20 – 30	25 – 30	8 – 12	35 – 50	5	0.18	23	10	MERV 6	NR
Pet shop	74 – 78	70 – 74	50 – 60	20 – 30	25 – 30	8 – 12	35 – 50	7.5	0.18	26	10	MERV 6	0.90
Banks	74 – 78	70 – 74	50 – 60	20 – 30	25 – 30	8 – 12	35 – 50	7.5	0.60	17	30	MERV 6	NR
Photo Studio	74 – 78	70 – 74	50 – 60	20 – 30	25 – 30	8 – 12	35 – 50	5	0.12	17	10	MERV 6	1.00
Supermarket	74 – 78	70 – 74	50 – 60	20 – 30	25 – 30	8 – 12	35 – 50	7.5	0.06	15	8	MERV 6	NR
Coin-op laundries	74 – 78	70 – 74	50 – 60	20 – 30	25 – 30	8 – 12	35 – 50	7.5	0.06	11	20	MERV 6	NR
Convenience stores	74 – 78	70 – 74	50 – 60	20 – 30	25 – 30	8 – 12	35 – 50	7.5	0.60	15	10	MERV 6	NR
Bakeries	74 – 78	70 – 74	50 – 60	20 – 30	25 – 30	8 – 12	35 – 50	7.5	0.60	19	13	MERV 6	NR
Public restrooms	74 – 78	70 – 74	50 – 60	20 – 30	25 – 30	8 – 12	35 – 50	7.5	0.60	NS	NS	MERV 6	50-70/Fixture

Category	Temperature		Relative Humidity		Supply Air		Sound	Ventilation			Occupancy	Filters	Exhaust
	Summer °F	Winter °F	Summer %	Winter %	Speed fpm	Changes / hour	NC or RC(N)	cfm / Person	cfm / ft^2	Default / Person	People / 1000 ft^2	Eff. % MERV	cfm or cfm/ft^2
Florists	74 - 78	70 - 74	50 - 60	20 - 30	25 - 30	8 - 12	35 - 50	7.5	0.60	11	20	MERV 6	NR
Gift shop	74 - 78	70 - 74	50 - 60	20 - 30	25 - 30	8 - 12	35 - 50	7.5	0.60	11	20	MERV 6	NR
Jewelry store	74 - 78	70 - 74	50 - 60	20 - 30	25 - 30	8 - 12	35 - 50	7.5	0.60	11	20	MERV 6	NR
Restaurants												MERV 6	NR
Kitchen	82 - 88	70 - 74	65 max	NR	30 - 50	12 - 15	40 - 50	5	0.06	6	10	MERV 6	0.70 / ft^2
Fast food	78	70 - 74	50	20 - 30	50	12 - 15	40 - 50	7.5	0.18	9	100	MERV 8	NR
Formal	74 - 78	70 - 74	55 - 60	20 -30	25 - 30	8 - 12	35 - 40	7.5	0.18	9	70	MERV 8	NR
Bars	74 - 78	70 - 74	50 - 60	20 – 30	30	15 – 20	35 - 50	7.5	0.18	9	100	MERV 8	NR
Cafeterias	78	70 - 74	50	20 -30	50	12 - 15	40 - 50	7.5	0.18	9	100	MERV 8	NR
Sports and Entertainment													
Auditoriums	74 - 78	70 - 74	50 - 60	20 - 30	30	6 - 8	35 - 50	5	0.06	5	150	35+	0.50
Bowling alleys	75 - 78	70 - 74	50 - 55	20 - 30	50 @ 6'	10 - 15	40 - 50	10	0.12	13	40	10 - 15	0.50
Gym / Sports complex	74 - 78	70 - 74	50 - 60	20 - 30	30	6 - 8	35 - 50	NS	0.30	NS	NS	35	0.50
Spectator areas	74 - 78	70 - 74	50 - 60	20 - 30	30	6 - 8	35 - 50	7.5	0.06	8	150	35	NS
Gambling Casinos	74 - 78	70 - 74	50 - 60	20 -30	30	6 - 8	35 - 50	7.5	0.18	9	120	35	NS
Libraries	68 - 72	68 - 72	40 - 55	40 - 55	25	8 - 12	35 - 45	5	0.12	17	10	35 - 60	NS
Museums - children's	68 - 72	68 - 72	40 - 55	40 - 55	25	8 - 12	35 - 45	7.5	0.12	11	40	35 - 60	NS
Museums - galleries	68 - 72	68 - 72	40 - 55	40 - 55	25	8 - 12	35 - 45	7.5	0.06	9	40	35 - 60	NS
Swimming pools	75 - 85	75 - 85	50 - 60	50 - 60	30	4 - 6	35 - 50	NS	0.48	NS	NS	SR	NS
Dance floors	74 - 78	70 - 74	50 - 60	20 - 30	30	6 - 8	35 - 50	20	0.06	21	100	35	NS
Health clubs													
Aerobics	74 - 78	70 - 74	50 - 60	20 - 30	30	6 - 8	35 - 50	20	0.06	22	40	35	0.50
Weight	74 - 78	70 - 74	50 - 60	20 - 30	30	6 - 8	35 - 50	20	0.06	26	10	35	0.50
Arcades	74 - 78	70 - 74	50 - 60	20 - 30	30	6 - 8	35 - 50	7.5	0.18	17	20	35	NS
Stage / studio	74 - 78	70 - 74	50 - 60	20 - 30	30	6 - 8	35 - 50	10	0.06	11	70	35	NS

Category	Temperature		Relative Humidity		Supply Air		Sound	Ventilation			Occupancy	Filters	Exhaust
	Summer °F	Winter °F	Summer %	Winter %	Speed fpm	Changes / hour	NC or RC(N)	cfm / Person	cfm / ft^2	Default / Person	People / 1000 ft^2	Eff. % MERV	cfm or cfm/ft^2
Transportation Centers / Correctional Facilities													
Waiting area	74 - 78	70 - 74	50 - 60	20 - 30	25 - 30	8 - 12	35 - 50	7.5	0.06	8	100	MERV 8	NS
Correctional													
Cell	78	72	50	30	25 - 30	6	35 - 50	5	0.12	10	25	MERV 8	1.00
Day room	78	72	50	30	25 - 30	6	35 - 50	5	0.60	7	30	MERV 8	NS
Guard Station	78	72	50	30	25 - 30	6	35 - 50	5	0.60	9	15	MERV 8	NS
Booking	78	72	50	30	25 - 30	8	35 - 50	7.5	0.60	9	50	MERV 8	NS
Assembly Buildings													
Places of worship	78	72	60	30	30 - 45	NS	30 - 40	5	0.06	5	150	MERV 8	NS
Courtroom	78	72	60	30	30 - 45	NS	30 - 40	5	0.06	6	70	MERV 8	NS
Post office	78	72	60	30	30 - 45	NS	30 - 40	5	0.06	6	70	MERV 8	NS
Education													
Classrooms													
Age 5 - 8	78	75	50 - 60	20 - 30	30 - 45	NS	30 - 40	10	0.12	15	25	MERV 8	NS
Age 9 +	78	72	50 - 60	20 - 30	30 - 45	NS	30 - 40	10	0.12	13	35	MERV 8	NS
Lecture	78	72	50 - 60	20 - 30	30 - 45	NS	30 - 40	7.5	0.06	8	65	MERV 8	NS
Fixed-seat lecture	78	72	50 - 60	20 - 30	30 - 45	NS	30 - 40	7.5	0.06	8	150	MERV 8	NS
Art	78	72	50 - 60	20 - 30	30 - 45	NS	30 - 40	10	0.18	19	20	MERV 8	0.70
Science Lab	78	72	50 - 60	20 - 30	30 - 45	NS	35 - 45	10	0.18	17	25	MERV 8	1.00
Wood / metal shops	78	72	50 - 60	20 - 30	30 - 45	NS	35 - 45	10	0.18	19	20	MERV 8	0.50
Computer lab	78	72	50 - 60	20 - 30	30 - 45	NS	30 - 40	10	0.12	15	25	MERV 8	NS
Media center	78	72	50 - 60	20 - 30	30 - 45	NS	30 - 40	10	0.12	15	25	MERV 8	NS
Hallways	80	72	50 - 60	20 - 30	30 - 45	NS	35 - 45	5	0.06	11	10	MERV 8	NS
Daycare	78	75	50 - 60	20 - 30	30 - 45	NS	30 - 40	10	0.18	17	25	MERV 8	NS
Music	78	72	50 - 60	20 - 30	30 - 45	NS	30 - 40	10	0.06	12	35	MERV 8	NS
Multi-use	78	72	50 - 60	20 - 30	30 - 45	NS	30 - 40	7.5	0.06	8	100	MERV 8	NS

Category	Temperature		Relative Humidity		Supply Air		Sound	Ventilation			Occupancy	Filters	Exhaust
	Summer °F	Winter °F	Summer %	Winter %	Speed fpm	Changes / hour	NC or RC(N)	cfm / Person	cfm / ft^2	Default / Person	People / 1000 ft^2	Eff. % MERV	cfm or cfm/ft^2
Health													
Dr. Office	74 - 78	70 - 74	50 - 60	20 - 30	25 - 45	4 - 10	30 - 45	5	0.06	17	5	30-60	NS
Exam rm.	75	75	60	30	NS	6	35 - 45	15	NS	NS	20	MERV 8	NS
Hospital													
Patient room	75	75	50	30	NS	6	25 - 35	25	NS	NS	10	MERV 8	25 to 50
Hallways	78	75	60	30	NS	4	35 - 45	5	0.06	11	10	MERV 6	NS
Exam room	75	75	60	30	NS	6	35 - 45	15	NS	NS	20	MERV 8	NS
Waiting room	74	74	60	30	NS	6	35 - 45	5	0.06	11	10	MERV 8	NS
Operating rooms	62 - 80	62 - 80	45 - 55	45 - 55	NS	25	35 - 45	30	NS	NS	20	SR*	SR*
Recovery / ICU	75	75	45 - 55	45 - 55	NS	6	35 - 45	15	NS	NS	20	SR*	SR*
Autopsy	74	74	60	30	NS	6	35 - 45	NS	.5	NS	20	SR*	SR*
Physical therapy	72 - 80	72 - 80	60	30	NS	6	35 - 45	15	NS	NS	20	SR*	SR*
Industrial													
Light Mfg	78	72	Max 60*	***	NS	***	To 50	15	NS	15	10	MERV 6	***
Med Mfg	78	60	Max 60*	***	NS	***	To 50	15	NS	15	5	MERV 6	***
Heavy Mfg	78	65	Max 60*	***	NS	***	To 75	15	NS	15	3	MERV 6	***
Warehouse	80 - 100	40 - 55	Max 60*	***	NS	1 - 4	To 75	NS	0.12	NS	NS	10 - 35	***

Data in this table is provided as a guideline only, local codes and standards must be complied with. The data is extracted from ASHRAE Standards, the ASHRAE handbook and a number of internal sources, and by no means covers all situations.

- SR – Depends on activity, material or process in the space
- NS – Not specified
- NR – Not required

Glossary

Word	**Definition**
air handling unit (AHU) / air handler	equipment, usually connected to ductwork, to move and condition air; contains components such as fans, cooling coils, heating coils, filters and dampers to control return, outdoor and exhaust air
air system	the components of a comfort air-conditioning system that treat the air to a space; generally these include the fan, supply duct system, exhaust, ventilation dampers, mixing dampers, filtration, cooling coils, heating devices, and humidification capabilities
apparatus dew point (ADP)	the dew point temperature that corresponds to the average effective coil surface temperature
blow-thru	air systems where the fan is located before the cooling coil and air is blown through the coil; the coil is under positive pressure
British thermal unit (Btu))	measure of heat energy; the heat energy of a Btu is approximately that required to raise the temperature of a pound of water from one degree Fahrenheit, from 59° F to 60° F
bypass factor	the ratio of the air that does not contact the coil to the total airflow
centrifugal compressor	nonpositive displacement refrigeration compressor, which depends for pressure rise, at least in part, on centrifugal forces developed by a rotating impeller.
circulate	Function of a comfort air-conditioning system to provide air motion within a space which is uniform and provides adequate mixing of supply and return air at room air velocities which would not be described as drafty or stuffy
CLO	unit of clothing insulation, defined as the thermal insulation necessary to keep a sitting person comfortable in normally ventilated room at 70° F and 50 percent relative humidity. In physical terms, the thermal resistance of one CLO = 0.88° F·ft^2 ·h/Btu
comfort (air conditioning)	Controlling simultaneously space conditions of temperature, humidity, air motion, mean radiant temperature, air purity, and air quality (ventilation/exhaust), for the purpose of satisfying the physical needs of the space occupants
compressor	a component in a refrigerating system that "pumps" the refrigerant and raises its temperature by increasing the pressure above the temperature of the condensing media
condenser	a device in a refrigerating system where refrigerant rejects the heat to the condensing media, either air or water
conduction	heat transfer by which heat is moved from molecule to molecule of a substance or between substances by a chain collision of those molecules
convection	heat transfer within a fluid by the movement of heated molecules from one place to another as the result of density differences
convective baseboard	heaters which create air motion by natural convective heat transfer without any fan; normally electric or hot water and located along the perimeter walls
cool	function of a comfort air-conditioning system to remove heat energy from a space adequate to overcome heat gains and maintain comfort levels
cooling load	the heat gains to a space above the heat losses that the cooling system needs to remove to maintain space temperature; includes both sensible and latent heat gain elements

dehumidifier	air cooler, absorption or adsorption device used to lower moisture content
dehumidify	function of a comfort air-conditioning system to remove moisture from the air in a space at a quantity that maintains space relative humidity at levels within the comfort envelop or as required for a process
diffuser	an air distribution component in the space or air outlet that delivers supply air such that appropriate mixing, conditioning and air motion occurs
draw-thru	air systems where the fan is located after the cooling coil and air is drawn through the coil; the coil is under negative pressure
ductwork	series of interconnected ducts and fittings that delivers air to a space from a remotely located fan then returns it to the fan
effective temperature	a set of temperature and humidity conditions that feel the same as the dry bulb temperature and 50 percent relative humidity
evaporation	change of state from liquid to vapor, from the body regulation of body temperature by the process of evaporation of perspiration
evaporator	a heat exchange coil within a mechanical refrigeration system where the refrigerant is evaporated to absorb heat from the contacting heat source
filter	function of a comfort air-conditioning system to remove particles, gaseous, and biological contaminants which may be in circulation
grille	a covering over the opening of a duct system, normally return air; does not greatly affect the room air distribution.
heat	a form of energy that can be transferred by conduction, convection, or radiation; only transferred from a warmer substance to a colder substance; function of a comfort air-conditioning system to provide heat energy to a space adequate to overcome heat losses and maintain comfort levels
heat pump	air conditioning heating/refrigerating system that will transfer heat; the condenser and evaporator may change roles to transfer heat in either direction by receiving the flow of air or other fluid; a heat pump is used to cool or heat.
heating load	the heat losses from a space above the heat gains that a heating system needs to add to maintain space temperature; it is only sensible
high side	parts subjected to approximately the condenser pressure; discharge side, from compressor discharge to metering device inlet
human comfort	space in which more that 80 percent of the occupants describe the space as comfortable with regard to temperature, humidity, air motion, air purity, air quality and effects of radiant temperature; conditions are a function of activity level, clothing, and time in the space
humidifier	device to add moisture to air
humidify, humidification	function of a comfort air-conditioning system to add moisture to the air in a space at a quantity that will maintain a space relative humidity at levels within the comfort envelope or as required for a process
humidity	the moisture content of air, measured in pounds/pound of dry air, or grains/pound of dry air
HVAC	heating, ventilating and air conditioning; standard abbreviation
infiltration	the leakage of outside air into a structure through doors, cracks, windows, and other openings.

low pressure side (low side)	portion of a refrigerating system operating at approximately the evaporator pressure; suction side, from metering device outlet to compressor suction
mechanical refrigeration system	a system that transfers heat from a colder place to a warmer place by raising the pressure and temperature of a volatile fluid above its condensing temperature; system consists of a refrigerant (volatile fluid), an evaporator (heat absorber), a condenser (heat rejector), a compressor (vapor pump), and a metering device (pressure reducing valve)
MET	unit of metabolic rate of people defined as 18.4 Btu/h·ft^2, which is equal to the energy produced per unit surface area of a seated person at rest. The surface area of an average man is about 19 ft^2
process air conditioning	controlling simultaneously, space conditions to provide temperature, humidity, air motion, air purity, air quality (ventilation/exhaust), and mean radiant temperature, primarily for the purpose of a product or production process in the space; if human comfort is provided, it is a secondary benefit
psychrometrics	the study of air and moisture mixtures.
radiation	transfer of heat energy by electromagnetic waves due to temperatures in the two surfaces; is a function of temperature difference, distance, angle between surfaces, and surface absorption
reciprocating compressor	positive-displacement compressor in which the change in internal volume of the compression chamber(s) is accomplished by the reciprocating motion of one or more pistons.
refrigerant	the medium of heat transfer which picks up heat by evaporating at a low temperature and pressure, and giving up heat on condensing at a higher temperature and pressure
refrigerant metering device	device that controls the flow of liquid refrigerant to an evaporator
register	combination grille and damper assembly over a supply air opening; normally located in the floor or wall
relative humidity	a measure of the moisture content in the air as a percentage of the maximum amount of moisture the air could hold at that dry bulb temperature
room air distribution	the process of providing proper air motion and proper air mixing of room, supply, and ventilation air in a space
rotary compressor	positive displacement compressor in which the change in internal volume of the compression chamber(s) is accomplished by the rotary motion of a positive displacement member(s).
screw compressor	rotary compressor relying on the intermeshing of two helical rotors to produce compression.
scroll compressor	positive-displacement compressor in which the reduction in internal volume of the compression chamber is accomplished by an orbiting scroll (involute spiral) within a mating stationary scroll.
sensible heat	the energy of molecular motion, measured with a standard thermometer; heat which causes a change in temperature
setback	reduction of space temperature for heating during hours when a building is unoccupied, or during periods when lesser demand is acceptable
setup	Increasing the space temperature setpoint for cooling during hours when a building is unoccupied, or during periods when lesser demand is acceptable
specific humidity	ratio of the mass of water to the total mass of dry air in a volume of moist air

swing	change in temperature above the cooling setpoint that is allowed for calculation such that the system is still considered to be in control of the space temperature
thermostat	temperature-sensitive device that automatically opens and closes an electric circuit to regulate the dry bulb temperature of the space with which it is associated
thermostatic expansion valve (TXV)	control device for regulating the flow of refrigerant into a cooling unit, actuated by the changes in evaporator pressure and superheat of the refrigerant leaving the cooling unit; the basic response is to the superheat
throttling range	change in controlled variable (temperature) required to move the thermostat from one of its extreme limits of travel to the other
transmission load	the heat gain or loss to a space that occurs due to a temperature difference between space and outdoor conditions or adjacent room conditions; may be a cooling or heating load
ventilation	process of supplying or removing air by natural or mechanical means to or from any space for the purpose of oxygen replacement or odor dilution; such air may or may not have been conditioned

Work Session Answers

Work Session 1

1. True
2. a), b), c), d)
3. 75° F or 68-78° F (range)
4. 50 percent or 30-60 percent (range)
5. b)
6. d)
7. b) and c)
8. False, comfort is also compromised
9. b), c), d)
10. d)
11. True
12. d)

Work Session 2

1. True
2. False
3. c), d)
4. a), c)
5. c)
6. b), c), d)
7. d)
8. True
9. b)
10. b)
11. b), d)
12. d)
13. b)
14. a), b)

Work Session 3

1. c), d)
2. a)
3. evaporator, compressor, condenser, metering device
4. b)
5. b), d)
6. a), d)
7. a)
8. b)
9. d)
10. c)
11. b)

Prerequisites:

None required.

Learning Objectives:

At the conclusion of this module, participants will be able to:

- Describe the six conditions that influence human comfort.
- List the four processes the body uses to transfer heat.
- Describe how a system designer can control the environment to maintain comfort conditions.
- Describe what environmental control parameters are required from comfort air-conditioning systems.
- List a reasonable range for each of the comfort parameters.
- List the basic components of the air system and the function of each.
- List the four principles that are used to make a mechanical refrigeration system work.
- List the four components of a mechanical refrigeration system and the principles used to create the refrigeration process.
- Explain what measurements can be done to calculate each of the environmental factors.
- Calculate the relative humidity.
- Explain the principle of effective temperature.
- Determine the CLO (clothing insulating value) and MET (Activity – metabolic rate) values.
- List the typical operating parameters of the components of an air system.
- Explain the difference between sensible and latent heat.
- Create a list of comfort design conditions for a project.

Supplemental Material:

Form No.	Color Book	Instructor Presentation	Title
TDP-103	796-027	797-027	Concepts of Air Conditioning
TDP-104	796-028	797-028	Equipment Introduction
TDP-105	796-029	797-029	Comfort Design Steps
TDP-400	796-037	797-037	Principles of Mechanical Refrigeration
TDP-201	796-030	797-030	Psychrometrics, Level 1, Fundamentals
TDP-507	796-048	797-048	Room Air Distribution

Instructor Information:

Each TDP topic is supported with a number of different items to meet the specific needs of the user. Instructor materials consist of a CD-ROM disk that includes a PowerPoint™ presentation with convenient links to all required support materials required for the topic. This always includes: slides, presenter notes, text file including work sessions and work session solutions, quiz and quiz answers. Depending upon the topic, the instructor CD may also include sound, video, spreadsheets, forms, or other material required to present a complete class. Self-study or student material consists of a text including work sessions and work session answers, and may also include forms, worksheets, calculators, etc.



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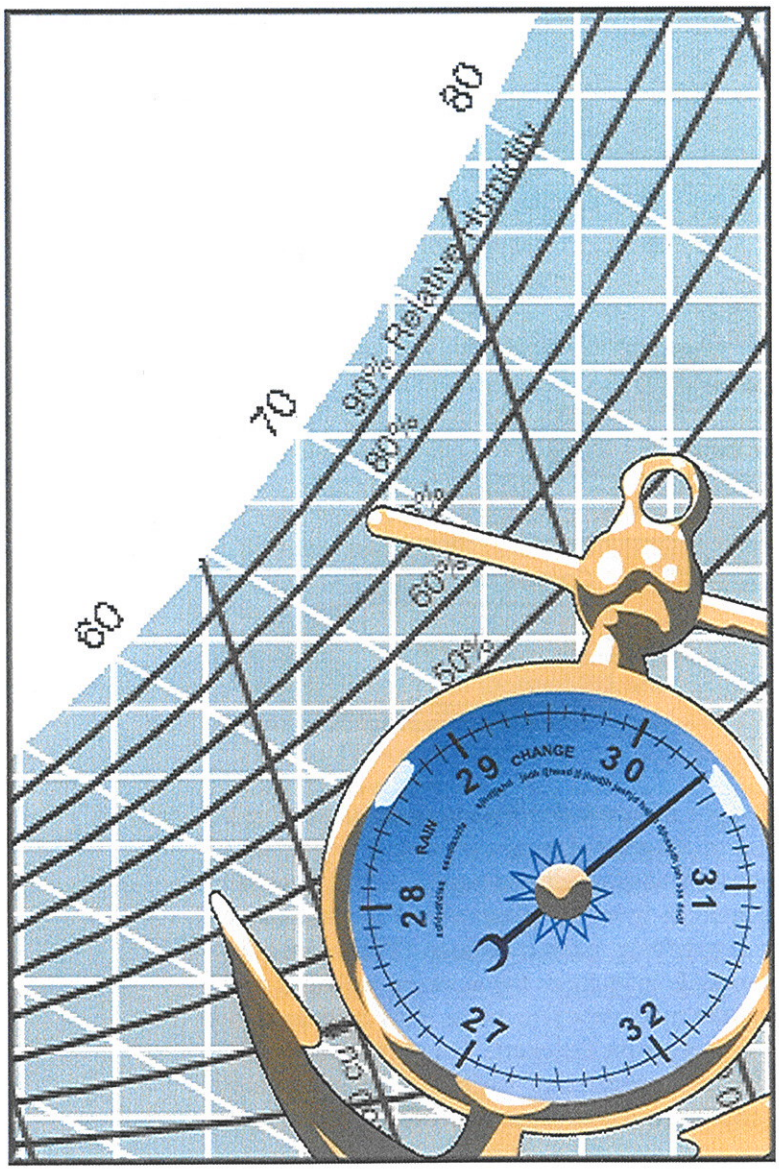


PSYCHROMETRICS

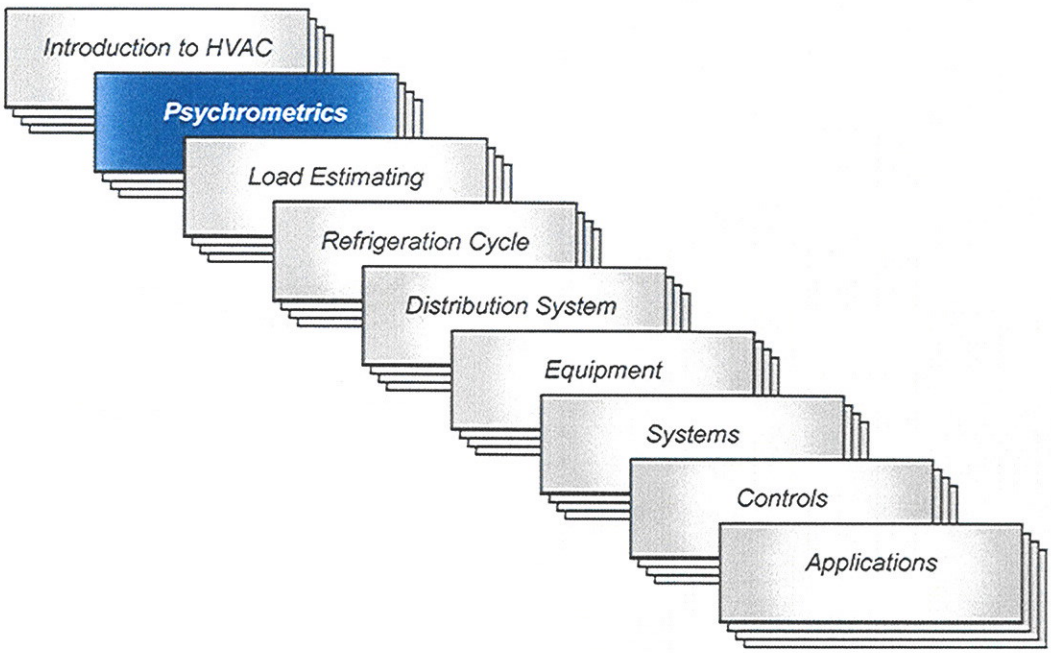
Psychrometrics

Level 1: Introduction

Technical Development Program

Technical Development Programs (TDP) are modules of technical training on HVAC theory, system design, equipment selection and application topics. They are targeted at engineers and designers who wish to develop their knowledge in this field to effectively design, specify, sell or apply HVAC equipment in commercial applications.

Although TDP topics have been developed as stand-alone modules, there are logical groupings of topics. The modules within each group begin at an introductory level and progress to advanced levels. The breadth of this offering allows for customization into a complete HVAC curriculum – from a complete HVAC design course at an introductory-level or to an advanced-level design course. Advanced-level modules assume prerequisite knowledge and do not review basic concepts.



Psychrometrics is the study of the air and water vapor mixture. Proficiency in the use of the psychrometric chart is an important tool for designers of air conditioning systems. Psychrometrics is required to properly calculate heating and cooling loads, select equipment, and design air distribution systems. While the topic is not complicated, it involves a number of formulas and their application; the psychrometric chart is useful in simplifying the calculations. This module is the first of four on the topic of psychrometrics. This module introduces the air-vapor mixture and how the psychrometric chart can be used to determine the mixture's properties. This module also explains how to plot the eight basic air conditioning processes on the chart. Other modules build on the information from this module to explain the psychrometrics of various air conditioning systems, analysis of part load and control methods, computerized psychrometrics, and the theory used to develop the chart.

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Introduction

Why does an air-conditioning design course begin with psychrometrics? In the computer-aided design world of today, is psychrometrics a necessary and practical topic to understand? The answer is that the principles of psychrometrics provide the key to understanding why the air conditioning industry exists and will help explain many of the processes and steps used in system design. It is so important, we have four TDP modules devoted to psychrometrics. This first module has four sections: properties of air and vapor, building the psychrometric chart, state points, and air conditioning processes. Other modules describe using psychrometrics to analyze processes and determine loads or airflows, using psychrometrics to evaluate performance of compound systems with the psychrometric chart or computer tools, and psychrometric formula and the theory used to construct the chart.

Many of the terms and concepts are used in daily conversation, yet we may not recognize them as psychrometrics. What does ***relative humidity*** really mean? How does a cooling coil remove water vapor? What causes air conditioning ducts to sweat? The answers to questions such as these depend upon the properties of air and water vapor and how they act together. Being able to analyze air conditioning systems with an understanding of these properties means better operating systems and lower costs.

The history of psychrometrics started on a foggy evening in 1902 on a train platform in Pittsburgh. A young engineer for Buffalo Forge Company was working on an air conditioning design problem involving a Brooklyn printer who was having a problem with color registration between printing press runs. Color printing was done at that time by running the paper through the presses for each primary color. The concentration of the various color dots gave the pictures their color. Since paper changes dimensionally with changes in the humidity, on some days, the colors were not lining up, leading to poor quality and wasted materials. On this foggy night, the young engineer observed the fog condensing on cold surfaces and determined that there was a relationship between temperature and humidity. As temperature dropped, the air could hold less moisture. It followed that a temperature could be reached where the air could hold no more moisture and a concept called ***dew point*** control was born. This understanding of dew point allowed him to solve the printer's problem. The young engineer, Willis Carrier, went on to mathematically describe the phenomena he observed that night and the science of psychrometrics was born.

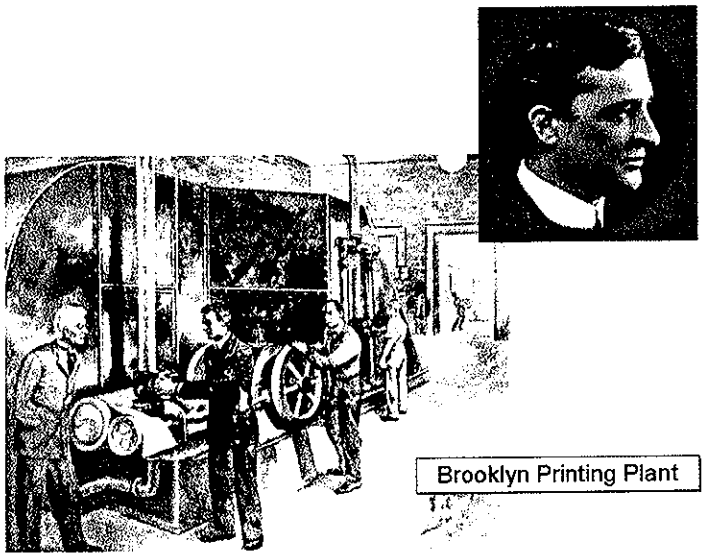


Figure 1

Dr. Carrier and the Brooklyn Printing Plant

The formulas that were developed were plotted on a chart that is the psychrometric chart. This chart is one of the most useful tools a system designer has to describe air conditioning processes.

What is Psychrometrics?

Psychrometrics is the study of the thermodynamic properties of moist air. In other words, if the air is to be conditioned, how can the amount of heat that must be added or removed and the amount of moisture that must be added or removed be determined? This is what we can learn from our study of psychrometrics.

Five uses for psychrometrics:

- Determine the temperature at which condensation will occur in walls or on a duct.
- Find all the properties of moist air by knowing any two conditions.
- Calculate the required airflow to the space and the equipment to satisfy the loads.
- Determine the sensible and total cooling load the unit needs to provide.
- Determine the coil depth and temperature to meet the design load conditions.

Properties of Air and Vapor

We will start at the beginning with air itself. Atmospheric air is a mixture of a number of gases. The two primary gases are nitrogen and oxygen. Nitrogen accounts for 78 percent of air's weight by volume and oxygen accounts 21 percent. The remaining 1 percent is trace amounts of other gases, but these do not appear in volumes significant enough to be a factor in psychrometric calculations.

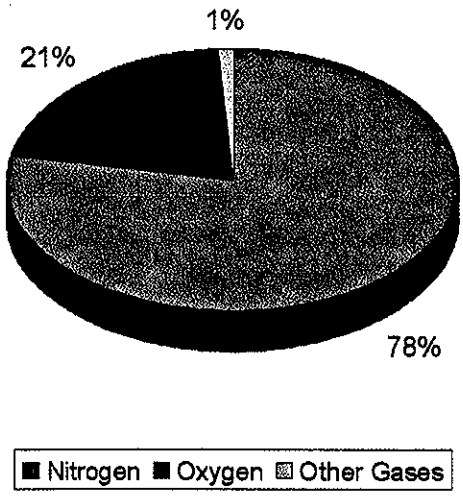


Figure 2

Composition of Dry Atmospheric Air

Atmospheric air has one other element in this mixture of gases commonly called air: water vapor. Water vapor is not present in large quantities in the atmosphere; however, it is a significant factor to those concerned with the field of psychrometrics and air conditioning.

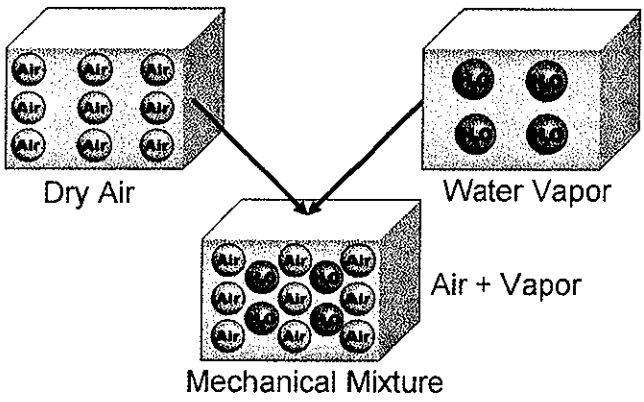


Figure 3

Atmospheric air is a mixture of dry air and water vapor.

How Air and Water Vapor are Measured

Air conditioning is the simultaneous control of temperature, humidity, cleanliness, and distribution. So, the first order of business in order to control temperature and humidity, is how they can be measured. Once temperature and humidity are determined, then the amount of each to be removed or added can be calculated.

Convention for the industry is to base calculations of air properties on pounds. Since air is a mixture, and not a compound, the amount of moisture in the mixture can change. Therefore, to have a common measuring point, ***moisture content*** is defined by comparing the moisture content at any point to dry air.

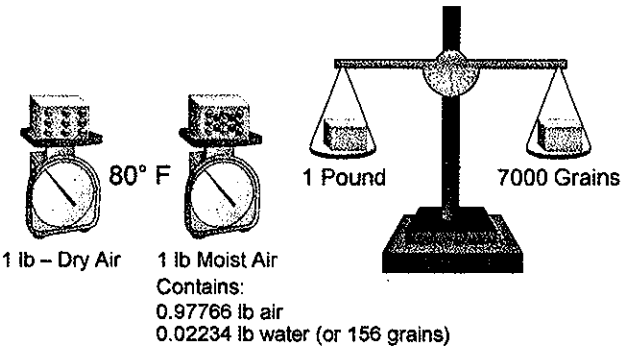


Figure 4

Psychrometric calculations are based on a pound of dry air.

The amount of actual water vapor present in a quantity of air is so small that it is measured in grains. It takes 7000 grains to make up one pound. Since one pound of air at 100° F, with all the water it can hold, contains 302.45 grains (about ½ ounce), this water does not have much bearing on the actual weight of the air. The actual final weight of a volume of air will be the sum of the air's dry weight and the weight of the water vapor it contains.

The unit of measurement

for moisture content is pounds of moisture per pound of dry air (lb_w / lb_{da}).

Note: to convert from pounds of moisture per pound of dry air to grains is:

$lb_w / lb_{da} \times 7000 = Grains$

Humidity and Its Sources

The common term for the water vapor that is in the air is humidity. Humidity has many sources. Evaporation from oceans, lakes, and rivers puts water into the air and forms clouds. Inside buildings, cooking, showers, people, open sources of water, and process work can add water vapor.

How can the exact amount of evaporated moisture be measured? Formulas are available that allow us to calculate the amount. However, the psychrometric chart makes it easy and provides a good way to visualize the process.

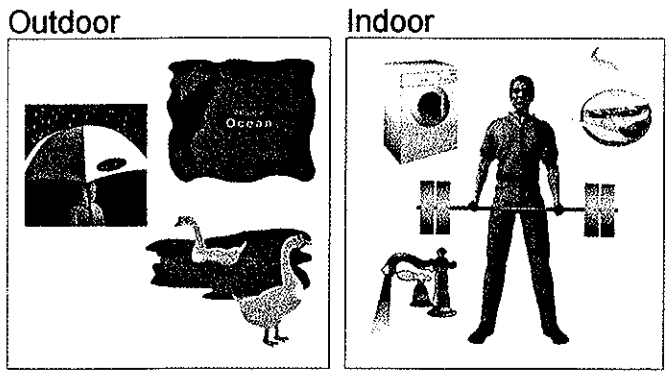


Figure 5

Water vapor in the air comes from many sources.

How the Air-Vapor Mixture Reacts

Two basic laws apply to the air and vapor mixture that make our calculations possible. First, within the range of comfort air conditioning, the mixture follows the ideal gas laws. Put simply, if two properties of either pressure, temperature, or volume, are known, the other one may be calculated. Second, the gases follow ***Dalton's law of partial pressures***. This means that air and the water vapor in the air occupy the same volume and are at the same pressure as if one alone were in the space, and the total pressure is the sum of each.

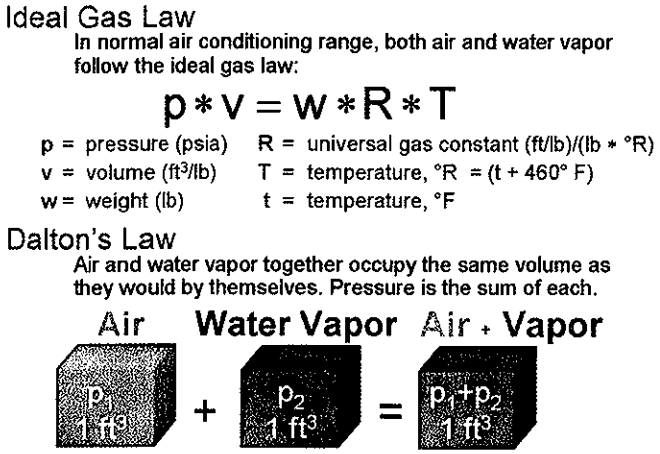


Figure 6

The ideal gas law and Dalton's Law control psychrometric calculations.

Temperature and Pressure

Our first air property, air temperature, can be easily determined with a standard thermometer. What about the second, pressure? What is air pressure?

Air pressure is often called barometric pressure.

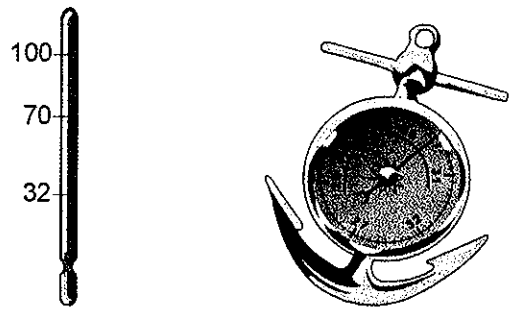


Figure 7

Air Temperature and Pressure

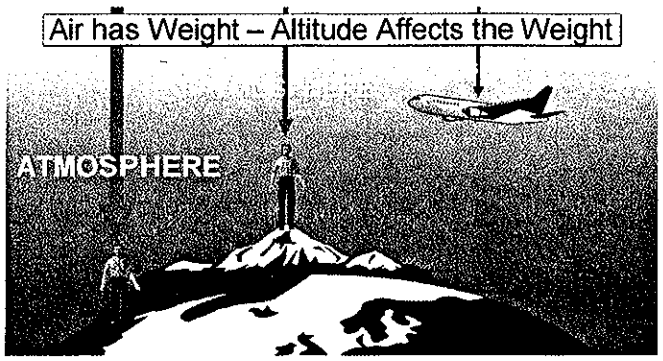


Figure 8

The weight of atmospheric air varies with elevation.

The daily weather report gives the barometric pressure. Air has weight, even though we may not recognize it as such. The barometer is a measure of the weight of the column of atmospheric air. Barometric pressure is usually measured in inches of mercury, (in. Hg). Notice that the weight is dependent on the elevation, the higher above sea level the lower the air pressure.

The air in a space where conditions are being calculated is dependent on barometric pressure. To account for the weight of atmospheric air, calculations use the absolute pressure. This is referred to as pressure in pounds per square inch absolute, written psia. At sea level, this is 29.921 in. Hg and converts to 14.696 psia; in Denver at 5000 feet elevation the pressure is 12.23 psia. Since the two laws depend on pressure, the charts also depend on pressure. To account for this, psychrometric charts are published for different elevations, sea

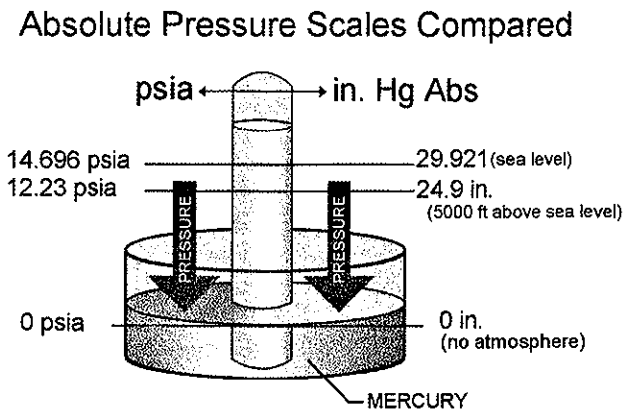


Figure 9

Absolute pressure is used in psychrometric calculations.

level, 2,500 feet, 5,000 feet, 7,500 feet, and 10,000 feet are common. Charts can be used for plus or minus 1,000 ft of chart elevation without correction.

Pressure measurements used in HVAC are sometimes in pounds per square inch gauge, psig or psi; these measurements are the difference above the atmospheric. For psychrometric calculations, all pressures are in psia.

Recall that in the daily weather reports the barometer changes from day to day for the same location. This is because air pressure is also dependent on the moisture in the air. Therefore, determining air pressure is dependent on elevation and moisture content.

Dalton's law said that the total pressure was the sum of the air pressure and water vapor pressure; so, which weighs more, dry air or moist air?

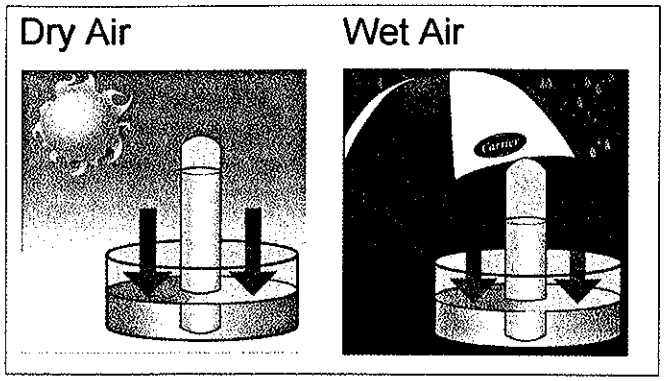


Figure 10

Which weighs more, dry air or wet air?

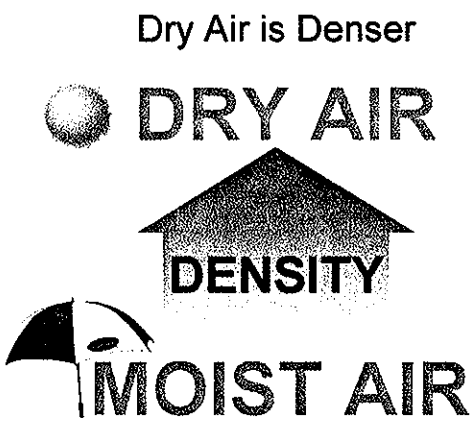


Figure 11

Dry air is denser than moist air.

Again, think about what happens in the weather report. When they say it will be a beautiful clear sunny day, there is a high-pressure front with a rising barometer. Conversely, a hurricane has a very low pressure. Therefore, the answer is that dry air weighs more. This is true because in a pound of atmospheric air the water vapor occupies a greater percentage of the volume and weighs less. This means the dry air is denser than the moist air.

Since calculations of air properties are dependent on the altitude, temperature, and moisture content, the industry has agreed on a set of conditions for the air called standard air. This is the point of reference we will use for our calculations. Standard air is defined as sea level, 59° F, and a barometer of 29.921 in. Hg, or 14.696 psia. The amount of moisture will be measured based on dry air.

Conditions of Standard Air

Standard air for psychrometrics is at sea level and is 59° F.

For most air conditioning work, sea level data may be used up to 1000 feet elevation with no correction.

Building the Psychrometric Chart

A psychrometric chart is a convenient way to determine properties of air and describe air conditioning processes. To create the chart, it is necessary to base the calculations on elevation; sea level is used for this discussion.

Since the behavior of temperature and humidity are predictable at atmospheric pressure and temperatures, different characteristic properties can be plotted on a graph. To start the chart it is necessary to define our vertical and horizontal axis.

Dry Bulb Temperature Scale

Our horizontal axis on the chart will represent an ordinary temperature scale called ***dry bulb temperature***. These lines can then be extended vertically so any point on the line is equal to that dry bulb temperature. The lines could cover any temperature range, but here we will use a range common for normal comfort calculations, 30° F to 120° F.

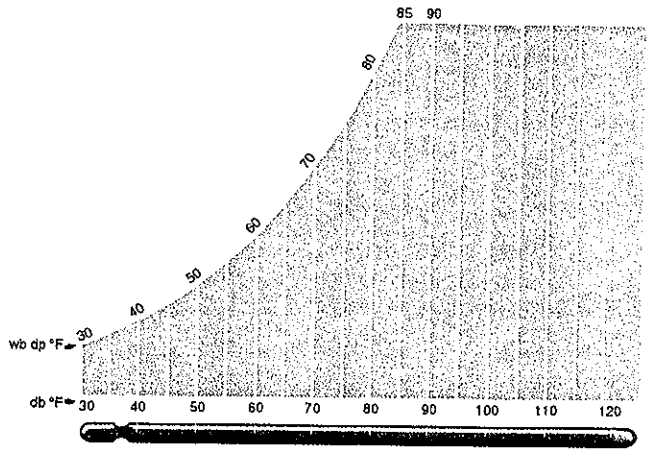


Figure 12

The horizontal scale is dry bulb temperature.

Specific Humidity Scale

Next, the vertical scale is made according to the amount of water vapor mixed with each pound of dry air. Since the amount of water vapor is small, the scale is plotted in grains of water vapor per pound of dry air at standard atmospheric pressure. Some charts plot water vapor in pounds of water per pound of dry air rather than grains. The vertical axis is called the specific humidity scale.

Not always grains

Some charts will plot specific humidity in pounds of water per pound of dry air (lb/lb). Some charts have both scales.

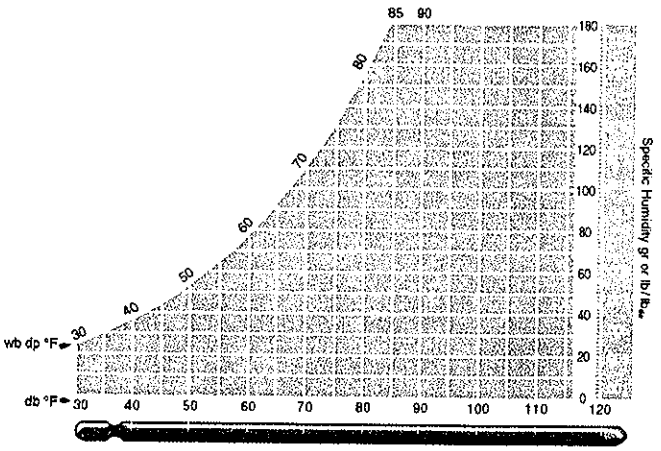


Figure 13

The vertical scale is specific humidity, a measure of the amount of water vapor in the air.

Now it is easy to locate many air and water vapor mixtures by using the chart. For example, air at 75° F dry bulb temperature is anywhere on the vertical line above 75° F, regardless of the humidity. Air with 60 grains of water vapor per pound of dry air is anywhere on the horizontal line at 60 grains. The air at 75° F and 60 grains is the point where these two lines meet.

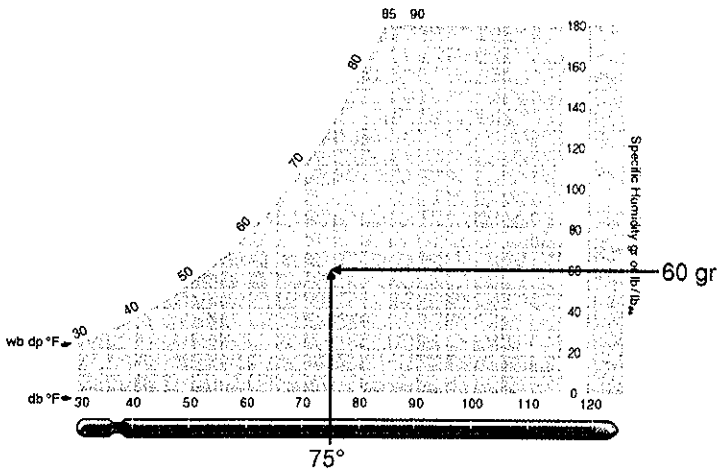


Figure 14

Locate a dry bulb and specific humidity point.

Dew Point and the Saturation Line

Suppose this air is then cooled – what happens? Observe the dew on the grass on a summer morning. The night air was cooled and water vapor in the air from the day before condensed on the grass. As the temperature dropped, the air could hold no more water vapor and so water condensed out of the air. This highlights the fact that the amount of water vapor that the air can hold is related to the air temperature.

As the air at 75° F and 60 grains has the temperature reduced, no water vapor is removed until the air reaches its point of maximum humidity. For this example, when the temperature is 53° F, any further cooling will now cause some water vapor to condense, because at 53° F the air can hold only 60 grains per pound of air. The temperature at which the moisture content or relative humidity has reached 100 percent is called the dew point. If the temperature drops below the dew point, say to 48° F, only 50 grains of water vapor remain in the air. Therefore, 10 grains of water vapor condenses. If the temperature drops still further to about 42° F, another 10 grains is condensed as only 40 grains remain in the air at this temperature.

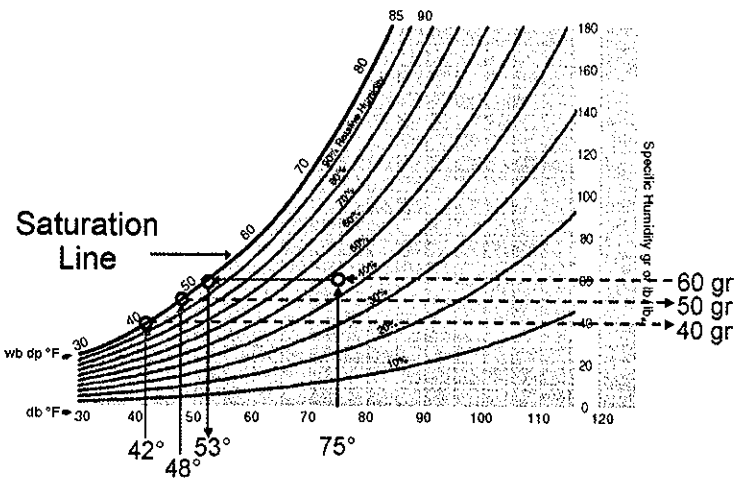


Figure 15

Saturation Line

Water vapor in the air at the dew point has a relative humidity of 100 percent and is said to be saturated.

A line that connects these and other 100 percent saturation points is known as the saturation line, which is the same as the 100 percent relative humidity line. This line gives the dew point temperatures and is called the saturation curve or saturation line. The dew point temperature for air depends upon the amount of water vapor present and is found on the psychrometric chart by moving horizontally over to the saturation curve and reading the temperature there.

To illustrate the use of dew point, we will check to see whether sweating occurs on a 55° F uninsulated supply air duct that runs through an unconditioned space. At a space temperature of 95° F dry bulb and 100 grains of water vapor, the dew point is 67° F. That means the 55° F duct cools the surrounding unconditioned air below the 67° F dew point, therefore, water vapor condenses. Moisture condenses not only on the duct, but also on any surface with a temperature below the dew point of the air. If water dripping is likely to cause damage, the duct should be wrapped with insulation then with a vapor barrier. Enough insulation should be used to prevent the outside surface temperature from dropping below the dew point of the surrounding air.

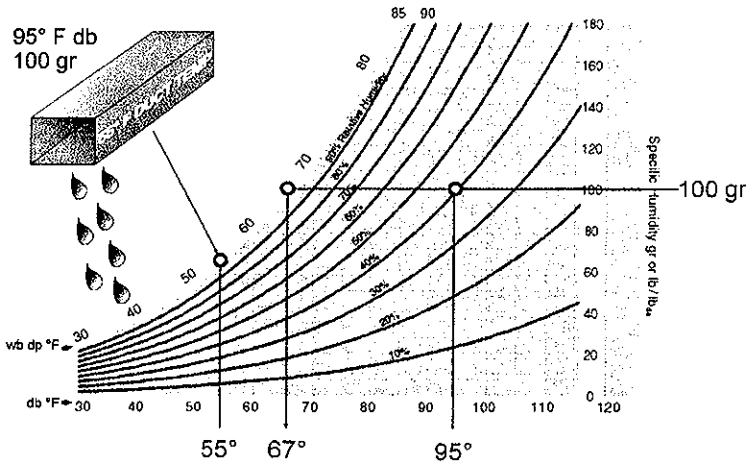


Figure 16

Determine dew point with conditions of a duct in an unconditioned space.

Relative Humidity Lines

The saturation curve indicates the 100 percent relative humidity line. Lines for partly saturated air look very much like the saturation line on the chart. These lines normally appear in increments of 10 percent and indicate the degree of saturation.

Relative humidity is defined as the amount of moisture in the air compared to the maximum amount that could be present at the same temperature. For example, air at 75° F dry bulb with 60 grains shows a relative humidity between the 40 and 50 percent lines on the chart.

Check this by following the 75° F dry bulb temperature line up to the saturation line could be used to check this. Here, air has 132 grains of water vapor. The relative humidity is approximately equal to 60 divided by 132, or 45 percent.

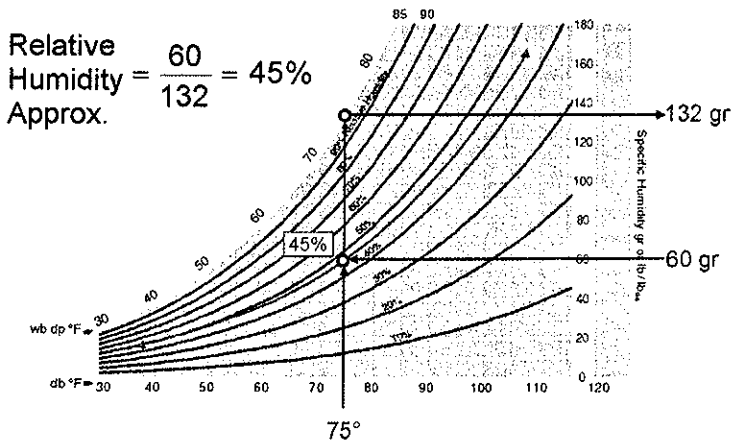


Figure 17

Relative humidity lines resemble the saturation curve.

One use for relative humidity lines is to determine the maximum allowable relative humidity permitted inside a house in winter without having moisture condense on the windows. If the window surface temperature is 35° F and the room temperature is 75° F, the maximum relative humidity can be found by starting at 35° F at the saturation line and moving across until the 75° F dry bulb temperature line is intersected. This point falls between 20 and 30 percent and is estimated at 23 percent relative humidity. Therefore, the maximum winter relative humidity is 23 percent and controls should be used to maintain this level.

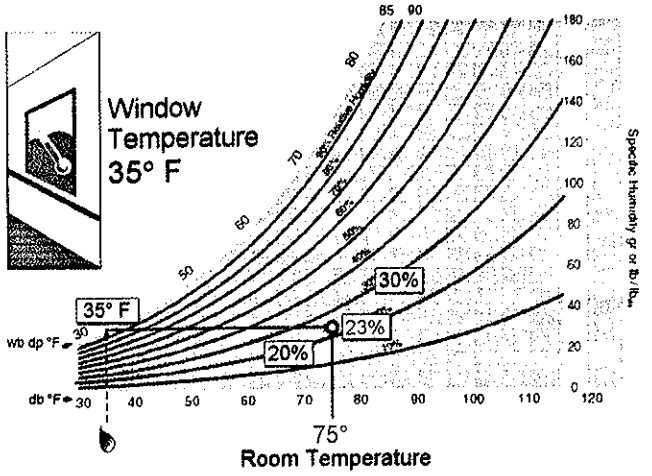


Figure 18

Relative humidity lines can be used to determine maximum winter humidity levels.

Wet Bulb Temperature Lines

Another term that is often used in air conditioning is ***wet bulb temperature***. To see how it is obtained, start with the same pound of air at 75° F dry bulb temperature and 60 grains of water vapor. Pass this air through a series of water sprays that use the same water repeatedly, except for the small amount that may evaporate. This device is called a saturator. As the air goes through the water spray, the temperature of the air drops because heat is absorbed to evaporate the atomized water. If the sprays are well designed, the air temperature drops, in this case, down to almost 61.5° F. At this temperature, it is saturated with almost 82 grains of water vapor. The temperature of the saturated air, after passing through the sprays is called the wet bulb temperature. In this case, 61.5° F is the wet bulb temperature of air at 75° F dry bulb temperature and 60 grains of water vapor.

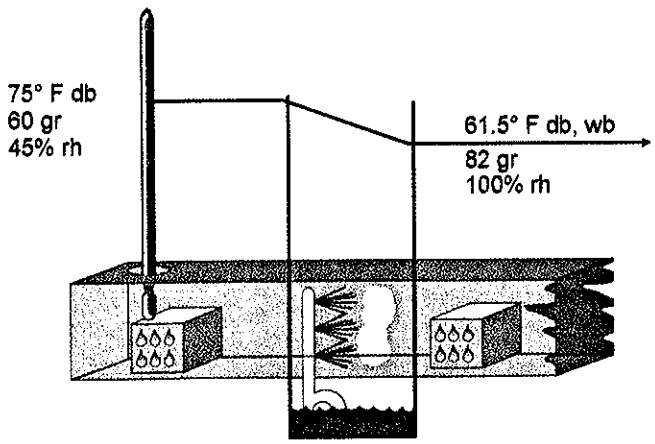
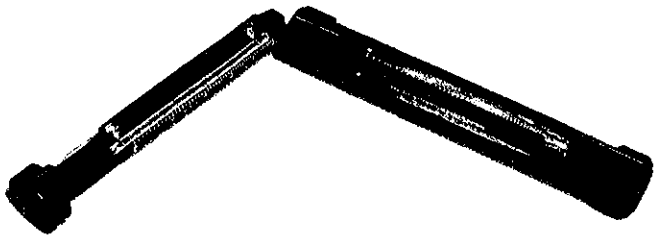


Figure 19

Water saturates the air when passed through a water spray saturator.

This experiment would be difficult to perform every time the wet bulb temperature was needed. Instead, a device called a sling psychrometer can be used more conveniently and gives quite accurate results. The sling psychrometer consists of two thermometers mounted in a frame and attached to a handle by means of a swivel. One thermometer has a wetted cotton wick wrapped around its mercury bulb. When the apparatus is whirled around, air is moved across the wick and some of the water is evaporated. This evaporation absorbs heat and causes the thermometer to register the wet bulb temperature. A dry bulb thermometer is usually mounted on the sling psychrometer so that a wet bulb/dry bulb comparison can be instantly taken. This piece of equipment provides a convenient way of determining the humidity condition in the air, since measuring the specific humidity or dew point directly is difficult to do.



- Avoid adverse conditions that can affect reading
- Moisten wick before procedure
- Rotate device at least 2 minutes
- Read device immediately after rotation

Figure 20

Wet bulb temperature is determined with a sling psychrometer.

This wet bulb process is also shown on the psychrometric chart. The initial unsaturated air started at 75° F with 60 grains and ended up saturated at 61.5° F with 82 grains. If these two points are connected, they form the 61.5° F wet bulb temperature line. In a similar manner, the wet bulb lines run diagonally from the lower right up to the saturation curve. All wet bulb temperatures are read at the saturation line.

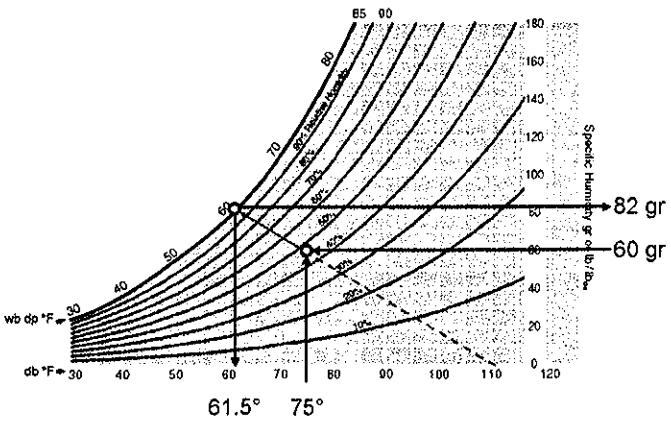


Figure 21

Wet bulb temperature lines run diagonally, intersecting the saturation curve at the wet bulb temperature.

To round out our understanding of the information we can get from a psychrometric chart, two other properties of the air need to be explained.

Specific Volume Lines

The first property is specific volume. Specific volume is defined as the number of cubic feet occupied by one pound of air at any given temperature and pressure. For example, one pound of air at 75° F dry bulb displaces a volume of 13½ cubic feet at sea level. If the air is heated to 95° F, it expands and takes up 14 cubic feet. Air, being a gas, will decrease in density as its temperature rises. If the air is cooled to 55° F, it occupies only 13 cubic feet, because the air is denser at lower temperatures. The lines for these specific volumes are shown on the psychrometric chart as almost vertical lines, which slant to the left. Specific volume is used primarily for checking fan performance and determining fan motor sizes for low and high temperature applications.

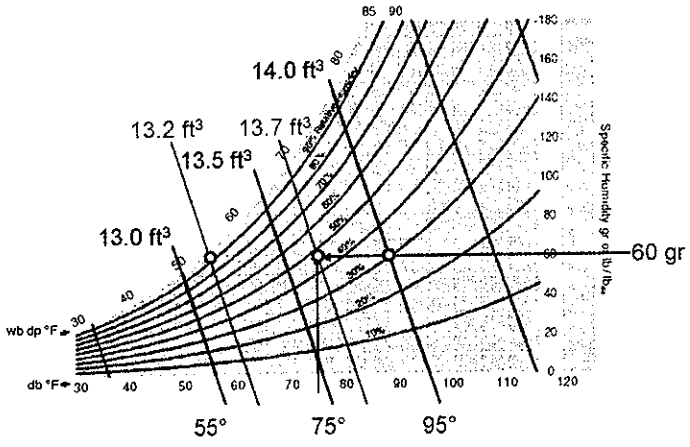


Figure 22

Specific Volume Lines

Enthalpy Scale (Total Heat Content)

Another property used in the air conditioning field is ***enthalpy***, or the total heat content of the air and water vapor mixture. Enthalpy is very useful in determining the amount of heat that is added to or removed from air in a given process.

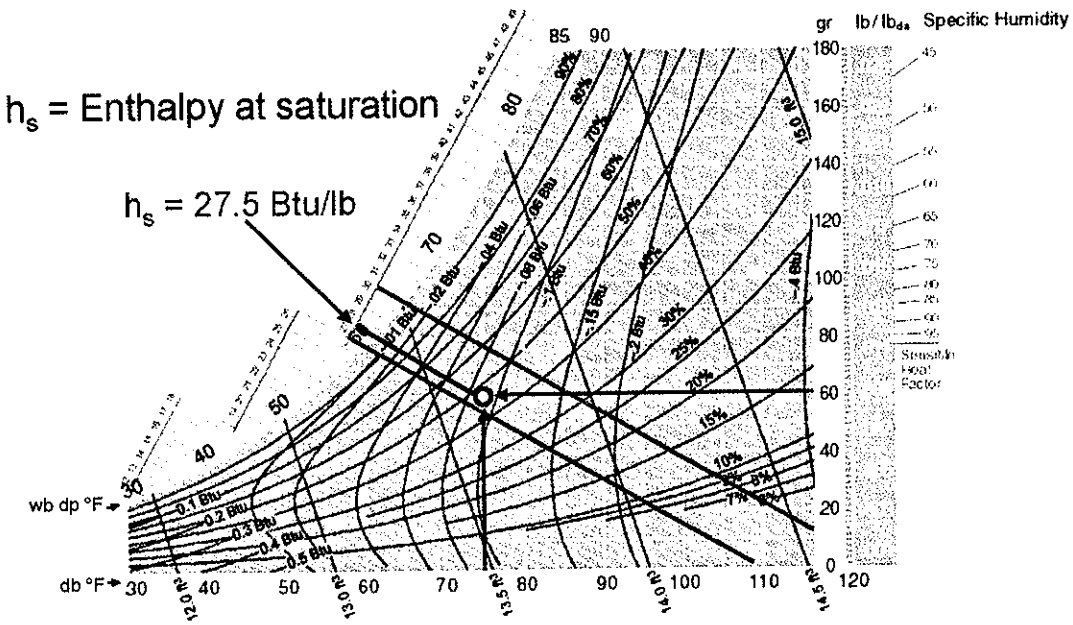


Figure 23

The enthalpy scale is an extension of the wet bulb lines.

It is found on the psychrometric chart by following along a wet bulb temperature line, past the saturation line, and out to the enthalpy scale. For example, air at 75° F dry bulb temperature and 60 grains of water vapor has an enthalpy of 27.50 Btu per pound of air. The enthalpy scale is shown at the extension of the wet bulb temperature lines and is read directly where the extended wet bulb line intersects the scale.

The enthalpy actually changes as the air becomes less saturated. This is shown on some charts with a deviation correction and by sloped enthalpy lines on other charts. For most comfort air conditioning calculations, the saturated enthalpy can be used without correction.

State Point

If all the lines that have been discussed are combined in one chart, it will look like the diagram. The chart now shows dry bulb temperature, specific humidity, dew point temperature, relative humidity, wet bulb temperature, specific volume and, enthalpy. When any two of these values are known, the exact condition of the air can be located on the chart and all other properties can be found from this one point. Such a point is sometimes referred to as a state point.

State Points

Intersection of any two values that define all other air properties.

Dry bulb temperature (db)

Wet bulb temperature (wb)

Dew point temperature (dp)

Specific humidity (gr or lb/lb$_{da}$)

Relative humidity % (rh)

Specific volume (v)

Enthalpy (h) Btu/lb

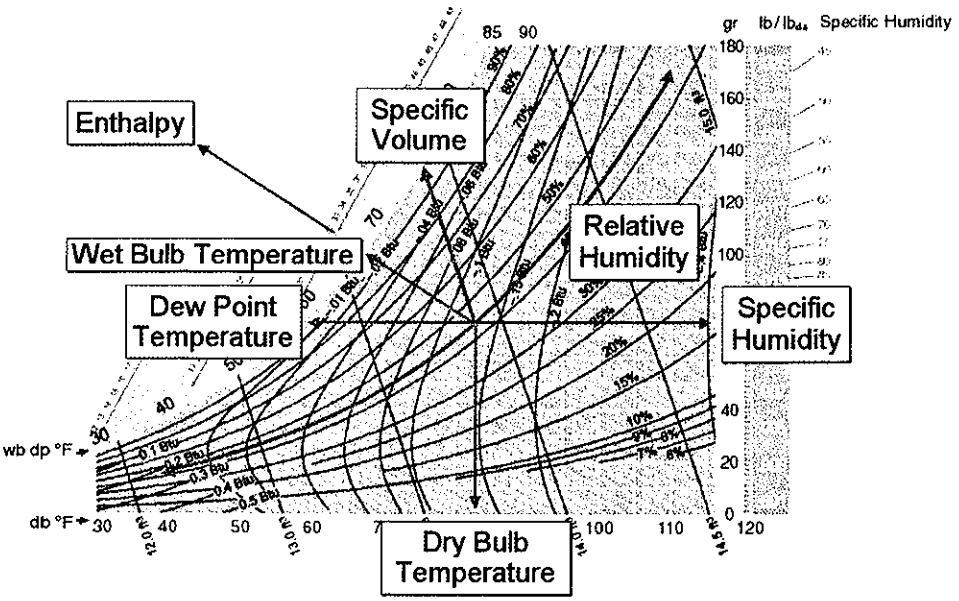


Figure 24

Seven properties can be found on the psychrometric chart.

Using the Psychrometric Chart

All the properties pertinent to most air conditioning calculations have now been defined using the psychrometric chart. The state point, or locating the properties using the two properties, now gives us a useful tool to evaluate conditions of the air at any point in the air conditioning process. Let's find the properties at four points common in an air conditioning system; room air, outdoor air, mixed return and outdoor air, and air leaving a cooling coil.

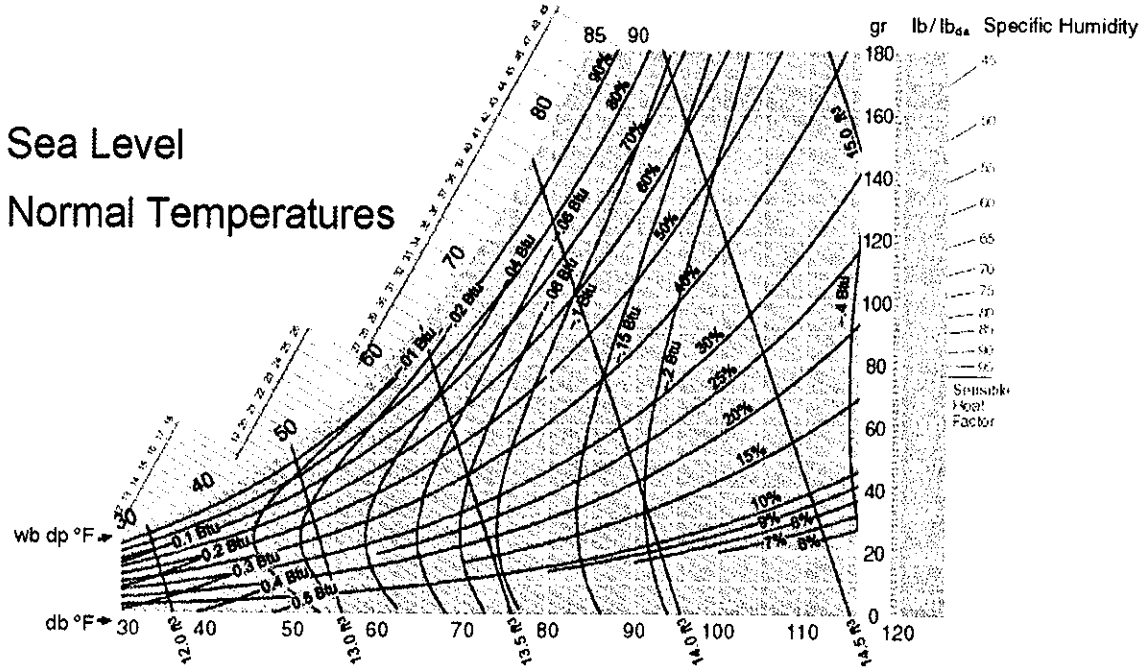


Figure 25

Complete Sea Level, Normal Temperature Psychrometric Chart

Examples Using State Points

First, room air conditions are normally given as a dry bulb temperature and a percent relative humidity, typically 75° F and 50 percent. To find the state point on the chart, we locate 75° F and follow the line vertically until it intersects the 50 percent relative humidity line. The other five air properties can then be read from this state point: wet bulb of 62.5° F, dew point of 55° F, specific humidity of 65 gr, air volume of 13.7 ft^3, and enthalpy of 28.1 Btu/lb.

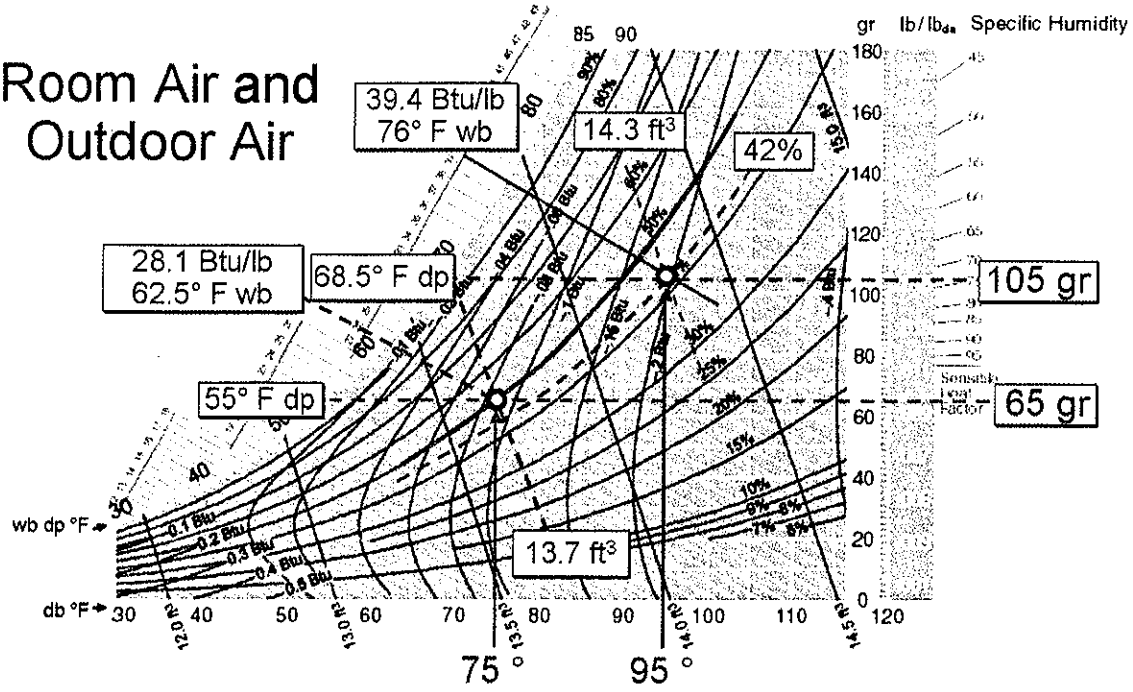


Figure 26

State Point Examples for Room Air and Outdoor Air

In a similar way, we can determine the other air properties at the outdoor condition, which is normally given as a dry bulb temperature and wet bulb temperature. For this example assume the state point conditions are 95° F dry bulb and 76° F wet bulb, the other properties are: relative humidity of 42 percent, dew point of 68.5° F, specific humidity of 105gr, air volume of 14.3 ft^3, and enthalpy of 39.4 Btu/lb.

Common air conditioning practice is to return air from the space to the unit and to mix that air with a portion of outdoor air. Using the last two examples, if 10 percent of the air is outdoor air and 90 percent of the room air are mixed the resulting mixed air state point conditions will be 78° F dry bulb and 64.7° F wet bulb. We explain how this is calculated in the next section of this module. Again, the properties can be determined by finding the state point and reading the other properties, which in this instance are: relative humidity of 50 percent, dew point of 57° F, specific

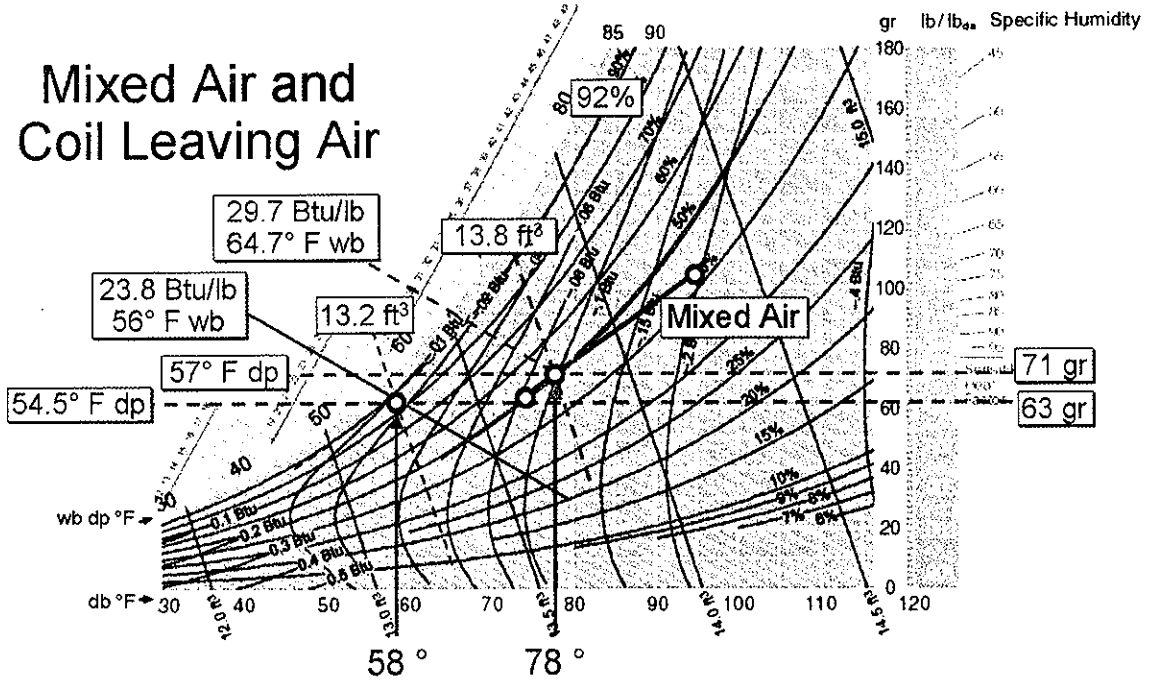


Figure 27

State point examples for mixed and coil leaving air

humidity of 71gr, air volume of 13.8 ft^3, and enthalpy of 29.7 Btu/lb.

Finally, the typical air conditions leaving the cooling coil can be found. Typical conditions are 58° F dry bulb and 56° F wet bulb. Finding this state point on the psychrometric chart, the other properties can be read: relative humidity of 90 percent, dew point of 54.5° F, specific humidity of 63 gr., air volume of 13.2 ft^3, and enthalpy of 23.8 Btu/lb.

We have now developed the psychrometric chart and learned how we can determine air properties using it. This is a good time to practice using the chart, Work Session 1 in the back of the book covers the skills covered so far.

Air Conditioning Processes

Air conditioning design is the application of a number of different psychrometric processes. For our purposes, a process could be defined as moving from one state point to another. To do this heat and moisture must be added or removed. In this section, we will discuss the eight basic air conditioning processes and how the chart is used to determine the heat and moisture added or removed.

Eight Basic Process Types

Starting at a condition on the chart, directional arrows show a change in a given direction. These represent the basic processes. Notice that as the condition changes either the dry bulb temperature, specific humidity, or both change. If the beginning and ending point are known, the chart can be used to determine how much heat and moisture change. Air mixing is also a typical air conditioning process and is included in this section as well.

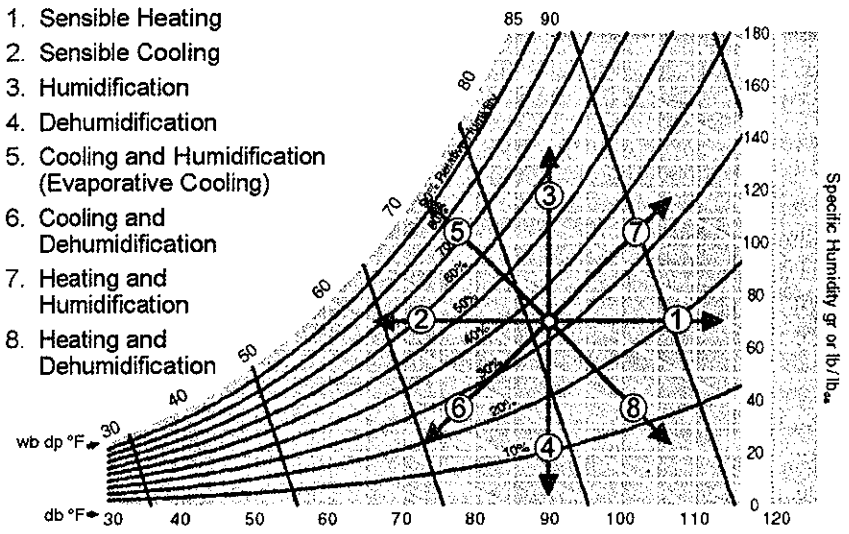


Figure 28

The Eight Basic Air Conditioning Processes

Each of the eight processes is familiar though we may not always recognize them by the process definition. The eight processes and a typical example are:

Sensible Heating – Residential gas furnace

Sensible Cooling – Cooling coil above the air dew point

Humidification – Steam humidifier in an air handler

Dehumidification – Dehumidifier

Evaporative Cooling or Cooling and Humidification – Swamp cooler

Cooling and Dehumidification – Cooling coil below the air dew point

Heating with Humidification – Winter heating with humidifier

Heating with Dehumidification – Chemical dehumidification wheels

Pure humidification and dehumidification are rare as some heating or cooling normally occurs in the process as well.

Sensible and Latent Heat Changes

The change in dry bulb temperature and specific humidity are referred to in air conditioning processes as sensible and latent heat changes.

Sensible heat changes result in a change in temperature and are indicated by a horizontal line on the psychrometric chart. Processes that increase dry bulb temperature are heating and those that decrease dry bulb temperature are cooling. As the dry bulb changes without a change in the specific humidity, notice that the wet bulb changes, but the dew point and specific humidity remains the same.

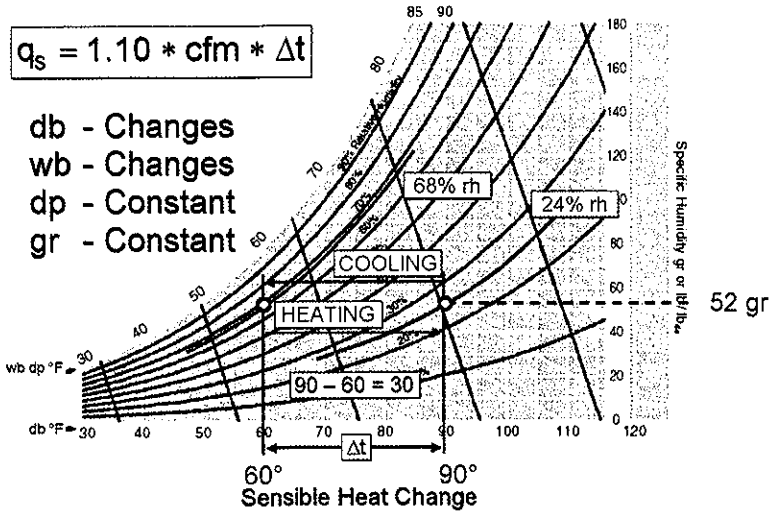
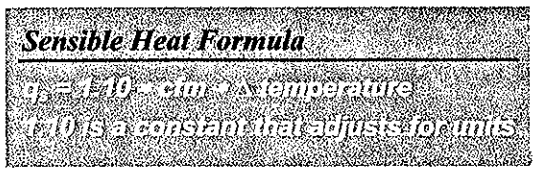


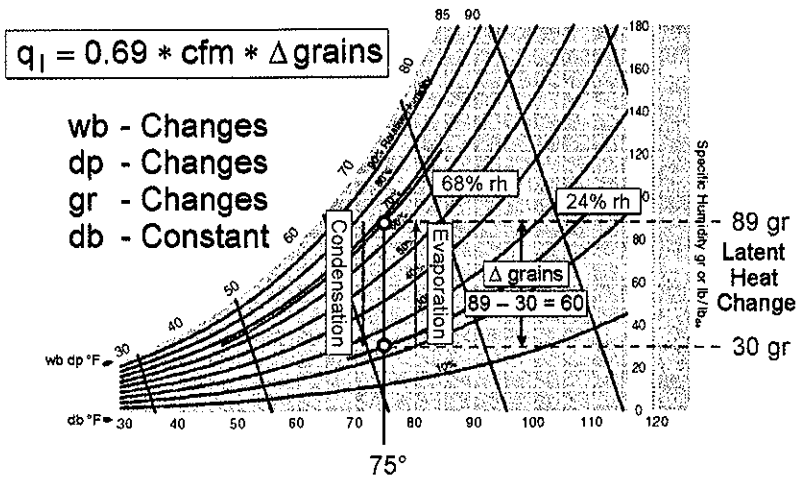
Figure 29

Sensible Heating or Cooling Processes

Once we know the change in dry bulb temperature, we can determine the sensible heat added or removed. Most air conditioning calculations are done using the volume flow rate, or cfm. With these two pieces of information, a simple formula may be used to determine the amount of sensible heating or cooling (q_s).



A latent heat change occurs when water is evaporated or condensed and the dry bulb temperature does not change. This shows up as a vertical line on the chart. Processes that increase specific humidity are ***humidification*** and those that decrease specific humidity are ***dehumidification***. As the specific humidity changes without a change in the dry bulb temperature, notice that the wet bulb, specific humidity and dew point change but the dry bulb remains the same.



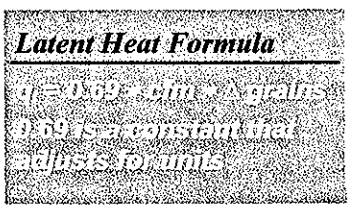


Figure 30

Latent Heating and Cooling Processes

Many air conditioning processes are a combination of both sensible and latent heat changes. The total heat is the sum of the sensible heat and the latent heat.

Enthalpy can be used to determine the total heat removed from a volume of air. Reading the scale between the two wet bulb lines does this. For example, air at 75° F dry bulb and 61.5° F wet bulb has an enthalpy of 27.5 Btu/lb. If this air is cooled and dehumidified to 55° F dry bulb and 51° F wet bulb, the enthalpy leaving the cooling coil is found to be 20.8 Btu/lb, Therefore, a total of 6.7 Btu is removed from each pound of air.

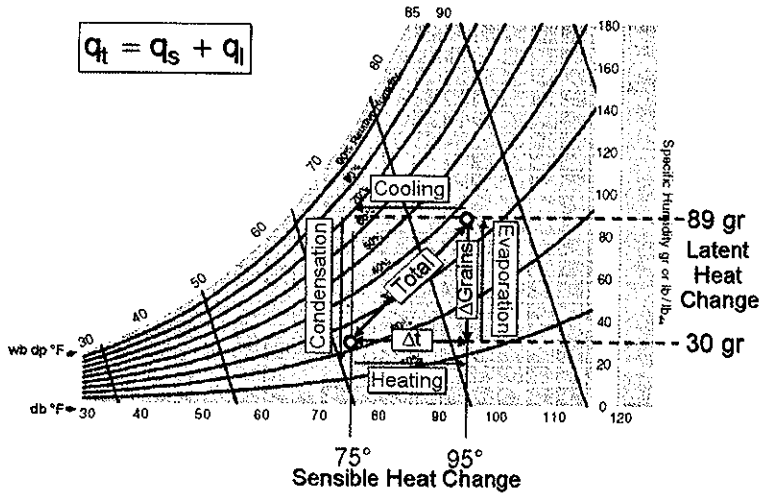


Figure 31

Total heat is sensible plus latent heat

If a triangle is drawn as shown, the vertical distance represents the amount of moisture removed - that is, latent heat. The horizontal distance represents the sensible cooling of the air. The enthalpy at the intersection of the vertical and horizontal lines is 25.8 Btu per pound. Therefore, the amount of latent heat removed is the difference between 27.5 and 25.8 or 1.7 Btu per pound. The sensible heat removed is the difference between 25.8 and 20.8, which equals 5 Btu per pound.

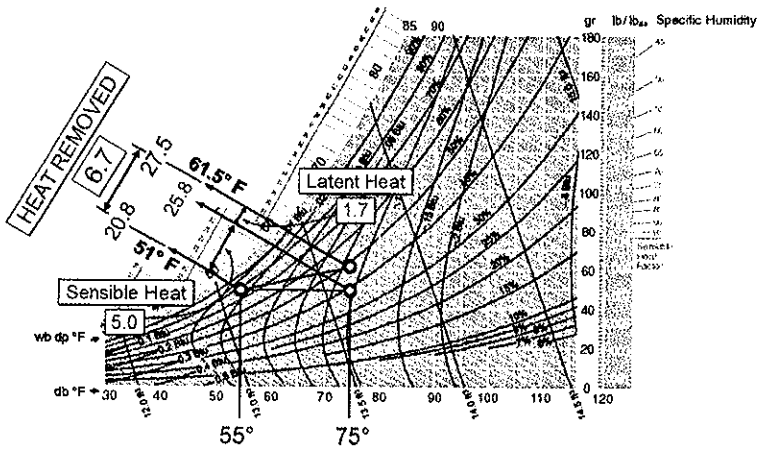


Figure 32

Enthalpy can be used to determine the total heat removed.,

When the enthalpy difference is used, we can use one additional formula to calculate the total capacity. The total capacity, sometimes called grand total heat is found by multiplying the airflow by a constant, 4.5, and the enthalpy difference.

By this using a simple formula:

GTH = 4.5 * cfm * Δh

The difference in enthalpy (Δh) between the time it enters and leaves a space or a coil can be used to determine the grand total heat (GTH) gained or lost, in Btuh.

For our example, the difference in enthalpy is 6.7 Btu/lb. If 1000 cfm of air is circulated over the coil, which removes this heat, then 30,150 Btuh is removed, as follows:

GTH = 4.5 * cfm * Δh

= 4.5 * 1000 * 6.7

= 30,150 Btuh

Total heat formula

$Q_t = 4.5 \times cfm \times \Delta enthalpy$

4.5 is a constant that adjusts for units

In other words, the coil provides 30,150 Btuh of total cooling capacity.

Sensible Heat Factor

If cooling is combined with dehumidification and a line is drawn showing the process, the air comes down the sloping line marked TOTAL HEAT. The amount of sensible heat and the amount of latent heat involved determines whether the line has a gentle slope or a steep slope. This combination of sensible and latent cooling occurs so frequently in air conditioning that the slope of this line has been named the sensible heat factor.

The mathematical definition of the sensible heat factor (SHF) is shown in Figure 33. If no latent heat change occurs, then the sensible heat factor is 1.0 and the line is horizontal - a pure sensible heat change process. If the sensible heat factor is 0.8, the line starts to slope. This means that 80 percent of the total heat change is sensible and 20 percent is latent. That is approximately the condition that exists in a department store air conditioning system. If the sensible heat factor is 0.7, the line is still steeper. This indicates more latent heat, or more water vapor change compared to sensible heat or temperature change. A system with this sensible heat factor would be used for a theater, church, or restaurant.

If the above process were reversed, it would be a heating and humidifying process. A heating coil to add sensible heat and a water spray to add humidity or latent heat could accomplish this.

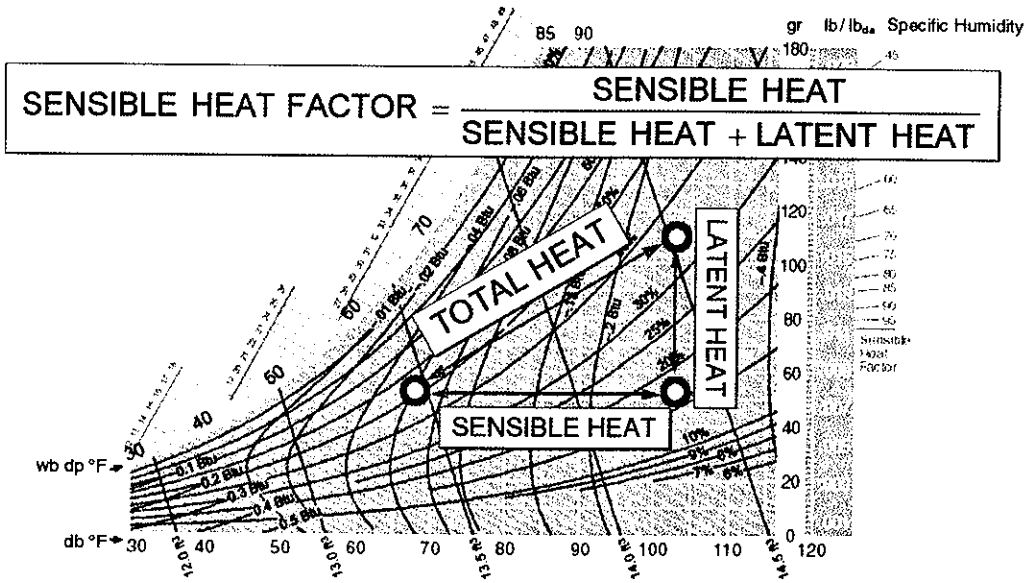


Figure 33

Sensible Heat Factor

For example, using the enthalpy calculated before, the total heat change is 6.7 Btu/lb, the sensible difference is 5 Btu/lb, and the latent difference is 1.7 Btu/lb. The SHF is then calculated by dividing the sensible heat difference by the total heat difference, which, in this example, is 0.75.

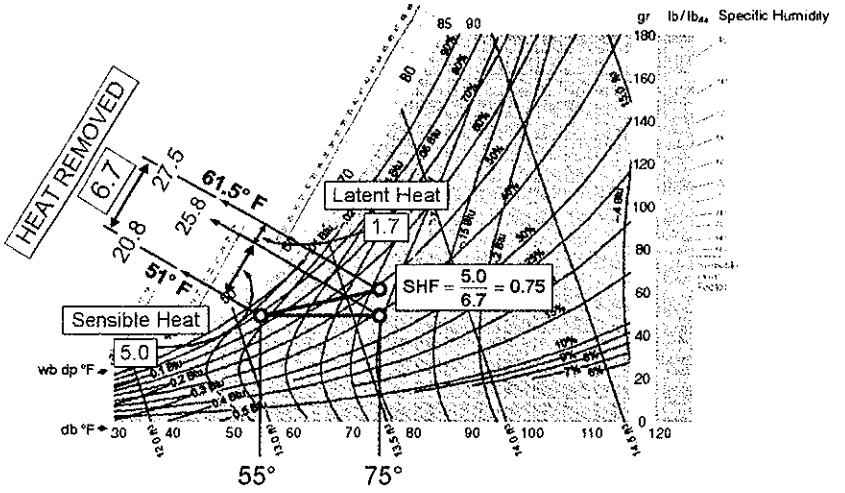


Figure 34

Example of Sensible Heat Factor Calculation

Sensible Heat Factor Scale

A convenient method for finding sensible heat can be found on the psychrometric chart. It is called the sensible heat factor scale. A small white circle printed on the chart at the 80° F dry bulb and the 50 percent relative humidity lines locates the pivot point of the scale.

To show the 0.90 sensible heat factor line for air at 75° F dry bulb and 60 grains of water vapor, take the following steps. First, get the slope of the 0.90 line by connecting 0.90 on the scale to the white circle. Draw a line parallel to this one passing through the air at 75° F and 60 grains. When the air is to be cooled and dehumidified, the apparatus dew point is found at the intersection of the sensible heat factor line and the saturation curve. In this case, it is 51° F. If the sensible heat factor is 0.80, the apparatus dew point, found by the same procedure, is 48° F.

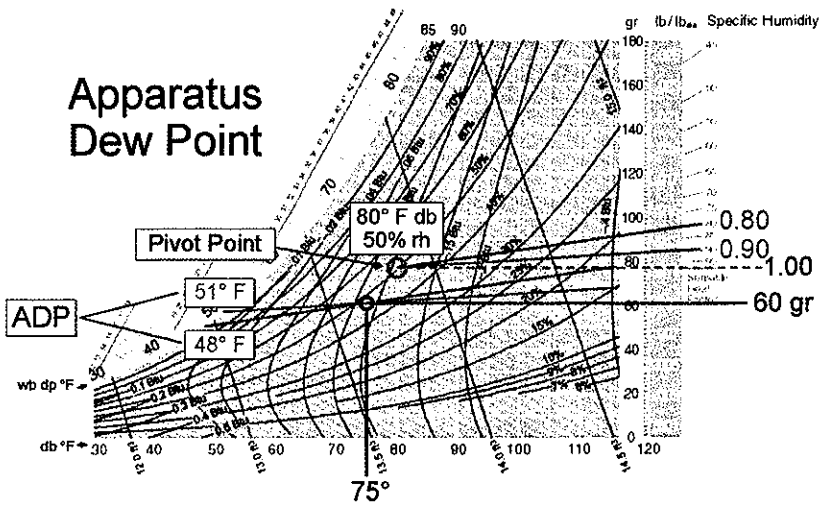


Figure 35

Use the sensible heat factor scale to find apparatus dew point.

The sensible heat factor is a very useful tool when making equipment selections. In combination with the psychrometric chart, it tells you the temperature at which the cooling coil must operate to handle the sensible and latent heat removal.

Sensible Heating and Cooling

A process that changes the sensible or dry bulb temperature without a change in the moisture content of the air is a sensible heating or cooling process.

To illustrate a sensible heating process, follow the example shown in the psychrometric chart in the figure. Air is heated by passing it over a heating coil. If the air starts out at 70° F dry bulb and 54° F wet bulb, its dew point is 40° F as obtained from the chart. After sensible heating to 100° F dry bulb, the dew point remains the same, because no water vapor has been added or condensed. The wet bulb temperature, however, has increased to 65° F. Also, notice that the relative humidity has decreased. This explains why relative humidity is high during early morning hours but decreases as the day gets warmer.

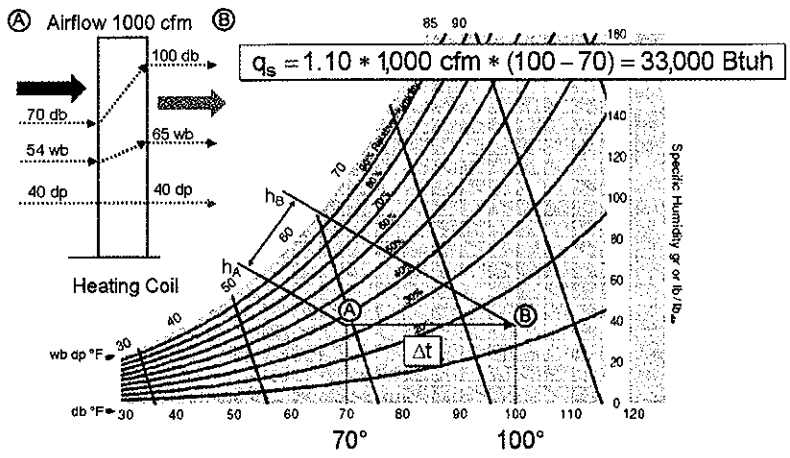


Figure 36

Sensible Heating Process

If the process airflow is 1000 cfm, the sensible heat equation can be used to determine the amount of heat that needs to be added to heat the air from 70° F to 100° F. In this example 33,000 Btuh of heat energy are required.

A hot water, steam heating, or electric heating coils are typical examples of this process.

If the process is reversed and the 100° F dry bulb and 40° F dew point air is cooled back to 70° F, we have a sensible cooling process. The wet bulb drops and the dew point remains the same. Notice that the heat energy added in the heating process and the heat energy subtracted cooling process are the same.

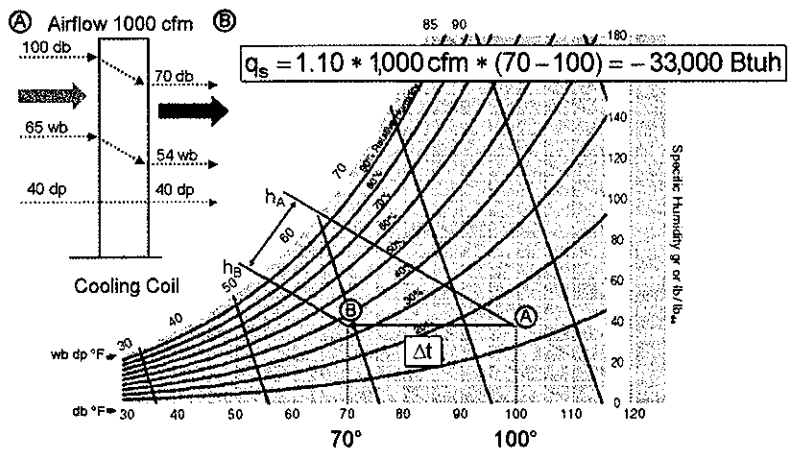


Figure 37

Sensible Cooling Process

The sensible cooling process often occurs when the surface temperature of a cooling coil is above the dew point.

Humidification and Dehumidification

When only moisture is removed with no change in the temperature, the process is dehumidification. In this example, 80° F air is entering and leaving the coil; the coil surface temperature is below the dew point of the air and moisture is removed. The dry bulb temperature remains the same and the wet bulb and dew point temperatures both decrease. The latent heat equation can be used with this example. It requires 27,600 Btuh of heat energy be removed to condense the 40 grains of moisture from the 1000 cfm of air.

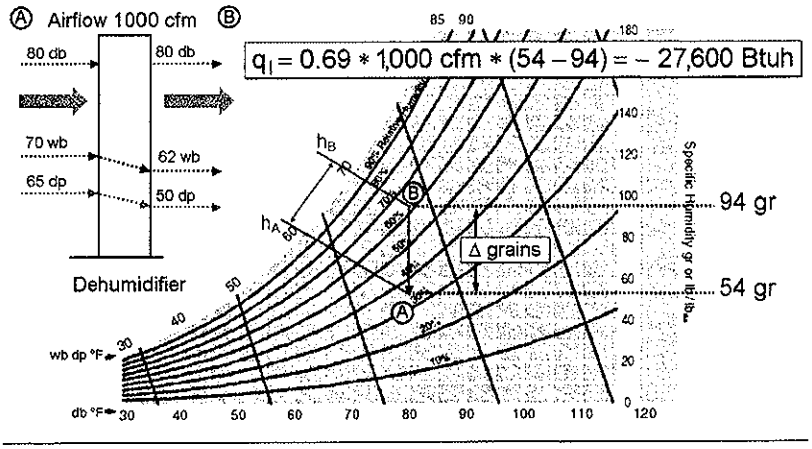


Figure 38

Dehumidification Process

This process is typical of what occurs with a dehumidifier some people use in a damp basement, during the summer. Removal of moisture only is not a common occurrence since most removal processes also tend to cool or heat the air as well.

If this process is reversed it is a humidification process. Sprays atomize water into the airstream to add moisture without affecting the dry bulb temperature. The latent heat equation can be used to determine how much heat energy must be added to convert the liquid water into water vapor without changing the temperature.

The humidification process is a typical air conditioning process, however, it is difficult to humidify without either cooling or heating the air as well.

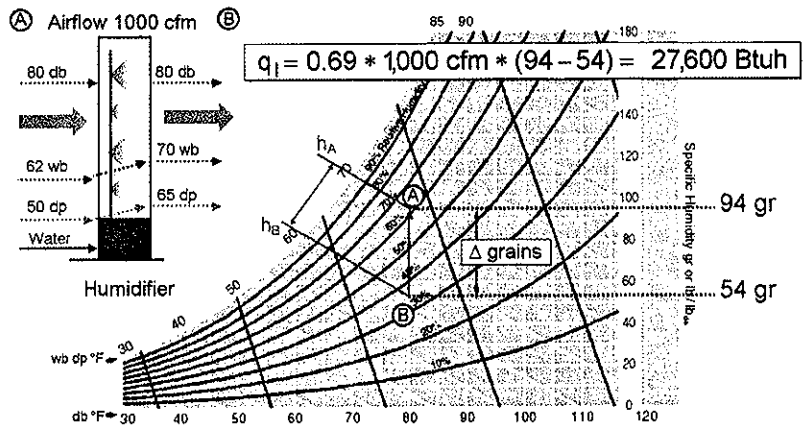


Figure 39

Humidification Process

Air Mixing

What happens when air at two different conditions is mixed? When recirculated room air is mixed with outdoor air, the mixture condition depends upon the conditions of the airstreams as they start out and the amount of each. The mixture's psychrometric coordinates fall on a straight line drawn to connect the state points of the airflows being mixed. If 1000 cfm of return air is mixed with 1000 cfm of outside air, the mixture is equally spaced between the two. If the outside dry bulb is 100° F, and the recirculated air temperature is 80° F, the mixture temperature is 90° F, 50 percent of the difference.

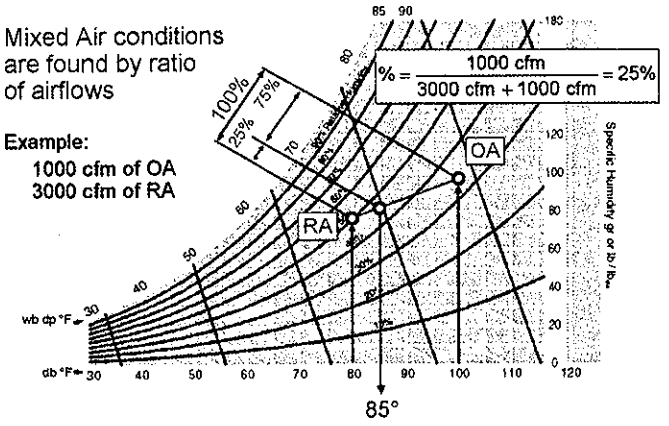


Figure 40

Mixing Return and Outdoor Air

Assume the following situation: 3000 cfm of this recirculated air is mixed with 1000 cfm of outdoor air. The mixture point ends up closer to the recirculated air's point because of the greater amount of recirculated air. Since, for all practical purposes the outdoor air represents 1/4 of the total volume of air, the mixture ends up at 1/4 the linear distance from the recirculated air's state point to the outdoor air's state point. The final temperature works out to be 85° F. Relative humidity, wet bulb temperature, grains of water vapor, and the mixture's dew point all can be found at the state point where 85° F meets the line connecting the return air and the outside air state points.

Finding Room Airflow

Air mixing has an important application: to determine the required quantity of cool, dehumidified supply air that must be delivered to a space to absorb the sensible and latent cooling load components. The supply air mixes with the room air in sufficient quantity to absorb the sensible and latent load. When the space heating and cooling load is calculated, rearranging the sensible heat formula and solving for airflow can be used to determine the required supply airflow. Load calculation programs yield three numbers: the sensible, latent, and total load requirements. The sensible load is used for determining the required room airflow. As long as the dew point is low enough the latent requirements will be met using the sensible load airflow.

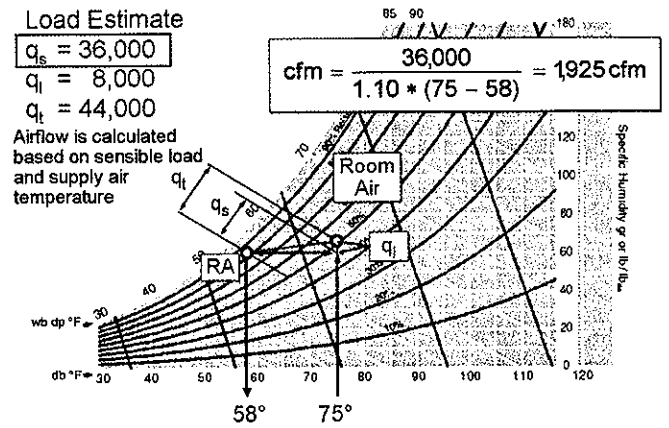


Figure 41

Calculating Room Airflow

An assumption needs to be made as to what the dry bulb temperature of the supply air will be in order to determine the supply airflow. In the example, a 58° F supply air temperature is assumed, which results in a required airflow of 1925 cfm.

Evaporative Cooling

Another process that is used in the air conditioning field is evaporative cooling. This is essentially the same as the wet bulb process. When the air goes through the spray, it loses sensible heat and picks up latent heat, thereby decreasing in dry bulb temperature and increasing in specific humidity. When no heat is added to or removed from the recirculated water, an ***adiabatic process*** is established, which is one where no heat enters or leaves the system. Therefore, the air condition moves up the wet bulb line at a constant enthalpy.

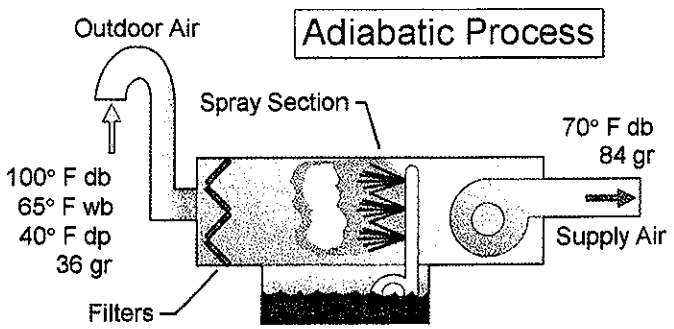


Figure 42

Evaporative Cooling with the Adiabatic Saturation Process

An example of evaporative cooling is the swamp cooler. It provides a crude but low-cost and simple means of using evaporative cooling to condition a space. The swamp cooler works best for arid climates, where substantial moisture can be added to the indoor air without creating excessive inside relative humidity. In addition, some applications require cooling with high humidity, such as the production areas of a textile mill. Overall, the swamp cooler has had limited success in residences because of the high humidity it produces, with the accompanying odor and building damage caused by mildew and mold growth.

The example shown follows the adiabatic saturation process. The entering air exchanges sensible heat for an equal amount of latent heat as it evaporates water sprayed into the airstream. As a result, the dry bulb of the air drops substantially, from 100° F to 70° F, as sensible heat is removed. However, the latent heat added to the air increases the moisture content substantially, from about 37 to 84 grains per pound of dry air. The distance the swamp cooler takes the entering air up the wet bulb line depends on the saturation efficiency of the spray section. In the example shown, it is 85.7 percent [(100° F - 70° F) / (100° F - 65° F)]. The greater the saturation efficiency, the lower

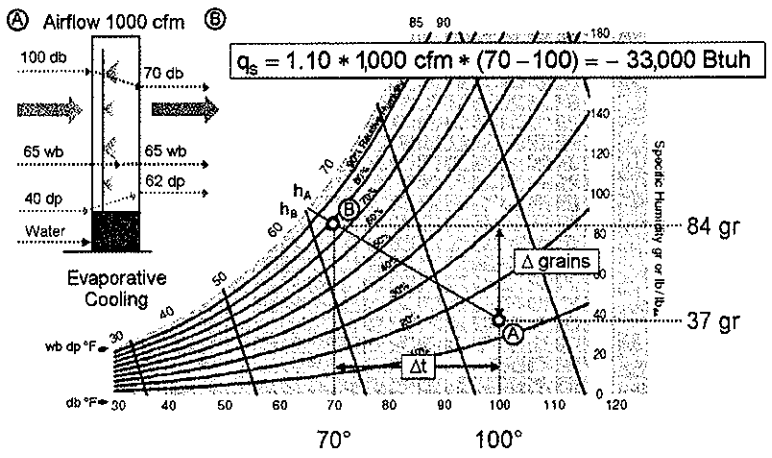


Figure 43

Evaporative Cooling Process

the leaving air dry bulb temperature, increasing sensible cooling capacity. Greater saturation efficiency also raises the leaving air specific humidity, increasing the latent cooling load added to the space. Since no heat is added or subtracted in the total process, the sensible heat loss is equal to the latent heat gain.

Cooling with Dehumidification

The sensible cooling process combined with the dehumidification is the process normally associated with air conditioning. This process is represented by diagonal movement on the chart, down and to the left. Both sensible heat and latent heat decrease. Dry bulb, wet bulb, dew point, specific humidity and enthalpy all decrease.

In this example, air at 80° F and 67° F enters a coil, which has a surface temperature below 47° F. As the air passes through the coil, the cold surface decreases the dry bulb temperature to 55° F. As the air reaches 100 percent saturation, the water vapor in the air condenses. The leaving air is at 51° F wet bulb and at 47° F dew point.

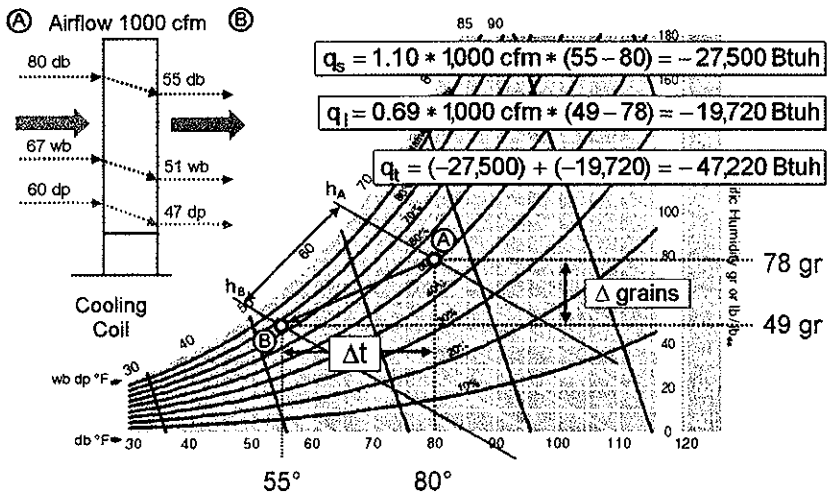


Figure 44

Cooling and Dehumidification Process

Both sensible and latent heat energy need to be removed. The sensible and latent heat formulas can be used to compute the total heat removal necessary. In this example, it required 47,220 Btuh of heat removal by the cooling coil for this cooling process, about a 4-ton unit.

An example of this would be an air conditioning coil, which reduces both the temperature and the moisture of the air passing through it.

Cooling Coils and the Bypass Factor

In order to understand the process of cooling and dehumidification it is necessary to understand cooling coils. Air cooling coils are multiple rows of copper tubes passing through either aluminum or copper fins. Performance is dependent on characteristics of the coil and the air passing through it. One important characteristic is the face area, which is the finned area length multiplied by height through which air flows. The coil face velocity is then the airflow through the coil divided by the face area. The other characteristics of the coil that influence performance are the number of rows of tubes in the airflow direction, the number of fins (fins/in.), and the temperature of the cooling fluid in the coil.

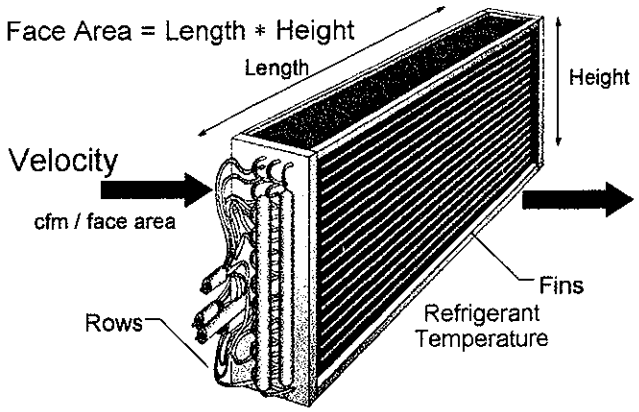


Figure 45

Characteristics of Cooling Coils

The mixing idea can be used to show how a cooling coil works. The figure illustrates one type of coil used for cooling and dehumidifying. Some of the air hits the tubes and some of it goes right through without hitting anything. The part that goes through freely is referred to as the bypass air, the remainder is the contact air. Let us assume that air enters the coil at 80° F dry bulb and 67° F wet bulb and that the coil surface temperature is 50° F. The air that hits the surface of the coil ends up saturated at a temperature of 50° F. The bypassed air is the same as when it started. After passing through the first row of tubes, the airstream is a mixture of bypassed and saturated conditions. If the bypass factor is 2/3 from this one-row coil, then the mixture is at 70° F, which is 2/3 the distance from the 50° F point to the 80° F point. If another row of cooling tubes is added, then less air bypasses the coil tubes. The bypass factor for the two-row coil might be close to 1/2. Air leaving the coil in this situation will be about 65° F. If a condition closer to saturation is required, more rows of tubes can be added. The name used for the coil's final average surface temperature is apparatus dew point. In the above case, the apparatus dew point is 50° F.

It is apparent that the number of rows and the temperature of the coil will change the coil performance by allowing the air to contact more surface area or a colder surface. The figure illustrates performance of a coil with constant air velocity and multiple rows ranging from 2 to 6 rows deep. It also has refrigerant temperatures of 40° F, 45° F, and 50° F. The more rows there are, the closer the coil comes to the saturation line, and the colder the refrigerant temperature the closer to saturation and with a lower leaving dew point temperature.

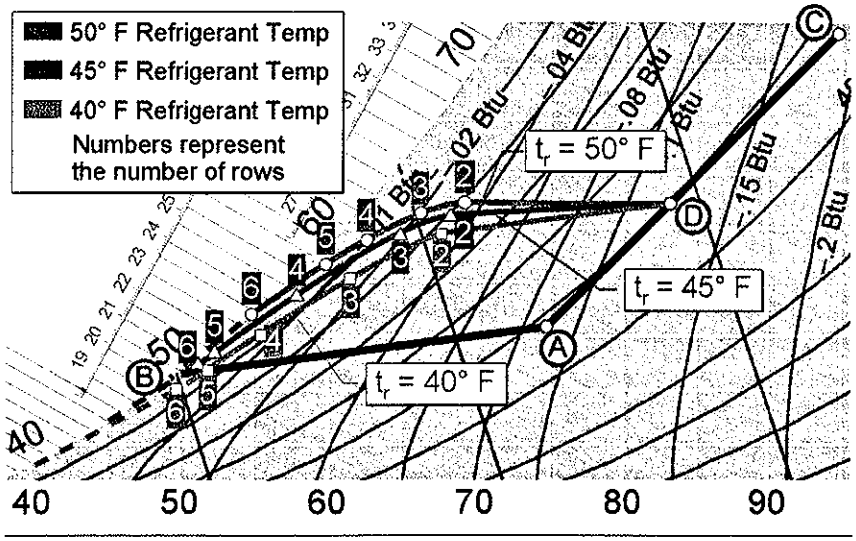


Figure 47

Cooling coil performance, varying rows and refrigerant temperature

The overall bypass factor for the complete cooling coil can be determined from the entering air conditions, leaving air conditions and the average surface temperature. In the example shown in the figure, the leaving air has a dry bulb temperature of 56° F. The overall bypass factor works out to be 0.20. The bypass factor for any coil depends upon the coil construction: that is, the number of tubes, size (face area), number of fins, and the tube and fin spacing.

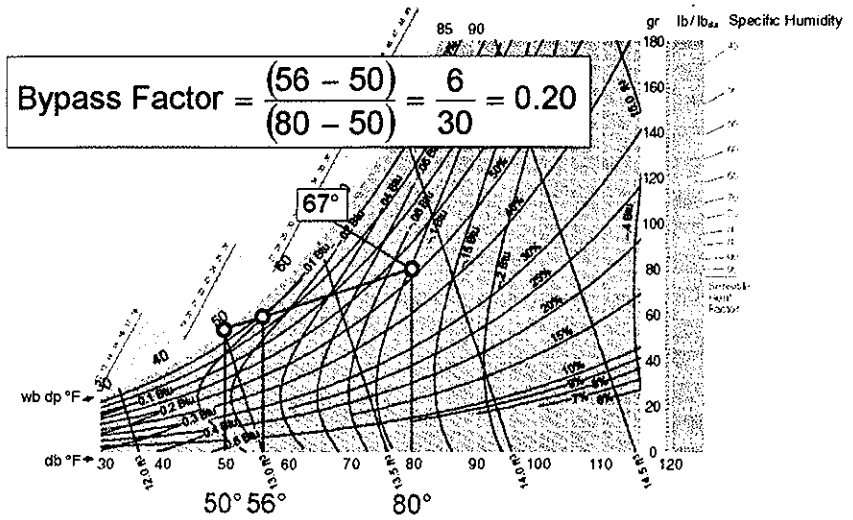


Figure 46

The bypass factor indicates coil performance.

One particular type of cooling coil shows the bypass values tabulated. Notice that each row added makes a smaller and smaller change in the bypass factor. Economically, it means that the sixth row of tubes in the coil is not as valuable as the second, third, or even fifth row.

ROWS	BYPASS FACTOR
2	0.31
3	0.18
4	0.10
5	0.06
6	0.03

Figure 48

Rows of Tubes and Bypass Factor

To lower the bypass factor:

- Increase the number of rows of coil
- Increase the number of fins
- Decrease the refrigerant temperature
- Decrease the velocity

Another condition, affecting the bypass factor is the velocity of the air through the coil. This is shown in the table by some typical bypass factors for various velocities. It can be seen that if smaller quantities of air are used with any one coil, the velocity and consequently the bypass factor is reduced. So, for a given airflow (cfm), the larger the coil, the lower the bypass factor.

AIR VELOCITY	BYPASS FACTOR
300 fpm	0.11
400 fpm	0.14
500 fpm	0.18
600 fpm	0.20

Figure 49

Air Velocity and Bypass Factor

The final characteristic of coil construction that influences bypass factor is the number of fins. Fin surface on a tube act to increase the effective area of the tube, increasing the heat transfer effectiveness. In comfort cooling coils typical fin spacing ranges from 8 to 14 fins per inch of tube. As shown in the table the greater the fins per inch, the lower the bypass factor. Since cooling coils are a wetted surface, water is condensing on and running over the fin surface, the coil fin spacing above 14 fins results in poor water drainage and possible water blowing off the fin surface and into the ductwork.

FINS PER INCH	BYPASS FACTOR
8	0.31
12	0.18
14	0.03

LOWER BYPASS FACTORS RESULT FROM:

- **Larger number of rows**
- **Lower air velocity**
- **More fins**

Figure 50

Fin Spacing and Bypass Factor

Different types of equipment have different bypass factors. In some equipment the system designer has choices as to the rows, fins, or face area and in others, the designer of the equipment has made the decision. If the rows, fins and face area are locked in for a piece of equipment the only options left for the system designer are to change the refrigerant temperature or the velocity (airflow). The figure illustrates typical ranges of bypass factor (BF) for typical air conditioning products.

- Packaged Units to 20 Tons
  - Rows 2 to 4
  - BF 0.18 to 0.07
- Packaged Units over 20 Tons
  - Rows 3 to 6
  - BF 0.32 to 0.03
- Packaged Air Handlers
  - Rows 3 or 4
  - BF 0.12 to 0.03
- Air Handlers
  - Rows 3 to 10
  - BF 0.12 to 0.002

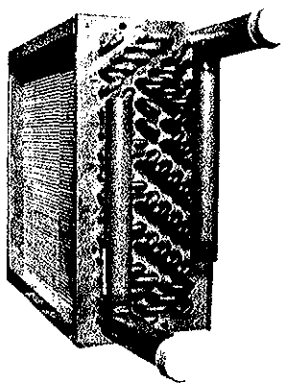


Figure 51

Typical Equipment Bypass Factors

How important is the bypass factor? Should it be high or low? There is no easy answer. Remember that a low coil bypass factor means a low air temperature leaving the coil.

The figure shows the impact of lower temperature supply air going to the room to pick up heat and water vapor, very much as a conveyor belt would do. For a 75° F room temperature, compare the heat absorbing capacity of the supply air at 55° F with air at 50° F.

The sensible heat pick up depends on the temperature difference, so the 50° F air with a 25° F difference can do a greater job than the 55° F air with only a 20° F difference. This is actually 25 percent greater, which means that it would take about 25 percent less air at 50° F to do the same job. Of course, this lower temperature obtained with a lower bypass factor would be desirable, for it would mean the possibility of smaller ducts to carry the air and a smaller fan and fan motor. Each would reduce the cost. However, there are some disadvantages too. To obtain the lower supply conditions may require the use of a larger cooling coil that would increase the initial cost. In addition, it may not be feasible to supply air at 50° F into a small room or office without causing discomfort. The limit of supply conditions depends upon how the air is brought in and the proximity of people to the outlets. For the most common applications of comfort air conditioning, on packaged products, cooling coils are three or four-row coils with bypass factors of 0.12 to 0.07.

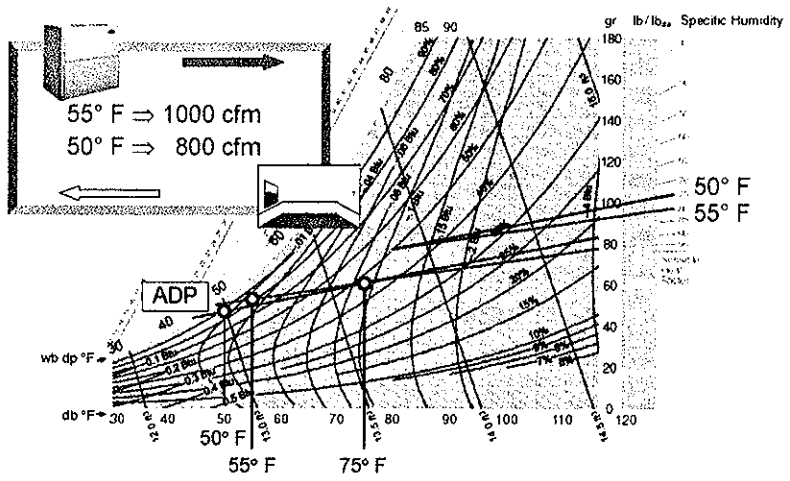


Figure 52

Example of Lower Supply Temperatures

Evaporative Cooling and Humidity Control

Evaporative cooling, as discussed previously, uses recirculating water sprays to saturate the air. We will elaborate on this principle in light of the knowledge we have acquired so far.

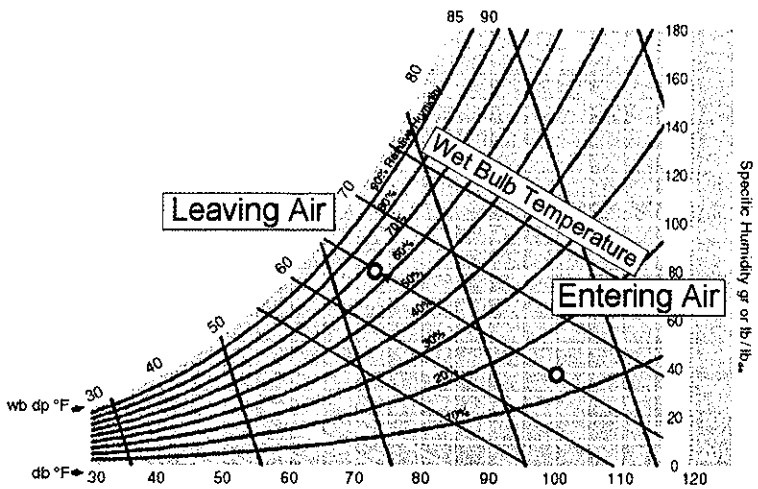


Figure 53

Evaporative Cooling Process

Assume that the temperature of the spray water and the leaving air is the same as the wet bulb temperature of the entering air. The air is cooled and humidified and becomes saturated at a temperature equal to the entering wet bulb. Figure 53 shows the way evaporative cooling appears on the psychrometric chart. The process takes place along the wet bulb line of the entering air and approaches the saturation line. The sensible heat given up is exactly equal to the latent heat required to saturate the air with moisture. If a continuous supply of spray water is available at a temperature below the dew point of the entering air, the air is cooled and dehumidified by the spray water. One way the spray water might be cooled below the dew point is by using a water chiller in a refrigeration system. Another method uses a cooling coil with recirculating water sprays. The water sprays improve the performance of the cooling coil during summer operation and provide close control of humidity as well as temperature. This process can be reversed in winter when it is desirable to heat and humidify the air. In this case, heat is added to the spray water to keep the wet bulb temperature of the leaving air above that of the entering air. The heated spray water is cooled, releasing heat and humidifying simultaneously.

A cooling tower acts as an evaporative cooler when the compression equipment is cycled off and there is no heat added to the condenser water loop by the condenser. Then the condenser water temperature entering and leaving the cooling tower will equalize, as shown here at 85° F. The tower will cool and saturate the air flowing through it just like the swamp cooler. In fact, under these zero-load conditions, with the condenser pump running, the psychrometric plot looks just the same as the swamp cooler.

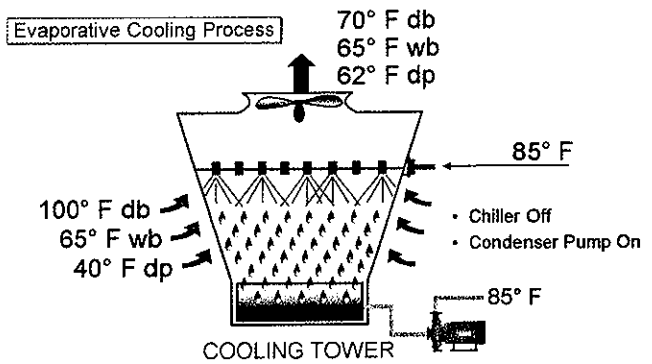


Figure 54

Cooling Tower – No Load

When operating with the compression equipment running, the cooling tower functions similar to an evaporative cooler with heat added to the spray water. The heat is added by the mechanical refrigeration system via the condenser. For example, when the outside air temperature is 100° F db and 65°F wb and the condenser water enters the tower at 95° F, a reasonable leaving air condition is 89° F db and 85° F wb. To accomplish this, the air passing through the tower has been greatly humidified, increasing in absolute humidity from 36 to 178 grains per pound of dry air. The outdoor air has also been slightly cooled, from 100° F to 89° F. At less than peak cooling conditions, as outside air dry bulb temperature drops, the outdoor air may increase somewhat in temperature rather than decreasing.

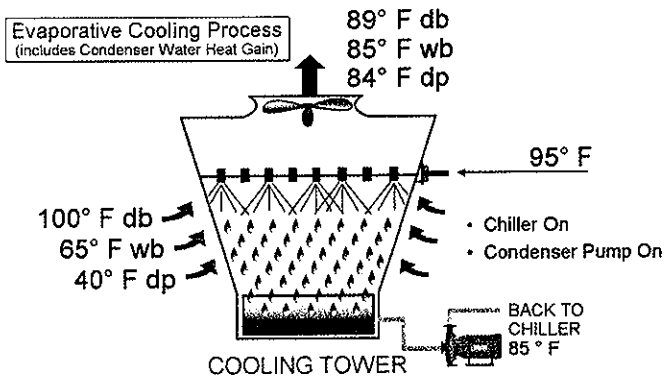


Figure 55

Cooling Tower – Peak Load

Heating and Humidification

The heating and humidification process is represented on the psychrometric chart as a diagonal line, moving up and to the right. Both the sensible heat and latent heat are increased. Dry bulb, wet bulb, dew point, specific humidity, and enthalpy all increase. Relative humidity may hold steady, decrease, or increase, depending on the amount of humidity added.

Heating and humidification is commonly practiced in comfort applications located in cold winter climates, particularly where outdoor ventilation air is introduced. At the air handling unit, a heat exchanger is combined with a pad, steam, or atomizing humidifier to achieve the desired level of humidification.

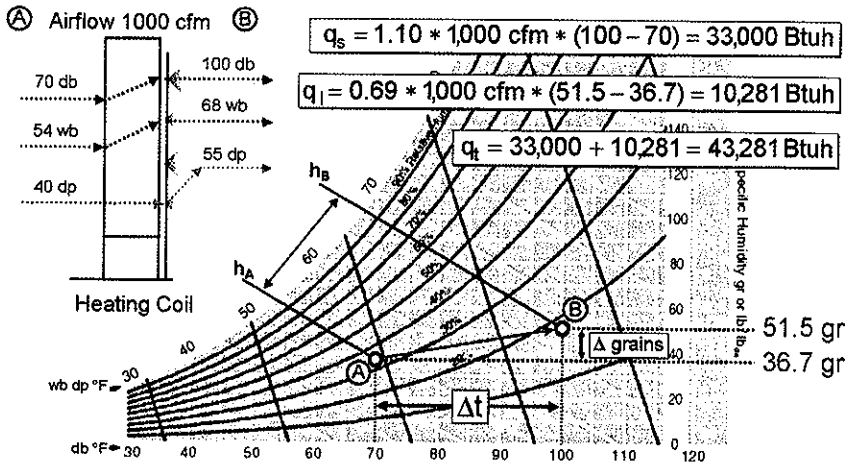


Figure 56

Heating and Humidification Process

A heating and humidification process is possible by use of hot water spray alone, if the water is hot enough. However, with substantial heating load this usually proves impractical.

Heating and Dehumidification

Heating and dehumidification, or sorbent dehumidification, is represented by diagonal movement on the chart, down and to the right. Latent heat is removed in exchange for a sensible heat addition. Theoretically, the process is adiabatic (constant enthalpy) but, in actual practice, the enthalpy climbs slightly.

Sorbent dehumidifiers are installed in the central air handling unit, and contain either a liquid absorbent, or a solid adsorbent, which is

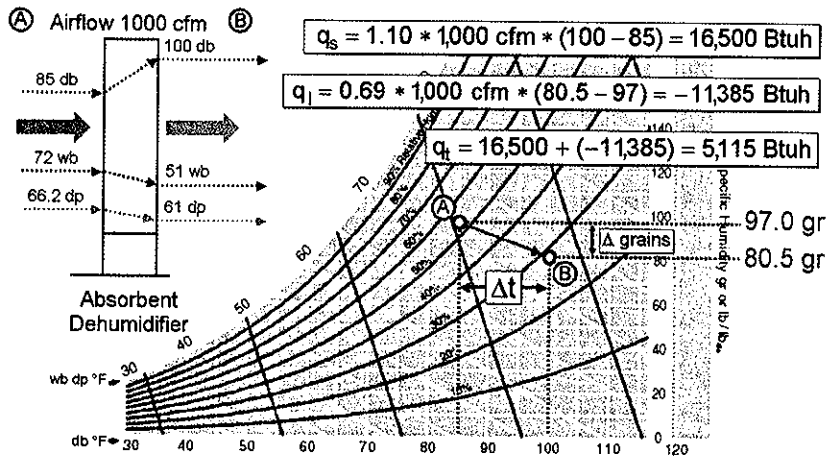


Figure 57

Heating and Dehumidification Process

exposed to the airstream. As the sorption process proceeds, the moisture in the air combines with the absorbent or adsorbent, condensing water from the air. As water is condensed, the latent heat of condensation is liberated, increasing the temperature of the airstream and the sorbent material.

The principles and processes discussed in the preceding two sections have identified how to find the properties of air and how the heat and moisture content change during air conditioning processes. These processes are all applied in products and applications regularly used in comfort air conditioning.

The principles of psychrometrics can be applied in another way. Temperature differences can be used when deciding whether to insulate ducts or whether to use more supply air. If 1000 cubic feet of air per minute at 55° F dry bulb temperature is needed to keep a room at 75° F, how much air is needed if the air temperature goes up to 57° F in an uninsulated duct before reaching the room?

The air has lost 2° F of the original 20° F temperature difference required to handle the sensible heat. This would indicate that 10 percent more air is needed and the decision is whether to use 1100 cfm or to insulate the duct.

Process Chart

Until now, processes have been dealt with as if each process happened independently. This concept is useful in evaluating the requirements of each piece of equipment. However, in an actual air conditioning application, the processes are part of a system and several processes are combined. In fact, the entire air conditioning process within a room from the heat Absorbed from the space, to the air delivered to the room, returned to the air conditioner, and then supplied back to the space is a system process.

It may be helpful to think of the process chart as following a molecule of air on its journey through the system. The process chart tracks the changes in state point conditions that occur in the air molecule as it undergoes each of the processes in the air conditioning system.

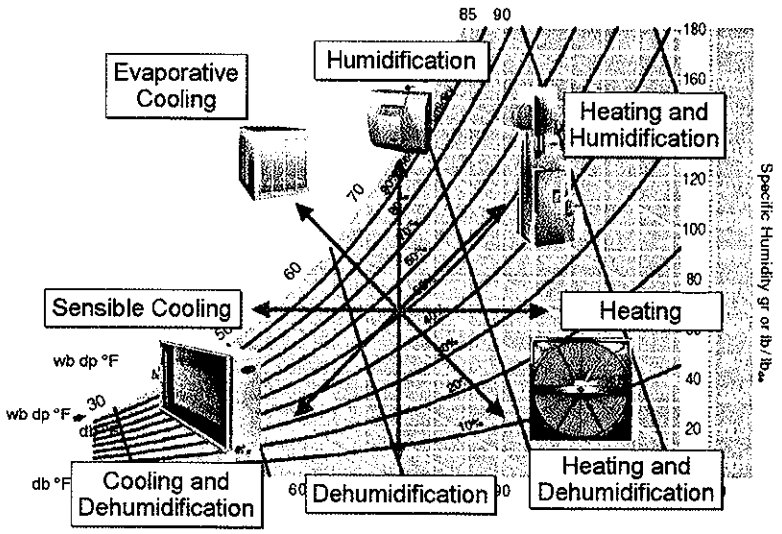


Figure 58

Process lines represent typical types of equipment.

It is advantageous to visualize this entire system of processes with a schematic diagram of the system and a system plot on a psychrometric chart. This diagram is sometimes referred to as an "H" diagram. This diagram, in conjunction with a system plot on the psychrometric chart, will be used in the next two modules to evaluate system performance.

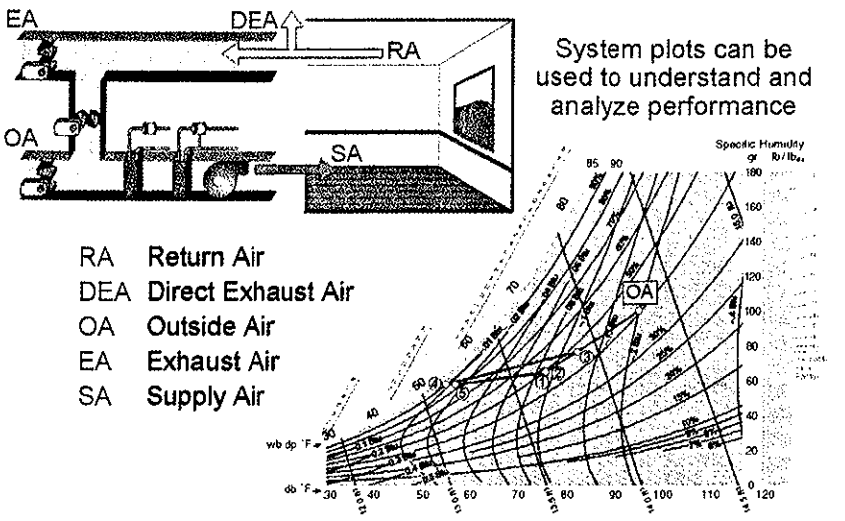


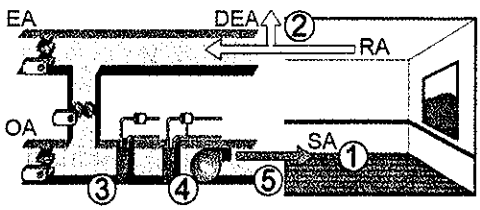
Figure 59

Visualize systems with an "H" diagram and a psychometric chart.

To see how processes work as a system, let's evaluate the basic room conditioning process. The air cycle of most commercial air conditioning systems has five process steps.

Starting in the room, a room control condition is generally assumed – normally something like 75° F, 50 percent rh. Start by plotting this state point from the diagram, "1," on the psychrometric chart. The required airflow is calculated as described, from the load estimate and the assumed supplied air temperature. The supply air absorbs the space sensible and latent heat loads in a heating and humidification process.

Air is then returned from the room to the air handler. As the air passes through the ductwork, it may pick up some heat as it passes through areas where the temperature is above return air temperature. Notice this is all-sensible gain and the specific humidity is unchanged. In this example, we increase it by 1° F. In some cases, a return air fan may be used and the heat from the fan will increase the return air temperature as well. This is state point "2" on the diagram and the point is plotted on the psychrometric chart and a process line, sensible heating, connects point "1" to point "2."



1. Air absorbs room load
2. Remainder returns to AHU
3. OA/RA mix in AHU
4. AHU produces cool air
5. Cool air passes through supply duct and air terminal or diffuser and mixes with room air

DEA Some air exhausted directly (locally), some air exfiltrates
EA Some RA exhausted at/near AHU
OA Outdoor air brought in for ventilation

Figure 60

The complete air cycle is shown on an H diagram.

Outdoor air is required for ventilation of the space and it is common practice in air conditioning systems to mix the return air and outdoor air as they enter the air handler. A portion of the return air is exhausted so that the return air and ventilation air equal 100 percent of the required airflow. In this case, we have 20 percent of the airflow that must be outdoor air to provide ventilation. The outdoor air condition can be plotted, state point "OA."

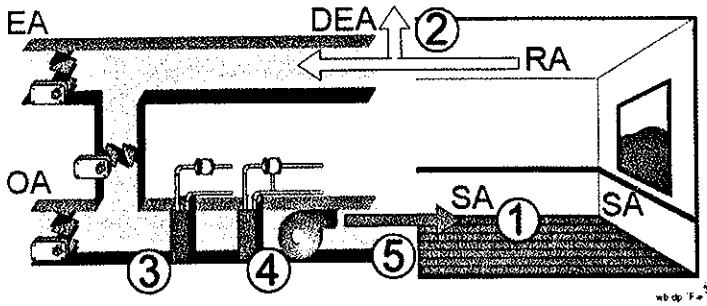
For this example, the outside air condition is 95° F dry bulb and 76° F wet bulb. Using the mixing equations, we can determine the condition of the mixed air, state point "3." This process results in heating and humidification of the return airstream.

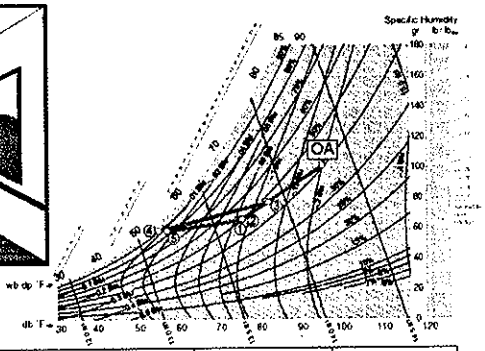
Next, a cooling coil cools the air. If the ADP and bypass factor of the equipment are assumed the condition of the air leaving the coiling coil is determined. This is the cooling and dehumidification process. This occurs at state point "4" on our system plot.

Air then passes through a fan, at state point "5," and the heat from the fan increases the temperature, once again, this is a sensible heating process.

The air is again supplied to the space and it absorbs the heat and moisture that are added to the air by people, lights, process, and solar and transmission gains.

The resulting conditions are back at the room condition state point "1."





Dry Bulb (°F)	Airflow	Dry Bulb (°F)	Wet Bulb (°F)	Rel. Humidity (%)	Humidity Ratio (gr/lb)	Enthalpy (Btu/lb)	Dew Point (°F)
Outdoor Air	600	90.4	72.8	43.3	93.35	36.38	65.1
Room Air	2658	75.0	62.5	50.0	64.92	28.15	55.1
Return Air	2058	78.3	63.7	44.8	64.92	28.95	55.1
Mixed Air	2658	81.0	65.9	45.0	71.34	30.63	57.7
Coil	2658	57.3	56.1	93.0	65.37	23.90	55.3
Supply	2658	58.0	56.4	90.7	65.37	24.07	55.3
Room	2658	75.0	62.5	50.0	64.92	28.15	55.1

Figure 61

Complete System Plot

This combination of an H diagram and a psychrometric chart system plot can be a powerful tool to evaluate system performance. As is evident from this discussion many assumptions about conditions at state points in the system are made based on the system configuration and capability. In the next modules, we use this approach to describe how changes in these characteristics will influence the system operation and conditions.

Summary

This module explained how atmospheric air is a mixture of gases, most importantly a compound mixture of dry air and water vapor, and how a graph, the psychrometric chart, can be used to determine the properties of the mixture. The module also described how psychrometrics is used to determine the air properties, load, and flow requirements of eight basic air conditioning processes. This information is a good start to understanding psychrometric calculations used in load estimating and equipment selection.

The next module develops further how to apply processes together into systems. If you wish to delve deeper into the development of the formula and the psychrometric chart, refer to the fourth module, Psychometrics, Level 4: Theory.

The principles discussed in this TDP module have many practical applications in the air conditioning industry. Review the five practical applications of psychrometrics presented previously, you should now be able to apply psychrometrics to all these situations. The second work session that follows is a good test of your grasp of the introductory concepts of psychrometrics. Psychrometrics is the backbone of air conditioning, and a thorough knowledge of the psychrometric chart is useful for efficient and economical air conditioning design.

Work Session 1

1. Using your psychrometric chart, find the proper values needed to fill in the blank spaces.

	db	wb	% rh	dp	W
A	75	65			
B	75		40		
C	75				80
D		65		55	
E		65	30		
F			30	55	

W = specific humidity, lb/lb of dry air

2. An air duct having a surface temperature of 60° F passes through a space at 90° F db and 75 wb.

   a. Will the duct sweat? Yes _____ No _____

   b. How do you explain this? _____

3. Air at 95° F db and 104 grains of moisture enters a saturator as shown on page 10 in the Building and Psychrometric Chart Section. The saturator is 100% effective. At what dry bulb and wet bulb temperature will the air leave the saturator? What will be its relative humidity?

   _____

4. If a house is maintained at 70° F db and 30 percent rh when the outdoor air temperature is +25° F, is there any need for a vapor barrier in the wall?

   _____

5. On a summer day at 7 a.m. the conditions outside are 70° F db and 80 percent rh. In mid-afternoon the outdoor temperature is 90° F db. If there has been no rain, what is the relative humidity when the db is 90° F? _____

6. The statement is made that the amount of water vapor needed to saturate a pound of air increases with the temperature of the air. How could you demonstrate this with the psychrometric chart?

   _____

   _____

   _____

7. The vapor in an air vapor mixture is saturated and there is 78 grains of moisture present.

   What is the db temperature? _____° F

   What is the wb temperature? _____° F

   What is the dp temperature? _____° F

Work Session 2

1. Air at 30° F db and 80 percent rh is sensibly heated to 75° F db by passing it over a heating coil. Show the process on a psychrometric chart and fill in the blank spaces below:

	db	wb	% rh	dp
Air at	30		80	
Heated to	75			

2. Air at 95° F db and 75° F wb is sensibly cooled to 80° F db by passing it over a cooling coil. Show the process on a psychrometric chart and fill in the blank spaces in the table below:

	db	wb	% rh	dp
Air at	95	75		
Cooled to	80			

3. Air at 80° F db and 50 percent rh is cooled and dehumidified to 50° F and 100 percent rh. How much sensible heat and latent heat is removed from 1000 cfm of this air?

   Sensible Heat Removed =1.10 *cfm * temperature change

   Latent Heat Removed =0.69 * cfm * grains of moisture removed

4. If 500 cfm of outdoor air at 96° F db and 76° F wb is mixed with 1500 cfm of return air at 80° F db and 50 percent rh, find the following properties of the mixture:

   a. Dry bulb _____ ° F
   b. Wet bulb _____ ° F
   c. Dew point _____ ° F
   d. Specific humidity _____ grains/lb.

5. Should the humidifier for a warm air furnace be located in the return air duct or in the warm air plenum or supply duct?

   Return Duct_____

   Supply Duct_____

   Explain why.

   _____

   _____

6. Air at 80° F db and 50 percent rh passes through a coil that has a bypass factor of 0.25 and is operating at 56° F apparatus dew point temperature. What will be the db and wb temperature of the air leaving the coil?

   db = _____ ° F wb = _____ ° F

7. What is the volume of one pound of dry air plus water vapor if its conditions are 95° F db and 75° F wb?

   v = _____ ft^3/lb dry air

8. Find the enthalpy of air whose dry bulb temperature is 76° F with 60 grains of moisture.
   _____ Btu/lb dry air

9. A room is maintained at 75° F db and 50 percent rh by air supplied from a cooling and dehumidifying coil whose leaving air temperature is 55° F db and 53° F wb. Find the sensible heat factor line along which the supply air is warming up. What percentage of the room load is sensible heat and what percentage is latent heat?

   SHF = _____

   % Sensible Heat = _____

   % Latent Heat = _____

Appendix

List of Symbols and Abbreviations

Symbols

cfm_{ba}	cfm of bypassed air, ft^3/m
cfm_{da}	cfm of dehumidified air, ft^3/m
cfm_{oa}	cfm of outdoor air, ft^3/m
cfm_{ra}	cfm of return air, ft^3/m
cfm_{sa}	cfm of supply air, ft^3/m
c_p	specific heat at constant pressure, Btu/lb * °F
c_{pa}	specific heat at constant pressure, air Btu/lb * °F
c_{pg}	specific heat at constant pressure, water Btu/lb * ° F
d	enthalpy deviation, Btu/lb
ρ	density, lb/ft^3
h_a	enthalpy of air, Btu/lb
h_{ADP}	enthalpy at ADP, Btu/lb
h_{ea}	entering air enthalpy, Btu/lb
h_{es}	enthalpy at effective surface temperature, Btu/lb
h_f	enthalpy of saturated liquid, Btu/lb
h_{fg}	enthalpy of evaporation or condensation, Btu/lb
h_g	enthalpy of saturated water vapor, Btu/lb
h_{la}	leaving air enthalpy, Btu/lb
h_{ma}	mixed air enthalpy, Btu/lb
h_{oa}	outdoor air enthalpy, Btu/lb
h_{rm}	room air enthalpy, Btu/lb
h_s	enthalpy of saturated air at dry bulb temperature, t″, Btu/lb
h_s'	enthalpy of saturated air at wet bulb temperature, t′, Btu/lb
h_{sa}	supply air enthalpy, Btu/lb
p	barometric pressure, psia, psfa, in. Hg
p_a	pressure of dry air, and partial pressure of dry air, psia
p_g	partial pressure of water vapor corresponding to the dry bulb temperature, t, psia
p_g'	partial pressure of water vapor corresponding to the dew point temperature, t′, psia
p_g''	partial pressure of water vapor corresponding to the wet bulb temperature, t″, psia
q	heat added or removed, Btuh
q_l	latent heat added or removed, Btuh
q_s	sensible heat added or removed, Btuh
q_t	total heat added or removed, Btuh
R	universal gas constant, 1545.32 $(lb_f/ft^2) * ft^3/(lb_{mole} * °R)$
R_a	gas constant for dry air
θ	relative humidity, %
R_w	gas constant for water vapor
s	entropy, $Btu/lb_{da} * °F$
T	absolute temperature ° R (t + 460° F)
t	dry bulb temperature, °F
t′	wet bulb temperature, °F
t″	dew point temperature, °F
t_{ADP}	temperature ADP, °F
t_{edb}	temperature entering dry bulb, °F
t_{es}	temperature effective surface, °F
t_{ew}	temperature entering water, °F
t_{ewb}	temperature entering wet bulb, °F
t_{ldb}	temperature leaving dry bulb, °F
t_{lw}	temperature leaving water, °F
t_{lwb}	temperature leaving wet bulb, °F
t_{ma}	temperature mixed outdoor and return air dry bulb, °F
t_{oa}	temperature outdoor air dry bulb, °F
t_{rm}	temperature room air dry bulb, °F
t_{sa}	temperature supply air, °F
v_a	specific volume of air ft^3/lb
v_g	specific volume of air, water vapor, ft^3/lb
v_w	specific volume of water, ft^3/lb

W	specific humidity, moisture content, lb/lb_{da} or gr
w	weight (mass), lb
W_{ADP}	specific humidity at ADP, moisture content, lb/lb_{da} or gr
W_{ea}	specific humidity of entering air, moisture content, lb/lb_{da} or gr
W_{es}	specific humidity at effective surface temperature, moisture content, lb/lb_{da} or gr
W_{la}	specific humidity of leaving air, moisture content, lb/lb_{da} or gr
W_{ma}	specific humidity of mixed air, moisture content, lb/lb_{da} or gr
W_{oa}	specific humidity of outdoor air, moisture content, lb/lb_{da} or gr
W_{rm}	specific humidity of room air, moisture content, lb/lb_{da} or gr
W_s	moisture content saturated at the wet bulb temperature, t, lb/lb_{da} or gr
W_s'	moisture content saturated at the dry bulb temperature, t', lb/lb_{da} or gr
W_{sa}	specific humidity of supply air, moisture content, lb/lb_{da} or gr
Δgr	moisture content difference, gr
Δh	enthalpy difference, Btu/lb
Δt	temperature difference, °F

Superscripts

$( )'$	values corresponding to the wet bulb temperature, t'
$( )''$	values corresponding to the dew point temperature, t''

Subscripts

a	dry air
ba	bypassed air conditions
da	dehumidified air conditions
ea	entering air conditions
es	effective surface
f	liquid water
fg	vaporization
g	saturated water
l	latent heat (used with q)
la	leaving air conditions
ma	mixed air conditions
oa	outdoor air conditions
p	constant pressure
r	room conditions
ra	return air conditions
s	saturated (used with h, p, t, W
s	sensible heat (used with q)
sa	supply air conditions
t	total heat (used with q)

Units

Btu	British thermal units
Btuh	British thermal units per hour
cfh	cubic feet per hour
cfm	cubic feet per minute
fpm	feet per minute
gpm	gallons per minute
gr	grains of moisture per pound of dry air
in. Hg	inches of mercury
lb	pounds
lb/lb_{da}	pounds of moisture per pound of dry air
psfa	pounds per square foot absolute
psia	pounds per square inch absolute

Abbreviations

ADP	apparatus dewpoint
BF	bypass factor
CF	contact factor
db	dry bulb
dp	dew point
ERLH	effective room latent heat, includes bypassed air latent
ERSH	effective room sensible heat, includes bypassed air sensible
ERTH	effective room total heat, included bypassed air sensible and latent
ESHF	effective room sensible heat factor
F	Fahrenheit degrees
R	Rankine degrees
rh	relative humidity
RLH	room latent heat
RSH	room sensible heat
RSHF	room sensible heat factor
RTH	room total heat
Sat. Eff.	saturation efficiency
SHF	sensible heat factor
wb	wet bulb

Thermodynamic Properties of Water At Saturation: U.S. Units

TEMP °F	ABSOLUTE PRESSURE psi	ABSOLUTE PRESSURE in. Hg	SPECIFIC VOLUME (ft^3/lb) Sat. Liquid v_f	SPECIFIC VOLUME (ft^3/lb) Evap. v_{fg}	SPECIFIC VOLUME (ft^3/lb) Sat. Vapor v_g	ENTHALPY (Btu/lb) Sat. Liquid h_f	ENTHALPY (Btu/lb) Evap. h_{fg}	ENTHALPY (Btu/lb) Sat. Vapor h_g	ENTROPY (Btu/lba/°F) Sat. Liquid s_f	ENTROPY (Btu/lba/°F) Evap. s_{fg}	ENTROPY (Btu/lba/°F) Sat. Vapor s_g	TEMP °F
-80	0.000116	0.000236	0.01732	1953234	1953234	-193.50	1219.19	1025.69	-0.4067	3.2112	2.8045	-80
-79	0.000125	0.000254	0.01732	1814052	1814052	-193.11	1219.24	1026.13	-0.4056	3.2028	2.7972	-79
-78	0.000135	0.000275	0.01732	1685445	1685445	-192.71	1219.28	1026.57	-0.4046	3.1946	2.7900	-78
-77	0.000145	0.000296	0.01732	1566663	1566663	-192.31	1219.33	1027.02	-0.4036	3.1864	2.7828	-77
-76	0.000157	0.000319	0.01732	1456752	1456752	-191.92	1219.38	1027.46	-0.4025	3.1782	2.7757	-76
-75	0.000169	0.000344	0.01733	1355059	1355059	-191.52	1219.42	1027.90	-0.4015	3.1700	2.7685	-75
-74	0.000182	0.000371	0.01733	1260977	1260977	-191.12	1219.46	1028.34	-0.4005	3.1620	2.7615	-74
-73	0.000196	0.000399	0.01733	1173848	1173848	-190.72	1219.51	1028.79	-0.3994	3.1538	2.7544	-73
-72	0.000211	0.000430	0.01733	1093149	1093149	-190.32	1219.55	1029.23	-0.3984	3.1459	2.7475	-72
-71	0.000227	0.000463	0.01733	1018381	1018381	-189.92	1219.59	1029.67	-0.3974	3.1379	2.7405	-71
-70	0.000245	0.000498	0.01733	949067	949067	-189.52	1219.63	1030.11	-0.3963	3.1299	2.7336	-70
-69	0.000263	0.000536	0.01733	884803	884803	-189.11	1219.66	1030.55	-0.3953	3.1220	2.7267	-69
-68	0.000283	0.000576	0.01733	825187	825187	-188.71	1219.71	1031.00	-0.3943	3.1142	2.7199	-68
-67	0.000304	0.000619	0.01734	769864	769864	-188.30	1219.74	1031.44	-0.3932	3.1063	2.7131	-67
-66	0.000326	0.000664	0.01734	718508	718508	-187.90	1219.78	1031.88	-0.3922	3.0985	2.7063	-66
-65	0.000350	0.000714	0.01734	670800	670800	-187.49	1219.81	1032.32	-0.3912	3.0908	2.6996	-65
-64	0.000376	0.000766	0.01734	626503	626503	-187.08	1219.85	1032.77	-0.3901	3.0830	2.6929	-64
-63	0.000404	0.000822	0.01734	585316	585316	-186.67	1219.88	1033.21	-0.3891	3.0753	2.6862	-63
-62	0.000433	0.000882	0.01734	547041	547041	-186.26	1219.91	1033.65	-0.3881	3.0677	2.6796	-62
-61	0.000464	0.000945	0.01734	511446	511446	-185.85	1219.94	1034.09	-0.3870	3.0600	2.6730	-61
-60	0.000498	0.001013	0.01734	478317	478317	-185.44	1219.98	1034.54	-0.3860	3.0525	2.6665	-60
-59	0.000533	0.001086	0.01735	447495	447495	-185.03	1220.01	1034.98	-0.3850	3.0450	2.6600	-59
-58	0.000571	0.001163	0.01735	418803	418803	-184.61	1220.03	1035.42	-0.3839	3.0374	2.6535	-58
-57	0.000612	0.001246	0.01735	392068	392068	-184.20	1220.06	1035.86	-0.3829	3.0299	2.6470	-57
-56	0.000655	0.001333	0.01735	367172	367172	-183.78	1220.08	1036.30	-0.3819	3.0225	2.6406	-56
-55	0.000701	0.001427	0.01735	343970	343970	-183.37	1220.12	1036.75	-0.3808	3.0150	2.6342	-55
-54	0.000750	0.001526	0.01735	322336	322336	-182.95	1220.14	1037.19	-0.3798	3.0077	2.6279	-54
-53	0.000802	0.001632	0.01735	302157	302157	-182.53	1220.16	1037.63	-0.3788	3.0004	2.6216	-53
-52	0.000857	0.001745	0.01735	283335	283335	-182.11	1220.18	1038.07	-0.3778	2.9931	2.6153	-52
-51	0.000916	0.001865	0.01736	265773	265773	-181.69	1220.21	1038.52	-0.3767	2.9858	2.6091	-51
-50	0.000979	0.001992	0.01736	249381	249381	-181.27	1220.23	1038.96	-0.3757	2.9786	2.6029	-50
-49	0.001045	0.002128	0.01736	234067	234067	-180.85	1220.25	1039.40	-0.3747	2.9714	2.5967	-49
-48	0.001116	0.002272	0.01736	219766	219766	-180.42	1220.26	1039.84	-0.3736	2.9642	2.5906	-48
-47	0.001191	0.002425	0.01736	206398	206398	-180.00	1220.28	1040.28	-0.3726	2.9570	2.5844	-47
-46	0.001271	0.002587	0.01736	193909	193909	-179.57	1220.30	1040.73	-0.3716	2.9500	2.5784	-46
-45	0.001355	0.002760	0.01736	182231	182231	-179.14	1220.31	1041.17	-0.3705	2.9428	2.5723	-45
-44	0.001445	0.002943	0.01736	171304	171304	-178.72	1220.33	1041.61	-0.3695	2.9358	2.5663	-44
-43	0.001541	0.003137	0.01737	161084	161084	-178.29	1220.34	1042.05	-0.3685	2.9288	2.5603	-43
-42	0.001642	0.003343	0.01737	151518	151518	-177.86	1220.36	1042.50	-0.3675	2.9219	2.5544	-42
-41	0.001749	0.003562	0.01737	142566	142566	-177.43	1220.37	1042.94	-0.3664	2.9149	2.5485	-41
-40	0.001863	0.003793	0.01737	134176	134176	-177.00	1220.38	1043.38	-0.3654	2.9080	2.5426	-40
-39	0.001984	0.004039	0.01737	126322	126322	-176.57	1220.39	1043.82	-0.3644	2.9011	2.5367	-39
-38	0.002111	0.004299	0.01737	118959	118959	-176.13	1220.40	1044.27	-0.3633	2.8942	2.5309	-38
-37	0.002247	0.004575	0.01737	112058	112058	-175.70	1220.41	1044.71	-0.3623	2.8874	2.5251	-37
-36	0.002390	0.004866	0.01738	105592	105592	-175.26	1220.41	1045.15	-0.3613	2.8806	2.5193	-36

Thermodynamic Properties of Water At Saturation: U.S. Units

TEMP °F	ABSOLUTE PRESSURE psi	ABSOLUTE PRESSURE in. Hg	SPECIFIC VOLUME (ft^3/lb) Sat. Liquid v_f	SPECIFIC VOLUME (ft^3/lb) Evap. v_{fg}	SPECIFIC VOLUME (ft^3/lb) Sat. Vapor v_g	ENTHALPY (Btu/lb) Sat. Liquid h_f	ENTHALPY (Btu/lb) Evap. h_{fg}	ENTHALPY (Btu/lb) Sat. Vapor h_g	ENTROPY (Btu/lba/°F) Sat. Liquid s_f	ENTROPY (Btu/lba/°F) Evap. s_{fg}	ENTROPY (Btu/lba/°F) Sat. Vapor s_g	TEMP °F
-35	0.002542	0.005175	0.01738	99522	99522	-174.83	1220.42	1045.59	-0.3603	2.8739	2.5136	-35
-34	0.002702	0.005502	0.01738	93828	93828	-174.39	1220.42	1046.03	-0.3592	2.8670	2.5078	-34
-33	0.002872	0.005848	0.01738	88489	88489	-173.95	1220.43	1046.48	-0.3582	2.8604	2.5022	-33
-32	0.003052	0.006213	0.01738	83474	83474	-173.51	1220.43	1046.92	-0.3572	2.8537	2.4965	-32
-31	0.003242	0.006600	0.01738	78763	78763	-173.07	1220.43	1047.36	-0.3561	2.8470	2.4909	-31
-30	0.003443	0.007009	0.01738	74341	74341	-172.63	1220.43	1047.80	-0.3551	2.8404	2.4853	-30
-29	0.003655	0.007441	0.01738	70187	70187	-172.19	1220.44	1048.25	-0.3541	2.8338	2.4797	-29
-28	0.003879	0.007898	0.01739	66282	66282	-171.74	1220.43	1048.69	-0.3531	2.8273	2.4742	-28
-27	0.004116	0.008380	0.01739	62613	62613	-171.30	1220.43	1049.13	-0.3520	2.8207	2.4687	-27
-26	0.004366	0.008890	0.01739	59161	59161	-170.86	1220.43	1049.57	-0.3510	2.8142	2.4632	-26
-25	0.004630	0.009428	0.01739	55915	55915	-170.41	1220.42	1050.01	-0.3500	2.8077	2.4577	-25
-24	0.004909	0.009995	0.01739	52861	52861	-169.96	1220.42	1050.46	-0.3489	2.8012	2.4523	-24
-23	0.005203	0.010594	0.01739	49986	49986	-169.51	1220.41	1050.90	-0.3479	2.7948	2.4469	-23
-22	0.005514	0.011226	0.01739	47281	47281	-169.07	1220.41	1051.34	-0.3469	2.7884	2.4415	-22
-21	0.005841	0.011892	0.01740	44733	44733	-168.62	1220.40	1051.78	-0.3459	2.7821	2.4362	-21
-20	0.006186	0.012595	0.01740	42333	42333	-168.16	1220.38	1052.22	-0.3448	2.7757	2.4309	-20
-19	0.006550	0.013336	0.01740	40073	40073	-167.71	1220.38	1052.67	-0.3438	2.7694	2.4256	-19
-18	0.006933	0.014117	0.01740	37943	37943	-167.26	1220.37	1053.11	-0.3428	2.7631	2.4203	-18
-17	0.007337	0.014939	0.01740	35934	35934	-166.81	1220.36	1053.55	-0.3418	2.7569	2.4151	-17
-16	0.007763	0.015806	0.01740	34041	34041	-166.35	1220.34	1053.99	-0.3407	2.7505	2.4098	-16
-15	0.008211	0.016718	0.01740	32256	32256	-165.90	1220.33	1054.43	-0.3397	2.7443	2.4046	-15
-14	0.008683	0.017678	0.01741	30572	30572	-165.44	1220.31	1054.87	-0.3387	2.7382	2.3995	-14
-13	0.009179	0.018689	0.01741	28983	28983	-164.98	1220.30	1055.32	-0.3377	2.7320	2.3943	-13
-12	0.009702	0.019753	0.01741	27483	27483	-164.52	1220.28	1055.76	-0.3366	2.7258	2.3892	-12
-11	0.010252	0.020873	0.01741	26067	26067	-164.06	1220.26	1056.20	-0.3356	2.7197	2.3841	-11
-10	0.010830	0.022050	0.01741	24730	24730	-163.60	1220.24	1056.64	-0.3346	2.7137	2.3791	-10
-9	0.011438	0.023288	0.01741	23467	23467	-163.14	1220.22	1057.08	-0.3335	2.7075	2.3740	-9
-8	0.012077	0.024590	0.01741	22274	22274	-162.68	1220.21	1057.53	-0.3325	2.7015	2.3690	-8
-7	0.012750	0.025958	0.01742	21147	21147	-162.21	1220.18	1057.97	-0.3315	2.6955	2.3640	-7
-6	0.013456	0.027396	0.01742	20081	20081	-161.75	1220.16	1058.41	-0.3305	2.6896	2.3591	-6
-5	0.014197	0.028906	0.01742	19074	19074	-161.28	1220.13	1058.85	-0.3294	2.6835	2.3541	-5
-4	0.014977	0.030493	0.01742	18121	18121	-160.82	1220.11	1059.29	-0.3284	2.6776	2.3492	-4
-3	0.015795	0.032159	0.01742	17220	17220	-160.35	1220.08	1059.73	-0.3274	2.6717	2.3443	-3
-2	0.016654	0.033908	0.01742	16367	16367	-159.88	1220.05	1060.17	-0.3264	2.6658	2.3394	-2
-1	0.017556	0.035744	0.01742	15561	15561	-159.41	1220.03	1060.62	-0.3253	2.6599	2.3346	-1
0	0.018502	0.037671	0.01743	14797	14797	-158.94	1220.00	1061.06	-0.3243	2.6541	2.3298	0
1	0.019495	0.039693	0.01743	14073	14073	-158.47	1219.97	1061.50	-0.3233	2.6482	2.3249	1
2	0.020537	0.041813	0.01743	13388	13388	-157.99	1219.93	1061.94	-0.3223	2.6425	2.3202	2
3	0.021629	0.044037	0.01743	12740	12740	-157.52	1219.90	1062.38	-0.3212	2.6366	2.3154	3
4	0.022774	0.046369	0.01743	12125	12125	-157.05	1219.87	1062.82	-0.3202	2.6309	2.3107	4
5	0.023975	0.048813	0.01743	11543	11543	-156.57	1219.83	1063.26	-0.3192	2.6252	2.3060	5
6	0.025233	0.051375	0.01743	10991	10991	-156.09	1219.79	1063.70	-0.3182	2.6195	2.3013	6
7	0.026552	0.054059	0.01744	10468	10468	-155.62	1219.76	1064.14	-0.3171	2.6137	2.2966	7
8	0.027933	0.056872	0.01744	9971	9971	-155.14	1219.72	1064.58	-0.3161	2.6081	2.2920	8
9	0.029379	0.059817	0.01744	9500	9500	-154.66	1219.69	1065.03	-0.3151	2.6024	2.2873	9
10	0.030894	0.062901	0.01744	9054	9054	-154.18	1219.65	1065.47	-0.3141	2.5968	2.2827	10
11	0.032480	0.066131	0.01744	8630	8630	-153.70	1219.61	1065.91	-0.3130	2.5912	2.2782	11
12	0.034140	0.069511	0.01744	8228	8228	-153.21	1219.56	1066.35	-0.3120	2.5856	2.2736	12
13	0.035878	0.073047	0.01745	7846	7846	-152.73	1219.52	1066.79	-0.3110	2.5801	2.2691	13
14	0.037696	0.076748	0.01745	7483	7483	-152.24	1219.47	1067.23	-0.3100	2.5745	2.2645	14

Thermodynamic Properties of Water At Saturation: U.S. Units

TEMP °F	ABSOLUTE PRESSURE psi	ABSOLUTE PRESSURE in. Hg	SPECIFIC VOLUME (ft^3/lb) Sat. Liquid v_f	SPECIFIC VOLUME (ft^3/lb) Evap. v_{fg}	SPECIFIC VOLUME (ft^3/lb) Sat. Vapor v_g	ENTHALPY (Btu/lb) Sat. Liquid h_f	ENTHALPY (Btu/lb) Evap. h_{fg}	ENTHALPY (Btu/lb) Sat. Vapor h_g	ENTROPY (Btu/lba/°F) Sat. Liquid s_f	ENTROPY (Btu/lba/°F) Evap. s_{fg}	ENTROPY (Btu/lba/°F) Sat. Vapor s_g	TEMP °F
15	0.039597	0.080621	0.01745	7139	7139	-151.76	1219.43	1067.67	-0.3089	2.5689	2.2600	15
16	0.041587	0.084671	0.01745	6811	6811	-151.27	1219.38	1068.11	-0.3079	2.5635	2.2556	16
17	0.043666	0.088905	0.01745	6501	6501	-150.78	1219.33	1068.55	-0.3069	2.5580	2.2511	17
18	0.045841	0.093332	0.01745	6205	6205	-150.30	1219.29	1068.99	-0.3059	2.5526	2.2467	18
19	0.048113	0.097960	0.01745	5924	5924	-149.81	1219.24	1069.43	-0.3049	2.5472	2.2423	19
20	0.050489	0.102796	0.01746	5657	5657	-149.32	1219.19	1069.87	-0.3038	2.5417	2.2379	20
21	0.052970	0.107849	0.01746	5404	5404	-148.82	1219.13	1070.31	-0.3028	2.5363	2.2335	21
22	0.055563	0.113128	0.01746	5162	5162	-148.33	1219.08	1070.75	-0.3018	2.5310	2.2292	22
23	0.058271	0.118641	0.01746	4932	4932	-147.84	1219.03	1071.19	-0.3008	2.5256	2.2248	23
24	0.061099	0.124398	0.01746	4714	4714	-147.34	1218.97	1071.63	-0.2997	2.5202	2.2205	24
25	0.064051	0.130408	0.01746	4506	4506	-146.85	1218.92	1072.07	-0.2987	2.5149	2.2162	25
26	0.067133	0.136684	0.01747	4308	4308	-146.35	1218.85	1072.50	-0.2977	2.5096	2.2119	26
27	0.070349	0.143233	0.01747	4119	4119	-145.85	1218.79	1072.94	-0.2967	2.5044	2.2077	27
28	0.073706	0.150066	0.01747	3940	3940	-145.35	1218.73	1073.38	-0.2956	2.4991	2.2035	28
29	0.077207	0.157195	0.01747	3769	3769	-144.85	1218.67	1073.82	-0.2946	2.4938	2.1992	29
30	0.080860	0.164632	0.01747	3606	3606	-144.35	1218.61	1074.26	-0.2936	2.4887	2.1951	30
31	0.084669	0.172387	0.01747	3450	3450	-143.85	1218.55	1074.70	-0.2926	2.4835	2.1909	31
32	0.088640	0.180474	0.01747	3302	3302	-143.35	1218.49	1075.14	-0.2915	2.4782	2.1867	32
32	0.08865	0.18049	0.01602	3302.07	3302.09	-0.02	1075.16	1075.14	0.0000	2.1867	2.1867	32
33	0.09229	0.18791	0.01602	3178.14	3178.16	0.99	1074.59	1075.58	0.0020	2.1812	2.1832	33
34	0.09607	0.19559	0.01602	3059.47	3059.49	2.00	1074.01	1076.01	0.0041	2.1755	2.1796	34
35	0.09998	0.20355	0.01602	2945.66	2945.68	3.00	1073.45	1076.45	0.0061	2.1700	2.1761	35
36	0.10403	0.21180	0.01602	2836.59	2836.61	4.01	1072.88	1076.89	0.0081	2.1645	2.1726	36
37	0.10822	0.22035	0.01602	2732.13	2732.15	5.02	1072.31	1077.33	0.0102	2.1590	2.1692	37
38	0.11257	0.22919	0.01602	2631.87	2631.89	6.02	1071.75	1077.77	0.0122	2.1535	2.1657	38
39	0.11707	0.23835	0.01602	2535.86	2535.88	7.03	1071.18	1078.21	0.0142	2.1481	2.1623	39
40	0.12172	0.24783	0.01602	2443.67	2443.69	8.03	1070.62	1078.65	0.0162	2.1427	2.1589	40
41	0.12654	0.25765	0.01602	2355.22	2355.24	9.04	1070.05	1079.09	0.0182	2.1372	2.1554	41
42	0.13153	0.26780	0.01602	2270.41	2270.43	10.04	1069.48	1079.52	0.0202	2.1319	2.1521	42
43	0.13669	0.27831	0.01602	2189.02	2189.04	11.04	1068.92	1079.96	0.0222	2.1265	2.1487	43
44	0.14203	0.28918	0.01602	2110.92	2110.94	12.05	1068.35	1080.40	0.0242	2.1212	2.1454	44
45	0.14755	0.30042	0.01602	2035.90	2035.92	13.05	1067.79	1080.84	0.0262	2.0978	2.1240	45
46	0.15326	0.31205	0.01602	1963.85	1963.87	14.05	1067.23	1081.28	0.0282	2.1105	2.1387	46
47	0.15917	0.32407	0.01602	1894.71	1894.73	15.06	1066.65	1081.71	0.0302	2.1052	2.1354	47
48	0.16527	0.33650	0.01602	1828.28	1828.30	16.06	1066.09	1082.15	0.0321	2.1000	2.1321	48
49	0.17158	0.34935	0.01602	1764.44	1764.46	17.06	1065.53	1082.59	0.0341	2.0947	2.1288	49
50	0.17811	0.36263	0.01602	1703.18	1703.20	18.06	1064.97	1083.03	0.0361	2.0895	2.1256	50
51	0.18484	0.37635	0.01602	1644.24	1644.26	19.06	1064.40	1083.46	0.0381	2.0843	2.1224	51
52	0.19181	0.39053	0.01603	1587.63	1587.65	20.07	1063.83	1083.90	0.0400	2.0791	2.1191	52
53	0.19900	0.40516	0.01603	1533.22	1533.24	21.07	1063.27	1084.34	0.0420	2.0739	2.1159	53
54	0.20643	0.42029	0.01603	1480.89	1480.91	22.07	1062.70	1084.77	0.0439	2.0689	2.1128	54
55	0.21410	0.43591	0.01603	1430.60	1430.62	23.07	1062.14	1085.21	0.0459	2.0637	2.1096	55
56	0.22202	0.45204	0.01603	1382.19	1382.21	24.07	1061.58	1085.65	0.0478	2.0586	2.1064	56
57	0.23020	0.46869	0.01603	1335.65	1335.67	25.07	1061.01	1086.08	0.0497	2.0536	2.1033	57
58	0.23864	0.48588	0.01603	1290.85	1290.87	26.07	1060.45	1086.52	0.0517	2.0485	2.1002	58
59	0.24735	0.50362	0.01603	1247.76	1247.78	27.07	1059.89	1086.96	0.0536	2.0435	2.0971	59
60	0.25635	0.52192	0.01604	1206.30	1206.32	28.07	1059.32	1087.39	0.0555	2.0385	2.0940	60
61	0.26562	0.54081	0.01604	1166.38	1166.40	29.07	1058.76	1087.83	0.0575	2.0334	2.0909	61
62	0.27519	0.56029	0.01604	1127.93	1127.95	30.07	1058.20	1088.27	0.0594	2.0284	2.0878	62
63	0.28506	0.58039	0.01604	1090.94	1090.96	31.07	1057.63	1088.70	0.0613	2.0235	2.0848	63
64	0.29524	0.60112	0.01604	1055.31	1055.33	32.07	1057.07	1089.14	0.0632	2.0186	2.0818	64

Thermodynamic Properties of Water At Saturation: U.S. Units

TEMP °F	ABSOLUTE PRESSURE psi	ABSOLUTE PRESSURE in. Hg	SPECIFIC VOLUME (ft^3/lb) Sat. Liquid v_f	SPECIFIC VOLUME (ft^3/lb) Evap. v_{fg}	SPECIFIC VOLUME (ft^3/lb) Sat. Vapor v_g	ENTHALPY (Btu/lb) Sat. Liquid h_f	ENTHALPY (Btu/lb) Evap. h_{fg}	ENTHALPY (Btu/lb) Sat. Vapor h_g	ENTROPY (Btu/lba/°F) Sat. Liquid s_f	ENTROPY (Btu/lba/°F) Evap. s_{fg}	ENTROPY (Btu/lba/°F) Sat. Vapor s_g	TEMP °F
65	0.30574	0.62249	0.01604	1020.98	1021.00	33.07	1056.50	1089.57	0.0651	2.0136	2.0787	65
66	0.31656	0.64452	0.01604	987.95	987.97	34.07	1055.94	1090.01	0.0670	2.0088	2.0758	66
67	0.32772	0.66724	0.01605	956.10	956.12	35.07	1055.37	1090.44	0.0689	2.0039	2.0728	67
68	0.33921	0.69065	0.01605	925.43	925.45	36.07	1054.81	1090.88	0.0708	1.9990	2.0698	68
69	0.35107	0.71478	0.01605	895.85	895.87	37.07	1054.24	1091.31	0.0727	1.9941	2.0668	69
70	0.36328	0.73964	0.01605	867.34	867.36	38.07	1053.68	1091.75	0.0746	1.9893	2.0639	70
71	0.37586	0.76526	0.01605	839.86	839.88	39.07	1053.11	1092.18	0.0765	1.9845	2.0610	71
72	0.38882	0.79164	0.01606	813.37	813.39	40.07	1052.54	1092.61	0.0783	1.9797	2.0580	72
73	0.40217	0.81883	0.01606	787.85	787.87	41.07	1051.98	1093.05	0.0802	1.9750	2.0552	73
74	0.41592	0.84682	0.01606	763.19	763.21	42.06	1051.42	1093.48	0.0821	1.9702	2.0523	74
75	0.43008	0.87564	0.01606	739.42	739.44	43.06	1050.86	1093.92	0.0840	1.9654	2.0494	75
76	0.44465	0.90532	0.01606	716.51	716.53	44.06	1050.29	1094.35	0.0858	1.9607	2.0465	76
77	0.45966	0.93587	0.01607	694.38	694.40	45.06	1049.72	1094.78	0.0877	1.9560	2.0437	77
78	0.47510	0.96732	0.01607	673.04	673.06	46.06	1049.16	1095.22	0.0896	1.9513	2.0409	78
79	0.49100	0.99968	0.01607	652.44	652.46	47.06	1048.59	1095.65	0.0914	1.9466	2.0380	79
80	0.50736	1.03298	0.01607	632.54	632.56	48.06	1048.02	1096.08	0.0933	1.9419	2.0352	80
81	0.52419	1.06725	0.01608	613.35	613.37	49.06	1047.45	1096.51	0.0951	1.9373	2.0324	81
82	0.54150	1.10250	0.01608	594.82	594.84	50.05	1019.90	1069.95	0.0970	1.9327	2.0297	82
83	0.55931	1.13877	0.01608	576.90	576.92	51.05	1046.33	1097.38	0.0988	1.9281	2.0269	83
84	0.57763	1.17606	0.01608	559.63	559.65	52.05	1045.76	1097.81	0.1006	1.9236	2.0242	84
85	0.59647	1.21442	0.01609	542.92	542.94	53.05	1045.19	1098.24	0.1025	1.9189	2.0214	85
86	0.61584	1.25385	0.01609	526.79	526.81	54.05	1044.62	1098.67	0.1043	1.9144	2.0187	86
87	0.63575	1.29440	0.01609	511.20	511.22	55.05	1044.06	1099.11	0.1061	1.9099	2.0160	87
88	0.65622	1.33608	0.01609	496.13	496.15	56.05	1043.49	1099.54	0.1080	1.9053	2.0133	88
89	0.67726	1.37892	0.01610	481.59	481.61	57.04	1042.93	1099.97	0.1098	1.9008	2.0106	89
90	0.69889	1.42295	0.01610	467.51	467.53	58.04	1042.36	1100.40	0.1116	1.8963	2.0079	90
91	0.72111	1.46820	0.01610	453.91	453.93	59.04	1041.79	1100.83	0.1134	1.8919	2.0053	91
92	0.74394	1.51468	0.01611	440.76	440.78	60.04	1041.22	1101.26	0.1152	1.8874	2.0026	92
93	0.76740	1.56244	0.01611	428.04	428.06	61.04	1040.65	1101.69	0.1170	1.8830	2.0000	93
94	0.79150	1.61151	0.01611	415.74	415.76	62.04	1040.08	1102.12	0.1188	1.8785	1.9973	94
95	0.81625	1.66189	0.01612	403.84	403.86	63.03	1039.52	1102.55	0.1206	1.8741	1.9947	95
96	0.84166	1.71364	0.01612	392.32	392.34	64.03	1038.95	1102.98	0.1224	1.8697	1.9921	96
97	0.86776	1.76678	0.01612	381.19	381.21	65.03	1038.38	1103.41	0.1242	1.8653	1.9895	97
98	0.89456	1.82134	0.01612	370.42	370.44	66.03	1037.81	1103.84	0.1260	1.8610	1.9870	98
99	0.92207	1.87736	0.01613	359.99	360.01	67.03	1037.23	1104.26	0.1278	1.8566	1.9844	99
100	0.95031	1.93485	0.01613	349.90	349.92	68.03	1036.66	1104.69	0.1296	1.8523	1.9819	100
101	0.97930	1.99387	0.01613	340.13	340.15	69.02	1036.10	1105.12	0.1314	1.8479	1.9793	101
102	1.00904	2.05443	0.01614	330.69	330.71	70.02	1035.53	1105.55	0.1332	1.8436	1.9768	102
103	1.03956	2.11667	0.01614	321.53	321.55	71.02	1034.96	1105.98	0.1349	1.8394	1.9743	103
104	1.07088	2.18034	0.01614	312.67	312.69	72.02	1034.38	1106.40	0.1367	1.8351	1.9718	104
105	1.10301	2.24575	0.01615	304.08	304.10	73.02	1033.81	1106.83	0.1385	1.8308	1.9693	105
106	1.13597	2.31285	0.01615	295.75	295.77	74.02	1033.24	1107.26	0.1402	1.8266	1.9668	106
107	1.16977	2.38168	0.01616	287.71	287.73	75.01	1032.67	1107.68	0.1420	1.8223	1.9643	107
108	1.20444	2.45226	0.01616	279.90	279.92	76.01	1032.10	1108.11	0.1438	1.8181	1.9619	108
109	1.23999	2.52464	0.01616	272.34	272.36	77.01	1031.53	1108.54	0.1455	1.8139	1.9594	109
110	1.27644	2.59885	0.01617	265.01	265.03	78.01	1030.95	1108.96	0.1473	1.8097	1.9570	110
111	1.31381	2.67494	0.01617	257.91	257.93	79.01	1030.38	1109.39	0.1490	1.8056	1.9546	111
112	1.35212	2.75293	0.01617	251.02	251.04	80.01	1029.80	1109.81	0.1508	1.8013	1.9521	112
113	1.39138	2.83288	0.01618	244.36	244.38	81.01	1029.23	1110.24	0.1525	1.7972	1.9497	113
114	1.43162	2.91481	0.01618	237.88	237.90	82.00	1028.66	1110.66	0.1543	1.7931	1.9474	114

Thermodynamic Properties of Water At Saturation: U.S. Units

TEMP °F	ABSOLUTE PRESSURE psi	ABSOLUTE PRESSURE in. Hg	SPECIFIC VOLUME (ft^3/lb) Sat. Liquid v_f	SPECIFIC VOLUME (ft^3/lb) Evap. v_{fg}	SPECIFIC VOLUME (ft^3/lb) Sat. Vapor v_g	ENTHALPY (Btu/lb) Sat. Liquid h_f	ENTHALPY (Btu/lb) Evap. h_{fg}	ENTHALPY (Btu/lb) Sat. Vapor h_g	ENTROPY (Btu/lba/°F) Sat. Liquid s_f	ENTROPY (Btu/lba/°F) Evap. s_{fg}	ENTROPY (Btu/lba/°F) Sat. Vapor s_g	TEMP °F
115	1.47286	2.99878	0.01619	231.61	231.63	83.00	1028.09	1111.09	0.1560	1.7890	1.9450	115
116	1.51512	3.08481	0.01619	225.53	225.55	84.00	1027.51	1111.51	0.1577	1.7849	1.9426	116
117	1.55842	3.17296	0.01619	219.63	219.65	85.00	1026.93	1111.93	0.1595	1.7807	1.9402	117
118	1.60277	3.26327	0.01620	213.91	213.93	86.00	1026.36	1112.36	0.1612	1.7767	1.9379	118
119	1.64820	3.35577	0.01620	208.35	208.37	87.00	1025.78	1112.78	0.1629	1.7727	1.9356	119
120	1.69474	3.45052	0.01620	202.97	202.99	88.00	1025.20	1113.20	0.1647	1.7685	1.9332	120
121	1.74240	3.54755	0.01621	197.74	197.76	89.00	1024.62	1113.62	0.1664	1.7645	1.9309	121
122	1.79117	3.64691	0.01621	192.67	192.69	90.00	1024.05	1114.05	0.1681	1.7605	1.9286	122
123	1.84117	3.74863	0.01622	187.76	187.78	90.99	1023.48	1114.47	0.1698	1.7565	1.9263	123
124	1.89233	3.85282	0.01622	182.97	182.99	91.99	1022.90	1114.89	0.1715	1.7525	1.9240	124
125	1.94470	3.95945	0.01623	178.34	178.36	92.99	1022.32	1115.31	0.1732	1.7485	1.9217	125
126	1.99831	4.06860	0.01623	173.84	173.86	93.99	1021.74	1115.73	0.1749	1.7446	1.9195	126
127	2.05318	4.18032	0.01623	169.47	169.49	94.99	1021.16	1116.15	0.1766	1.7406	1.9172	127
128	2.10934	4.29465	0.01624	165.23	165.25	95.99	1020.58	1116.57	0.1783	1.7367	1.9150	128
129	2.16680	4.41165	0.01624	161.10	161.12	96.99	1020.00	1116.99	0.1800	1.7327	1.9127	129
130	2.22560	4.53136	0.01625	157.10	157.12	97.99	1019.42	1117.41	0.1817	1.7288	1.9105	130
131	2.28576	4.65384	0.01625	153.21	153.23	98.99	1018.84	1117.83	0.1834	1.7249	1.9083	131
132	2.34730	4.77914	0.01626	149.44	149.46	99.99	1018.26	1118.25	0.1851	1.7210	1.9061	132
133	2.41025	4.90730	0.01626	145.76	145.78	100.99	1017.68	1118.67	0.1868	1.7171	1.9039	133
134	2.47463	5.03839	0.01627	142.21	142.23	101.99	1017.09	1119.08	0.1885	1.7132	1.9017	134
135	2.54048	5.17246	0.01627	138.74	138.76	102.99	1016.51	1119.50	0.1902	1.7093	1.8995	135
136	2.60782	5.30956	0.01627	135.37	135.39	103.98	1015.94	1119.92	0.1919	1.7055	1.8974	136
137	2.67667	5.44975	0.01628	132.10	132.12	104.98	1015.36	1120.34	0.1935	1.7017	1.8952	137
138	2.74707	5.59308	0.01628	128.92	128.94	105.98	1014.77	1120.75	0.1952	1.6978	1.8930	138
139	2.81903	5.73961	0.01629	125.83	125.85	106.98	1014.19	1121.17	0.1969	1.6940	1.8909	139
140	2.89260	5.88939	0.01629	122.82	122.84	107.98	1013.60	1121.58	0.1985	1.6903	1.8888	140
141	2.96780	6.04250	0.01630	119.90	119.92	108.98	1013.02	1122.00	0.2002	1.6865	1.8867	141
142	3.04465	6.19897	0.01630	117.05	117.07	109.98	1012.43	1122.41	0.2019	1.6826	1.8845	142
143	3.12320	6.35888	0.01631	114.29	114.31	110.98	1011.85	1122.83	0.2035	1.6789	1.8824	143
144	3.20345	6.52229	0.01631	111.60	111.62	111.98	1011.26	1123.24	0.2052	1.6751	1.8803	144
145	3.28546	6.68926	0.01632	108.98	109.00	112.98	1010.68	1123.66	0.2068	1.6715	1.8783	145
146	3.36924	6.85984	0.01632	106.43	106.45	113.98	1010.09	1124.07	0.2085	1.6677	1.8762	146
147	3.45483	7.03410	0.01633	103.96	103.98	114.98	1009.50	1124.48	0.2101	1.6640	1.8741	147
148	3.54226	7.21211	0.01633	101.55	101.57	115.98	1008.91	1124.89	0.2118	1.6603	1.8721	148
149	3.63156	7.39393	0.01634	99.20	99.22	116.98	1008.33	1125.31	0.2134	1.6566	1.8700	149
150	3.72277	7.57962	0.01634	96.92	96.94	117.98	1007.74	1125.72	0.2151	1.6529	1.8680	150
151	3.81591	7.76925	0.01635	94.70	94.72	118.99	1007.14	1126.13	0.2167	1.6492	1.8659	151
152	3.91101	7.96289	0.01635	92.54	92.56	119.99	1006.55	1126.54	0.2184	1.6455	1.8639	152
153	4.00812	8.16061	0.01636	90.44	90.46	120.99	1005.96	1126.95	0.2200	1.6419	1.8619	153
154	4.10727	8.36247	0.01636	88.39	88.41	121.99	1005.37	1127.36	0.2216	1.6383	1.8599	154
155	4.20848	8.56854	0.01637	86.39	86.41	122.99	1004.78	1127.77	0.2233	1.6346	1.8579	155
156	4.31180	8.77890	0.01637	84.45	84.47	123.99	1004.19	1128.18	0.2249	1.6310	1.8559	156
157	4.41725	8.99360	0.01638	82.56	82.58	124.99	1003.60	1128.59	0.2265	1.6274	1.8539	157
158	4.52488	9.21274	0.01638	80.71	80.73	125.99	1003.00	1128.99	0.2281	1.6238	1.8519	158
159	4.63472	9.43637	0.01639	78.92	78.94	126.99	1002.41	1129.40	0.2297	1.6203	1.8500	159
160	4.74680	9.66460	0.01639	77.176	77.192	127.99	1001.82	1129.81	0.2314	1.6166	1.8480	160
161	4.86120	9.89740	0.01640	75.472	75.488	128.99	1001.23	1130.22	0.2330	1.6131	1.8461	161
162	4.97780	10.13500	0.01640	73.813	73.829	130.00	1000.62	1130.62	0.2346	1.6095	1.8441	162
163	5.09690	10.37740	0.01641	72.197	72.213	131.00	1000.03	1131.03	0.2362	1.6060	1.8422	163
164	5.21830	10.62460	0.01642	70.620	70.636	132.00	999.43	1131.43	0.2378	1.6025	1.8403	164

Thermodynamic Properties of Water At Saturation: U.S. Units

TEMP °F	ABSOLUTE PRESSURE psi	ABSOLUTE PRESSURE in. Hg	SPECIFIC VOLUME (ft^3/lb) Sat. Liquid v_f	SPECIFIC VOLUME (ft^3/lb) Evap. v_{fg}	SPECIFIC VOLUME (ft^3/lb) Sat. Vapor v_g	ENTHALPY (Btu/lb) Sat. Liquid h_f	ENTHALPY (Btu/lb) Evap. h_{fg}	ENTHALPY (Btu/lb) Sat. Vapor h_g	ENTROPY (Btu/lba/°F) Sat. Liquid s_f	ENTROPY (Btu/lba/°F) Evap. s_{fg}	ENTROPY (Btu/lba/°F) Sat. Vapor s_g	TEMP °F
165	5.34220	10.87680	0.01642	69.085	69.101	133.00	998.84	1131.84	0.2394	1.5989	1.8383	165
166	5.46850	11.13400	0.01643	67.588	67.604	134.00	998.24	1132.24	0.2410	1.5954	1.8364	166
167	5.59740	11.39630	0.01643	66.130	66.146	135.00	997.64	1132.64	0.2426	1.5919	1.8345	167
168	5.72870	11.66380	0.01644	64.707	64.723	136.01	997.04	1133.05	0.2442	1.5884	1.8326	168
169	5.86270	11.93660	0.01644	63.320	63.336	137.01	996.44	1133.45	0.2458	1.5850	1.8308	169
170	5.99930	12.21480	0.01645	61.970	61.986	138.01	995.84	1133.85	0.2474	1.5815	1.8289	170
171	6.13860	12.49830	0.01646	60.650	60.666	139.01	995.24	1134.25	0.2490	1.5780	1.8270	171
172	6.28060	12.78740	0.01646	59.364	59.380	140.01	994.65	1134.66	0.2506	1.5745	1.8251	172
173	6.42530	13.08210	0.01647	58.112	58.128	141.02	994.04	1135.06	0.2521	1.5712	1.8233	173
174	6.57290	13.38250	0.01647	56.888	56.904	142.02	993.44	1135.46	0.2537	1.5677	1.8214	174
175	6.72320	13.68860	0.01648	55.695	55.711	143.02	992.84	1135.86	0.2553	1.5643	1.8196	175
176	6.87650	14.00060	0.01648	54.533	54.549	144.02	992.24	1136.26	0.2569	1.5609	1.8178	176
177	7.03270	14.31860	0.01649	53.398	53.414	145.03	991.62	1136.65	0.2585	1.5574	1.8159	177
178	7.19180	14.64260	0.01650	52.291	52.307	146.03	991.02	1137.05	0.2600	1.5541	1.8141	178
179	7.35390	14.97270	0.01650	51.210	51.226	147.03	990.42	1137.45	0.2616	1.5507	1.8123	179
180	7.51910	15.30910	0.01651	50.154	50.171	148.04	989.81	1137.85	0.2632	1.5473	1.8105	180
181	7.68740	15.65180	0.01651	49.126	49.143	149.04	989.20	1138.24	0.2647	1.5440	1.8087	181
182	7.85890	16.00080	0.01652	48.121	48.138	150.04	988.60	1138.64	0.2663	1.5406	1.8069	182
183	8.03350	16.35640	0.01653	47.141	47.158	151.05	987.98	1139.03	0.2679	1.5372	1.8051	183
184	8.21140	16.71850	0.01653	46.185	46.202	152.05	987.38	1139.43	0.2694	1.5340	1.8034	184
185	8.39260	17.08740	0.01654	45.250	45.267	153.05	986.77	1139.82	0.2710	1.5306	1.8016	185
186	8.57700	17.46300	0.01654	44.339	44.356	154.06	986.16	1140.22	0.2725	1.5273	1.7998	186
187	8.76490	17.84550	0.01655	43.448	43.465	155.06	985.55	1140.61	0.2741	1.5240	1.7981	187
188	8.95620	18.23500	0.01656	42.578	42.595	156.07	984.93	1141.00	0.2756	1.5207	1.7963	188
189	9.15100	18.63160	0.01656	41.729	41.746	157.07	984.32	1141.39	0.2772	1.5174	1.7946	189
190	9.34930	19.03530	0.01657	40.901	40.918	158.07	983.71	1141.78	0.2787	1.5142	1.7929	190
191	9.55120	19.44640	0.01658	40.091	40.108	159.08	983.10	1142.18	0.2803	1.5108	1.7911	191
192	9.75670	19.86480	0.01658	39.300	39.317	160.08	982.49	1142.57	0.2818	1.5076	1.7894	192
193	9.96590	20.29070	0.01659	38.527	38.544	161.09	981.86	1142.95	0.2834	1.5043	1.7877	193
194	10.17880	20.72420	0.01659	37.773	37.790	162.09	981.25	1143.34	0.2849	1.5011	1.7860	194
195	10.39550	21.16530	0.01660	37.035	37.052	163.10	980.63	1143.73	0.2864	1.4979	1.7843	195
196	10.61600	21.61430	0.01661	36.314	36.331	164.10	980.02	1144.12	0.2880	1.4946	1.7826	196
197	10.84040	22.07120	0.01661	35.611	35.628	165.11	979.40	1144.51	0.2895	1.4914	1.7809	197
198	11.06870	22.53610	0.01662	34.923	34.940	166.11	978.78	1144.89	0.2910	1.4882	1.7792	198
199	11.30100	23.00910	0.01663	34.251	34.268	167.12	978.16	1145.28	0.2926	1.4850	1.7776	199
200	11.53740	23.49040	0.01663	33.593	33.610	168.13	977.53	1145.66	0.2941	1.4818	1.7759	200
201	11.77790	23.98000	0.01664	32.951	32.968	169.13	976.92	1146.05	0.2956	1.4786	1.7742	201
202	12.02250	24.47800	0.01665	32.323	32.340	170.14	976.29	1146.43	0.2971	1.4755	1.7726	202
203	12.27130	24.98470	0.01665	31.709	31.726	171.14	975.67	1146.81	0.2986	1.4723	1.7709	203
204	12.52440	25.50000	0.01666	31.110	31.127	172.15	975.05	1147.20	0.3002	1.4691	1.7693	204
205	12.78190	26.02410	0.01667	30.523	30.540	173.16	974.42	1147.58	0.3017	1.4660	1.7677	205
206	13.04360	26.55710	0.01667	29.948	29.965	174.16	973.80	1147.96	0.3032	1.4628	1.7660	206
207	13.30990	27.09910	0.01668	29.387	29.404	175.17	973.17	1148.34	0.3047	1.4597	1.7644	207
208	13.58060	27.65030	0.01669	28.839	28.856	176.18	972.54	1148.72	0.3062	1.4566	1.7628	208
209	13.85580	28.21080	0.01669	28.302	28.319	177.18	971.92	1149.10	0.3077	1.4535	1.7612	209
210	14.13570	28.78060	0.01670	27.778	27.795	178.19	971.29	1149.48	0.3092	1.4504	1.7596	210
211	14.42265	29.36475	0.01671	27.271	27.288	179.20	970.66	1149.86	0.3107	1.4473	1.7580	211
212	14.70960	29.94890	0.01671	26.763	26.780	180.20	970.03	1150.23	0.3122	1.4442	1.7564	212
213	15.00605	30.55260	0.01672	26.277	26.294	181.21	969.40	1150.61	0.3137	1.4411	1.7548	213
214	15.30250	31.15630	0.01673	25.790	25.807	182.22	968.76	1150.98	0.3152	1.4380	1.7532	214

Thermodynamic Properties of Water At Saturation: U.S. Units

TEMP °F	ABSOLUTE PRESSURE psi	ABSOLUTE PRESSURE in. Hg	SPECIFIC VOLUME (ft^3/lb) Sat. Liquid v_f	SPECIFIC VOLUME (ft^3/lb) Evap. v_{fg}	SPECIFIC VOLUME (ft^3/lb) Sat. Vapor v_g	ENTHALPY (Btu/lb) Sat. Liquid h_f	ENTHALPY (Btu/lb) Evap. h_{fg}	ENTHALPY (Btu/lb) Sat. Vapor h_g	ENTROPY (Btu/lba/°F) Sat. Liquid s_f	ENTROPY (Btu/lba/°F) Evap. s_{fg}	ENTROPY (Btu/lba/°F) Sat. Vapor s_g	TEMP °F
215	15.60885	31.77995	0.01674	25.326	25.343	183.23	968.13	1151.36	0.3167	1.4350	1.7517	215
216	15.91520	32.40360	0.01674	24.861	24.878	184.24	967.49	1151.73	0.3182	1.4319	1.7501	216
217	16.23155	33.04775	0.01675	24.416	24.433	185.25	966.86	1152.11	0.3197	1.4288	1.7485	217
218	16.54790	33.69190	0.01676	23.970	23.987	186.25	966.23	1152.48	0.3212	1.4257	1.7469	218
219	16.87460	34.35685	0.01677	23.544	23.561	187.26	965.59	1152.85	0.3227	1.4227	1.7454	219
220	17.20130	35.02180	0.01677	23.117	23.134	188.27	964.95	1153.22	0.3241	1.4197	1.7438	220
221	17.53860	35.70870	0.01678	22.708	22.725	189.28	964.31	1153.59	0.3256	1.4167	1.7423	221
222	17.87590	36.39560	0.01679	22.299	22.316	190.29	963.67	1153.96	0.3271	1.4136	1.7407	222
223	18.22400	37.10435	0.01680	21.908	21.925	191.30	963.03	1154.33	0.3286	1.4106	1.7392	223
224	18.57210	37.81310	0.01680	21.516	21.533	192.31	962.39	1154.70	0.3301	1.4076	1.7377	224
225	18.93130	38.54445	0.01681	21.141	21.158	193.32	961.75	1155.07	0.3316	1.4031	1.7362	225
226	19.29050	39.27580	0.01682	20.765	20.782	194.33	961.10	1155.43	0.3330	1.3986	1.7347	226
227	19.66105	40.03030	0.01683	20.405	20.422	195.34	960.46	1155.80	0.3345	1.3972	1.7332	227
228	20.03160	40.78480	0.01683	20.045	20.062	196.35	959.81	1156.16	0.3359	1.3957	1.7316	228
229	20.41385	41.56300	0.01684	19.700	19.717	197.36	959.17	1156.53	0.3374	1.3913	1.7302	229
230	20.79610	42.34120	0.01684	19.355	19.372	198.37	958.52	1156.89	0.3389	1.3868	1.7287	230
231	21.19020	43.14365	0.01685	19.024	19.041	199.38	957.88	1157.26	0.3404	1.3839	1.7272	231
232	21.58430	43.94610	0.01686	18.692	18.709	200.39	957.23	1157.62	0.3418	1.3809	1.7257	232
233	21.99065	44.77335	0.01687	18.374	18.391	201.40	956.58	1157.98	0.3433	1.3780	1.7242	233
234	22.39700	45.60060	0.01688	18.056	18.073	202.41	955.93	1158.34	0.3447	1.3751	1.7227	234
235	22.81575	46.45330	0.01689	17.751	17.768	203.43	955.28	1158.70	0.3462	1.3722	1.7213	235
236	23.23450	47.30600	0.01689	17.446	17.463	204.44	954.62	1159.06	0.3476	1.3693	1.7198	236
237	23.66610	48.18465	0.01690	17.153	17.170	205.45	953.97	1159.42	0.3491	1.3679	1.7184	237
238	24.09770	49.06330	0.01691	16.860	16.877	206.46	953.31	1159.77	0.3505	1.3664	1.7169	238
239	24.54230	49.96855	0.01692	16.579	16.596	207.48	952.65	1160.13	0.3520	1.3621	1.7155	239
240	24.98690	50.87380	0.01692	16.297	16.314	208.49	951.99	1160.48	0.3534	1.3577	1.7140	240
241	25.44485	51.80620	0.01693	16.027	16.044	209.50	951.34	1160.84	0.3549	1.3549	1.7126	241
242	25.90280	52.73860	0.01694	15.757	15.774	210.51	950.68	1161.19	0.3563	1.3520	1.7111	242
243	26.37445	53.69885	0.01695	15.498	15.515	211.53	950.02	1161.55	0.3578	1.3492	1.7097	243
244	26.84610	54.65910	0.01695	15.238	15.255	212.54	949.36	1161.90	0.3592	1.3463	1.7083	244
245	27.33165	55.64775	0.01696	14.989	15.006	213.56	948.70	1162.25	0.3607	1.3434	1.7069	245
246	27.81720	56.63640	0.01697	14.739	14.756	214.57	948.03	1162.60	0.3621	1.3405	1.7055	246
247	28.31705	57.65405	0.01698	14.499	14.516	215.59	947.36	1162.95	0.3635	1.3391	1.7041	247
248	28.81690	58.67170	0.01698	14.259	14.276	216.60	946.69	1163.29	0.3649	1.3377	1.7026	248
249	29.33130	59.71905	0.01699	14.029	14.046	217.62	946.03	1163.64	0.3664	1.3335	1.7012	249
250	29.84570	60.76640	0.01700	13.798	13.815	218.63	945.36	1163.99	0.3678	1.3293	1.6998	250
251	30.37500	61.84410	0.01701	13.576	13.594	219.65	944.69	1164.34	0.3692	1.3265	1.6985	251
252	30.90430	62.92180	0.01702	13.355	13.372	220.66	944.02	1164.68	0.3706	1.3237	1.6971	252
253	31.44885	64.03045	0.01703	13.141	13.159	221.68	943.35	1165.03	0.3721	1.3209	1.6957	253
254	31.99340	65.13910	0.01703	12.928	12.945	222.69	942.68	1165.37	0.3735	1.3181	1.6943	254
255	32.55345	66.27940	0.01704	12.714	12.732	223.71	942.01	1165.71	0.3749	1.3154	1.6930	255
256	33.11350	67.41970	0.01705	12.501	12.518	224.72	941.33	1166.05	0.3763	1.3126	1.6916	256
257	33.68940	68.59230	0.01706	12.312	12.329	225.74	940.65	1166.39	0.3778	1.3112	1.6903	257
258	34.26530	69.76490	0.01707	12.123	12.140	226.76	939.97	1166.73	0.3792	1.3097	1.6889	258
259	34.35745	70.97045	0.01708	11.932	11.950	227.78	939.29	1167.07	0.3806	1.3056	1.6876	259
260	34.44960	72.17600	0.01708	11.742	11.759	228.79	938.61	1167.40	0.3820	1.3015	1.6862	260
261	35.55825	73.41525	0.01709	11.559	11.576	229.81	937.93	1167.74	0.3834	1.2988	1.6849	261
262	36.66690	74.65450	0.01710	11.376	11.393	230.83	937.25	1168.08	0.3848	1.2960	1.6835	262
263	37.29245	75.92810	0.01711	11.200	11.217	231.85	936.56	1168.41	0.3862	1.2933	1.6822	263
264	37.91800	77.20170	0.01712	11.024	11.041	232.87	935.87	1168.74	0.3876	1.2905	1.6808	264

Thermodynamic Properties of Water At Saturation: U.S. Units

TEMP °F	ABSOLUTE PRESSURE psi	ABSOLUTE PRESSURE in. Hg	SPECIFIC VOLUME (ft^3/lb) Sat. Liquid v_f	SPECIFIC VOLUME (ft^3/lb) Evap. v_{fg}	SPECIFIC VOLUME (ft^3/lb) Sat. Vapor v_g	ENTHALPY (Btu/lb) Sat. Liquid h_f	ENTHALPY (Btu/lb) Evap. h_{fg}	ENTHALPY (Btu/lb) Sat. Vapor h_g	ENTROPY (Btu/lba/°F) Sat. Liquid s_f	ENTROPY (Btu/lba/°F) Evap. s_{fg}	ENTROPY (Btu/lba/°F) Sat. Vapor s_g	TEMP °F
265	38.56075	78.51035	0.01713	10.854	10.871	233.89	935.19	1169.08	0.3890	1.2878	1.6795	265
266	39.20350	79.81900	0.01714	10.684	10.701	234.90	934.51	1169.41	0.3904	1.2851	1.6781	266
267	39.86380	81.16340	0.01715	10.520	10.538	235.92	933.82	1169.74	0.3918	1.2837	1.6768	267
268	40.52410	82.50780	0.01715	10.357	10.374	236.94	933.13	1170.07	0.3932	1.2823	1.6755	268
269	41.20235	83.88875	0.01716	10.199	10.217	237.96	932.44	1170.40	0.3946	1.2783	1.6742	269
270	41.88060	85.26970	0.01717	10.042	10.059	238.98	931.74	1170.72	0.3960	1.2743	1.6729	270
271	42.57710	86.68780	0.01718	9.890	9.907	240.01	931.05	1171.05	0.3974	1.2716	1.6716	271
272	43.27360	88.10590	0.01719	9.738	9.755	241.03	930.35	1171.38	0.3988	1.2689	1.6703	272
273	43.98880	89.56200	0.01720	9.591	9.609	242.05	929.65	1171.70	0.4002	1.2662	1.6690	273
274	44.70400	91.01810	0.01721	9.445	9.462	243.07	928.95	1172.02	0.4016	1.2635	1.6677	274
275	45.43815	92.51285	0.01722	9.303	9.321	244.09	928.26	1172.35	0.4030	1.2609	1.6664	275
276	46.17230	94.00760	0.01722	9.162	9.179	245.11	927.56	1172.67	0.4044	1.2582	1.6651	276
277	46.92585	95.54185	0.01723	9.026	9.043	246.14	926.86	1172.99	0.4058	1.2569	1.6639	277
278	47.67940	97.07610	0.01724	8.890	8.907	247.16	926.15	1173.31	0.4071	1.2555	1.6626	278
279	48.45270	98.65055	0.01725	8.758	8.776	248.18	925.45	1173.63	0.4085	1.2516	1.6613	279
280	49.22600	100.22500	0.01726	8.627	8.644	249.20	924.74	1173.94	0.4099	1.2476	1.6600	280
281	50.01940	101.84040	0.01727	8.500	8.517	250.23	924.03	1174.26	0.4113	1.2450	1.6588	281
282	50.81280	103.45580	0.01728	8.373	8.390	251.25	923.32	1174.57	0.4127	1.2423	1.6575	282
283	51.62670	105.11295	0.01729	8.251	8.268	252.28	922.61	1174.89	0.4141	1.2397	1.6563	283
284	52.44060	106.77010	0.01730	8.129	8.146	253.30	921.90	1175.20	0.4154	1.2371	1.6550	284
285	53.27545	108.46980	0.01731	8.011	8.028	254.33	921.19	1175.51	0.4168	1.2345	1.6538	285
286	54.11030	110.16950	0.01731	7.893	7.910	255.35	920.47	1175.82	0.4182	1.2318	1.6525	286
287	54.96640	111.91255	0.01732	7.778	7.796	256.38	919.76	1176.13	0.4196	1.2305	1.6513	287
288	55.82250	113.65560	0.01733	7.664	7.681	257.40	919.04	1176.44	0.4209	1.2291	1.6500	288
289	56.70025	115.44275	0.01734	7.554	7.571	258.43	918.32	1176.75	0.4223	1.2253	1.6488	289
290	57.57800	117.22990	0.01735	7.444	7.461	259.45	917.60	1177.05	0.4236	1.2215	1.6476	290
291	58.47785	119.06200	0.01736	7.337	7.355	260.48	916.88	1177.36	0.4250	1.2189	1.6464	291
292	59.37770	120.89410	0.01737	7.231	7.248	261.51	916.15	1177.66	0.4264	1.2163	1.6451	292
293	60.30005	122.77195	0.01738	7.128	7.146	262.54	915.43	1177.96	0.4278	1.2137	1.6439	293
294	61.22240	124.64980	0.01739	7.026	7.043	263.56	914.70	1178.26	0.4291	1.2111	1.6427	294
295	62.16760	126.57425	0.01740	6.926	6.944	264.59	913.97	1178.56	0.4305	1.2086	1.6415	295
296	63.11280	128.49870	0.01741	6.827	6.844	265.62	913.24	1178.86	0.4318	1.2060	1.6402	296
297	64.08130	130.47060	0.01742	6.731	6.748	266.65	912.51	1179.16	0.4332	1.2035	1.6390	297
298	65.04980	132.44250	0.01743	6.635	6.652	267.68	911.77	1179.45	0.4345	1.2009	1.6378	298
299	66.04195	134.46260	0.01744	6.542	6.560	268.71	911.04	1179.75	0.4359	1.1996	1.6366	299
300	67.03410	136.48270	0.01745	6.450	6.467	269.74	910.30	1180.04	0.4372	1.1982	1.6354	300

Thermodynamic Properties of Moist Air: U.S. Units

(STANDARD ATMOSPHERIC PRESSURE, 29.921 in. Hg)

TEMP °F	w_s lbw/lba	VOLUME (ft³/lb) Sat. Liquid v_a	Evap. v_{as}	Sat. Vapor v_s	ENTHALPY (Btu/lb) Sat. Liquid h_a	Evap. h_{as}	Sat. Vapor h_s	ENTROPY (Btu/lb$_{da}$/°F) Sat. Liquid s_a	Evap. s_{as}	Sat. Vapor s_s	CONDENSED WATER HW Btu/lb	S_w Btu/lb/°F	VAPOR PRESS. p_s in. Hg	TEMP °F
-80	0.0000049	9.553	0.000	9.553	-19.221	0.006	-19.215	-0.04594	0.00002	-0.04592	-193.45	-0.4067	0.000236	-80
-79	0.0000053	9.579	0.000	9.579	-18.980	0.005	-18.975	-0.04531	0.00002	-0.04529	-193.06	-0.4056	0.000255	-79
-78	0.0000057	9.604	0.000	9.604	-18.740	0.006	-18.734	-0.04468	0.00002	-0.04466	-192.66	-0.4046	0.000275	-78
-77	0.0000062	9.629	0.000	9.629	-18.500	0.007	-18.493	-0.04405	0.00002	-0.04403	-192.27	-0.4036	0.000296	-77
-76	0.0000067	9.655	0.000	9.655	-18.259	0.007	-18.252	-0.04342	0.00002	-0.04340	-191.87	-0.4025	0.000319	-76
-75	0.0000072	9.680	0.000	9.680	-18.019	0.008	-18.011	-0.04279	0.00002	-0.04277	-191.47	-0.4015	0.000344	-75
-74	0.0000078	9.705	0.000	9.705	-17.778	0.008	-17.770	-0.04217	0.00002	-0.04215	-191.07	-0.4005	0.000371	-74
-73	0.0000084	9.731	0.000	9.731	-17.538	0.009	-17.529	-0.04155	0.00003	-0.04152	-190.68	-0.3994	0.000400	-73
-72	0.0000090	9.756	0.000	9.756	-17.298	0.010	-17.288	-0.04093	0.00003	-0.04090	-190.27	-0.3984	0.000430	-72
-71	0.0000097	9.781	0.001	9.782	-17.057	0.010	-17.047	-0.04031	0.00003	-0.04028	-189.87	-0.3974	0.000463	-71
-70	0.0000104	9.807	0.000	9.807	-16.817	0.011	-16.806	-0.03969	0.00003	-0.03966	-189.47	-0.3963	0.000498	-70
-69	0.0000112	9.832	0.000	9.832	-16.577	0.012	-16.565	-0.03907	0.00003	-0.03904	-189.07	-0.3953	0.000536	-69
-68	0.0000120	9.857	0.001	9.858	-16.336	0.012	-16.324	-0.03846	0.00003	-0.03843	-188.66	-0.3943	0.000576	-68
-67	0.0000129	9.883	0.000	9.883	-16.096	0.013	-16.083	-0.03785	0.00004	-0.03781	-188.26	-0.3932	0.000619	-67
-66	0.0000139	9.908	0.000	9.908	-15.856	0.015	-15.841	-0.03724	0.00004	-0.03720	-187.85	-0.3922	0.000665	-66
-65	0.0000149	9.933	0.001	9.934	-15.616	0.016	-15.600	-0.03663	0.00004	-0.03659	-187.44	-0.3912	0.000714	-65
-64	0.0000160	9.959	0.000	9.959	-15.375	0.016	-15.359	-0.03602	0.00005	-0.03597	-187.04	-0.3901	0.000766	-64
-63	0.0000172	9.984	0.000	9.984	-15.135	0.018	-15.117	-0.03541	0.00005	-0.03536	-186.63	-0.3891	0.000822	-63
-62	0.0000184	10.009	0.001	10.010	-14.895	0.019	-14.876	-0.03481	0.00005	-0.03476	-186.22	-0.3881	0.000882	-62
-61	0.0000198	10.035	0.000	10.035	-14.654	0.020	-14.634	-0.03420	0.00005	-0.03415	-185.81	-0.3870	0.000945	-61
-60	0.0000212	10.060	0.000	10.060	-14.414	0.022	-14.392	-0.03360	0.00006	-0.03354	-185.39	-0.3860	0.001013	-60
-59	0.0000227	10.085	0.001	10.086	-14.174	0.024	-14.150	-0.03300	0.00006	-0.03294	-184.98	-0.3850	0.001086	-59
-58	0.0000243	10.111	0.000	10.111	-13.933	0.025	-13.908	-0.03240	0.00007	-0.03233	-184.57	-0.3839	0.001163	-58
-57	0.0000260	10.136	0.001	10.137	-13.693	0.027	-13.666	-0.03180	0.00007	-0.03173	-184.15	-0.3829	0.001246	-57
-56	0.0000279	10.161	0.001	10.162	-13.453	0.029	-13.424	-0.03121	0.00008	-0.03113	-183.74	-0.3819	0.001333	-56
-55	0.0000298	10.187	0.000	10.187	-13.213	0.031	-13.182	-0.03061	0.00008	-0.03053	-183.32	-0.3808	0.001427	-55
-54	0.0000319	10.212	0.001	10.213	-12.972	0.033	-12.939	-0.03002	0.00009	-0.02993	-182.90	-0.3798	0.001526	-54
-53	0.0000341	10.237	0.001	10.238	-12.732	0.035	-12.697	-0.02943	0.00009	-0.02934	-182.48	-0.3788	0.001632	-53
-52	0.0000365	10.263	0.000	10.263	-12.492	0.038	-12.454	-0.02884	0.00010	-0.02874	-182.06	-0.3778	0.001745	-52
-51	0.0000390	10.288	0.001	10.289	-12.251	0.040	-12.211	-0.02825	0.00011	-0.02814	-181.64	-0.3767	0.001865	-51
-50	0.0000416	10.313	0.001	10.314	-12.011	0.043	-11.968	-0.02766	0.00011	-0.02755	-181.22	-0.3757	0.001992	-50
-49	0.0000445	10.339	0.001	10.340	-11.771	0.046	-11.725	-0.02708	0.00012	-0.02696	-180.80	-0.3747	0.002128	-49
-48	0.0000475	10.364	0.001	10.365	-11.531	0.050	-11.481	-0.02649	0.00013	-0.02636	-180.37	-0.3736	0.002272	-48
-47	0.0000507	10.389	0.001	10.390	-11.290	0.053	-11.237	-0.02591	0.00014	-0.02577	-179.95	-0.3726	0.002425	-47
-46	0.0000541	10.415	0.001	10.416	-11.050	0.056	-10.994	-0.02533	0.00015	-0.02518	-179.52	-0.3716	0.002587	-46
-45	0.0000577	10.440	0.001	10.441	-10.810	0.060	-10.750	-0.02475	0.00016	-0.02459	-179.10	-0.3705	0.002760	-45
-44	0.0000615	10.465	0.001	10.466	-10.570	0.065	-10.505	-0.02417	0.00017	-0.02400	-178.67	-0.3695	0.002943	-44
-43	0.0000656	10.491	0.001	10.492	-10.329	0.068	-10.261	-0.02359	0.00017	-0.02342	-178.24	-0.3685	0.003137	-43
-42	0.0000699	10.516	0.001	10.517	-10.089	0.073	-10.016	-0.02302	0.00019	-0.02283	-177.81	-0.3675	0.003343	-42
-41	0.0000744	10.541	0.002	10.543	-9.849	0.078	-9.771	-0.02244	0.00020	-0.02224	-177.38	-0.3664	0.003562	-41
-40	0.0000793	10.567	0.001	10.568	-9.609	0.083	-9.526	-0.02187	0.00021	-0.02166	-176.95	-0.3654	0.003793	-40
-39	0.0000844	10.592	0.001	10.593	-9.368	0.088	-9.280	-0.02130	0.00023	-0.02107	-176.52	-0.3644	0.004039	-39
-38	0.0000898	10.617	0.002	10.619	-9.128	0.094	-9.034	-0.02073	0.00024	-0.02049	-176.08	-0.3633	0.004299	-38
-37	0.0000956	10.643	0.001	10.644	-8.888	0.100	-8.788	-0.02016	0.00025	-0.01991	-175.65	-0.3623	0.004575	-37
-36	0.0001017	10.668	0.002	10.670	-8.648	0.107	-8.541	-0.01959	0.00027	-0.01932	-175.21	-0.3613	0.004866	-36
-35	0.0001081	10.693	0.002	10.695	-8.407	0.113	-8.294	-0.01902	0.00028	-0.01874	-174.78	-0.3603	0.005175	-35
-34	0.0001150	10.719	0.002	10.721	-8.167	0.120	-8.047	-0.01846	0.00030	-0.01816	-174.34	-0.3592	0.005502	-34
-33	0.0001222	10.744	0.002	10.746	-7.927	0.128	-7.799	-0.01790	0.00032	-0.01758	-173.90	-0.3582	0.005848	-33
-32	0.0001298	10.769	0.003	10.772	-7.687	0.136	-7.551	-0.01733	0.00034	-0.01699	-173.46	-0.3572	0.006214	-32
-31	0.0001379	10.795	0.002	10.797	-7.447	0.145	-7.302	-0.01677	0.00036	-0.01641	-173.02	-0.3561	0.006601	-31

Thermodynamic Properties of Moist Air: U.S. Units

(STANDARD ATMOSPHERIC PRESSURE, 29.921 in. Hg)

		VOLUME (ft^3/lb)			ENTHALPY (Btu/lb)			ENTROPY (Btu/lb_{da}/°F)			CONDENSED WATER		VAPOR PRESS.	
TEMP °F	w_s lbw/lba	Sat. Liquid v_a	Evap. v_{as}	Sat. Vapor v_s	Sat. Liquid h_a	Evap. h_{as}	Sat. Vapor h_s	Sat. Liquid s_a	Evap. s_{as}	Sat. Vapor s_s	HW Btu/lb	S_W Btu/lb/°F	p_s in. Hg	TEMP °F
-30	0.0001465	10.820	0.002	10.822	-7.206	0.153	-7.053	-0.01621	0.00038	-0.01583	-172.58	-0.3551	0.007009	-30
-29	0.0001555	10.845	0.003	10.848	-6.966	0.163	-6.803	-0.01565	0.00040	-0.01525	-172.14	-0.3541	0.007442	-29
-28	0.0001650	10.871	0.002	10.873	-6.726	0.173	-6.553	-0.01510	0.00043	-0.01467	-171.70	-0.3531	0.007898	-28
-27	0.0001751	10.896	0.003	10.899	-6.486	0.184	-6.302	-0.01454	0.00045	-0.01409	-171.25	-0.3520	0.008381	-27
-26	0.0001858	10.921	0.003	10.924	-6.245	0.194	-6.051	-0.01399	0.00048	-0.01351	-170.81	-0.3510	0.008890	-26
-25	0.0001970	10.947	0.003	10.950	-6.005	0.207	-5.798	-0.01343	0.00050	-0.01293	-170.36	-0.3500	0.009428	-25
-24	0.0002088	10.972	0.004	10.976	-5.765	0.220	-5.545	-0.01288	0.00053	-0.01235	-169.92	-0.3489	0.009995	-24
-23	0.0002214	10.997	0.004	11.001	-5.525	0.233	-5.292	-0.01233	0.00057	-0.01176	-169.47	-0.3479	0.010594	-23
-22	0.0002346	11.022	0.005	11.027	-5.284	0.246	-5.038	-0.01178	0.00060	-0.01118	-169.02	-0.3469	0.011226	-22
-21	0.0002485	11.048	0.004	11.052	-5.044	0.261	-4.783	-0.01123	0.00063	-0.01060	-168.57	-0.3459	0.011893	-21
-20	0.0002632	11.073	0.005	11.078	-4.804	0.277	-4.527	-0.01069	0.00067	-0.01002	-168.12	-0.3448	0.012595	-20
-19	0.0002786	11.098	0.005	11.103	-4.564	0.293	-4.271	-0.01014	0.00071	-0.00943	-167.67	-0.3438	0.013336	-19
-18	0.0002950	11.124	0.005	11.129	-4.324	0.311	-4.013	-0.00960	0.00075	-0.00885	-167.21	-0.3428	0.014117	-18
-17	0.0003121	11.149	0.006	11.155	-4.084	0.330	-3.754	-0.00905	0.00079	-0.00826	-166.76	-0.3418	0.014939	-17
-16	0.0003303	11.174	0.006	11.180	-3.843	0.348	-3.495	-0.00851	0.00083	-0.00768	-166.30	-0.3407	0.015806	-16
-15	0.0003493	11.200	0.006	11.206	-3.603	0.368	-3.235	-0.00797	0.00088	-0.00709	-165.85	-0.3397	0.016718	-15
-14	0.0003694	11.225	0.007	11.232	-3.363	0.390	-2.973	-0.00743	0.00093	-0.00650	-165.39	-0.3387	0.017679	-14
-13	0.0003905	11.250	0.007	11.257	-3.123	0.413	-2.710	-0.00689	0.00098	-0.00591	-164.93	-0.3377	0.018690	-13
-12	0.0004128	11.276	0.007	11.283	-2.882	0.435	-2.447	-0.00635	0.00103	-0.00532	-164.47	-0.3366	0.019754	-12
-11	0.0004362	11.301	0.008	11.309	-2.642	0.460	-2.182	-0.00582	0.00109	-0.00473	-164.01	-0.3356	0.020873	-11
-10	0.0004608	11.326	0.009	11.335	-2.402	0.487	-1.915	-0.00528	0.00114	-0.00414	-163.55	-0.3346	0.022050	-10
-9	0.0004867	11.351	0.009	11.360	-2.162	0.515	-1.647	-0.00475	0.00121	-0.00354	-163.09	-0.3335	0.023289	-9
-8	0.0005139	11.377	0.009	11.386	-1.922	0.544	-1.378	-0.00422	0.00128	-0.00294	-162.63	-0.3325	0.024591	-8
-7	0.0005425	11.402	0.010	11.412	-1.681	0.573	-1.108	-0.00369	0.00135	-0.00234	-162.17	-0.3315	0.025959	-7
-6	0.0005726	11.427	0.011	11.438	-1.441	0.606	-0.835	-0.00316	0.00142	-0.00174	-161.70	-0.3305	0.027397	-6
-5	0.0006041	11.453	0.011	11.464	-1.201	0.640	-0.561	-0.00263	0.00149	-0.00114	-161.23	-0.3294	0.028907	-5
-4	0.0006373	11.478	0.012	11.490	-0.961	0.675	-0.286	-0.00210	0.00157	-0.00053	-160.77	-0.3284	0.030494	-4
-3	0.0006722	11.503	0.013	11.516	-0.721	0.713	-0.008	-0.00157	0.00165	0.00008	-160.30	-0.3274	0.032160	-3
-2	0.0007088	11.529	0.013	11.542	-0.480	0.751	0.271	-0.00105	0.00174	0.00069	-159.83	-0.3264	0.033909	-2
-1	0.0007472	11.554	0.014	11.568	-0.240	0.792	0.552	-0.00052	0.00182	0.00130	-159.36	-0.3253	0.035744	-1
0	0.0007875	11.579	0.015	11.594	0.000	0.835	0.835	0.00000	0.00192	0.00192	-158.89	-0.3243	0.037671	0
1	0.0008298	11.604	0.016	11.620	0.240	0.881	1.121	0.00052	0.00202	0.00254	-158.42	-0.3233	0.039694	1
2	0.0008742	11.630	0.016	11.646	0.480	0.928	1.408	0.00104	0.00213	0.00317	-157.95	-0.3223	0.041814	2
3	0.0009207	11.655	0.017	11.672	0.721	0.978	1.699	0.00156	0.00224	0.00380	-157.47	-0.3212	0.044037	3
4	0.0009695	11.680	0.019	11.699	0.961	1.030	1.991	0.00208	0.00235	0.00443	-157.00	-0.3202	0.046370	4
5	0.0010207	11.706	0.019	11.725	1.201	1.085	2.286	0.00260	0.00246	0.00506	-156.52	-0.3192	0.048814	5
6	0.0010743	11.731	0.020	11.751	1.441	1.143	2.584	0.00311	0.00259	0.00570	-156.05	-0.3182	0.051375	6
7	0.0011306	11.756	0.022	11.778	1.681	1.203	2.884	0.00363	0.00272	0.00635	-155.57	-0.3171	0.054060	7
8	0.0011895	11.782	0.022	11.804	1.922	1.266	3.188	0.00414	0.00286	0.00700	-155.09	-0.3161	0.056872	8
9	0.0012512	11.807	0.024	11.831	2.162	1.332	3.494	0.00466	0.00300	0.00766	-154.61	-0.3151	0.059819	9
10	0.0013158	11.832	0.025	11.857	2.402	1.402	3.804	0.00517	0.00315	0.00832	-154.13	-0.3141	0.062901	10
11	0.0013835	11.857	0.027	11.884	2.642	1.475	4.117	0.00568	0.00330	0.00898	-153.65	-0.3130	0.066131	11
12	0.0014544	11.883	0.027	11.910	2.882	1.551	4.433	0.00619	0.00347	0.00966	-153.17	-0.3120	0.069511	12
13	0.0015286	11.908	0.029	11.937	3.123	1.630	4.753	0.00670	0.00363	0.01033	-152.68	-0.3110	0.073049	13
14	0.0016062	11.933	0.031	11.964	3.363	1.714	5.077	0.00721	0.00381	0.01102	-152.20	-0.3100	0.076751	14

Thermodynamic Properties of Moist Air: U.S. Units

(STANDARD ATMOSPHERIC PRESSURE, 29.921 in. Hg)

		VOLUME (ft^3/lb)			ENTHALPY (Btu/lb)			ENTROPY (Btu/lb_{da}/°F)			CONDENSED WATER		VAPOR PRESS.	
TEMP °F	w_s lbw/lba	Sat. Liquid v_a	Evap. v_{as}	Sat. Vapor v_s	Sat. Liquid h_a	Evap. h_{as}	Sat. Vapor h_s	Sat. Liquid s_a	Evap. s_{as}	Sat. Vapor s_s	HW Btu/lb	S_w Btu/lb/°F	p_s in. Hg	TEMP °F
15	0.0016874	11.959	0.032	11.991	3.603	1.801	5.404	0.00771	0.00400	0.01171	-151.71	-0.3089	0.080623	15
16	0.0017724	11.984	0.034	12.018	3.843	1.893	5.736	0.00822	0.00419	0.01241	-151.22	-0.3079	0.084673	16
17	0.0018613	12.009	0.036	12.045	4.084	1.988	6.072	0.00872	0.00440	0.01312	-150.74	-0.3069	0.088907	17
18	0.0019543	12.035	0.037	12.072	4.324	2.088	6.412	0.00923	0.00460	0.01383	-150.25	-0.3059	0.093334	18
19	0.0020515	12.060	0.039	12.099	4.564	2.193	6.757	0.00973	0.00482	0.01455	-149.76	-0.3049	0.097962	19
20	0.0021531	12.085	0.042	12.127	4.804	2.303	7.107	0.01023	0.00505	0.01528	-149.27	-0.3038	0.102798	20
21	0.0022592	12.110	0.044	12.154	5.044	2.418	7.462	0.01073	0.00529	0.01602	-148.78	-0.3028	0.107849	21
22	0.0023703	12.136	0.046	12.182	5.285	2.537	7.822	0.01123	0.00554	0.01677	-148.28	-0.3018	0.113130	22
23	0.0024863	12.161	0.048	12.209	5.525	2.662	8.187	0.01173	0.00580	0.01753	-147.79	-0.3008	0.118645	23
24	0.0026073	12.186	0.051	12.237	5.765	2.793	8.558	0.01223	0.00607	0.01830	-147.30	-0.2997	0.124396	24
25	0.0027339	12.212	0.053	12.265	6.005	2.930	8.935	0.01272	0.00636	0.01908	-146.80	-0.2987	0.130413	25
26	0.0028660	12.237	0.056	12.293	6.246	3.072	9.318	0.01322	0.00665	0.01987	-146.30	-0.2977	0.136684	26
27	0.0030039	12.262	0.059	12.321	6.486	3.222	9.708	0.01371	0.00696	0.02067	-145.81	-0.2967	0.143233	27
28	0.0031480	12.287	0.062	12.349	6.726	3.378	10.104	0.01420	0.00728	0.02148	-145.31	-0.2956	0.150066	28
29	0.0032984	12.313	0.065	12.378	6.966	3.541	10.507	0.01470	0.00761	0.02231	-144.81	-0.2946	0.157198	29
30	0.0034552	12.338	0.068	12.406	7.206	3.711	10.917	0.01519	0.00796	0.02315	-144.31	-0.2936	0.164631	30
31	0.0036190	12.363	0.072	12.435	7.447	3.888	11.335	0.01568	0.00832	0.02400	-143.80	-0.2926	0.172390	31
32	0.0037895	12.389	0.075	12.464	7.687	4.073	11.760	0.01617	0.00870	0.02487	-143.30	-0.2915	0.180479	32
32*	0.003790	12.389	0.075	12.464	7.687	4.073	11.760	0.01617	0.00870	0.02487	0.02	0.0000	0.18050	32
33	0.003947	12.414	0.078	12.492	7.927	4.243	12.170	0.01665	0.00905	0.02570	1.03	0.0020	0.18791	33
34	0.004109	12.439	0.082	12.521	8.167	4.420	12.587	0.01714	0.00941	0.02655	2.04	0.0041	0.19559	34
35	0.004277	12.464	0.086	12.550	8.408	4.602	13.010	0.01763	0.00977	0.02740	3.05	0.0061	0.20356	35
36	0.004452	12.490	0.089	12.579	8.648	4.793	13.441	0.01811	0.01016	0.02827	4.05	0.0081	0.21181	36
37	0.004633	12.515	0.093	12.608	8.888	4.990	13.878	0.01860	0.01055	0.02915	5.06	0.0102	0.22035	37
38	0.004820	12.540	0.097	12.637	9.128	5.194	14.322	0.01908	0.01096	0.03004	6.06	0.0122	0.22920	38
39	0.005014	12.566	0.101	12.667	9.369	5.404	14.773	0.01956	0.01139	0.03095	7.07	0.0142	0.23835	39
40	0.005216	12.591	0.105	12.696	9.609	5.624	15.233	0.02004	0.01183	0.03187	8.07	0.0162	0.24784	40
41	0.005424	12.616	0.110	12.726	9.849	5.851	15.700	0.02052	0.01229	0.03281	9.08	0.0182	0.25765	41
42	0.005640	12.641	0.115	12.756	10.089	6.086	16.175	0.02100	0.01275	0.03375	10.08	0.0202	0.26781	42
43	0.005863	12.667	0.119	12.786	10.330	6.330	16.660	0.02148	0.01324	0.03472	11.09	0.0222	0.27831	43
44	0.006094	12.692	0.124	12.816	10.570	6.582	17.152	0.02196	0.01374	0.03570	12.09	0.0242	0.28918	44
45	0.006334	12.717	0.129	12.846	10.810	6.843	17.653	0.02244	0.01425	0.03669	13.09	0.0262	0.30042	45
46	0.006581	12.743	0.134	12.877	11.050	7.114	18.164	0.02291	0.01479	0.03770	14.10	0.0282	0.31206	46
47	0.006838	12.768	0.140	12.908	11.291	7.394	18.685	0.02339	0.01534	0.03873	15.10	0.0302	0.32408	47
48	0.007103	12.793	0.146	12.939	11.531	7.684	19.215	0.02386	0.01592	0.03978	16.10	0.0321	0.33651	48
49	0.007378	12.818	0.152	12.970	11.771	7.985	19.756	0.02433	0.01651	0.04084	17.10	0.0341	0.34937	49
50	0.007661	12.844	0.157	13.001	12.012	8.294	20.306	0.02480	0.01712	0.04192	18.11	0.0361	0.36264	50
51	0.007955	12.869	0.164	13.033	12.252	8.616	20.868	0.02528	0.01774	0.04302	19.11	0.0381	0.37636	51
52	0.008259	12.894	0.171	13.065	12.492	8.949	21.441	0.02575	0.01840	0.04415	20.11	0.0400	0.39054	52
53	0.008573	12.920	0.177	13.097	12.732	9.293	22.025	0.02622	0.01907	0.04529	21.11	0.0420	0.40518	53
54	0.008897	12.945	0.184	13.129	12.973	9.648	22.621	0.02668	0.01977	0.04645	22.11	0.0439	0.42030	54
55	0.009233	12.970	0.192	13.162	13.213	10.016	23.229	0.02715	0.02048	0.04763	23.11	0.0459	0.43592	55
56	0.009580	12.995	0.200	13.195	13.453	10.397	23.850	0.02762	0.02122	0.04884	24.11	0.0478	0.45205	56
57	0.009938	13.021	0.207	13.228	13.694	10.790	24.484	0.02808	0.02198	0.05006	25.11	0.0497	0.46870	57
58	0.010309	13.046	0.216	13.262	13.934	11.197	25.131	0.02855	0.02277	0.05132	26.11	0.0517	0.48589	58
59	0.010692	13.071	0.224	13.295	14.174	11.618	25.792	0.02901	0.02358	0.05259	27.11	0.0536	0.50363	59

Thermodynamic Properties of Moist Air: U.S. Units

(STANDARD ATMOSPHERIC PRESSURE, 29.921 in. Hg)

		VOLUME (ft^3/lb)			ENTHALPY (Btu/lb)			ENTROPY (Btu/lb_{da}/°F)			CONDENSED WATER		VAPOR PRESS.	
TEMP °F	w_s lbw/lba	Sat. Liquid v_a	Evap. v_{as}	Sat. Vapor v_s	Sat. Liquid h_a	Evap. h_{as}	Sat. Vapor h_s	Sat. Liquid s_a	Evap. s_{as}	Sat. Vapor s_s	HW Btu/lb	S_w Btu/lb/°F	p_s in. Hg	TEMP °F
60	0.011087	13.096	0.233	13.329	14.415	12.052	26.467	0.02947	0.02442	0.05389	28.11	0.0555	0.52193	60
61	0.011496	13.122	0.242	13.364	14.655	12.502	27.157	0.02994	0.02528	0.05522	29.12	0.0575	0.54082	61
62	0.011919	13.147	0.251	13.398	14.895	12.967	27.862	0.03040	0.02617	0.05657	30.11	0.0594	0.56032	62
63	0.012355	13.172	0.261	13.433	15.135	13.447	28.582	0.03086	0.02709	0.05795	31.11	0.0613	0.58041	63
64	0.012805	13.198	0.270	13.468	15.376	13.942	29.318	0.03132	0.02804	0.05936	32.11	0.0632	0.60113	64
65	0.013270	13.223	0.281	13.504	15.616	14.455	30.071	0.03178	0.02902	0.06080	33.11	0.0651	0.62252	65
66	0.013750	13.248	0.292	13.540	15.856	14.984	30.840	0.03223	0.03003	0.06226	34.11	0.0670	0.64454	66
67	0.014246	13.273	0.304	13.577	16.097	15.529	31.626	0.03269	0.03107	0.06376	35.11	0.0689	0.66725	67
68	0.014758	13.299	0.314	13.613	16.337	16.094	32.431	0.03315	0.03214	0.06529	36.11	0.0708	0.69065	68
69	0.015286	13.324	0.326	13.650	16.577	16.677	33.254	0.03360	0.03325	0.06685	37.11	0.0727	0.71479	69
70	0.015832	13.349	0.339	13.688	16.818	17.279	34.097	0.03406	0.03438	0.06844	38.11	0.0746	0.73966	70
71	0.016395	13.375	0.351	13.726	17.058	17.901	34.959	0.03451	0.03556	0.07007	39.11	0.0765	0.76528	71
72	0.016976	13.400	0.364	13.764	17.299	18.542	35.841	0.03496	0.03677	0.07173	40.11	0.0783	0.79167	72
73	0.017575	13.425	0.378	13.803	17.539	19.204	36.743	0.03541	0.03802	0.07343	41.11	0.0802	0.81882	73
74	0.018194	13.450	0.393	13.843	17.779	19.889	37.668	0.03586	0.03930	0.07516	42.11	0.0821	0.84684	74
75	0.018833	13.476	0.406	13.882	18.020	20.595	38.615	0.03631	0.04063	0.07694	43.11	0.0840	0.87567	75
76	0.019491	13.501	0.422	13.923	18.260	21.323	39.583	0.03676	0.04199	0.07875	44.10	0.0858	0.90533	76
77	0.020170	13.526	0.437	13.963	18.500	22.076	40.576	0.03721	0.04339	0.08060	45.10	0.0877	0.93589	77
78	0.020871	13.551	0.454	14.005	18.741	22.851	41.592	0.03766	0.04484	0.08250	46.10	0.0896	0.96733	78
79	0.021594	13.577	0.469	14.046	18.981	23.652	42.633	0.03811	0.04633	0.08444	47.10	0.0914	0.99970	79
80	0.022340	13.602	0.487	14.089	19.222	24.479	43.701	0.03855	0.04787	0.08642	48.10	0.0933	1.03302	80
81	0.023109	13.627	0.505	14.132	19.462	25.332	44.794	0.03900	0.04944	0.08844	49.10	0.0951	1.06728	81
82	0.023902	13.653	0.522	14.175	19.702	26.211	45.913	0.03944	0.05108	0.09052	50.10	0.0970	1.10252	82
83	0.024720	13.678	0.542	14.220	19.943	27.119	47.062	0.03986	0.05278	0.09264	51.09	0.0988	1.13882	83
84	0.025563	13.703	0.561	14.264	20.183	28.055	48.238	0.04033	0.05448	0.09481	52.09	0.1006	1.17608	84
85	0.026433	13.728	0.582	14.310	20.424	29.021	49.445	0.04077	0.05626	0.09703	53.09	0.1025	1.21445	85
86	0.027329	13.754	0.602	14.356	20.664	30.017	50.681	0.04121	0.05809	0.09930	54.09	0.1043	1.25388	86
87	0.028254	13.779	0.624	14.403	20.905	31.044	51.949	0.04165	0.05998	0.10163	55.09	0.1061	1.29443	87
88	0.029208	13.804	0.646	14.450	21.145	32.105	53.250	0.04209	0.06192	0.10401	56.09	0.1080	1.33613	88
89	0.030189	13.829	0.669	14.498	21.385	33.197	54.582	0.04253	0.06392	0.10645	57.09	0.1098	1.37893	89
90	0.031203	13.855	0.692	14.547	21.626	34.325	55.951	0.04297	0.06598	0.10895	58.08	0.1116	1.42298	90
91	0.032247	13.880	0.717	14.597	21.866	35.489	57.355	0.04340	0.06810	0.11150	59.08	0.1134	1.46824	91
92	0.033323	13.905	0.742	14.647	22.107	36.687	58.794	0.04384	0.07028	0.11412	60.08	0.1152	1.51471	92
93	0.034433	13.930	0.769	14.699	22.347	37.924	60.271	0.04427	0.07253	0.11680	61.08	0.1170	1.56248	93
94	0.035577	13.956	0.795	14.751	22.588	39.199	61.787	0.04471	0.07484	0.11955	62.08	0.1188	1.61154	94
95	0.036757	13.981	0.823	14.804	22.828	40.515	63.343	0.04514	0.07723	0.12237	63.08	0.1206	1.66196	95
96	0.037972	14.006	0.852	14.858	23.069	41.871	64.940	0.04558	0.07967	0.12525	64.07	0.1224	1.71372	96
97	0.039225	14.032	0.881	14.913	23.309	43.269	66.578	0.04601	0.08220	0.12821	65.07	0.1242	1.76685	97
98	0.040516	14.057	0.912	14.969	23.550	44.710	68.260	0.04644	0.08480	0.13124	66.07	0.1260	1.82141	98
99	0.041848	14.082	0.944	15.026	23.790	46.198	69.988	0.04687	0.08747	0.13434	67.07	0.1278	1.87745	99
100	0.043219	14.107	0.977	15.084	24.031	47.730	71.761	0.04730	0.09022	0.13752	68.07	0.1296	1.93492	100
101	0.044634	14.133	1.010	15.143	24.271	49.312	73.583	0.04773	0.09306	0.14079	69.07	0.1314	1.99396	101
102	0.046090	14.158	1.045	15.203	24.512	50.940	75.452	0.04816	0.09597	0.14413	70.06	0.1332	2.05447	102
103	0.047592	14.183	1.081	15.264	24.752	52.621	77.373	0.04859	0.09897	0.14756	71.06	0.1349	2.11661	103
104	0.049140	14.208	1.118	15.326	24.993	54.353	79.346	0.04901	0.10207	0.15108	72.06	0.1367	2.18037	104

Thermodynamic Properties of Moist Air: U.S. Units

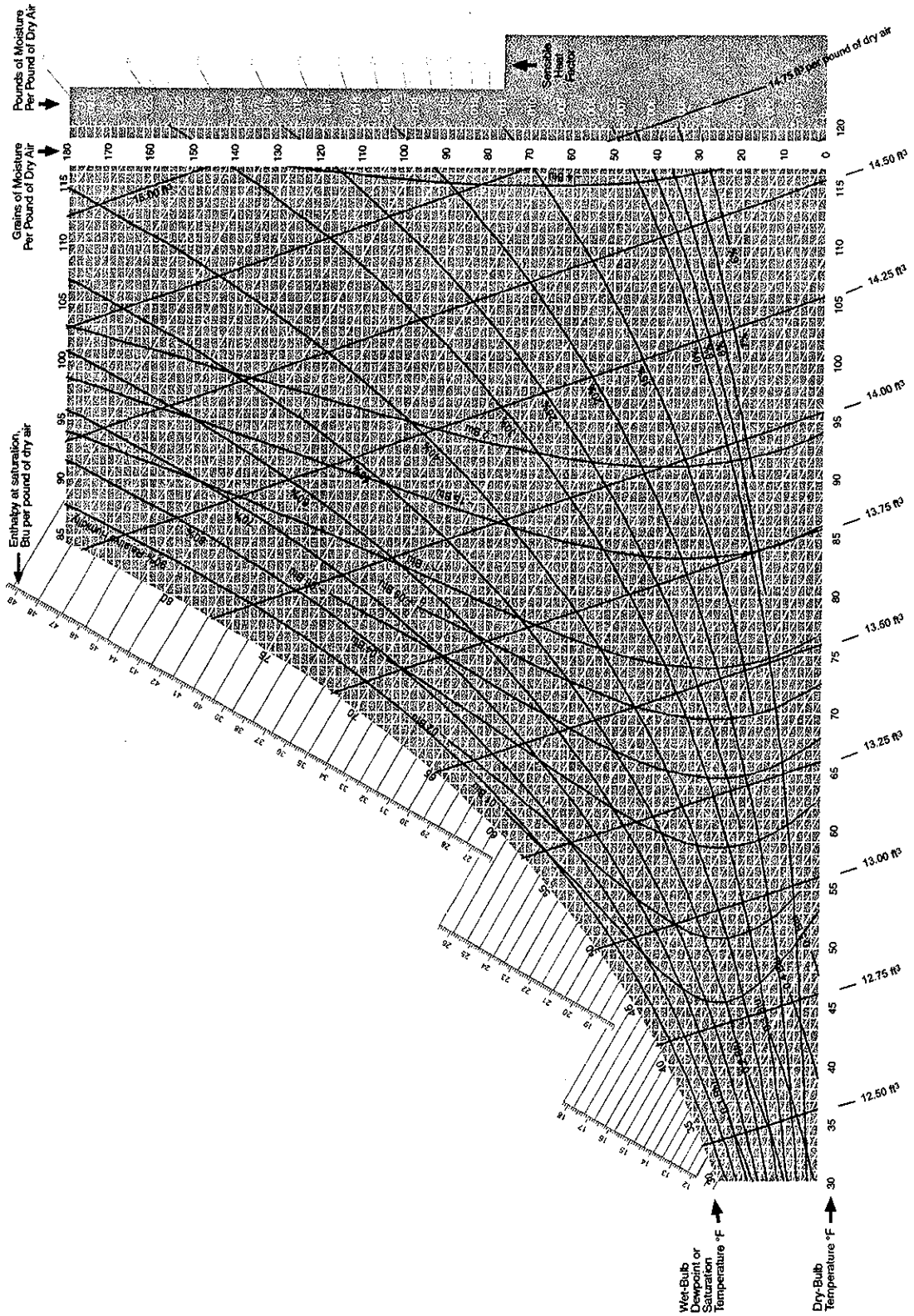
(STANDARD ATMOSPHERIC PRESSURE, 29.921 in. Hg)

		VOLUME (ft^3/lb)			ENTHALPY (Btu/lb)			ENTROPY (Btu/lb_{da}/°F)			CONDENSED WATER		VAPOR PRESS.	
TEMP °F	w_s lbw/lba	Sat. Liquid v_a	Evap. v_{as}	Sat. Vapor v_s	Sat. Liquid h_a	Evap. h_{as}	Sat. Vapor h_s	Sat. Liquid s_a	Evap. s_{as}	Sat. Vapor s_s	HW Btu/lb	S_W Btu/lb/°F	p_s in. Hg	TEMP °F
105	0.050737	14.234	1.156	15.390	25.233	56.142	81.375	0.04944	0.10525	0.15469	73.06	0.1385	2.24581	105
106	0.052383	14.259	1.196	15.455	25.474	57.986	83.460	0.04987	0.10852	0.15839	74.06	0.1402	2.31297	106
107	0.054077	14.284	1.237	15.521	25.714	59.885	85.599	0.05029	0.11189	0.16218	75.06	0.1420	2.38173	107
108	0.055826	14.309	1.279	15.588	25.955	61.844	87.799	0.05071	0.11537	0.16608	76.05	0.1438	2.45232	108
109	0.057628	14.335	1.322	15.657	26.195	63.866	90.061	0.05114	0.11894	0.17008	77.05	0.1455	2.52473	109
110	0.059486	14.360	1.367	15.727	26.436	65.950	92.386	0.05156	0.12262	0.17418	78.05	0.1473	2.59891	110
111	0.061401	14.385	1.414	15.799	26.677	68.099	94.776	0.05198	0.12641	0.17839	79.05	0.1490	2.67500	111
112	0.063378	14.411	1.461	15.872	26.917	70.320	97.237	0.05240	0.13032	0.18272	80.05	0.1508	2.75310	112
113	0.065411	14.436	1.511	15.947	27.158	72.602	99.760	0.05282	0.13434	0.18716	81.05	0.1525	2.83291	113
114	0.067512	14.461	1.562	16.023	27.398	74.964	102.362	0.05324	0.13848	0.19172	82.04	0.1543	2.91491	114
115	0.069676	14.486	1.615	16.101	27.639	77.396	105.035	0.05366	0.14274	0.19640	83.04	0.1560	2.99883	115
116	0.071908	14.512	1.669	16.181	27.879	79.907	107.786	0.05408	0.14713	0.20121	84.04	0.1577	3.08488	116
117	0.074211	14.537	1.726	16.263	28.120	82.497	110.617	0.05450	0.15165	0.20615	85.04	0.1595	3.17305	117
118	0.076586	14.562	1.784	16.346	28.361	85.169	113.530	0.05492	0.15630	0.21122	86.04	0.1612	3.26335	118
119	0.079036	14.587	1.845	16.432	28.601	87.927	116.528	0.05533	0.16111	0.21644	87.04	0.1629	3.35586	119
120	0.081560	14.613	1.906	16.519	28.842	90.770	119.612	0.05575	0.16605	0.22180	88.04	0.1647	3.45052	120
121	0.084169	14.638	1.971	16.609	29.083	93.709	122.792	0.05616	0.17115	0.22731	89.04	0.1664	3.54764	121
122	0.086860	14.663	2.037	16.700	29.323	96.742	126.065	0.05658	0.17640	0.23298	90.03	0.1681	3.64704	122
123	0.089633	14.688	2.106	16.794	29.564	99.868	129.432	0.05699	0.18181	0.23880	91.03	0.1698	3.74871	123
124	0.092500	14.714	2.176	16.890	29.805	103.102	132.907	0.05740	0.18740	0.24480	92.03	0.1715	3.85298	124
125	0.095456	14.739	2.250	16.989	30.045	106.437	136.482	0.05781	0.19315	0.25096	93.03	0.1732	3.95961	125
126	0.098504	14.764	2.326	17.090	30.286	109.877	140.163	0.05823	0.19906	0.25729	94.03	0.1749	4.06863	126
127	0.101657	14.789	2.404	17.193	30.527	113.438	143.965	0.05864	0.20518	0.26382	95.03	0.1766	4.18046	127
128	0.104910	14.815	2.484	17.299	30.767	117.111	147.878	0.05905	0.21149	0.27054	96.03	0.1783	4.29477	128
129	0.108270	14.840	2.569	17.409	31.008	120.908	151.916	0.05946	0.21799	0.27745	97.03	0.1800	4.41181	129
130	0.111738	14.865	2.655	17.520	31.249	124.827	156.076	0.05986	0.22471	0.28457	98.03	0.1817	4.53148	130
131	0.115322	14.891	2.744	17.635	31.489	128.881	160.370	0.06027	0.23163	0.29190	99.02	0.1834	4.65397	131
132	0.119023	14.916	2.837	17.753	31.730	133.066	164.796	0.06068	0.23876	0.29944	100.02	0.1851	4.77919	132
133	0.122855	14.941	2.934	17.875	31.971	137.403	169.374	0.06109	0.24614	0.30723	101.02	0.1868	4.90755	133
134	0.126804	14.966	3.033	17.999	32.212	141.872	174.084	0.06149	0.25375	0.31524	102.02	0.1885	5.03844	134
135	0.130895	14.992	3.135	18.127	32.452	146.505	178.957	0.06190	0.26161	0.32351	103.02	0.1902	5.17258	135
136	0.135124	15.017	3.242	18.259	32.693	151.294	183.987	0.06230	0.26973	0.33203	104.02	0.1919	5.30973	136
137	0.139494	15.042	3.352	18.394	32.934	156.245	189.179	0.06271	0.27811	0.34082	105.02	0.1935	5.44985	137
138	0.144019	15.067	3.467	18.534	33.175	161.373	194.548	0.06311	0.28707	0.35018	106.02	0.1952	5.59324	138
139	0.148696	15.093	3.585	18.678	33.415	166.677	200.092	0.06351	0.29573	0.35924	107.02	0.1969	5.73970	139
140	0.153538	15.118	3.707	18.825	33.656	172.168	205.824	0.06391	0.30499	0.36890	108.02	0.1985	5.88945	140
141	0.158643	15.143	3.835	18.978	33.897	177.857	211.754	0.06431	0.31456	0.37887	109.02	0.2002	6.04256	141
142	0.163748	15.168	3.967	19.135	34.138	183.754	217.892	0.06471	0.32447	0.38918	110.02	0.2019	6.19918	142
143	0.169122	15.194	4.103	19.297	34.379	189.854	224.233	0.06511	0.33470	0.39981	111.02	0.2035	6.35898	143
144	0.174694	15.219	4.245	19.464	34.620	196.182	230.802	0.06551	0.34490	0.41081	112.02	0.2052	6.52241	144
145	0.180467	15.244	4.393	19.637	34.860	202.740	237.600	0.06591	0.35587	0.42218	113.02	0.2068	6.68932	145
146	0.186460	15.269	4.546	19.815	35.101	209.550	244.651	0.06631	0.36724	0.43395	114.02	0.2085	6.86009	146
147	0.192668	15.295	4.704	19.999	35.342	216.607	251.949	0.06671	0.37861	0.44611	115.02	0.2101	7.03435	147
148	0.199110	15.320	4.869	20.189	35.583	223.931	259.514	0.06710	0.45871	0.45871	116.02	0.2118	7.21239	148
149	0.205792	15.345	5.040	20.385	35.824	231.532	267.356	0.06750	0.40384	0.47174	117.02	0.2134	7.39413	149

Thermodynamic Properties of Moist Air: U.S. Units

(STANDARD ATMOSPHERIC PRESSURE, 29.921 in. Hg)

TEMP °F	w_s lbw/lba	VOLUME (ft³/lb) Sat. Liquid v_a	Evap. v_{as}	Sat. Vapor v_s	ENTHALPY (Btu/lb) Sat. Liquid h_a	Evap. h_{as}	Sat. Vapor h_s	ENTROPY (Btu/lb$_{da}$/°F) Sat. Liquid s_a	Evap. s_{as}	Sat. Vapor s_s	CONDENSED WATER HW Btu/lb	S_W Btu/lb/°F	VAPOR PRESS. p_s in. Hg	TEMP °F
150	0.212730	15.370	5.219	20.589	36.064	239.426	275.490	0.06790	0.41656	0.48524	118.02	0.2151	7.57977	150
151	0.219945	15.396	5.403	20.799	36.305	247.638	283.943	0.06829	0.43017	0.49925	119.02	0.2167	7.76958	151
152	0.227429	15.421	5.596	21.017	36.546	256.159	292.705	0.06868	0.44507	0.51375	120.02	0.2184	7.96306	152
153	0.235218	15.446	5.797	21.243	36.787	265.029	301.816	0.06908	0.45973	0.52881	121.02	0.2200	8.16087	153
154	0.243309	15.471	6.006	21.477	37.028	274.245	311.273	0.06947	0.47494	0.54441	122.02	0.2216	8.36256	154
155	0.251738	15.497	6.223	21.720	37.269	283.849	321.118	0.06986	0.49078	0.56064	123.02	0.2233	8.56871	155
156	0.260512	15.522	6.450	21.972	37.510	293.849	331.359	0.07025	0.50724	0.57749	124.02	0.2249	8.77915	156
157	0.269644	15.547	6.686	22.233	37.751	304.261	342.012	0.07065	0.52434	0.59499	125.02	0.2265	8.99378	157
158	0.279166	15.572	6.933	22.505	37.992	315.120	353.112	0.07104	0.54216	0.61320	126.02	0.2281	9.21297	158
159	0.289101	15.598	7.190	22.788	38.233	326.452	364.685	0.07143	0.56073	0.63216	127.02	0.2297	9.43677	159
160	0.299450	15.623	7.459	23.082	38.474	338.263	376.737	0.07181	0.58007	0.65188	128.02	0.2314	9.6648	160
161	0.310270	15.648	7.740	23.388	38.715	350.610	389.325	0.07220	0.60025	0.67245	129.02	0.2330	9.8978	161
162	0.321560	15.673	8.034	23.707	38.956	363.501	402.457	0.07259	0.62129	0.69388	130.03	0.2346	10.1353	162
163	0.333360	15.699	8.341	24.040	39.197	376.978	416.175	0.07298	0.64325	0.71623	131.03	0.2362	10.3776	163
164	0.345720	15.724	8.664	24.388	39.438	391.095	430.533	0.07337	0.66622	0.73959	132.03	0.2378	10.6250	164
165	0.358650	15.749	9.001	24.750	39.679	405.865	445.544	0.07375	0.69022	0.76397	133.03	0.2394	10.8771	165
166	0.372200	15.774	9.355	25.129	39.920	421.351	461.271	0.07414	0.71535	0.78949	134.03	0.2410	11.1343	166
167	0.386390	15.800	9.726	25.526	40.161	437.578	477.739	0.07452	0.74165	0.81617	135.03	0.2426	11.3965	167
168	0.401310	15.825	10.117	25.942	40.402	454.630	495.032	0.07491	0.76924	0.84415	136.03	0.2442	11.6641	168
169	0.416980	15.850	10.527	26.377	40.643	472.554	513.197	0.07529	0.79821	0.87350	137.04	0.2458	11.9370	169
170	0.433430	15.875	10.959	26.834	40.884	491.372	532.256	0.07567	0.82858	0.90425	138.04	0.2474	12.2149	170
171	0.450790	15.901	11.414	27.315	41.125	511.231	552.356	0.07606	0.86058	0.93664	139.04	0.2490	12.4988	171
172	0.469050	15.926	11.894	27.820	41.366	532.138	573.504	0.07644	0.89423	0.97067	140.04	0.2506	12.7880	172
173	0.488290	15.951	12.401	28.352	41.607	554.160	595.767	0.07682	0.92962	1.00644	141.04	0.2521	13.0823	173
174	0.508670	15.976	12.937	28.913	41.848	577.489	619.337	0.07720	0.96707	1.04427	142.04	0.2537	13.3831	174
175	0.530190	16.002	13.503	29.505	42.089	602.140	644.229	0.07758	1.00658	1.08416	143.05	0.2553	13.6894	175
176	0.552940	16.027	14.103	30.130	42.331	628.197	670.528	0.07796	1.04828	1.12624	144.05	0.2569	14.0010	176
177	0.577100	16.052	14.741	30.793	42.572	655.876	698.448	0.07834	1.09253	1.17087	145.05	0.2585	14.3191	177
178	0.602740	16.078	15.418	31.496	42.813	685.260	728.073	0.07872	1.13943	1.21815	146.05	0.2600	14.6430	178
179	0.630020	16.103	16.139	32.242	43.054	716.525	759.579	0.07910	1.18927	1.26837	147.06	0.2616	14.9731	179
180	0.659110	16.128	16.909	33.037	43.295	749.871	793.166	0.07947	1.24236	1.32183	148.06	0.2632	15.3097	180
181	0.690120	16.153	17.730	33.883	43.536	785.426	828.962	0.07985	1.29888	1.37873	149.06	0.2647	15.6522	181
182	0.723310	16.178	18.609	34.787	43.778	823.487	867.265	0.08023	1.35931	1.43954	150.06	0.2663	16.0014	182
183	0.758850	16.204	19.551	35.755	44.019	864.259	908.278	0.08060	1.42397	1.50457	151.07	0.2679	16.3569	183
184	0.797030	16.229	20.564	36.793	44.260	908.061	952.321	0.08098	1.49332	1.57430	152.07	0.2694	16.7190	184
185	0.838170	16.254	21.656	37.910	44.501	955.262	999.763	0.08135	1.56797	1.64932	153.07	0.2710	17.0880	185
186	0.882510	16.280	22.833	39.113	44.742	1006.150	1050.892	0.08172	1.64834	1.73006	154.08	0.2725	17.4634	186
187	0.930570	16.305	24.111	40.416	44.984	1061.314	1106.298	0.08210	1.73534	1.81744	155.08	0.2741	17.8462	187
188	0.982720	16.330	25.498	41.828	45.225	1121.174	1166.399	0.08247	1.82963	1.91210	156.08	0.2756	18.2357	188
189	1.039510	16.355	27.010	43.365	45.466	1186.382	1231.848	0.08284	1.93221	2.01505	157.09	0.2772	18.6323	189
190	1.101540	16.381	28.661	45.042	45.707	1257.614	1303.321	0.08321	2.04412	2.12733	158.09	0.2787	19.0358	190
191	1.169650	16.406	30.476	46.882	45.949	1335.834	1381.783	0.08359	2.16684	2.25043	159.09	0.2803	19.4468	191
192	1.244710	16.431	32.477	48.908	46.190	1422.048	1468.238	0.08396	2.30193	2.38589	160.10	0.2818	19.8652	192
193	1.327880	16.456	34.695	51.151	46.431	1517.582	1564.013	0.08433	2.45143	2.53576	161.10	0.2834	20.2913	193
194	1.420290	16.481	37.161	53.642	46.673	1623.757	1670.430	0.08470	2.61738	2.70208	162.11	0.2849	20.7244	194
195	1.523960	16.507	39.928	56.435	46.914	1742.879	1789.793	0.08506	2.80332	2.88838	163.11	0.2864	21.1661	195
196	1.640700	16.532	43.046	59.578	47.155	1877.033	1924.188	0.08543	3.01244	3.09787	164.12	0.2880	21.6152	196
197	1.772990	16.557	46.580	63.137	47.397	2029.069	2076.466	0.08580	3.24914	3.33494	165.12	0.2895	22.0714	197
198	1.924720	16.583	50.635	67.218	47.638	2203.464	2251.102	0.08617	3.52030	3.60647	166.13	0.2910	22.5367	198
199	2.099750	16.608	55.315	71.923	47.879	2404.668	2452.547	0.08653	3.83276	3.91929	167.13	0.2926	23.0092	199
200	2.304540	16.633	60.793	77.426	48.121	2640.084	2688.205	0.08690	4.19787	4.28477	168.13	0.2941	23.4906	200



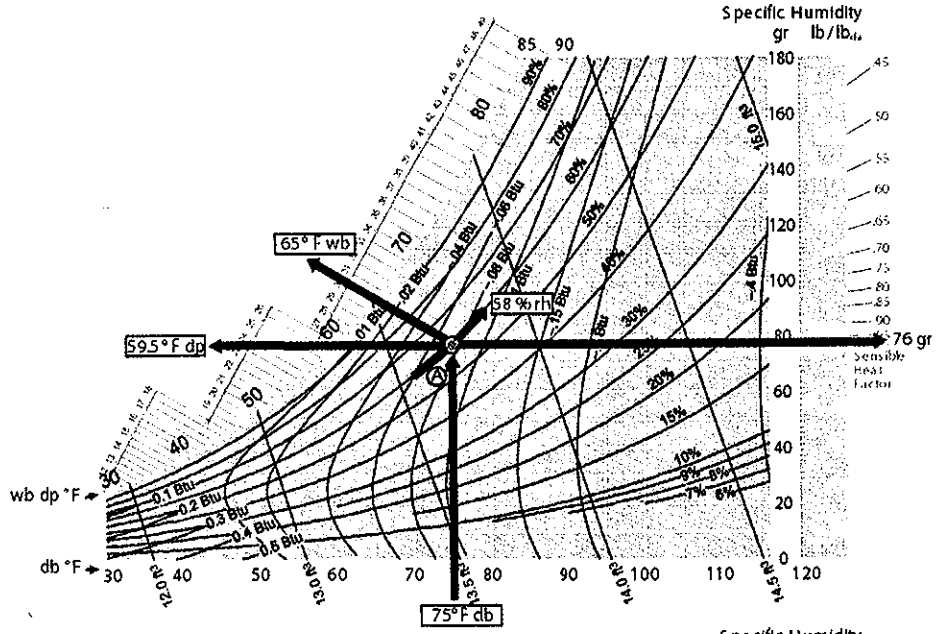
Psychrometric Chart, Normal Temperature, Sea Level

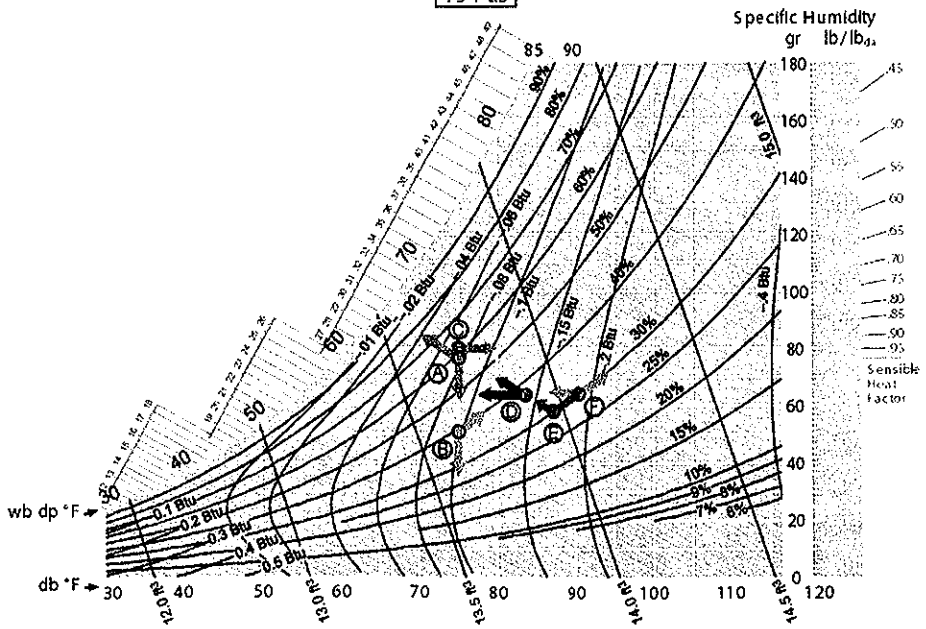
Work Session 1 Answers

1.

	db	wb	% rh	dp	W
A	75	65	**58**	**59.5**	**76**
B	75	**59.8**	40	**49.5**	**52.5**
C	75	**65.7**	**61**	**61.8**	80
D	**82.5**	65	**38**	55	**64.5**
E	**86.7**	65	30	**51.5**	**57**
F	**90.5**	**67.5**	30	55	**64.5**

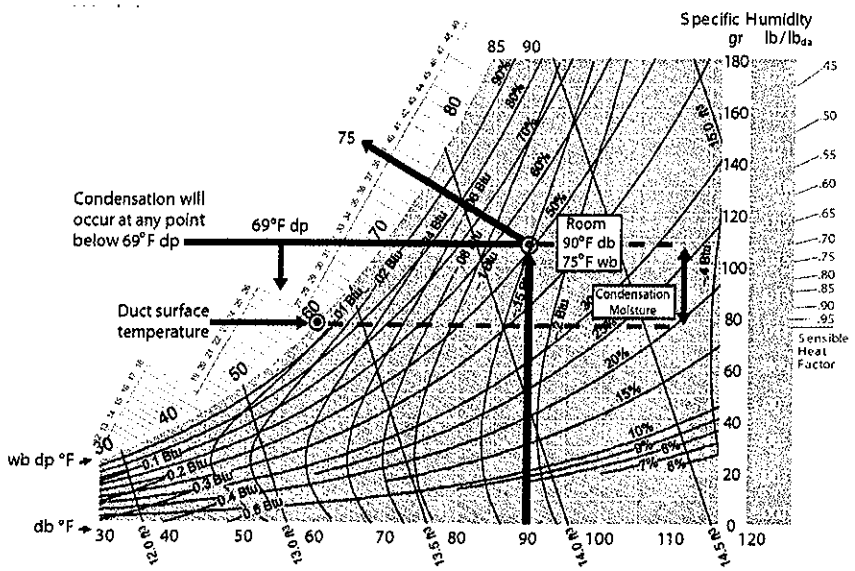
W= specific humidity lb/lb dry air



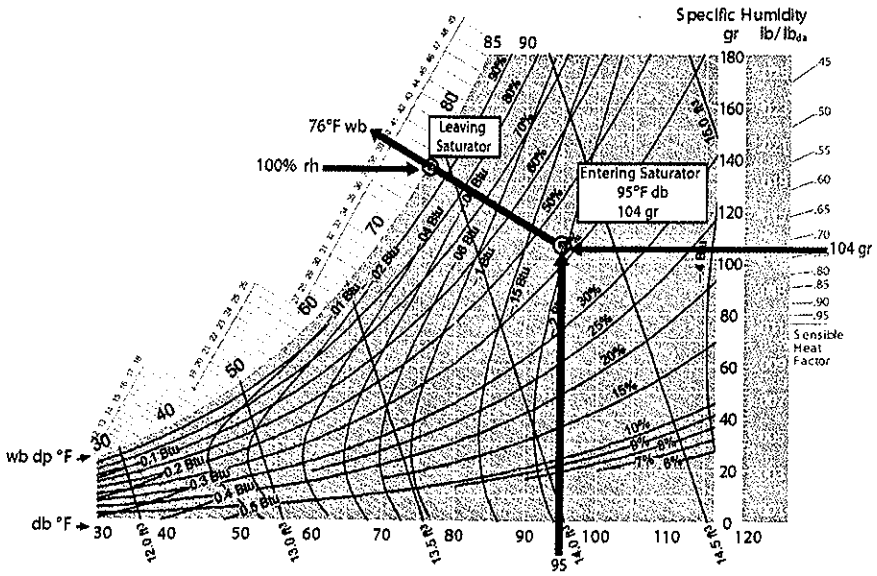


2. a. Yes

   b. Air at 90° F db and 75° F wb has a dew point of approximately 69° F. Any object in the space with a temperature of less than 69° F will cool the moisture below its saturation temperature and condensation will occur.

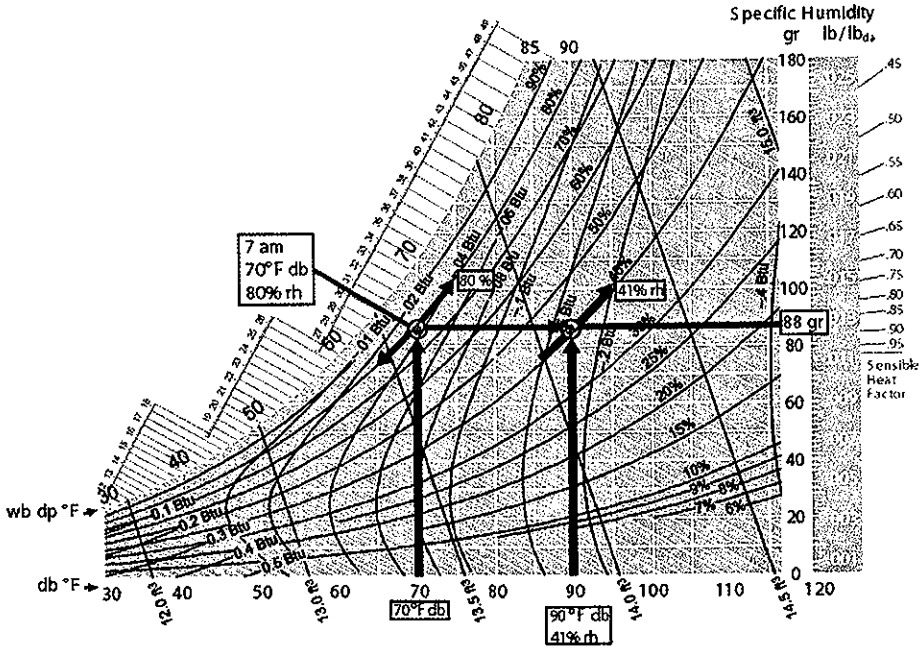


3. The air vapor mixture leaving the saturator will be saturated at 76° F. The relative humidity at saturation is 100%.

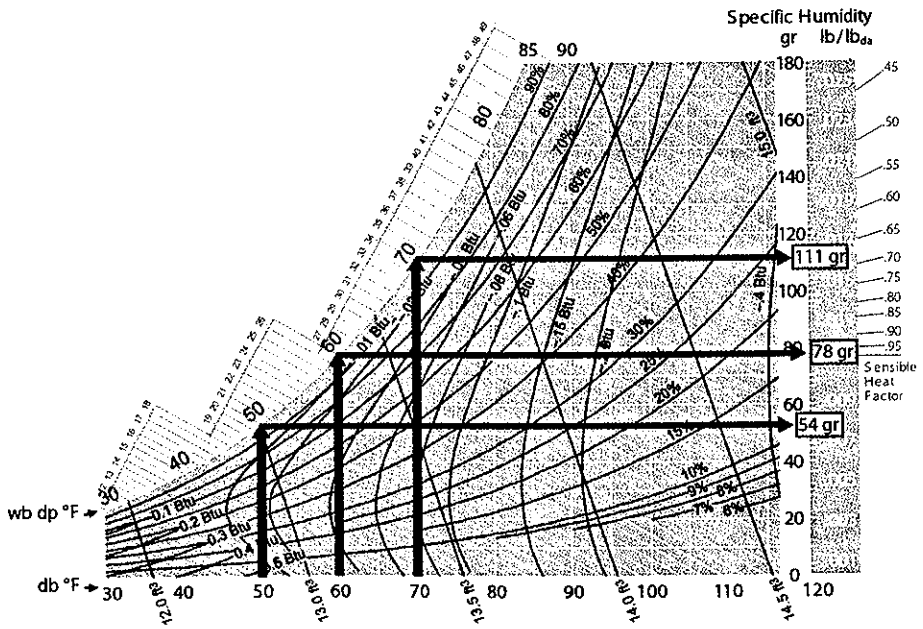


4. The dew point at 70° F and 30% relative humidity is 37° F. Water vapor will go from the higher dew point area to the lower one. Therefore, the vapor will try to pass through the wall and will condense within the wall when cooled below the dew point temperature. A vapor seal on the warm side of the wall is needed.

5. When the temperature is 90° F, the relative humidity is 41%. If there is no precipitation, the specific humidity, remains constant through out the day. The relative humidity will vary during the day with the dry bulb.

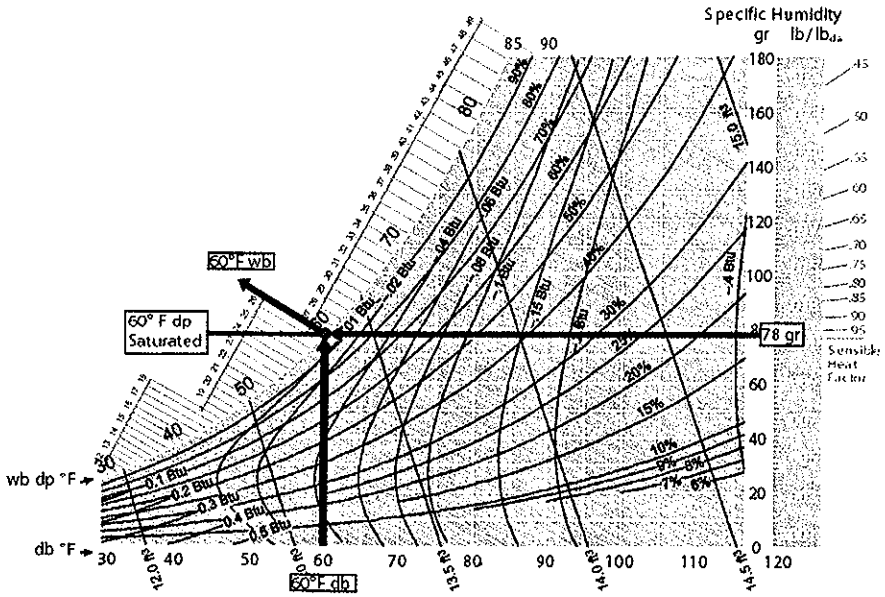


6.



7. What is the db temperature? 60° F
   What is the wb temperature? 60° F
   What is the dp temperature? 60° F

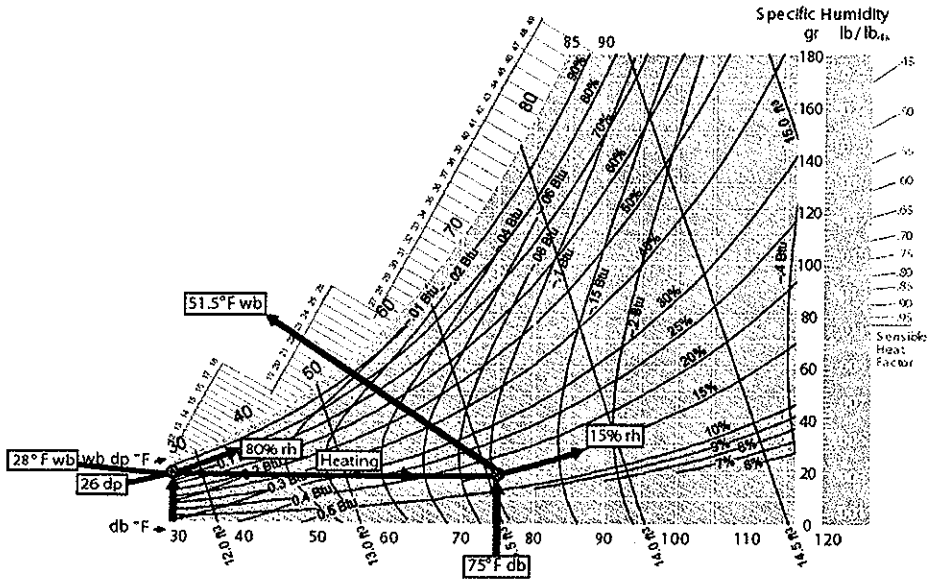
   When the vapor in an air-vapor mixture is saturated, the dry bulb, wet bulb and dew point temperature establishes the amount of water vapor present.



Work Session 2 Answers

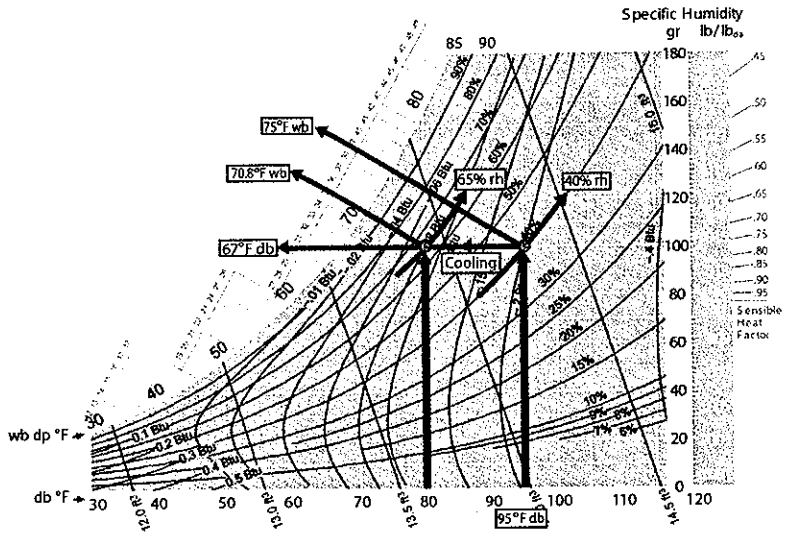
1. When air is heated with no moisture added, the process is a horizontal line on the chart as from point 1 to 2. Heating air at constant moisture content reduces the relative humidity.

	db	wb	% rh	dp
Air at	30	**28**	80	**26**
Heated to	75	**51.5**	**15**	**26**



2. When air is cooled without condensing moisture, the process is a horizontal line on the chart as from point 1 to 2. Cooling air at constant moisture content increases the relative humidity.

	db	wb	% rh	dp
Air at	95	75	**40**	**67**
Cooled to	80	**70.8**	**65**	**67**

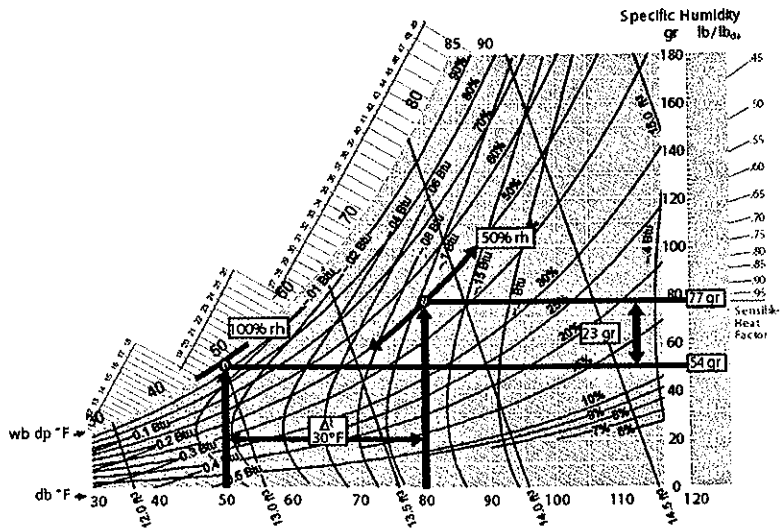


3. Sensible Heat Removed = 1.10 * cfm * temperature change
= 1.10 * 1,000 cfm * (80° F-50° F)
= 1.10 * 1,000 cfm * 30 = 33,000 Btuh

The factor 1.10 used in the sensible heat formula is a combination figure, which converts cfm to lbs per hour, and also the amount of heat required to raise 1 lb of air 1 degree Fahrenheit.

Latent Heat Removed = 0.69 * cfm * grains of moisture removed
= 0.69 * 1,000 cfm * (77 gr – 54 gr)
= 0.69 * 1,000 cfm * 23 = 15,870 Btuh

The factor 0.69 used in the latent heat formula is also a combination figure, which converts cfm to lbs per hour, and also takes into account the amount of heat removed to condense moisture.

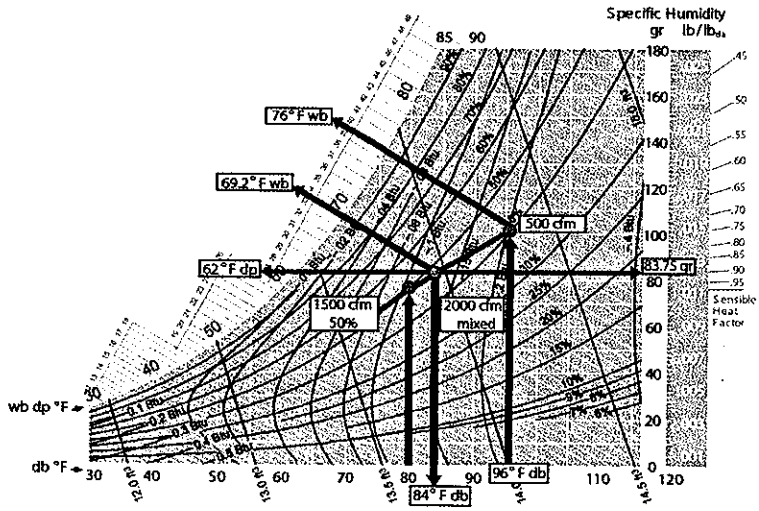


4.

a.	Dry bulb	84° F
b.	Wet bulb	69.2° F
c.	Dew point	62° F
d.	Specific humidity	83.75 gr/lb

The mixture will be closest to the larger air quantity. In this case, it is the 80° F db condition. The total cfm is in the mixture is 500 + 1500 or 2000 cfm. Five hundred cfm of outdoor air represents 500/2000 or ¼ of the mixture. Therefore, the mixture will be located ¼ of the way up from point 1 on line 1-2. The temperature difference between point 1 (80° F db) and point 2 (96° F db) is 96-80 or 16 degrees. One-quarter of 16 is 4 degrees. Therefore, the mixture is at 80° F +4° F or 84° F db or point 3.

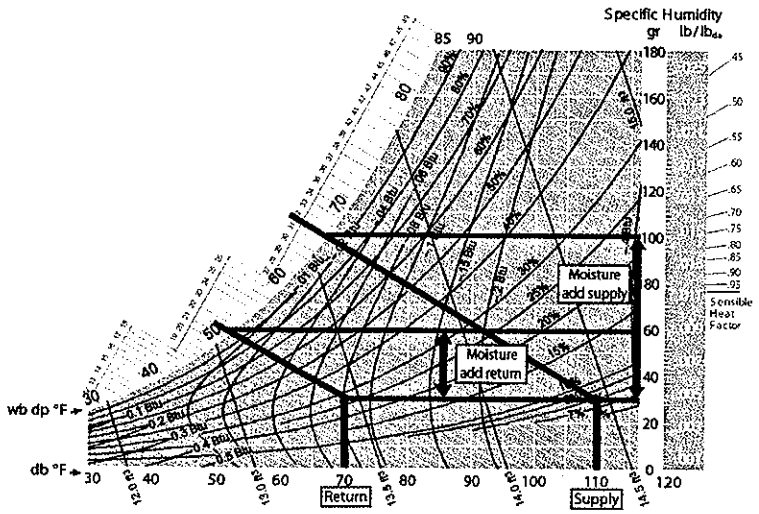
By locating 84° F on line 1-2, we can determine the other properties by using the chart.



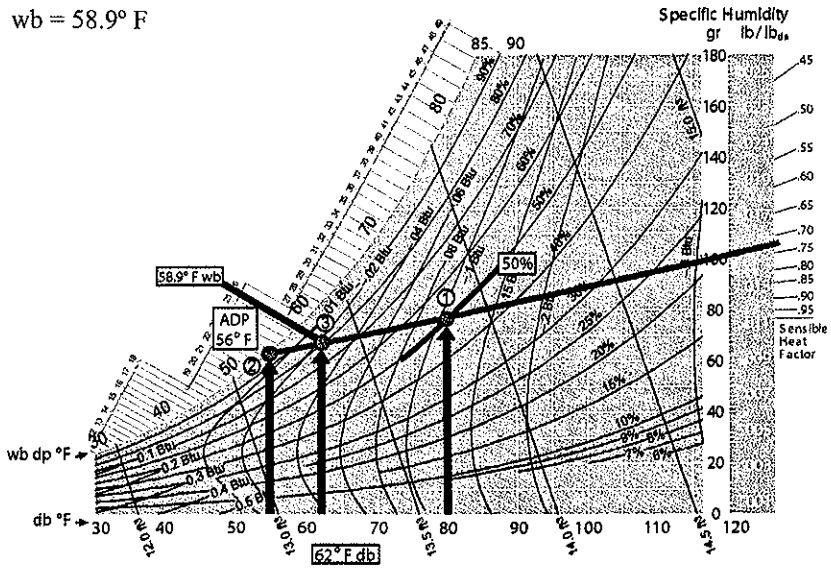
Note: The temperature of the mixture can also be determined by calculation as follows:

$$t_m = \frac{(cfm_1 \times t_1) + (cfm_2 \times t_2)}{cfm_1 + cfm_2} = \frac{(1500 \times 80) + (500 \times 96)}{1500 + 500} = \frac{120{,}000 + 48{,}000}{2000} = \frac{168}{2} = 84°F$$

5. The humidifier should be located on the supply side in the warm air from the furnace. Warm air has a greater capacity for moisture than cool air. Therefore, water evaporates more readily in the warm air stream. See the chart below. On the return side, the air can only absorb (60 gr – 30 gr) on the supply (100 gr – 30 gr) or 70 gr.



6. db = 62° F wb = 58.9° F



The bypass factor of 0.25 means that 75% of the 80° F db and 50% rh will contact the coil and be cooled to 56° F saturation (point 2), 25% of the air at point 1 will pass through the coil unchanged. The air leaving the coil will be a mixture of air at these two conditions.

db leaving the coil = 56° F +1/4(80° F+56° F)
= 56 + 6 = 62° F (point 3)

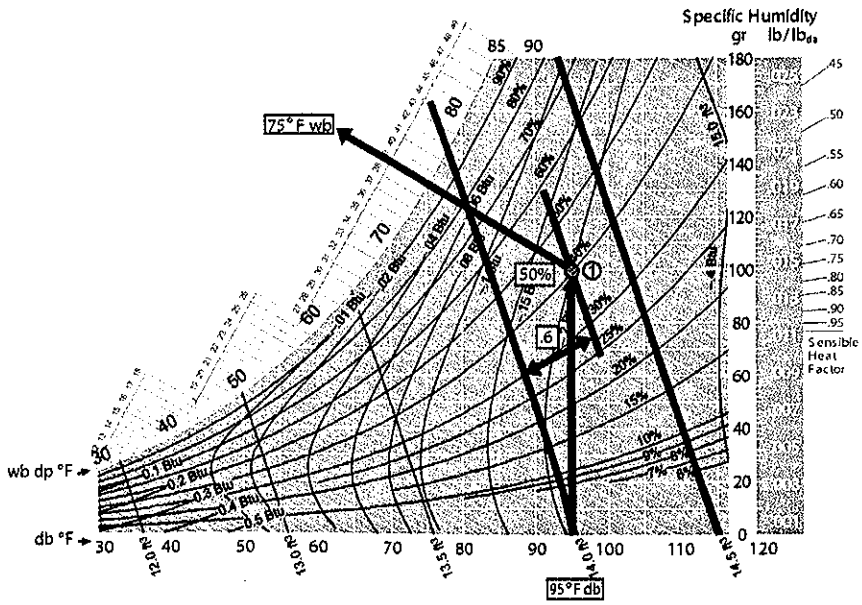
From the chart, the wb leaving the coil is 58.9° F.

This could also be checked by using the mixing equation: $t_m = \frac{(0.75 * 56) + (0.25 * 80)}{1.0}$

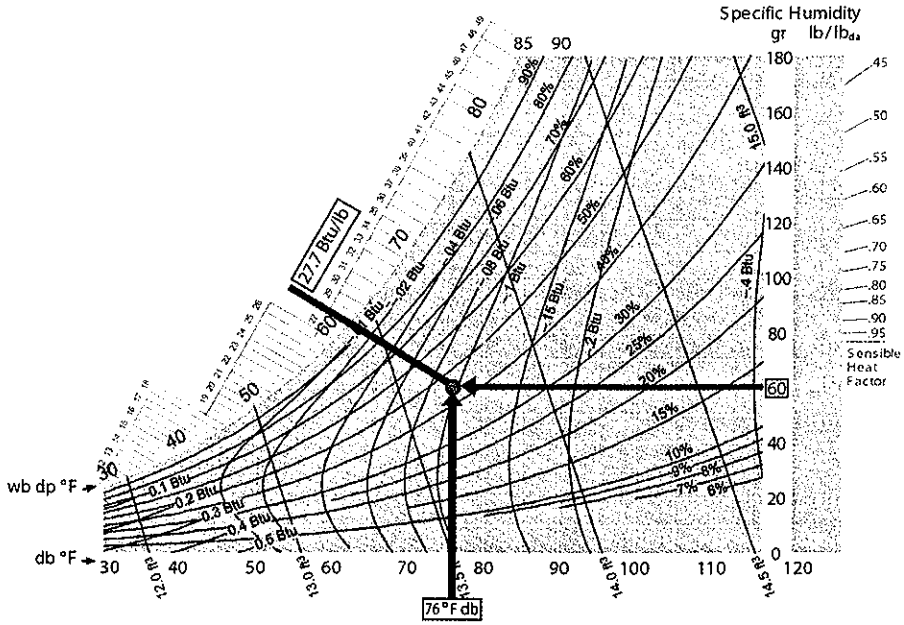
$t_m = 42 + 20 = 62$

7. Air at 95° F db and 75° F wb falls between the 14 and 14.5 cu ft/lb volume lines. Point 1 is estimated to be 0.6 of the distance to the left of the 14.0 cu ft line.

Volume at point 1 = 14 + 0.6 x (14.5-14) = 14.3 cu ft/lb



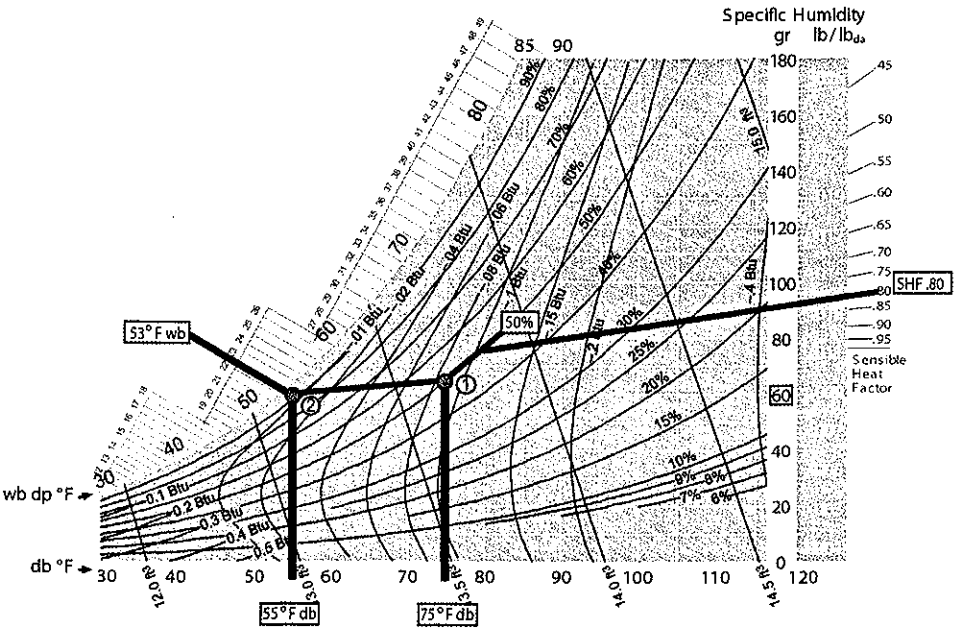
8. To determine enthalpy of air at condition 1, follow from point 1 parallel to a wb line and read enthalpy Btu/lb dry air on the scale to the left of the saturation line. In this case it is 27.7 Btu/lb_{da}. Per the recommendation on page 13, we have not included a correction for enthalpy deviation.



9. Draw a line connecting points 1 and 2. Then draw a line parallel to line 1-2 through the pivot point dot at 80° F db and 50% rh to the sensible heat factor scale. Read 0.8.

   Room sensible heat is 80% of the room total heat.

   Room latent is 100% -80% or 20% of the room total heat.



Glossary

adiabatic process	thermodynamic process during which no heat is extracted from or added to the system
Dalton's Law of Partial Pressure	each constituent of a mixture of gases behaves thermodynamically as if it alone occupied the space. The sum of the individual pressures of the constituents equals the total pressure of the mixture.
dehumidification	removal of water vapor from air.
dew point	temperature at which water reaches the saturation point (100% relative humidity)
dry bulb temperature	temperature of air on a thermometer that measures sensible heat energy
enthalpy	thermodynamic quantity equal to the sum of the internal energy of a system plus the product of the pressure-volume work done on the system. $h = E + pv$, where h = enthalpy or total heat content, E = internal energy of the system, p = pressure, and v = volume.
humidification	the process of adding water vapor to the air
moisture content	the weight of water vapor, expressed in pounds or grains, associated with one pound of dry air. One pound of dry air is equal to 7000 grains
relative humidity	the ratio of the amount of vapor contained in the air to the greatest amount the air could hold at that temperature. Normally expressed as percentage.
superheat	extra heat in a vapor when at a temperature higher than the saturation temperature corresponding to its pressure.
wet bulb temperature	temperature indicated by a psychrometer when the bulb of one thermometer is covered with a water-saturated wick over which air is caused to flow at approximately 900 ft/min (4.5 m/s) to reach an equilibrium temperature of water evaporating into air, when the heat of vaporization is supplied by the sensible heat of the air.

Notes

Prerequisites:

This module assumes the participant has an understanding of industry terminology, basic concepts of the air conditioning, and the mechanical refrigeration process. The following TDPs are good reference for this material:

Form No.	Book Cat. No.	Instructor CD Cat. No.	Title
TDP-102	796-026	797-026	ABCs of Air Conditioning

Learning Objectives:

After completing this module, participants will be able to:

- Understand the properties of air and water vapor mixtures.
- Build the psychrometric chart.
- Use the psychrometric chart to determine the properties of an air/water vapor mixture.
- Use the psychrometric chart to understand the basic air conditioning processes.
- Understand how the processes can be combined into a system using a system plot diagram and psychrometric chart.

Supplemental Material:

Form No.	Book Cat. No.	Instructor CD Cat. No.	Title
TDP-202	796-031	797-031	Psychrometrics, Level 2: Process
TDP-203	796-032	797-032	Psychrometrics, Level 3: Application
TDP-204	796-033	797-033	Psychrometrics, Level 4: Theory
	796-018		Normal Temperature, Vinyl 11" x 17" Chart
	796-013		SHF Alignment Ruler
	796-016		Pad of 25 Paper Normal Temp 11" x 17" Charts

Instructor Information:

Each TDP topic is supported with a number of different items to meet the specific needs of the user. Instructor materials consist of a CD-ROM disk that includes a PowerPoint™ presentation with convenient links to all required support materials required for the topic. This always includes: slides, presenter notes, text file including work sessions and work session solutions, quiz and quiz answers. Depending upon the topic, the instructor CD may also include sound, video, spreadsheets, forms, or other material required to present a complete class. Self-study or student material consists of a text including work sessions and work session answers, and may also include forms, worksheets, calculators, etc.



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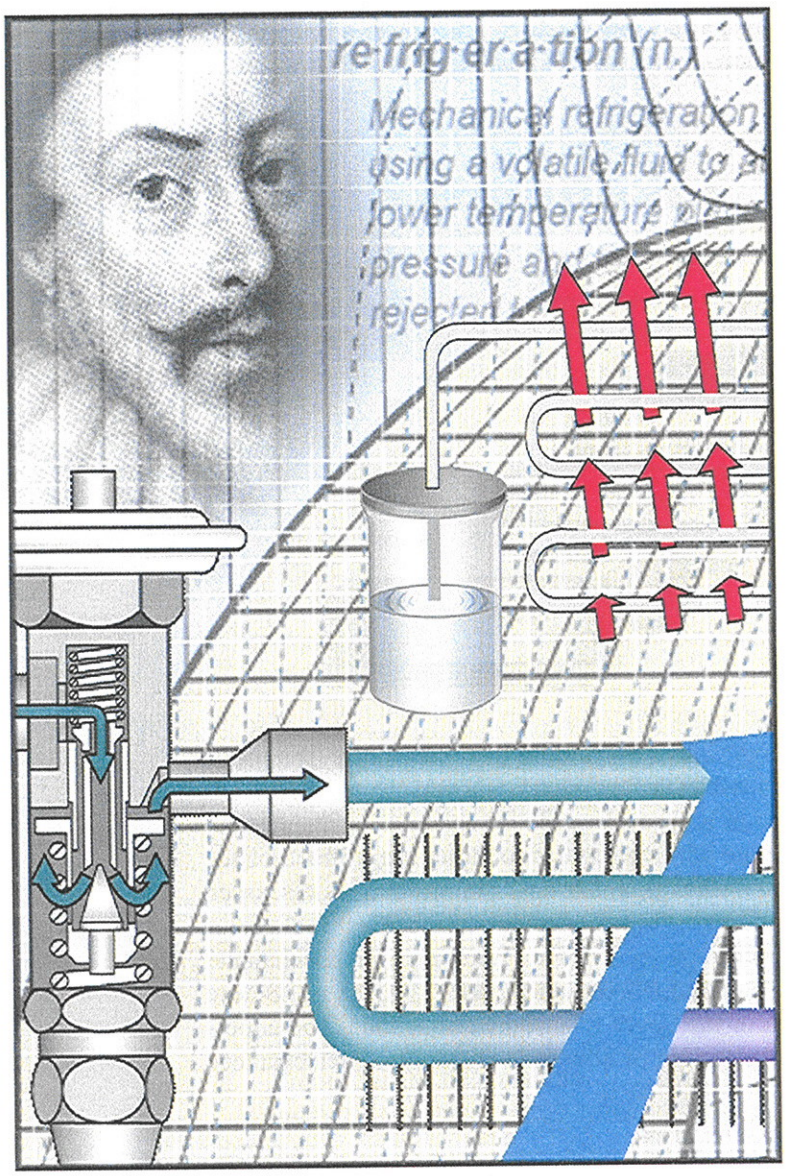


REFRIGERATION CYCLE

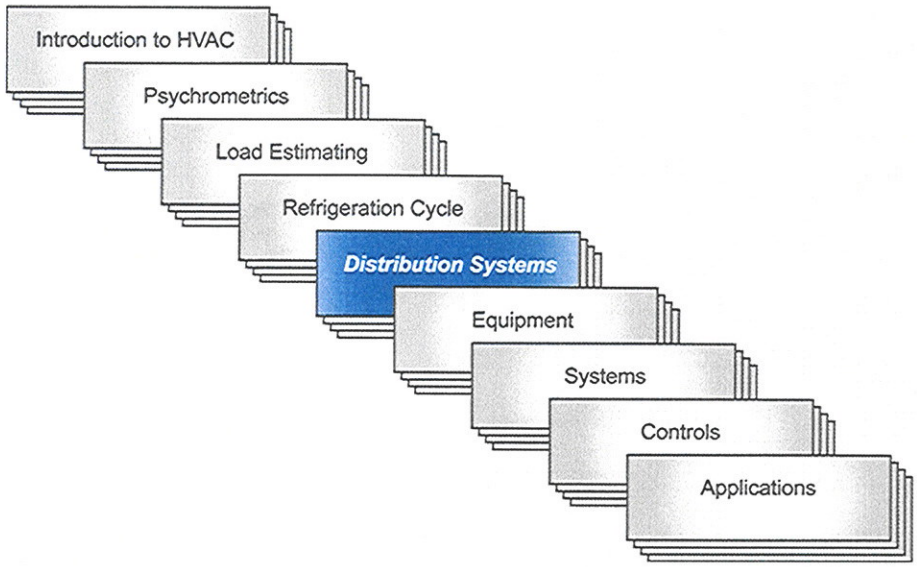
Principles of Mechanical Refrigeration

Level 1: Introduction

Technical Development Program

Technical Development Programs (TDP) are modules of technical training on HVAC theory, system design, equipment selection and application topics. They are targeted at engineers and designers who wish to develop their knowledge in this field to effectively design, specify, sell or apply HVAC equipment in commercial applications.

Although TDP topics have been developed as stand-alone modules, there are logical groupings of topics. The modules within each group begin at an introductory level and progress to advanced levels. The breadth of this offering allows for customization into a complete HVAC curriculum – from a complete HVAC design course at an introductory-level or to an advanced-level design course. Advanced-level modules assume prerequisite knowledge and do not review basic concepts.



Air conditioning is all about moving heat energy, either by adding or removing it from one place and moving it to another. This module deals with the way heat is moved from a place of lower temperature to a place of higher temperature in a process called mechanical refrigeration. This process is used in preserving the food we eat and for comfort air conditioning. Much of the equipment discussed in other TDP modules dealing with equipment uses the principles discussed in this TDP. A designer needs a thorough understanding of the concepts of mechanical refrigeration to create the best performing and cost effective projects. The Principles of Mechanical Refrigeration is divided into two modules, Level 1: Introduction, and Level 2: Analysis. Before proceeding to the equipment TDPs, the information in the Level 1: Introduction, should be understood. Level 2: Analysis, will provide a better understanding of how to evaluate unit performance and select refrigeration components.

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Introduction

The process of mechanical refrigeration has changed our daily lives. Thanks to mechanical refrigeration the variety, quality, and safety of the foods we eat has improved; and the buildings where we live and work are more comfortable and productive. The mechanical refrigeration cycle is the process of how heat is moved. This module will explain the process.

This book is the first of two modules on the topic of mechanical refrigeration. The first step is to understand the terminology used in mechanical refrigeration, and understand how four basic principles are applied to cause the movement of heat. These principles can then be applied to the refrigeration process that uses four basic components to accomplish moving heat – the evaporator, the compressor, the condenser, and the metering device. The process also requires a fluid, called a refrigerant, to move heat through the system. This module describes what constitutes a good refrigerant and how to determine its properties anywhere in the refrigeration cycle.

The second module of Principles of Mechanical Refrigeration deals with analyzing the refrigeration cycle. The pressure-enthalpy diagram (p-h) is a useful tool used by system designers. The construction of this diagram will be explained and the function of each refrigeration system component is explained using the p-h diagram. Finally, this second module will use the p-h diagram to evaluate several system variations and evaluate their performance.

It is important for a designer to understand the operation of a refrigeration system. While the designer may not actually design the refrigeration system, as is typical with packaged units, proper selection and application depends on a working knowledge of mechanical refrigeration. This module will develop an understanding of the mechanical refrigeration system and demonstrate how to analyze system performance.

Definition

Heat is a form of energy, and energy is simply the potential to do work. The first law of thermodynamics states that energy can neither be created nor destroyed, we only transferred heat energy from one place to another. The total heat energy into a system must therefore always equal the heat energy out. The second law of thermodynamics says that energy can only flow from a level of higher potential to a level of lower potential. Therefore, heat transfer is the movement of heat energy from a higher temperature substance to a lower temperature substance.

Mechanical Refrigeration

is the process of using a volatile fluid to absorb heat from a lower temperature place, raising the fluid's pressure and temperature so it can be rejected to a higher temperature place.

The refrigeration process becomes the process of providing a substance at a lower temperature to which heat can flow from a higher temperature substance.

History of Refrigeration

The earliest use of a cold substance for taking away heat was using ice or snow. The Chinese were among the first to learn that ice made drinks cooler and tastier. Early Greek and Roman slaves were used to carry snow from mountaintops to be stored in pits for later use in creating frozen delicacies. In France during the 16th century, ice and snow were used to cool beverages, and frozen dishes became popular.

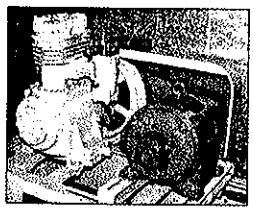
In 1626, Francis Bacon was the first to experiment with refrigeration for preserving foods. He experimented with a chicken stuffed with snow to see if it would preserve it; but it was not until the discovery of the microscope in 1683 that tangible results were obtained. With the microscope, scientist learned about bacteria, enzymes, and molds. They discovered that these microscopic organisms multiply with heat, but below 50° F they seem to hibernate.



Natural Refrigeration



Francis Bacon



Mechanical Refrigeration

Figure 1

The History of Mechanical Refrigeration

Lower temperature does not kill these organisms, but it does control their growth. Food could now be maintained in its fresh state by the use of cold instead of preserving it through drying, smoking, or salting. The use of cold, or refrigeration for food preservation grew to the worldwide business it is today.

For nearly a hundred years, all refrigeration for foods was provided by nature through the use of ice or snow. It was not until 1775 that experiments were made to create lower temperatures artificially, but these first experiments never got beyond the laboratory. In 1834 the first patent was granted on a mechanical refrigeration machine. This was a British patent, and a quotation from it is of interest because in principle it describes the mechanical refrigeration cycle as it is used today:

> *What I claim is an arrangement whereby I am enabled to use volatile fluids for the purpose of producing the cooling or freezing of fluids, and yet at the same time constantly condensing such volatile fluids and bringing them again and again into operation without waste.*

In 1855, the first mechanical refrigeration machine was put to use in the United States, in Cleveland, Ohio, where it was used for the artificial making of ice. About 1900, electricity began to enter homes, and with the development of the small electric motor, the "ice plant" was brought into the home.

It was recognized that the same cooling used to preserve foods could be used to cool the air and improve comfort. Over the next 100 years the use of mechanical refrigeration for comfort and food preservation grew dramatically to the point it became universally accepted. Now refrigeration machines are made in sizes up to hundreds of horsepower, and multiple machines are used for systems totaling several thousands of horsepower in a single installation – all with the same principle as described in that original patent.

The mechanical refrigeration cycle can be classified into two broad categories, comfort cooling and process cooling. Both involve the process of removing heat energy from a place where it is not wanted and rejecting it to a place where it is not objectionable.

In general, the concepts presented here apply equally well to either comfort or process applications. There are some differences however.

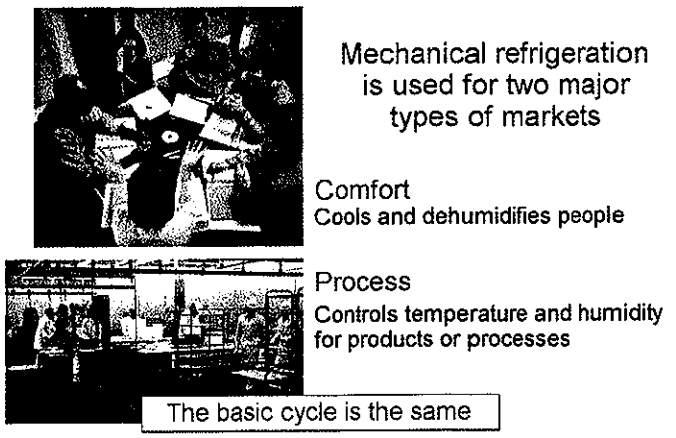


Figure 2

There are two major categories of mechanical refrigeration.

Systems used for comfort, often referred to as air conditioning, operate at higher refrigeration system temperatures than most process or refrigeration applications like food storage. In this module reference to refrigeration means the mechanical process of refrigeration not necessarily lower temperature mechanical refrigeration in food or process application. The TDP text will clearly distinguish between comfort applications and low-temperature or process applications.

Basic Principles

To understand the way mechanical refrigeration works, we must first understand how heat moves. The process can then be explained by four fundamental principles. Heat transfer must first be understood so the four principles of refrigeration can be explained.

Heat Transfer

Heat energy can move by one of three methods of heat transfer.

Conduction is heat transfer from one molecule to another either in the same material or two materials in contact with each other. We have most likely experienced this when we grabbed a spoon that has been in a hot pan of food on the stove. First, the spoon got hot from the contact with the hot food (conduction), then the spoon handle got hot as heat energy moved up the spoon (conduction), and finally, our hand got hot when we touched the spoon (also conduction). This principle is used extensively in refrigeration, for example the refrigerant in the tubes is in contact with the tubes and heat transfer takes place from the refrigerant to the tube to the fluid that surrounds the tube.

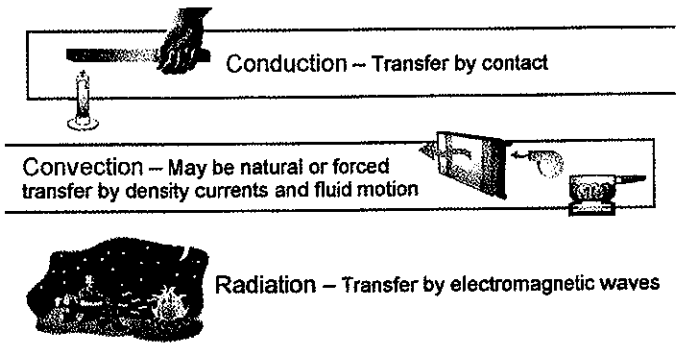
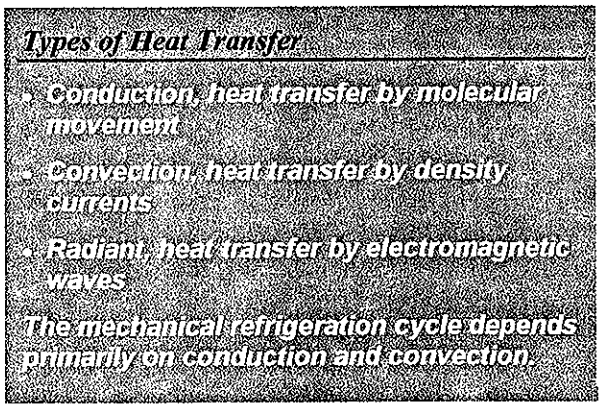


Figure 3

There are three methods of transferring heat.

Convection is the second method of heat transfer that takes place by flow movement. Convection can further be classified into one of two categories, natural or forced. Natural convection is the movement that occurs from the density differences in a fluid. If you observe a pot of water heating on a stove, as the water heats you will notice a turbulence begin to occur in the pot. This is the result of the difference in density between the warm water and the cooler water. This same phenomena can be observed when we stand on a chair in a room on a hot day, the air near the ceiling is warmer than near the floor because the lower density of the hot air caused it to rise toward the ceiling. If a force, fan or pump, is used to cause movement, it is called forced convection. This process is also used extensively in mechanical refrigeration when a fan is used to force air over a coil.



The final heat transfer method is radiant heat transfer, which is the transfer of heat from a warmer surface to a colder surface by electromagnetic waves. This transfer does not warm the fluids in between the surfaces but only the surfaces themselves. We have all experienced this with the heat energy we get from the sun or the way we can be warmed by a fire without the air around us actually being warmed. Radiant heat transfer is not used in the basic refrigeration cycle.

Measuring Heat

The primary device used to measure heat energy is a thermometer. The thermometer works by a fluid in a sealed tube responding to the degree of molecular movement in the fluid that surrounds the thermometer. As substances are heated, they contain more heat energy; therefore, they have a greater amount of movement. If all heat energy were removed, molecular motion would stop, a point referred to as absolute zero.

The readings on a thermometer are a relative scale. Two common types are in general use. The Fahrenheit scale was developed with the freezing point of water as 32° F and the boiling point as 212° F and was based on the theory that absolute zero was 0° F. Scientists later discovered that absolute zero was actually -459.7° F on the Fahrenheit scale. The other scale based on the freezing point and boiling point of water was the Celsius scale. This scale established 0° C as the freezing point of water and 100° C as the boiling point. Absolute zero on the Celsius scale is -273° C.

A thermometer measures molecular motion

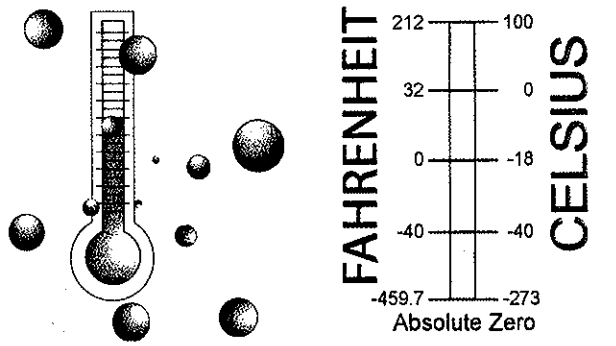


Figure 4

Thermometers measure molecular motion, typically in Fahrenheit or Celsius.

Two variations of these temperature scales also are used which have the same scale increments as the Fahrenheit and Celsius scale but starting at absolute zero; these are the Rankin and Kelvin scales, respectively.

When we measure heat energy with a thermometer we are measuring the heat's intensity. It is also important to understand that heat energy must be considered as a quantity as well. For example, a container with one pound of water at 100° F has twice the heat intensity as one with two pounds of water at 50° F. However, both the one-pound and two-pound containers have the same quantity of heat energy. Both intensity and quantity heat values are important in refrigeration.

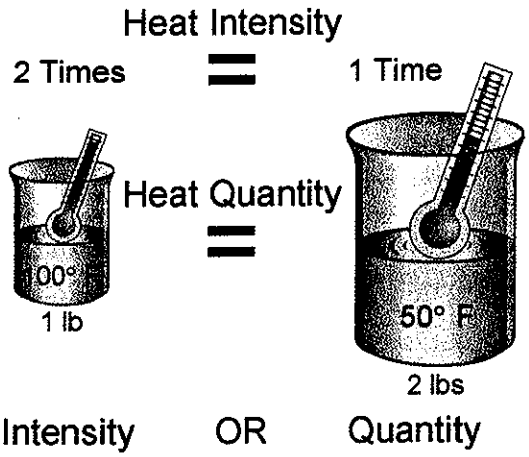


Figure 5

Heat energy has both intensity and quantity.

To explain the four principles used in mechanical refrigeration, we will use water as our refrigerant since it is common and familiar. Water can be a refrigerant, but not a very good one. Characteristics of refrigerants are explained later in this module.

As explained before, heat energy causes internal movement of the molecules within the substance. The amount of movement is an indication of the amount of heat energy. A thermometer responds to this internal movement. This is a measurement of heat intensity.

There is no device that can directly read heat quantity, so an arbitrary unit of measure has been defined to measure heat energy quantity, the Btu or British thermal unit. It is defined as the quantity of heat necessary to raise the temperature of one pound of water one degree Fahrenheit. The term Btu can be used if heat is being added or removed.

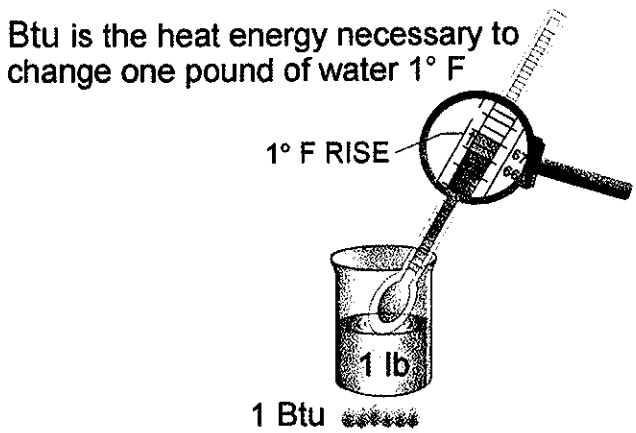


Figure 6

The British thermal unit measures the quantity of heat energy.

Sensible Heat

Heat energy is stored in a substance in two forms. The first of these is sensible heat energy. The heat energy associated with molecular movement is referred to as sensible heat energy. The temperature measured with a thermometer is a measure of this sensible heat energy.

We can plot this on a graph with temperature on one axis and enthalpy Btu/lb (we will define enthalpy shortly), on the other axis. For each Btu of sensible heat added per pound of water the line on our graph will rise one degree on the temperature scale. For example, 10 Btu will raise the temperature of the water from 32° F to 42° F. This same process is true up to 212° F, at which point the water begins to boil.

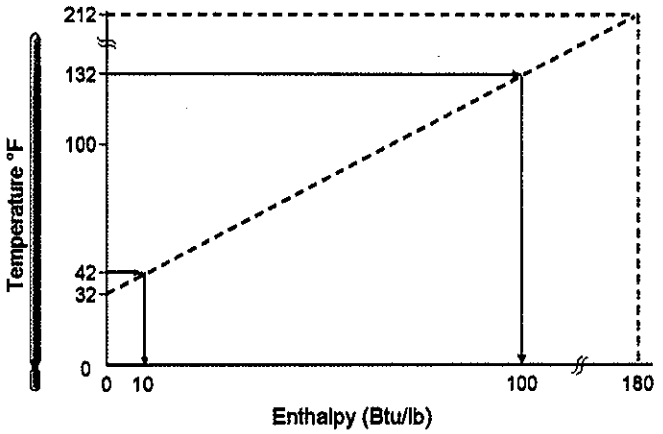


Figure 7

A plot of the sensible heat process

Latent Heat

The second form of stored heat energy is latent heat energy.

The "iceboxes" of the past were periodically stocked with a block of ice. The ice melted and the pan of water below the box had to be emptied on schedule to avoid overflow.

In melting from a solid to a liquid, the ice absorbed heat energy from the foods within the icebox. This energy is its latent heat of fusion, which is 144 Btu per pound.

The water created by the melting ice was discarded even though it was cold, because it had limited refrigeration capacity. This is because 1 Btu raises the temperature of 1 pound of water one degree. Therefore, when 1 pound of water starting at 32° F absorbs 10 Btu, its temperature rises ten degrees to 42° F. The same 1 pound of ice at 32° F melting to water at 32° F would absorb 144 Btu of heat energy. The water has a reduced cooling effect.

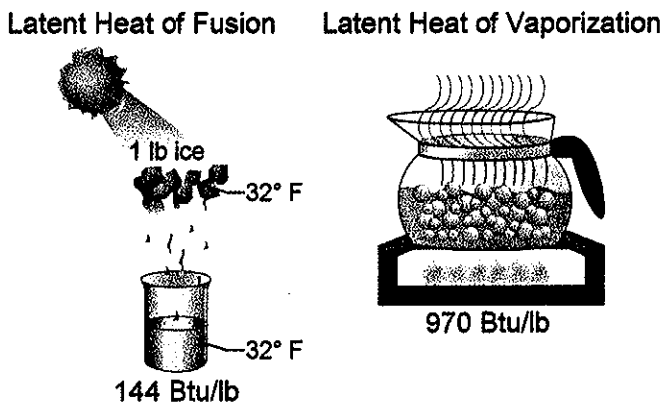


Figure 8

Latent heat energy is the energy required to change the state of a substance from a solid to a liquid or from a liquid to a vapor.

Latent heat energy is the energy required to change the state of a substance from solid to liquid or liquid to vapor. This heat energy transfer takes place at a constant temperature and cannot be measured by a thermometer.

If the temperature of one pound of water at 32° F is raised to the point at which water will boil we will add 180 Btu (212° F - 32° F = 180° F, each degree equals one Btu). Latent heat energy must then be added to change the water from a liquid into a vapor. This energy is the latent heat of vaporization. The latent heat of vaporization for water at sea level requires 970 Btu/lb to change from a liquid to a vapor.

A thermometer does not measure latent heat energy, but we can still use the heat quantity measurement of a Btu to measure latent heat energy. Again we can use our graph to demonstrate these principles. Notice we have negative values below 32° F since 32° F is the reference point we established for our graph. When the ice melted and the water boiled, the quantity of energy changed but the temperature did not. The combined sensible and latent heat energy quantity is called enthalpy and is a measure of total heat energy.

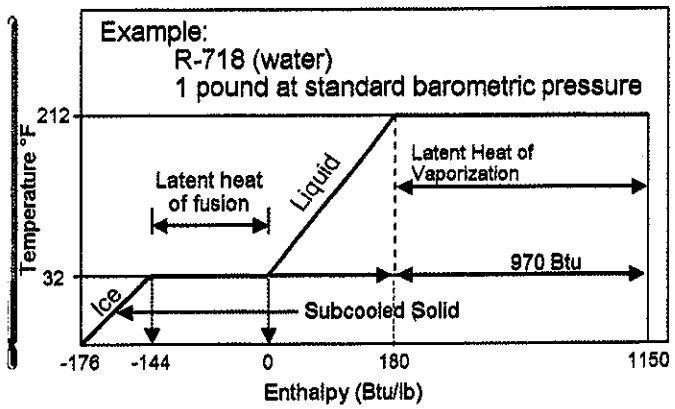


Figure 9

A plot of the latent heat process.

Heat can still be added to the vapor after it has completely boiled. This heat continues to raise the temperature of the vapor and is called superheat. For example if the vapor is heated by 30° F, from 212° F to 242° F, the vapor is said to have 30° of superheat.

In the superheat region, the vapor expands slightly in volume as its temperature is raised. Moreover, the specific heat of the vapor is different than the same fluid in liquid state. For example, it takes only 0.45 Btu to raise 1 pound of water vapor 1° F. If a pound of steam is superheated by 30° F, then 30 * 0.45, or 13.5 Btu, is required.

Our chart can now be used to explain several important refrigeration terms. As ice melts the latent heat involved is the heat of fusion, the latent heat involved in changing a liquid to a vapor is the latent heat of vaporization and is sometimes also referred to as the latent heat of condensation when the

Superheat / Subcooling

Superheat – temperatures above the saturated vapor temperature

Subcooling – temperatures below the saturated liquid temperature

Saturated vapor and saturated liquid temperature are a function of the refrigerant and the pressure.

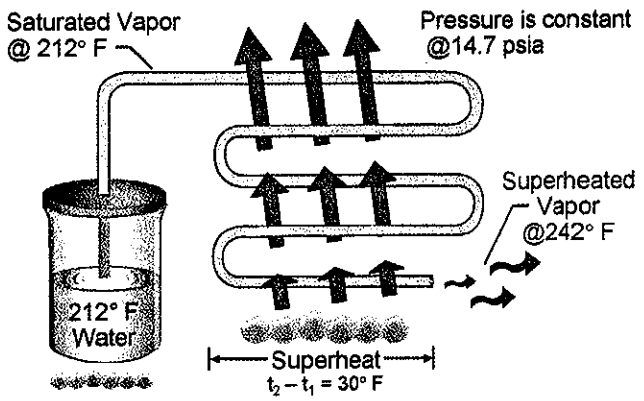


Figure 10

Superheat is heat energy above the saturated vapor.

process is condensing the vapor to a liquid. Enthalpy is a measure of the total heat content and includes both the latent and sensible heat energy. When the vapor is heated above the boiling point it is referred to as superheated. Also when the liquid is at a temperature below the boiling point it is referred to as subcooled. Like superheat is the number of degrees above the boiling point, subcooling is the number of degrees below the boiling point. For example a liquid at 180° F could be said to have 32° of subcooling.

Two other important points are on our chart. The point at which the liquid just begins to turn to a vapor is called saturated liquid. When heat is added to a fluid so it has all evaporated, it reaches a point identified as saturated vapor. It is saturated with all the heat it can contain at that pressure and still remain at the saturation temperature. The change of state has been completed, and any additional heat results in a rise in temperature of the vapor. This additional heat is called superheat. Between saturated liquid and saturated vapor is a mixture of liquid and vapor.

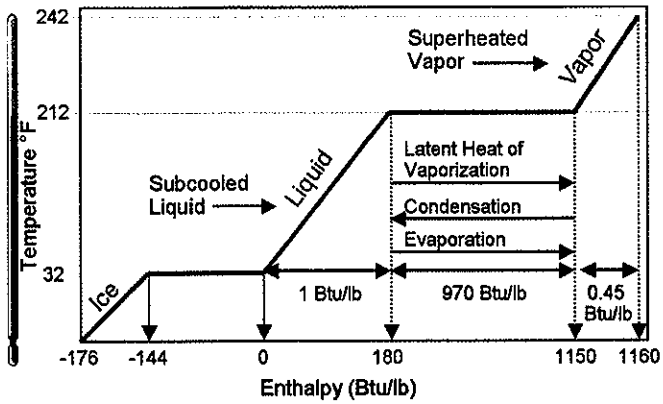


Figure 11

The greatest heat energy occurs during phase change.

Notice the heat energy involved in each step of the process: 970 Btu/lb are required to change liquid to a vapor, 144 Btu/lb are required to change solid ice to liquid, 1 Btu is required to change the liquid temperature 1 degree, and 0.45 Btu/lb are required to change the vapor temperature 1 degree. The greatest heat energy is in changing from a liquid to a vapor or vapor to a liquid.

A ton of refrigeration

is the heat energy required to melt one ton (2000 lb) of ice in 24 hours. In common terms this equates to 288,000 Btu per day, 12,000 Btu per hour and 200 Btu per minute. The Btu is not time dependent; when time rate of change is considered, we use Btuh (Btu per hour) or tons.

One final term related to the transfer of heat needs to be defined. In all the heat flow we have discussed to this point, the time it took to transfer a Btu was not a consideration. The heat flow was the same if it was a day or a minute. When the rate of time for heat flow is important, another term is used. The common measurement is a refrigeration ton. The refrigeration ton is the amount of heat energy required to melt one ton (2000 lb) of ice in 24 hours. We saw that the latent heat of fusion was 144 Btu per pound this would then be 2000 lb * 144 Btu/lb and would equal 288,000 Btu per day. For most of our analyses, it is more convenient to use smaller time increments. We normally rate per hour, (one ton or 288,000/24 hours equals 12,000 Btu per hour), or rate 200 per minute.

When heat flows are time dependent, we use the term tons or the terms ton or Btu per hour, Btuh.

Principle 1 – Heat Flows from Warm to Cold

The first basic principle of mechanical refrigeration is that heat flow can only take place if a temperature difference exists. Heat energy only moves from a warmer substance to a cooler substance. In the context of air conditioning, this means to remove heat from a space. The temperature of the fluid used to absorb the heat must be colder than the space temperature. Conversely, for the fluid to give up its heat, it must be warmer than the air or water to which it is rejecting heat.

Also notice that the greater the difference in intensity between the two substances, the greater the heat flow will be.

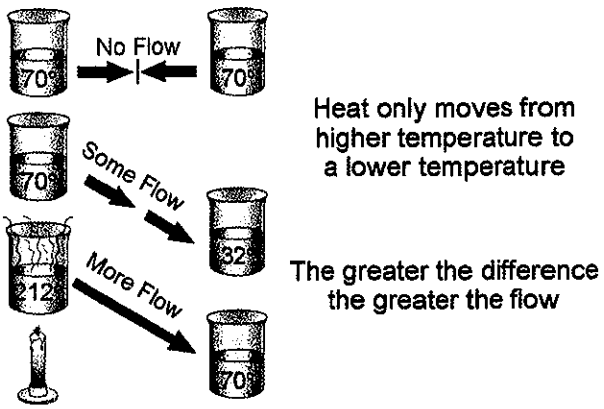


Figure 12

Heat moves from a warmer substance to a cooler substance.

Principle 2 – Phase Changes Require Large Amounts of Energy

The change of state is important to the mechanical refrigeration cycle for two reasons. First, the change absorbs a relatively large amount of heat per pound of substance; and second, this change of state takes place at a constant temperature.

This is the second basic principle of mechanical refrigeration systems. From a practical standpoint, this principle means the greatest amount of heat energy is transferred when the refrigerant changes state. The greatest amount of heat energy is in the change from liquid to vapor, and it is also much easier to transport a liquid than to transport a solid.

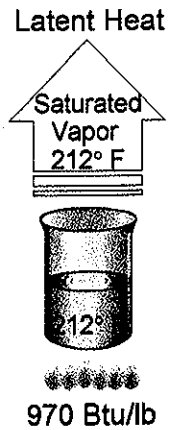


Figure 13

A large amount of energy is required to change the state of matter.

Principle 3 – Boiling Temperature is a Function of Pressure

Since we are familiar with the properties of water, and its behavior is similar to commonly used refrigerants, water is used here to demonstrate the boiling process.

If one pound of water at 32° F is heated, its temperature increases one degree for each Btu added. This process continues until the water reaches its boiling point, or boiling temperature. The pressure over the water determines this boiling temperature. In an open container, the pressure over the water is atmospheric pressure. If the process occurs at a higher elevation the pressure is less and the boiling temperature is lower. In a closed container, the pressure in the container controls the boiling temperature. Notice that the temperature required to boil the water changes with pressure, as does the amount of heat energy required. The lower the pressure the greater the heat energy between saturated liquid and saturated vapor; but the lower the boiling temperature.

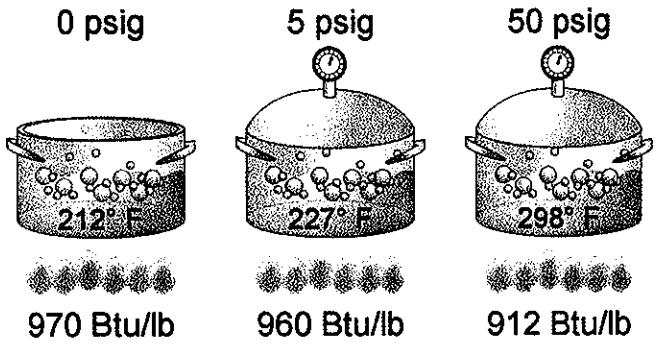


Figure 14

Pressure affects the boiling point.

Measuring Pressure

To begin the discussion of pressure, it is necessary to understand pressure measurements. Pressure readings are a comparison of pressures to a datum point.

The column of air in the atmosphere has weight that exerts a pressure on surfaces below. The weight of the air column will change depending on the elevation. Higher elevations will have lower atmospheric pressure. It is customary practice to use the weight of the column at sea level as a datum point to describe other pressures. The pressure of the column of air is called the barometric pressure.

When a pressure reading is taken, it is the difference above or below the sea level datum point and is called gage pressure or psig, often, just psi. If the pressure associated with the column of air is included in the reading, it is called pounds per square inch absolute, or psia.

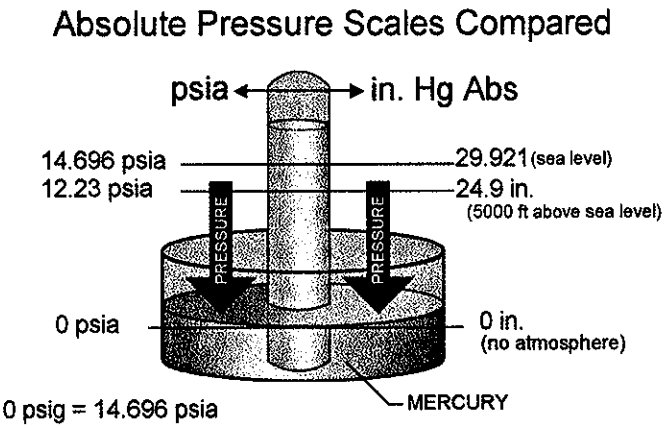


Figure 15

Measuring Pressure

Change of State

The temperature at which a fluid changes state is dependent on the pressure.

At sea level the pressure is 29.921 in. of mercury (Hg) or 14.696 psia. This point is often referred to as standard barometer. At a 5000-foot elevation, the pressure is 12.23 psia.

Most calculations in refrigeration use psia.

At the boiling point, either the pressure or the temperature establishes the other. Water at the boiling temperature for a given pressure is identified as saturated liquid. It is saturated with all the heat it can contain at that pressure and still remain a liquid. As more heat is added, it evaporates into a vapor until all the liquid is a vapor; it is then at saturated vapor. When the process is reversed and heat is removed from a vapor, the vapor begins to condense at the saturated vapor point until it is all a liquid at the saturated liquid point. This reverse process is the condensing process.

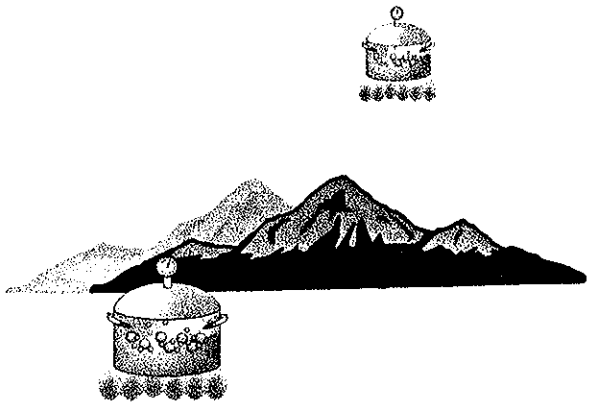


Figure 16

Changing pressure changes the point at which matter changes state.

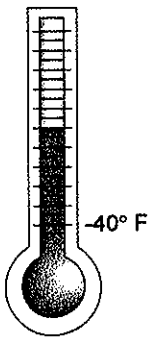
The third principle of mechanical refrigeration is that the temperature at which a fluid changes state is dependent on the pressure. In mechanical refrigeration this means that by changing the pressure, the temperature at which the change occurs can be controlled.

Refrigerant Enthalpy

To this point 32° F was our datum point. Actually, all matter contains heat or energy down to absolute zero (-460° F), but, for convenience purposes, other datum lines are selected arbitrarily.

Because water freezes at 32° F, the datum line for water could be taken at 32° F. Any values of heat content below 32° F are thus negative. With most common refrigerants the datum would be much lower.

To avoid the use of heat contents with negative values down to lower temperatures, the heat content of most refrigerants is taken at a datum line of -40° F. The figure shows the boiling points of some of the most common refrigerants at atmospheric pressure.



Water	212° F
HFC-134a	-15° F
HCFC-22	-41° F
HFC-410A	-62° F

Figure 17

Different refrigerants boil at different temperatures.

Total energy in a fluid involves not only the internal energy represented by heat, but also the external energy or work required to reach that state. The sum of these energies is called enthalpy. Since it is the more accurate term, enthalpy is used in refrigerant tables and in this text to indicate total heat content.

Principle 4 – Flow Occurs because of a Pressure Difference

The fourth principle of mechanical refrigeration is that flow of a liquid or vapor will only occur if a pressure difference exists. As with heat flow, if no differential exists, flow will not occur. Also, flow will not occur from a point of lower pressure to one of higher pressure unless mechanical work is added: Flow can be caused by static difference, as with a column of liquid, by pressure difference, or by mechanical work creating positive or negative pressure.

In mechanical refrigeration this means a pump is required to move the vapor from the low-pressure side of the system to the high-pressure side of the system.

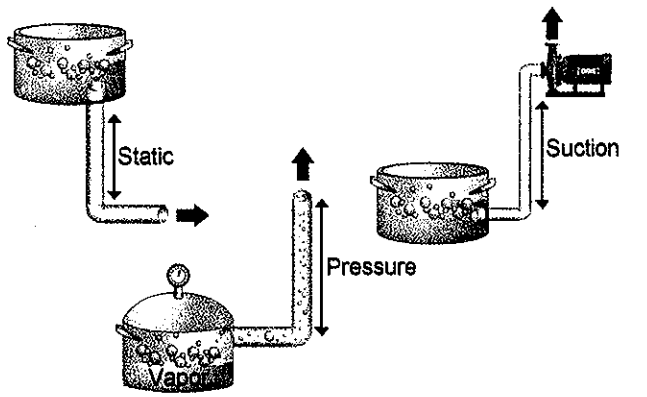


Figure 18

Flow only occurs when there is a difference in pressure.

Summary of the Four Basic Principles

The four basic principles can be summarized as:

- Heat energy will only flow from a warmer temperature to a cooler temperature and the greater the difference, the higher the rate of flow.
- Two types of heat energy exist in a substance, sensible and latent. A larger amount of energy is required to change states in the latent heat energy.
- The temperature at which a change of state occurs is dependent on the pressure. By controlling the pressure, we can control the boiling point.
- Flow of a liquid or vapor will occur only if a pressure difference exists, unless mechanical work is added.

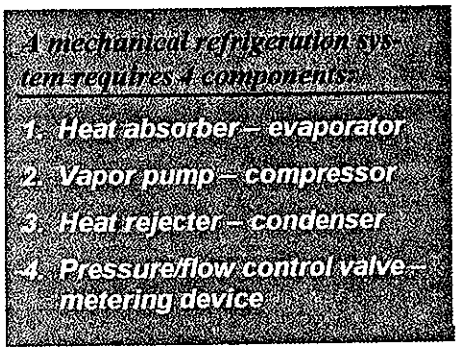
With these four principles the steps of the mechanical refrigeration cycle can easily be explained.

The Mechanical Refrigeration Cycle

As was mentioned in the introduction, all mechanical refrigeration systems require four basic components to form a complete cycle. In this section the basic components will be described with the function they serve in the process. Several other components are typically found in refrigeration systems and are identified with their function and location in the system.

Four Components

The mechanical refrigeration cycle requires four components for its operation to be a continuous cycle. First is the evaporator, a section where the refrigerant absorbs the heat from the fluid that is being cooled. Second is a device that raises the vapor's pressure to a temperature where it can be condensed, the compressor. Third is the condenser, a section where the absorbed heat is rejected from the refrigerant. Fourth is a device that controls the flow and drops the refrigerant pressure back to the temperature required in the evaporator, the metering device.



Refrigerant

The first thing required for a system to remove heat is a refrigerant. A refrigerant is the fluid that absorbs and transports the heat energy. For the refrigerant to deliver the most heat transfer capability, it must change from a vapor to a liquid at a reasonable pressure and temperature to take advantage of the latent heat capacity.

Many fluids can be refrigerants. The chart below shows properties of several refrigerants with the pressures associated with a few of the common temperatures found in comfort and process refrigeration. These pressures and temperatures define the work the mechanical refrigeration system must do to perform the function of moving heat. This text will use R-22 and R-410A to explain the refrigeration cycle. Notice the difference in pressures associated with these two refrigerants. Properties of a good refrigerant are discussed later in this module. A comparison chart of refrigerants is included in TDP-401, Principles of Mechanical Refrigeration, Level 2, Analysis.

A refrigerant is a fluid that absorbs heat and changes from vapor to liquid phase at reasonable pressures and temperatures as encountered in mechanical refrigeration.

	PRESSURE psia					
°F	Water	HCFC-22	HFC-410A	HFC-134a	CO_2	Propane
-40	0.00186	15.26	26	7.43	145.77	16.1
0	0.0185	38.73	64	21.62	305.80	38.4
40	0.122	82.28	132	49.70	567.50	78.6
100	0.950	210.70	340	138.80	X	188.6
130	2.225	311.60	500	213.40	X	273.3
212	14.696	*CP	*CP	587.20	X	X

*Critical Point, pressure psia

Figure 19

Refrigerants and their pressures at common comfort and process refrigeration temperatures.

Evaporator

If a volatile refrigerant is available in liquid form, it will absorb heat in evaporating or changing to a vapor. It is this change of state that accomplishes the cooling effect in a refrigeration cycle.

Since heat will only flow from a warmer substance to a cooler substance, the temperature of the evaporator must be less than the temperature of the substance to be cooled. For instance, if heat and moisture are to be removed from air, the temperature of the coil must be below the dew point temperature of the air. If the dew point temperature is 55° F the refrigerant temperature in the evaporator must be at least a few degrees colder than 55° F for latent heat removal from the air to occur. If the air is in a frozen food case which needs to be kept at 0° F, then the temperature must be at least a few degrees below zero.

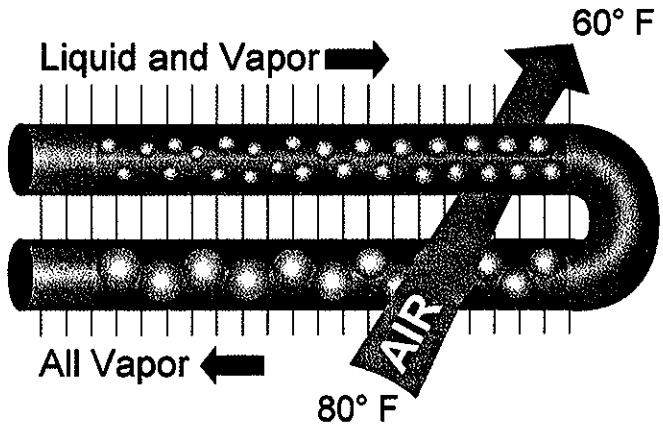


Figure 20

An evaporator absorbs heat.

If liquid refrigerant is allowed to expand through a cooling coil, which is open to the atmosphere, the refrigerant absorbs heat from the surrounding air and evaporates at a temperature corresponding to atmospheric pressure. With R-22, this would be at -41.4° F at standard atmospheric pressure, and for R-410A this would be -60.8° F.

Since it is evaporation that accomplishes the greatest cooling, that part of the equipment in which the cooling is done and the evaporation takes place is called the "evaporator."

The evaporator and metering valve alone would provide refrigeration or cooling. However, such an arrangement would be very expensive to operate because the refrigerant would be wasted. Therefore, the balance of the cycle or equipment is needed and used to recover the refrigerant, so that "*the volatile fluids are brought again and again into operation without waste*," to quote again from the original patent.

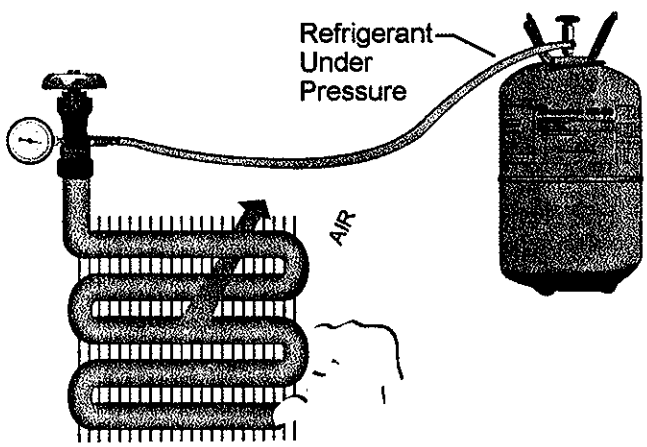


Figure 21

An open cycle requires only two components.

In order for the refrigerant to be used over and over, a closed cycle is required. Two other components are required if the refrigerant is to be returned to a state where it is ready once again to be used in the evaporation process. These are a vapor pump, compressor, to increase the pressure and therefore the temperature high enough that a temperature difference exits between the fluid to be used to absorb the rejected heat and the refrigerant. The other is a heat exchanger where the heat to be rejected can be adsorbed by the heat rejection fluid and the refrigerant vapor can be condensed back to a liquid. Since the vapor is being condensed this heat exchanger is called the condenser.

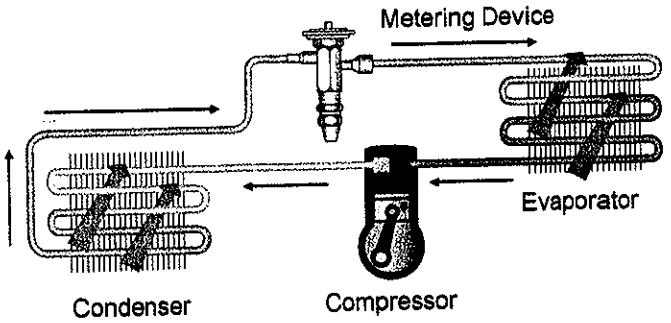


Figure 22

A closed cycle requires four components.

The system can be divided into two sections based on the pressure in that part of the system. The evaporator is at a low pressure so that a low temperature exists to absorb heat. It is often referred to as the low-pressure side of the system or low side. The condenser on the other hand is at a high pressure, so its temperature is high enough to reject heat to the condensing fluid. The condenser side of the system is referred to as the high-pressure side or high side. The compressor and metering device divide the high-pressure and low-pressure sides of the system since a pressure change takes place within each of these components.

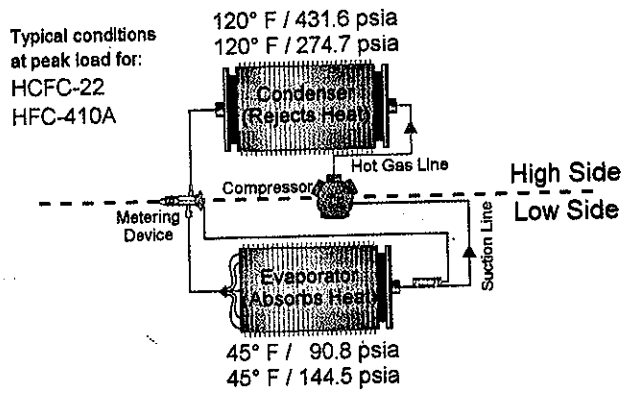


Figure 23

Refrigeration systems have two pressure zones, the low-pressure side (low side) and the high-pressure side (high side).

Then the function of the evaporator is to absorb heat energy at a rate that will cool or maintain the temperature of the fluid to be cooled, normally air or water, to match the load requirements of the space or process. Since the most heat transfer is accomplished with a change from liquid to vapor, the evaporation process is primarily used to absorb the heat load. For reasons to be discussed shortly, the refrigerant enters the evaporator as a cold mixture of vapor and mostly liquid. As it progresses through the evaporator, the refrigerant boils, absorbing heat until all the liquid has turned to a cold vapor. The point at which this occurs is called the saturated evaporator temperature, SET.

SET and SST:

SET is Saturated Evaporator Temperature and is the temperature that corresponds to the pressure leaving the evaporator.

SST is Saturated Suction Temperature and is the temperature that corresponds to the pressure at the inlet of the compressor.

SST is less than the SET by the temperature corresponding the pressure loss in the interconnecting suction line.

It is common practice on most systems to allow the refrigerant to absorb some additional heat above the saturated evaporator temperature. Refrigerant vapor leaves the evaporator as a cold, superheated vapor. Superheat is often required to assure that liquid will not be returned to the compressor. As was demonstrated earlier, the superheated vapor has much lower heat absorbing capacity than the evaporation process, therefore, the superheat region in the coil should be kept to a reasonable minimum.

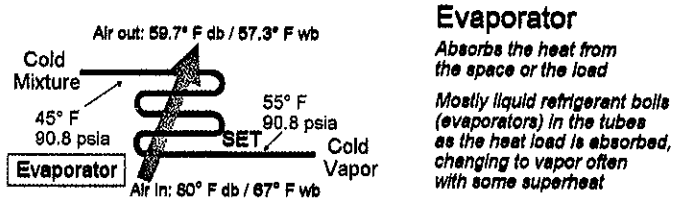


Figure 24

Evaporator Function

Compressor

If the refrigerant vapor from the evaporator were merely passed into a chamber or tank, the pressure in that chamber would rise until the pressure in the chamber and that of the refrigerant supply reached equilibrium. The pressure would correspond to ambient temperature and the cooling effect would stop. Therefore, it is necessary to continually pull off the refrigerant vapor as rapidly as it is formed to maintain proper pressure and hence, saturation or boiling temperature in the evaporator. This is one function of the compressor.

If the load increases and boils off refrigerant faster than the compressor withdraws it, the pressure and temperature rise. Thus, a refrigeration cycle "rides with the load," until compressor capacity and evaporator load reaches a balance.

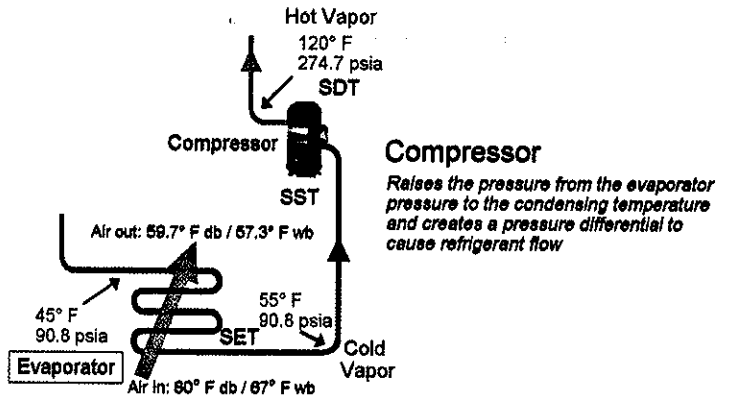


Figure 25

Compressor Function

To convert the refrigerant to a liquid so that it may be used again, heat must be removed so it will condense. To do this, some fluid, normally water or air, to which the rejected heat can flow, must be used. For heat flow to occur, the temperature of the refrigerant must be higher than the temperature of the fluid. A second function of the compressor, therefore, is to raise the temperature of the refrigerant above the temperature of the condensing fluid (air or water). By compressing the refrigerant, the pressure and temperature are increased. The temperature is raised above the saturated liquid temperature, and this permits the fluid to cool the refrigerant vapor and condense it to a liquid state.

Work of Compression

Refrigerant leaves the evaporator and enters the compressor as either a saturated or superheated vapor. Work needs to be added to the vapor to raise its pressure. The work done in compressing the vapor raises the vapor temperature. The refrigerant absorbs this heat during compression, and leaves the compressor in a superheated state with more energy, and hence more heat than it had when it entered the compressor. This increase in heat is referred to as the "heat of compression," and is equivalent to the work done on the vapor as it is being compressed.

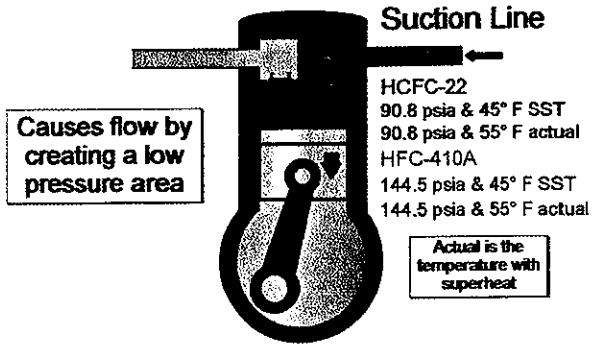


Figure 26

Compressor suction creates flow.

SCT and SDT

SCT is Saturated Condensing Temperature and is the boiling temperature that corresponds to the pressure leaving the condenser.

SDT is Saturated Discharge Temperature and is the boiling temperature that corresponds to the pressure at the outlet of the compressor.

SDT is greater than the SCT by the temperature corresponding the pressure loss in the interconnecting hot gas line.

The first function of the compressor, therefore, is to draw the cold vapor from the evaporator. The second function is to provide adequate work energy to raise the pressure such that the temperature will be high enough to reject heat to the heat rejection fluid and condense the refrigerant. The vapor will leave the compressor as a highly superheated hot vapor.

The heat of compression varies only slightly with different refrigerants, but varies greatly with different levels of temperature and pressure between which the cycle is operating. For example, at temperature levels of comfort air conditioning, namely 45° F saturated in the evaporator and the 105° F saturated commonly associated with water-cooled condensers, this work of compression amounts to approximately 1.0 horsepower per ton of refrigeration. For a frozen food cabinet, on the other hand, with -10° F in the evaporator and 120° F in an air-cooled condenser, the compressor would require almost 3.0 horsepower per ton.

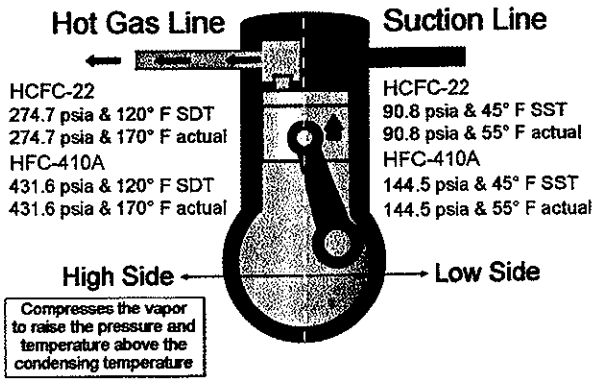


Figure 27

Compressor Discharge – Temperature and pressure are raised.

Condenser

In the condenser, heat is removed from the refrigerant by providing a condensing fluid at lower temperature to which the heat can flow and be dissipated. The condenser removes heat taken on by the refrigerant in the evaporator, and it also removes the heat put into the refrigerant during the compression process.

The first step in the heat removal process lowers the temperature of the superheated refrigerant until it reaches the saturation point at the pressure existing in the condenser. Further removal of heat then affects a change of state and condenses the refrigerant to a liquid so that it can be used over again in the cycle.

The temperature of the fluid (air or water) to which heat flows in the condenser establishes the temperature at which condensation takes place. As it takes on heat from the refrigerant, the temperature of the condensing fluid will rise. To maintain the flow of heat, the temperature of the refrigerant in the condenser must always be greater than the temperature of the condensing fluid.

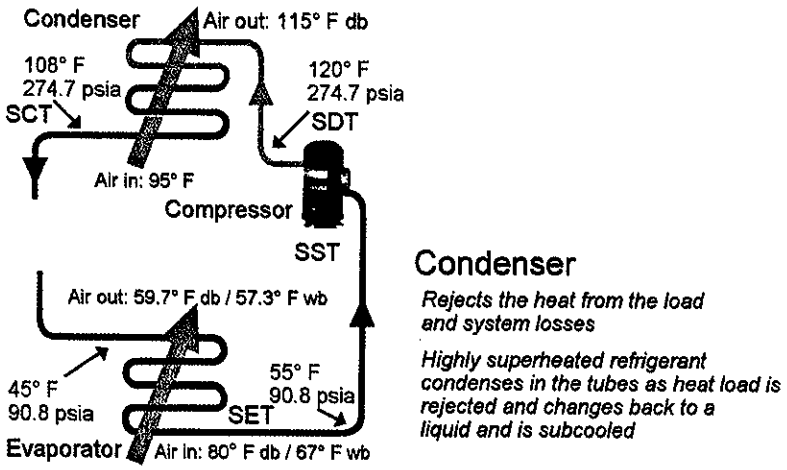


Figure 28

The condenser rejects heat in three forms – desuperheating, condensing and subcooling.

A final portion of the condenser is used to lower the temperature of the liquid refrigerant below the saturated condensing temperature providing subcooling. When the liquid is cooled below the saturated liquid temperature at a given pressure, the liquid is referred to as being subcooled. Most condensers are arranged so as to provide some subcooling of the liquid refrigerant as it leaves the condenser. If this subcooling is retained until the liquid reaches the metering device, then it reduces the percent flash gas and increases the refrigeration effect of each pound of refrigerant circulated. Flash gas will be explained in the next section.

Subcooling can increase system capacity by 1% for each degree of subcooling provided. The amount of available subcooling is limited by the condensing surface and the temperature of the condensing fluid. Like the superheating in the evaporator, the greatest heat transfer occurs in the condensing process not in the subcooling process.

Subcooling also assures that only liquid refrigerant will enter the metering device. The impacts of subcooling will be expanded in the next TDP module, TDP-401, Principles of Mechanical Refrigeration, Level 2: Analysis.

In an air-cooled condenser, the refrigerant passes back-and-forth through the coil several times; and to accomplish flow, there must be a pressure difference, or pressure drop. In most condensers, this will be approximately 10 psi or less.

A portion of the tubes must provide desuperheating for the evaporator superheat and the heat of compression work. Desuperheating represents between 10 percent and 20 percent of the total heat removal in the condenser, and since the density of the vapor is light, the pressure drop in this section will be small.

A section of the coil must also provide for the subcooling.

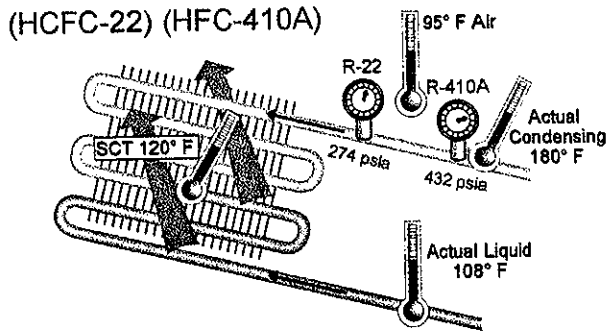


Figure 29

Air-Cooled Condenser – provides desuperheating, condensing and subcooling.

With water-cooled condensers, the refrigerant is often in the shell, and there is little variation in pressure and saturated condensing temperature throughout the shell. The difference between the saturated condensing temperature and the temperature of the condensing fluid as it leaves the condenser is referred to as the leaving difference.

Water-cooled condensers generally have a lower saturated condensing temperature than air-cooled since the condensing fluid can be provided at a lower entering temperature. In the next module of this series, the impact of lower condensing temperature on cycle performance will be evaluated.

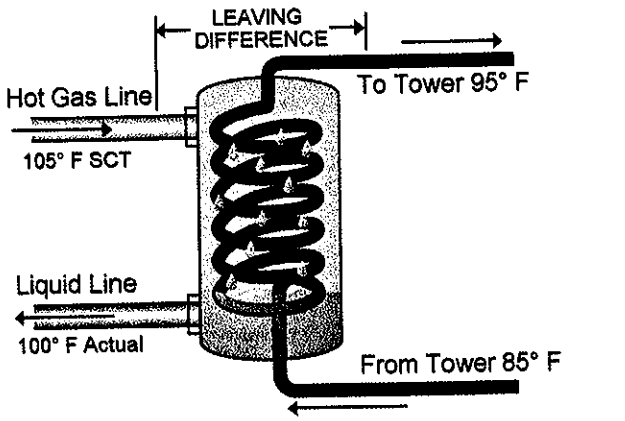


Figure 30

Water-Cooled Condenser – Less subcooling can be provided.

Metering Device

When liquid refrigerant flows from the higher pressure of the condenser to the lower pressure of the evaporator, some type of metering device must control its rate of flow. The pressure drop across the metering device reduces from the condenser pressure to the evaporator pressure. This metering device might be a thermal expansion valve, a hand valve, a float valve, capillary tube or a fixed orifice. In any of these, the refrigerant expands as it enters the area of lower pressure. This pressure reduction process is often referred to as throttling.

Metering device function

can be explained with the perfect gas equation:

$$pv = nRt$$

p pressure
v volume
n weight
R a constant
t temperature

The weight and constant will remain the same and the volume is nearly constant. If no heat comes from external sources, then temperature must change.

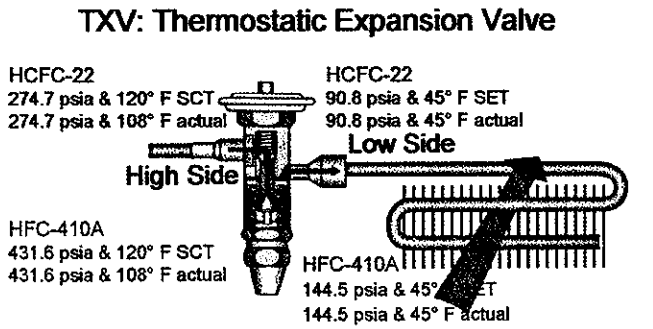


Figure 31

Metering Device

The pressure drop and expansion takes place quickly and in a relatively short distance through an orifice. Hence it is practical to assume that there is neither a heat gain nor a heat loss from the refrigerant in this process. Since the refrigerant will now boil at a lower temperature, a small portion of the refrigerant will evaporate into a vapor and absorb the required heat of vaporization from the surrounding refrigerant, in turn cooling that refrigerant to the evaporator saturated temperature. The portion of gas, which becomes a vapor is called flash gas.

The metering device serves to regulate the flow of subcooled warm liquid, dropping the pressure and thereby cooling the refrigerant to the evaporator saturation temperature. The refrigerant leaving the metering device is a cold mixture of liquid with a small amount of vapor.

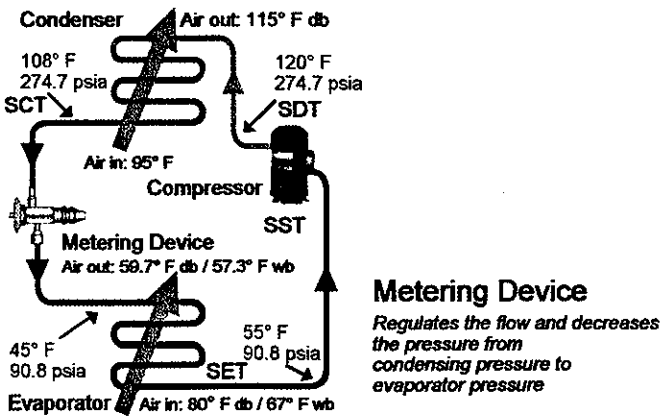


Figure 32

The metering device regulates the flow of subcooled warm liquid.

Refrigerant Lines

Refrigerant lines interconnect each of the four basic components of the mechanical refrigeration system. The line that connects the evaporator to the compressor is the suction line. The suction line carries a low-temperature and low-pressure vapor that occupies a relatively large specific volume. Therefore, the suction line is the largest of the interconnecting lines. It is also at a low temperature that may well have a temperature below the dew point of areas the line passes through. This could cause the line to sweat and the heat gain in the line would also reduce compressor capacity. For these reasons the suction line is usually insulated with a rubber-based insulating material.

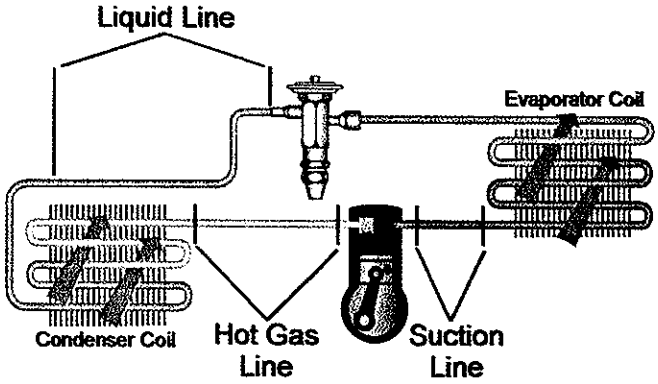


Figure 33

Refrigerant Lines

Refrigerant lines:

Suction lines are the largest and coldest and are normally insulated.

Hot gas lines are the hottest and smaller than the suction line but are not normally insulated.

Liquid lines are the smallest and are warm, so they are not normally insulated. However some types of units (some duct-free splits) use the pressure drop of the liquid line as part of the metering device. In this case, these lines must be insulated.

The line that connects the compressor to the condenser is the hot gas line. Sometimes this line is also called the discharge line. The hot gas line is carrying a high-pressure high-temperature vapor. The specific volume is smaller and the line size is smaller than the suction line. Heat loss from the line is not detrimental to the systems performance, and unless the very hot line would represent a safety hazard, the line is not insulated.

The liquid line connects the condenser and the metering device. The volume occupied by the liquid is the least of the lines and the line size is the smallest. Since the temperature of the liquid line is normally above the ambient temperature of the space the line goes through, it is not normally insulated.

The metering device is mounted as close to the evaporator inlet as is feasible and does not have a special line.

Refrigerant Cycle Accessories

Refrigerant systems may have a number of other components in the system to protect the four basic components, enhance the cycle performance, increase the operating range, and provide for special applications. This section will briefly identify the components and their use. For additional information on the selection, use, and application of these devices, see TDP-403, Expansion Devices and Refrigeration Specialties.

System Protectors

Three components are used in refrigeration systems that are classified as system protectors: filter-driers, moisture-liquid indicators, and mufflers.

The first component is a filter-drier. Filter-driers are normally located in the liquid line, however, in some applications they are located in the suction line. The purpose of the filter-drier is to filter out of the refrigerant both solid contaminants such as metal filings, flux, and dirt, and soluble contaminants such as acid, water, resins, and wax. These contaminants may be introduced into the system during manufacture of the unit, during installation or servicing, or through operation. The filter-drier consists of shell that contains screens and a desiccant material that will absorb the contaminants from the refrigerant. Filter-driers may be a sealed shell style or may be a shell with a replaceable core. Suction line filter-driers are used after a compressor motor failure to remove acids and debris that may have entered the system. Suction line filter-driers are also common on low-temperature refrigeration systems.

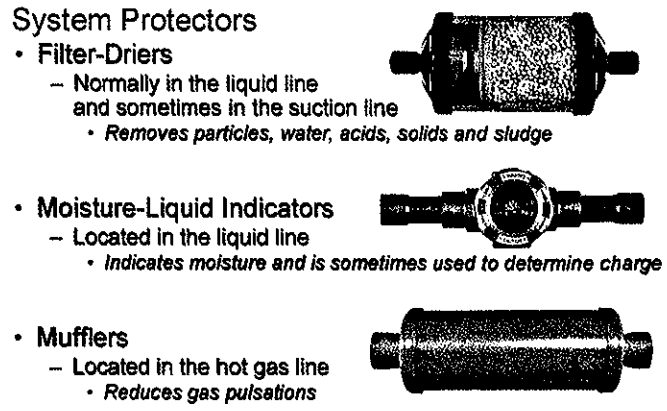


Figure 34

Refrigeration System Protectors

The second device is a moisture-liquid indicator, sometimes called a sight glass. It is located in the liquid line near the metering device. A moisture-liquid indicator is a view port with a chemically sensitive center eye. The color of the eye will change to indicate if the system is contaminated by moisture. Moisture in a refrigeration system can cause freezing in expansion devices, corrosion, and chemical damage. The area around the eye is clear and when the system is operating if flash gas bubbles are seen in the refrigerant flow it is an indication of a liquid line restriction or a low system charge. While the moisture-liquid indicator could be used for evaluating the charging of the system, it is recommended that charging be accurately accomplished by weighing in the system charge or using the appropriate superheat or subcooling method. Using the moisture-liquid indicator to add charge is not recommended since it is impossible to determine how close the system is to the correct charge.

The third device that may be classified as a system protector is a muffler. A muffler is an expansion chamber located in the hot gas line on the discharge side of the compressor. The purpose is to reduce gas pulsations from the compressor. Gas pulsations can cause noise and vibration and

in severe cases the vibration may cause premature failure of refrigerant lines. The muffler often has a series of baffles that help to break up the acoustic waves and gas pulsations, thus reducing the sound. Most reciprocating style compressors use mufflers because of the natural pulsation caused by the alternate opening and closing of the discharge valves. Scroll and many other styles of compressors normally only apply mufflers in situations where the characteristics are such that line vibration or sound generation is a problem.

The muffler is an additional pressure drop in the hot gas line. If the muffler is not provided as part of the manufacturer's package, the pressure drop needs to be included when sizing the hot gas line.

Storage Devices

There are two types of storage device encountered in refrigeration systems, accumulators, and receivers.

An accumulator is a refrigerant storage device located in the suction line before the compressor. The function of the accumulator is to prevent liquid refrigerant from entering the compressor. Since many compressors are positive displacement vapor pumps with small clearance volume, liquid, which does not compress, will do severe damage to the compressor valves and other internal components if it gets into the compressor. As ambient conditions change, so does the amount of charge required in the system. The accumulator provides a place for the excess liquid charge to accumulate. For the accumulator to function properly, it must provide two important features. It must allow adequate vapor flow to the compressor at all load conditions, and it must allow adequate oil return to the compressor. The inlet is high in the accumulator chamber. The outlet is a U-tube that comes from the top of the chamber down to the bottom and back up and out the top. Oil in the accumulator will settle to the bottom of the chamber. Oil return holes in the suction return U-tube pick up the oil from the bottom of the chamber and return it in the vapor to the compressor. Because of the wide changes in charge requirements for heat pump systems, heat pumps have accumulators.

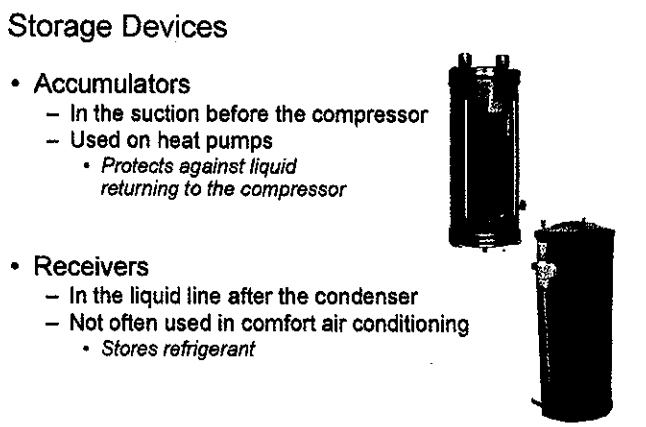


Figure 35

Refrigerant Storage Devices

The second type of storage device is a receiver. Receivers are located in the liquid line after the condenser. The function of the receiver is to hold liquid refrigerant during operation. Receivers are used on most low-temperature refrigeration systems but are not recommended for use in air conditioning-duty equipment. In comfort cooling applications, the range of operating temperature is limited compared to low-temperature refrigeration, and it is desirable to limit the system liquid charge. Because of this, the use of receivers is discouraged.

Performance Devices

Three devices are sometimes used to improve system performance in a mechanical refrigeration system. These are desuperheaters, subcoolers, and economizers.

A desuperheater is a heat exchanger located in the hot gas line. Water for domestic or process hot water requirements is circulated through the desuperheater, picking up some of the heat from the compressor discharge and doing a portion of the work normally done by the condenser. This essentially uses wasted heat to handle a supplemental load and improves condenser system efficiency. These are common on some water source heat pump units. These may be a brazed plate style heat exchanger or a tube-in-tube style.

A subcooler is a heat exchanger located in the liquid line after the condenser to provide additional subcooling. Cold water is circulated through the heat exchanger picking up heat from the liquid refrigerant. The additional subcooling improves system efficiency since it reduces flash gas. These heat exchangers may also be brazed plate, shell, or tube-in-tube.

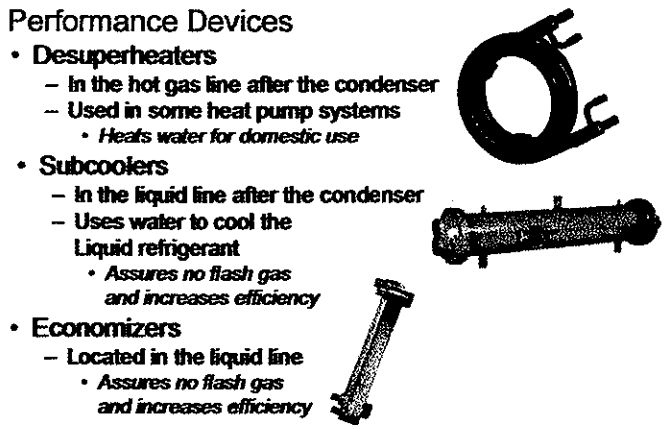


Figure 36

Devices that enhance refrigeration system performance.

The third device is an economizer. The economizer is a heat exchanger located in the liquid line that expands a small amount of refrigerant at an intermediate point between the saturated suction and condensing temperature and uses this to cool the remaining refrigerant. This reduces the amount of flash gas and improves overall unit performance. This system is used primarily with low-temperature refrigeration systems. Economizers are also used on some multi-stage centrifugal compressors.

The performance of these three devices is explained with the p-h diagram in TDP-401, Principles of Mechanical Refrigeration, Level 2: Analysis.

System Pressure Regulators

Refrigeration systems may employ pressure regulators to control system operation by maintaining or limiting pressure at points in the refrigeration cycle. Four types of pressure regulators are used: upstream, downstream, hot gas bypass, and head pressure control.

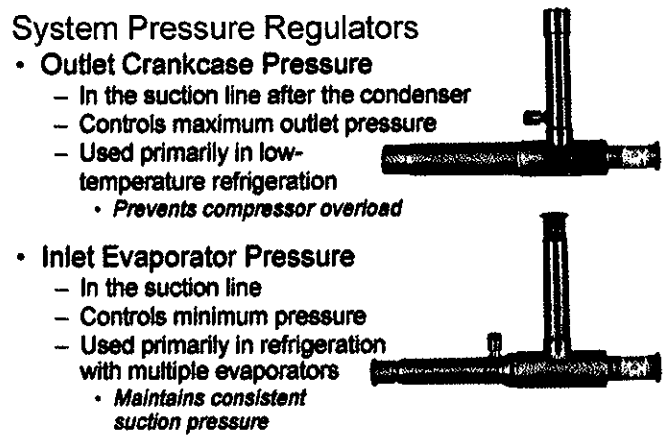


Figure 37

Refrigerant Pressure Regulators

Upstream pressure regulators are used to keep the evaporator temperature above some minimum level by limiting the pressure to a predetermined minimum. These regulators are typically located in the suction line after the evaporator. Typical uses are for multiple evaporators on the same compressor, a situation encountered in low-temperature refrigeration. For example, a system where refrigeration cases are to be kept at 45° F, 20° F, and 0° F all on the same compressor system. An inlet pressure regulator will hold the pressure in each evaporator so that the temperature in each of the cases is held constant at the required pressure. These devices are also used in some chiller applications to limit the evaporator temperature in the cooler to prevent water temperatures from dropping below freezing. Other applications include frost protection on fan coils and defrost applications.

Downstream pressure regulators are primarily used to prevent compressor motor overload. The valve throttles the suction gas flow during high load conditions limiting the current draw of the compressor. These are located in the suction line before the compressor suction.

Air-cooled condensers require a minimum condensing temperature for year round operation. As the ambient temperature drops, the condenser becomes more effective and the saturated condensing temperature drops.

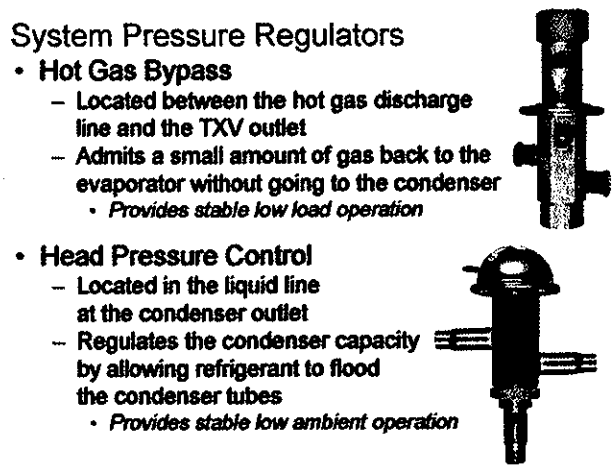


Figure 38

Hot Gas Bypass and Head Pressure Control Regulators

A minimum condensing pressure needs to be maintained to:

- Maintain liquid subcooling and prevent liquid line flashing.
- Provide adequate pressure to the TXV such that adequate pressure drop is present across the valve.
- Provide proper pressure for systems operating with hot gas bypass or hot gas defrost.
- Provide adequate temperatures for heat reclaim systems.

Head pressure control valves function to maintain the required condensing pressure by by-passing discharge gas to the receiver and reducing liquid flow from the condenser. These pressure valve systems may use a combination of inlet and outlet pressure regulators and 3-way bypass regulators. Head pressure control valves are also used with heat reclaim coils where the hot gas is diverted to a coil, which provides some heating or humidity control with condenser heat that would normally be rejected.

The other common application of pressure regulators is hot gas bypass. The hot gas bypass valve allows a small amount of vapor from the hot gas line from the compressor to be injected into the evaporator coil directly after the TXV. This is done to allow the unit to operate at reduced loads. In units without sufficient steps of unloading or no unloading, it allows operation to lower steps of capacity.

These devices are covered in greater detail in TDP-403, Expansion Devices and Refrigeration Specialties.

Valves

Refrigeration systems also use a number of valves to control flow, protect the system, or help in unit servicing. These can be described as hand valves which are used for service, solenoid valves used for flow control during operation, check valves to control flow in one direction only, and relief valves used to protect the system from excessive pressures.

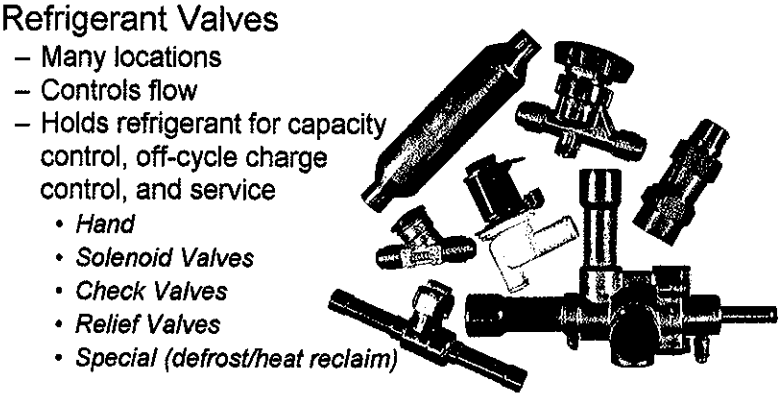


Figure 39

Refrigerant Valves Types

Temperature and Pressure Controls

Refrigeration systems may also require a number of pressure and temperature activated switches that are used to control the system or to protect it from undesirable operating conditions. Pressure switches are commonly used for high-pressure protection on the compressor discharge, and for low pressure or loss of charge protection on the suction. Temperature switches are used for protection on the hot gas or compressor discharge to protect from excessive superheat, and on the evaporator to protect from frost build up on the evaporator. Other pressure and temperature controls may be used to control operation by maintaining a refrigerant temperature or pressure. Some examples are: control of condenser pressure for head pressure control, and control of evaporator case temperature at a constant saturated suction. Oil pressure control may also be used on some units to protect against loss of compressor oil.

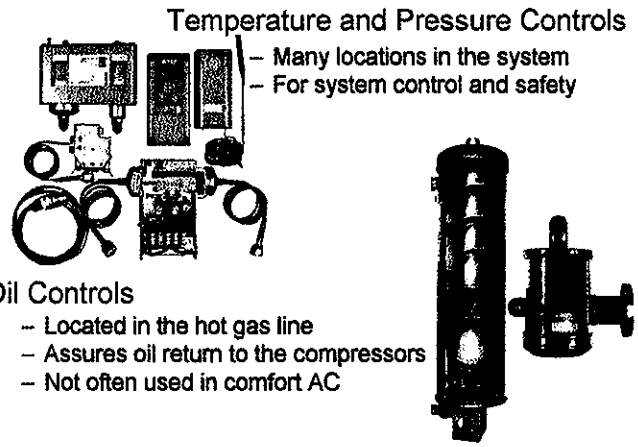


Figure 40

Refrigerant Control, System and Oil

The final component that may be encountered in a refrigeration system is an oil control system. On low-temperature refrigeration systems, too much oil may be in circulation and be pulled from the compressor crankcase. To be sure the oil is returned, an oil separator removes oil from circulation so it can be returned to the compressor.

All the components described in the refrigerant cycle accessories section are covered in greater detail in TDP-403, Expansion Devices and Refrigeration Specialties.

Reverse Cycle Heat Pump

A refrigeration cycle serves to move heat from a low temperature to a higher temperature. When the purpose is that of cooling to a lower temperature, it is referred to as the refrigeration cycle. However, the same cycle can be used to take heat from a source at a low temperature and deliver it at a higher temperature where it can be used for heating. When its purpose is that of heating, it is referred to as a heat pump or reverse cycle system.

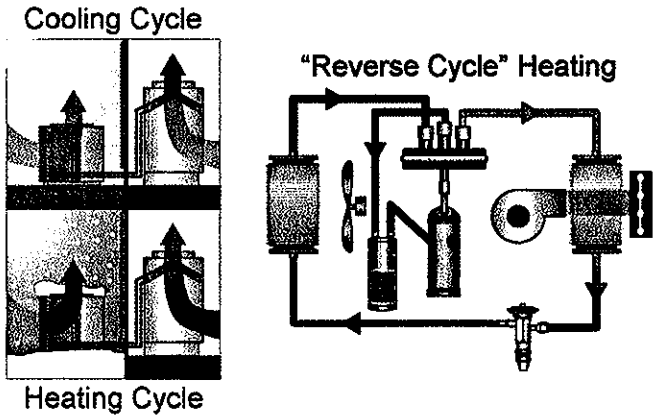


Figure 41

Reverse Cycle Heat Pump

The fluid used for the condenser and the evaporator may be water or air. Units that use air for the condensing and evaporator fluid are called air-to-air heat pumps and units which use air for the evaporator and water for the condensing fluid are water-to-air or water source heat pumps. Other types of systems are also used, for example, water-to-water. The most common type in use today, the air-to-air heat pump, has become popular for residential applications in some parts of the country since the same unit provides heating and cooling.

The feature that most distinguishes a heat pump from the typical refrigeration system we have discussed so far is that it is reversible. This allows the same unit to provide cooling in summer and heating in winter; also at a relatively efficient level. A reversing valve is provided which switches the compressor discharge from the outdoor coil in cooling to the indoor coil in heating.

As was pointed out before, air contains heat down to absolute zero so we can always "cool" it a little more, raising its temperature in the refrigeration process and gaining some heating benefit. In reality this process works best at outside temperatures from the room temperature down to about 32° F. Below 32° F, the moisture in the outdoor air begins to freeze on the outdoor coil and the unit requires a means of eliminating the frost build-up on the coil which, otherwise will become blocked, greatly impeding heat transfer. A means of defrost is provided to overcome this situation. This is usually accomplished by switching the unit back into cooling to use the outdoor coil as a condenser again to remove the frost by melting it. This of course means that the indoor coil is in cooling removing a portion of the condenser heat just added. At some point, the heat added to the space from the heat pump and removed by defrost does not provide adequate heat for the space. In that case, electric heaters may supplement the heat pump capacity.

The components required for a heat pump are much the same as for a traditional air conditioning unit. The same four basic components are used, however, the function of the coils is reversible and the same coils must function as both evaporator and condenser. To allow the operation to be reversed, a reversing valve is added to the basic components. Also to help control the wide range of required refrigerant charge encountered with heat pumps, a storage device called an accumulator (as discussed previously) is used.

A heat pump system has the same four basic components but adds a Reversing Valve and Accumulator

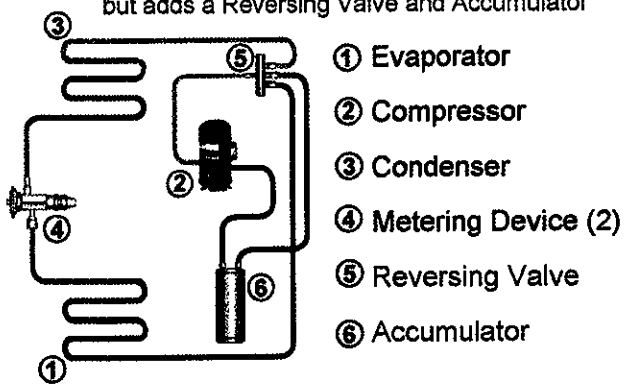


Figure 42

Heat Pump Components

During the cooling cycle, hot gas from the compressor discharge is directed to the outdoor coil by the reversing valve. Here, the hot gas is condensed into liquid as the gas gives up heat to the colder air passing over the coil. Liquid refrigerant then passes through a metering device that causes a drop in its pressure and temperature. The cold liquid-vapor mixture then enters the indoor coil where it evaporates. During evaporation, heat is removed from the warmer indoor air as it passes over the coil. The indoor air is thus cooled to condition the space. The cool refrigerant vapor is then directed to the accumulator where any unevaporated liquid refrigerant is trapped. Cool refrigerant vapor is then drawn into the compressor where its temperature and pressure are increased so the cycle can complete itself.

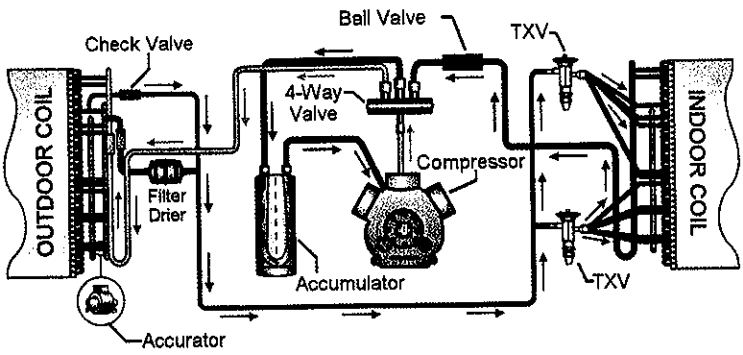


Figure 43

Heat Pump in Cooling Mode

When heat is required, the reversing valve is repositioned. The hot compressor discharge gas is now directed by the reversing valve to the indoor coil, which will act as a condenser. Air is then heated as it passes over the indoor coil and heats the space. Liquid refrigerant now flows in the opposite direction through the metering device to the outdoor coil that now acts as the evaporator. Heat is extracted from the outdoor air as it passes over the coil as the cold liquid refrigerant evaporates. The cold vapor then is drawn through the reversing valve into the accumulator where again liquid refrigerant is trapped. Vapor is drawn from the accumulator into the compressor and the cycle repeats itself.

In reversing the system from heating to cooling, some of the traditional components must have special features to allow for reverse flow. For example, the metering device must be capable of metering flow in both directions. When piston style metering is used, a specially designed orifice is used that allows bi-directional flow. With TXVs, it is common practice to locate an expansion device on both the indoor and outdoor coil. When refrigerant flow is in the opposite direction, either specially designed bi-directional flow valves are used or a bypass around the TXV with a check valve is used.

In addition to the reversing valve and metering devices, there are other design considerations that make the heat pump different from the conventional air conditioning unit. The compressor is specially designed to operate over the wide range of compressor ratios encountered in the heating mode. Both the indoor and outdoor coils are specially designed because they must both evaporate and condense refrigerant. For example, the indoor coil, when acting as a condenser in the heating cycle, must have more than normal surface area to keep the condensing temperature at reasonable levels. Further, most heat pumps are equipped with a suction line accumulator to prevent liquid from reaching the compressor, particularly in the defrost mode of operation. Finally, heat pumps are equipped with crankcase heaters on the compressor.

In the second module of this TDP series, the performance of heat pumps is described on the p-h diagram and the performance factors are calculated. Additional information on reverse cycle units can be found in the TDP-405, Reverse Cycle Refrigeration.

Refrigerants

Water as a Refrigerant

The four general principles of refrigeration were introduced at the beginning of this module using water as the volatile fluid or refrigerant. Actually, water has several desirable properties as a refrigerant. It is low in cost, readily available, and it is entirely safe, being non-toxic and non-flammable; it also has a large heat of vaporization. Although water is used as the refrigerant in steam-jet and in absorption refrigeration machines, it is not a practical refrigerant for a mechanical vapor compression refrigeration cycle.

Water has been used to explain many of the concepts already introduced.

Is water a good refrigerant?

Positive	Negative
• Non-toxic	• Low coefficient of performance
• Non-flammable	• High boiling and freezing point
• No Ozone Depletion Potential (ODP) or Global Warming Potential (GWP)	• Poor oil compatibility
• Abundant	• Potential for corrosive action
• Low cost	• Low pressure and leak potential
• Stable	
• High heat of vaporization	

Water is the refrigerant used in absorption chillers.

Figure 44

The pros and cons of using water as a refrigerant.

At standard atmospheric pressures water is limited to a minimum temperature of 32° F, for below this, the water freezes. To obtain water cool enough for most air conditioning purposes would require operating at extremely low pressures or high vacuums, which are difficult to obtain with positive displacement compression equipment. The specific volume of water vapor is large at the low pressures required, and such volumes are impractical with positive displacement machines, and centrifugal compressors are unable to maintain the required vacuum.

Therefore, it is desirable to use other volatile fluids whose overall characteristics and properties make them more practical for a particular purpose.

Desirable Characteristics

Commercially used refrigerants should have certain desirable characteristics. The following are among the more important ones, and their order of importance varies, depending on use and on the specific requirements of the application.

One of the most important considerations is that the refrigerant needs to be chemically stable over the operating range of pressures and temperatures where it will be applied.

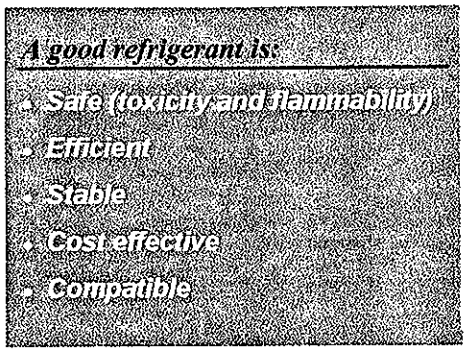
Safe • Efficient • Stable • Cost Effective • Compatible

1. Non-toxic and non-flammable
2. Reasonable operating pressures
3. Leakage resistance
4. Large heat of vaporization
5. Relatively low specific volume
6. Low liquid specific heat (reduced flash gas)
7. Easy to detect leaks
8. Compatible with oils (vapor side)
9. High coefficient of heat transfer
10. Easy to handle and cost effective
11. Non-corrosive and chemically stable
12. Environmentally sound

Figure 45

Characteristics that are desirable in a refrigerant.

The refrigerant needs to be safe, particularly since the refrigerant may be able to leak into an occupied space. ASHRAE Standard 15 defines limits on the use and safety precautions to be taken with different refrigerants. To be safe, the refrigerant should be non-flammable and non-toxic.



The pressures corresponding to the saturation temperatures available with normal condensing fluids should not be excessive, so as to eliminate the need for extremely heavy construction.

The pressures corresponding to the saturated suction temperatures required for most air conditioning and refrigeration processes should be above atmospheric pressure to avoid inward leakage of air and water vapor. Air and water vapor brought into the system degrade performance and are a potential corrosion problem.

A relatively large heat of vaporization is desirable so that the required capacities can be obtained with the smallest mass flow of refrigerant.

The vapor should have a relatively low specific volume, as this volume establishes the required size or displacement of the compressor. This property is more important to the positive displacement compressor than it is to the centrifugal machine, which is inherently a large volume, low-pressure pump.

The refrigerant should have a low specific heat in the liquid state so that less heat will be required to cool the liquid from the condensing temperature down to the evaporation temperature. The heat required for this cooling results in "flash gas," and subtracts from the refrigeration effect or cooling capacity of the refrigerant circulated.

The refrigerant should be easily detected by leak detection equipment to locate leaks in the system.

The refrigerant should be compatible with compressor lubricants and oils, and should not detract from their effectiveness as lubricants. The refrigerant and oil also need to miscible, particularly in the refrigerant vapor phase, so that oil in circulation in the system is returned to the compressor.

The coefficients of heat transfer should be conducive to good heat transfer rates. The refrigerant should also have low viscosity to minimize pumping losses.

The refrigerant should be readily available, low in cost, and easily handled.

The refrigerant should be non-corrosive to the metals usually employed in a hermetic refrigeration system and compatible with the materials used for gaskets and in hermetic motor construction.

The refrigerant should be environmentally sound. It has been determined that when refrigerants containing the chlorine atom are released into the atmosphere they tend to breakdown the earth's natural ozone layer. This ozone forms a protective layer to shield us from the sun's UV rays. Different refrigerants also impact the amount of greenhouse gas produced from their use and need to be evaluated on their global warming potential.

The best choice of a refrigerant should be one that is safe, efficient, stable, cost effective, and compatible.

Refrigerant Choices

There is not a perfect refrigerant that meets all of the criteria described; selecting a refrigerant is a compromise. There are a limited number of substances that can be used as a refrigerant. These could be divided into two broad classes, organic compounds and inorganic compounds.

Refrigerants can be classified into several groups based on composition

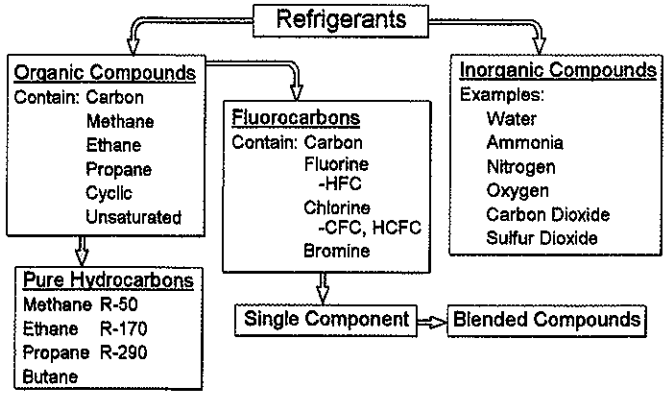


Figure 46

Classes of Refrigerants

Inorganic compounds are compounds like water, which we have used for a large portion of our text. Some of the others are the refrigerants used for many of the early refrigeration systems like carbon dioxide, and ammonia. Ammonia is still used extensively in low-temperature refrigeration work. These refrigerants do not present a safety risk as far as flammability is concerned; but, like ammonia, they do have toxicity issues and corrosion potential issues. As we explained earlier, water is used as a refrigerant but is a poor choice for the vapor compression cycle. Some inorganic compounds, like carbon dioxide, require extremely high pressures. While still used in some applications, these factors have limited the use of inorganic refrigerants in favor of organic substances.

Organic refrigerants contain carbon. The organic group can be divided into two, pure hydrocarbons and fluorocarbons. The pure hydrocarbons are refrigerants like methane, also called natural gas, ethane, propane, and butane. These may be recognized as the ones used for a number of combustion applications, like propane and methane for heating and butane for lighters. This group of refrigerants is all flammable. Though readily available, the flammability issue has limited their use. However, because of the trend away from fluorocarbon refrigerants for environmental reasons, this group of refrigerants is seeing renewed interest. Current technology is to use these in sealed systems limiting the amount of refrigerant, and to use them to cool a secondary fluid to be circulated to the actual load, in much the same way a chilled water chiller operates.

That leaves the choice of refrigerants to the fluorocarbons. This group of refrigerants consists of the ones that contain fluorine, chlorine, or bromine along with the carbon atom. Depending on the amount of chlorine and fluorine, these refrigerants are commonly called CFC, HCFC, and HFC.

Fluorocarbon Refrigerants

The group of fluorocarbon refrigerants has two categories of refrigerants, single element compounds and blends of single element compounds.

The single element refrigerants consist of a carbon atom bonded to one or more of the fluorine, chlorine, or bromine atoms. Refrigerants such as R-11 and R-12 contain both chlorine and fluorine and are referred to as CFC. These refrigerants have a high concentration of the chlorine atom. Refrigerants like R-22 and R-123a also contain the chlorine atom but in a smaller concentration than CFCs. These refrigerants are HCFCs. Refrigerants like R-134a contain fluorine and no chlorine atoms and are called HFCs.

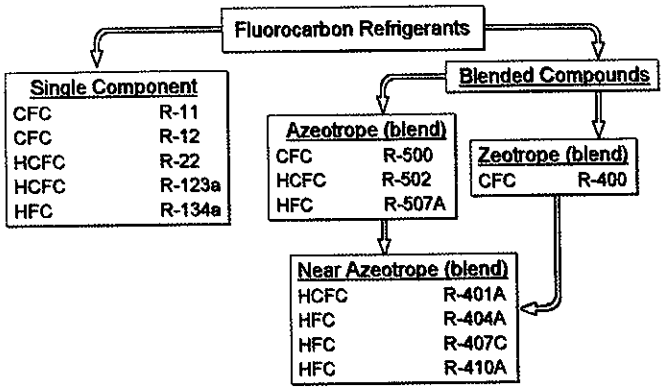
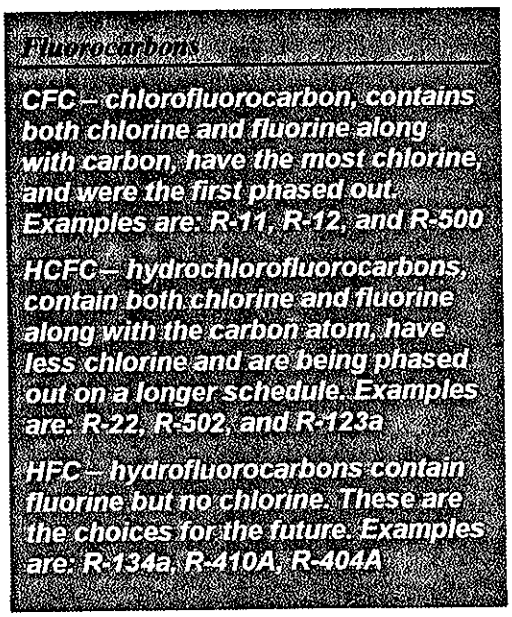


Figure 47

Fluorocarbon Refrigerants



To develop refrigerants that have more desirable properties, the single component refrigerants are mixed or "blended" in varying proportions. This mixing is done to achieve a mixture of refrigerants that has the required heat transfer properties for an application. In blended refrigerants, the refrigerants do not chemically bond; each of the refrigerants retains it own properties. Like a mix of oil and water, these refrigerants can be separated back to their original refrigerant.

The blended refrigerants are classified into two groups depending on how closely the blend behaves like a single element refrigerant. Refrigerants that act like a single compound refrigerant are called azeotropic blends. Refrigerants that vary from the performance of a single element compound are called zeotropic blends. Zeotropic blends that act much like a single compound are called near azeotrope. In zeotropic blends, the refrigerants have different saturated liquid and saturated vapor boiling points due to the influences of the different refrigerants. This phenomena, referred to as glide, results in different pressure/temperature saturated conditions in the liquid and vapor phases. The liquid-saturated condition is called bubble point and the vapor-saturated condition is called dew point.

Blended refrigerants can be CFCs like R-500 or HCFCs like R-502 or HFCs like R-404A, R-407C and R-410A. Refrigerants R-404A and R-410A are near azeotropic refrigerants.

Research has indicated that refrigerants containing chlorine are major contributors to the breakdown of the earth's natural ozone layer. As a result, world's communities have established a phase out of chlorine based refrigerants. The CFC refrigerants are currently banned from new unit production and replacement refrigerant for existing machines is being ramped down to force their eventual replacement. The HCFC have less chlorine and are also being phased-out but on a longer schedule.

Additional information on refrigerants can be found in TDP-402, Refrigerants.

Common Refrigerants

With phase out and performance issues, what are the refrigerants of choice?

Refrigerant R-22 is one of the most commonly used refrigerants in vapor compression equipment today, but its use is being phased out because of environmental concerns. For long-term choices, R-134a is becoming the choice for chiller and mobile applications. R-410A and R-407C are becoming the choices for packaged units and residential and commercial split system units. Refrigerant R-404A is the choice for low-temperature refrigeration.

Summary

In this module the four principles that explain how mechanical refrigeration works have been described. The principles were applied through four basic system components to have a cycle that could use a volatile fluid and "*and bringing them again and again into operation without waste.*" The components of the refrigeration systems have been described, including how the system can be used for cooling and for heating. The location and basic function of the refrigeration specialty components sometimes applied with a refrigeration system were identified. In addition, the terms associated with vapor compression refrigeration have been described.

In the next module (TDP-401, Principles of Mechanical Refrigeration, Level 2: Analysis) the components will be described with the use of the pressure-enthalpy diagram. This will allow the calculation of the performance, selection, and evaluation of operation of each component of the system. The p-h diagram will also be used to show the impacts of various system options and of design operation. A good understanding of these issues will allow a designer to understand the impact design decisions have on system performance.

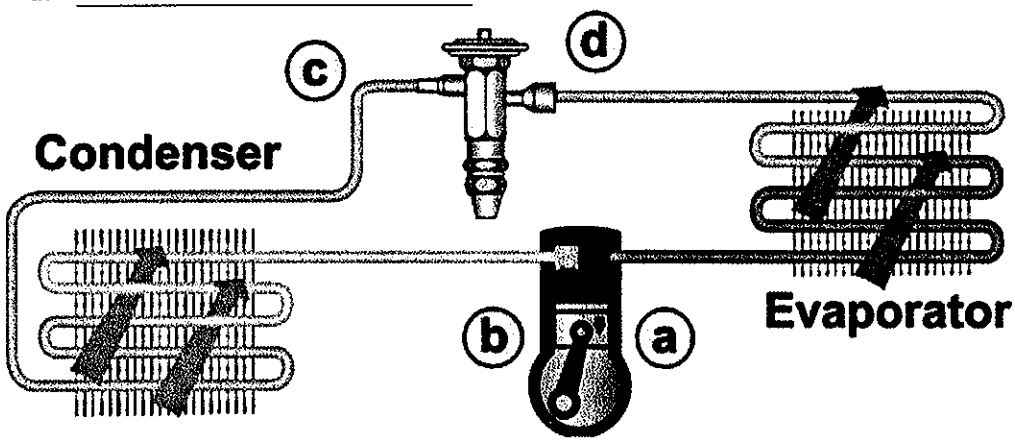
Before proceeding to the next module, Mechanical Refrigeration, Level 2: Analysis, it is a good time to review the material presented in this module. A work session is provided to check your understanding of the material.

Work Session

1. How many Btu are required for each of the following (assume sea level):
   a. Heat 1 pound of water from 60° F to 80° F? _____
   b. Cool 2 pounds of water from 95° F to 85° F? _____
   c. Heat 5 pounds of water from 40° F to 280° F? _____
   d. Cool 2 pounds of water from 40° F to 32° F ice? _____
   e. Heat 1 pound of ice at 32° F to steam at 212° F? _____

2. What is the warmest temperature the refrigerant can be to maintain a deli case cooler at 22° F? _____

3. Identify on the chart below the relative temperature and liquid or vapor condition of the refrigerant at each of the following points.
   a. _____
   b. _____
   c. _____
   d. _____



4. Explain the terms superheat and subcooling?
   a. _____
   b. _____

5. Identify each of the refrigerants as CFC, HCFC, HFC, pure hydrocarbon, inorganic or not a refrigerant

   a. Carbon Dioxide _____
   b. R-410A _____
   c. R-11 _____
   d. R-718 _____
   e. Ammonia _____
   f. R-22 _____
   g. R-123a _____
   h. R-134a _____
   i. Water _____

6. The four principles which can explain how the mechanical refrigeration cycle operate are:

   a. _____
   b. _____
   c. _____
   d. _____

7. List the four components every mechanical refrigeration cycle must contain to be a continuous process:

   a. _____
   b. _____
   c. _____
   d. _____

8. Explain azeotropic, zeotropic, and near azeotropic refrigerants

   _____
   _____
   _____

9. What are the key differences required for a system to operate as a heat pump?

_____

_____

10. Identify the refrigerant line each of the following components is located in:

a. Accumulator _____
b. Receiver _____
c. Moisture-liquid indicator _____
d. Filter-drier _____
e. Muffler _____
f. Head pressure control valve _____

11. What are the two key methods of heat transfer used in mechanical refrigeration?

a. _____
b. _____

12. Describe why the temperature of the vapor entering the compressor does not match the saturation temperature at the compressor? _____

_____

13. The _____ molecule in refrigerants is responsible for the breakdown of the earth's protective ozone layer.

14. Describe the terms

a. Flash gas _____
b. Heat of compression _____
c. Enthalpy _____
d. Desuperheating _____

15. True or False? A thermometer can measure the total heat quantity in a vapor? _____

Notes

Notes

Appendix

Work Session Answers

1.
   a. 20 Btu $(80 - 60) * 1$
   b. 20 Btu $(95 - 85) * 2$
   c. 5740.6 Btu $((212 - 40) * 5) + (970 * 5) + ((240 - 212) * .45 * 5)$
   d. 304 Btu $((40 - 32) * 2) + (144 * 2)$
   e. 1294 Btu $((144 * 1) * 2) + ((212 - 32) * 1) + (970 * 1)$
      ice water vapor

2. Temperature must be less than 22° F

3.
   a. cold superheated vapor
   b. hot highly superheated vapor
   c. warm subcooled liquid
   d. cold liquid / vapor mix

4.
   a. Superheat is heat added above the saturated vapor temperature
   b. Subcooling is temperature below the saturated liquid temperature

5.
   a. inorganic
   b. HFC
   c. CFC
   d. Inorganic (water)
   e. Inorganic
   f. HCFC
   g. HCFC
   h. HFC
   i. Inorganic

6.
   a. Heat will only flow from a higher temperature to a lower temperature
   b. The greatest heat energy is in a change of state
   c. Temperature is a function of pressure
   d. Flow only occurs if a pressure difference exists

7.
   a. Evaporator – (heat absorber)
   b. Compressor – (vapor pump)
   c. Condenser – (heat rejecter)
   d. Metering device – (pressure reducing valve)

8.
   a. Azeotropic – mix or blend of refrigerants where the mixture does not act as a single refrigerant.
   b. Zeotropic – mixture of refrigerants where the mixture acts like a single refrigerant
   c. Near azeotropic – zeotropic blend of refrigerants which approaches being like a azeotropic blend

9. A heat pump needs an accumulator, reversing valve, bi-flow expansion valves, special coil circuiting, special compressor, and crankcase heater on the compressor.

10.
   a. Suction line before the compressor
   b. Liquid line after the condenser
   c. Liquid line before the metering device
   d. Liquid line before the metering device
   e. Hot gas line after the compressor
   f. Liquid line after the condenser

11.
   a. Conduction, i.e. transfer of heat from refrigerant to tube
   b. Convection, both natural and forced, i.e. transfer of heat from the evaporator coil to the air

12. The vapor is superheated to prevent liquid refrigerant from entering the compressor

13. Chlorine

14.
   a. Flash gas is the small part of the refrigerant which changes to a vapor at the expansion valve and cools the rest of the refrigerant
   b. Heat of compression is the work (heat energy) put in by the compressor in raising the pressure of the vapor above the saturated condensing temperature
   c. Enthalpy is total heat content and is the sum of sensible and latent heat
   d. Desuperheating is the process of removing the superheat from the vapor to the saturated vapor temperature. This heat in the condenser is the superheat added in the evaporator and the heat of compression.

15. False. A thermometer can only measure sensible heat, not total heat.

Prerequisites:

None, however, a basic understanding of the concepts of comfort air conditioning is recommended and is provided in TDP-102, ABCs of Comfort, Cat. No. 796-026.

Learning Objectives:

At the conclusion of this module, participants will be able to:

- Identify the terms commonly used to describe the mechanical refrigeration process.
- List the four basic principles that are applied in a mechanical refrigeration cycle.
- Describe the methods of heat transfer used in the refrigeration cycle.
- Describe the functions of the four parts of refrigeration systems.
- Explain the characteristics required of a refrigerant.
- Describe the location and use of refrigerant specialty components.
- Explain the operational principle of a heat pump.

Supplemental Material:

Form No.	Book Cat. No.	Instructor CD Cat. No.	Title
TDP-401	796-084	797-084	Principles of Mechanical Refrigeration, Level 2: Analysis
TDP-402	796-038	797-038	Refrigerants
TDP-403	796-039	797-039	Expansion Devices and Refrigeration Specialties
TDP-405	796-041	797-041	Reverse Cycle Refrigeration
TDP-501	796-042	797-042	Refrigerant Piping

Instructor Information:

Each TDP topic is supported with a number of different items to meet the specific needs of the user. Instructor materials consist of a CD-ROM disk that includes a PowerPoint™ presentation with convenient links to all required support materials required for the topic. This always includes: slides, presenter notes, text file including work sessions and work session solutions, quiz and quiz answers. Depending upon the topic, the instructor CD may also include sound, video, spreadsheets, forms, or other material required to present a complete class. Self-study or student material consists of a text including work sessions and work session answers, and may also include forms, worksheets, calculators, etc.



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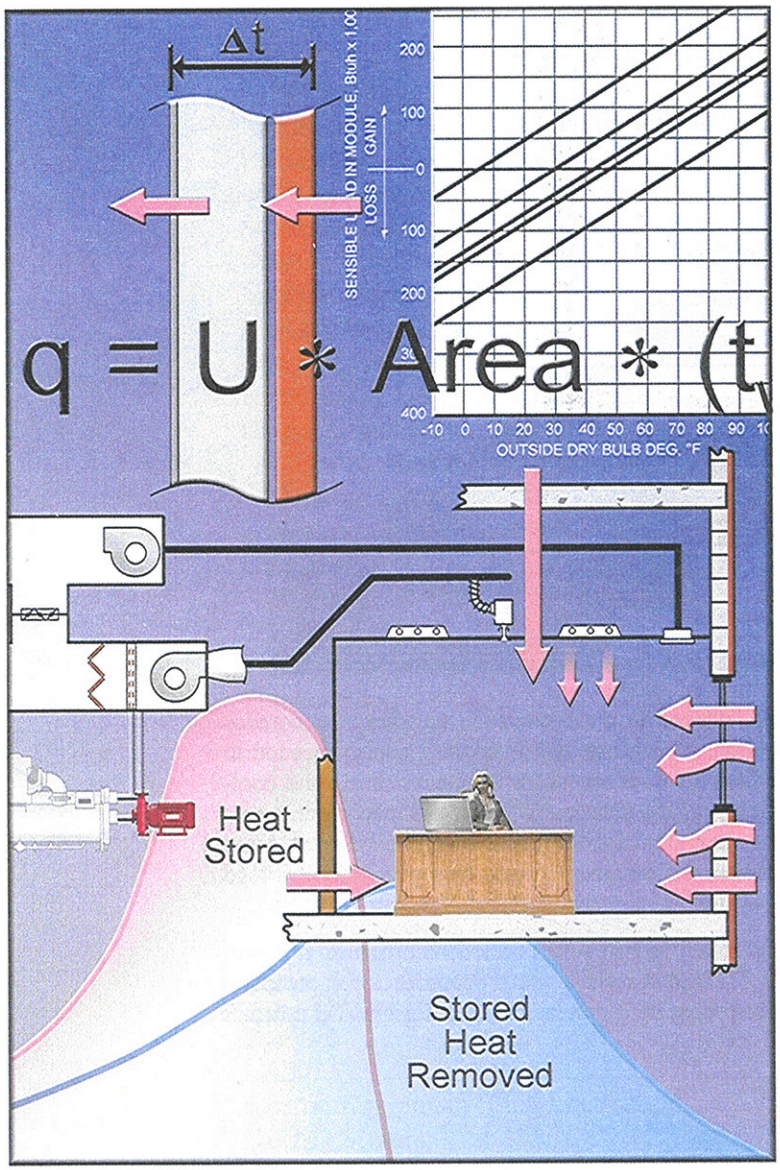


COMMERCIAL LOAD ESTIMATING

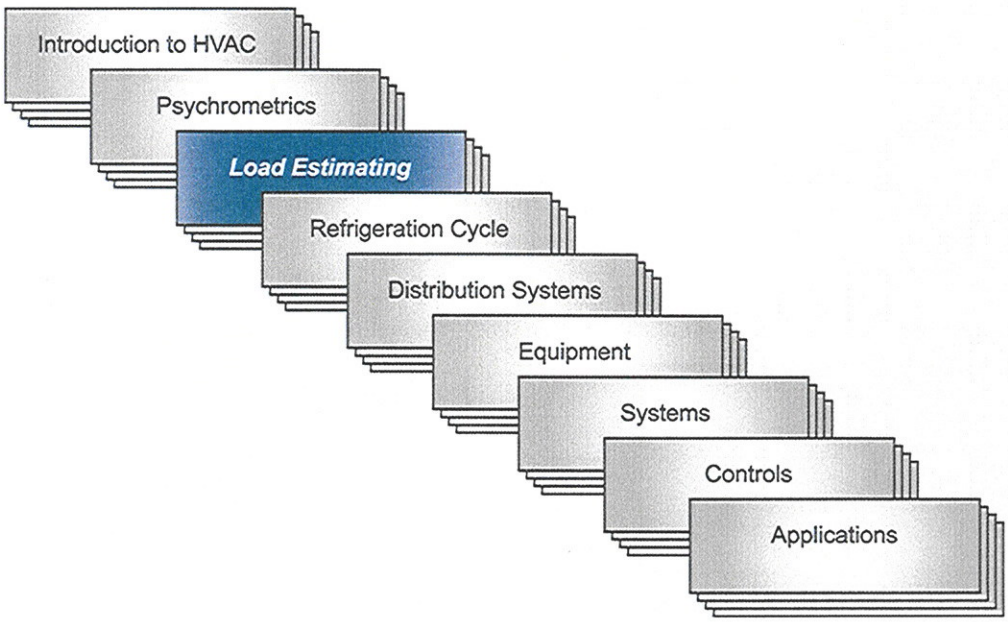
Load Estimating

Level 2: Fundamentals

Technical Development Program

Technical Development Programs (TDP) are modules of technical training on HVAC theory, system design, equipment selection and application topics. They are targeted at engineers and designers who wish to develop their knowledge in this field to effectively design, specify, sell or apply HVAC equipment in commercial applications.

Although TDP topics have been developed as stand-alone modules, there are logical groupings of topics. The modules within each group begin at an introductory level and progress to advanced levels. The breadth of this offering allows for customization into a complete HVAC curriculum – from a complete HVAC design course at an introductory-level or to an advanced-level design course. Advanced-level modules assume prerequisite knowledge and do not review basic concepts.



The fundamentals of commercial load estimating are needed to understand the various load components that go into making a practical estimate of the amount of heating and/or cooling energy needed to condition a building. Done properly, a load estimate provides the data necessary to select heating and cooling equipment that can condition the spaces within a building. If the characteristics of the load components for the building and the HVAC system are known, then an analysis of the application can be used to come up with the correct load and equipment selections to complete the design. Along with psychrometrics, load estimating establishes the foundation upon which HVAC system design and operation occur.

Load Estimating, Level 2: Fundamentals is the second in a four-part series on load estimating. It is preceded by an overview of the topic and followed by two TDPs that review the procedures for completing block and zone load estimates and refinements required for preparing a system-based design load estimate.

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Introduction

This Technical Development Program (TDP) on commercial load estimating presents the fundamentals needed to understand the various ***load components*** that go into making a practical estimate of the amount of ***heating*** and/or cooling energy needed to provide comfort conditioning of a building. Once a project begins to take shape, the job requirements (a scope of work) can be tabulated for the HVAC portion. Given this data, it is possible to perform a ***load estimate*** for the selection of the heating and cooling equipment that will condition the various building spaces. In the earliest stages, the load estimate will tell the designer how big the job is, either in terms of cooling capacity, expressed as ***tons of refrigeration*** or MBtuh (1000s of ***Btuh***, or in terms of airflow in cfm. ***Airflow*** refers to how much conditioned air needs to be supplied to the various ***rooms***, or ***spaces***, to maintain the temperature, humidity, or CO_2 ***set points***.

If the designer knows the elements of the loads for the building and the features of the HVAC system, then the application can be analyzed regardless of the calculation method used, and correct equipment selections can be made to complete the design.

If a simple, ***rule of thumb*** load estimate has been used during preliminary design, then later on in the design process the building and HVAC loads should be calculated in detail. When completed, all the necessary parameters for sizing the selected ***packaged equipment*** (rooftops units and split systems) or ***applied equipment*** (chillers, boilers, and air handlers), along with terminals and air diffusion devices will be available to the designer.

Importance of Load Estimating

An accurate load estimate is needed to get the process of designing, installing, and operating a project off to a good start. The load estimate numbers provide the data for a host of subsequent calculations, selections, and decisions. Among these items are: HVAC system selection; equipment selections for fans, coils, and pumps; duct and electrical feeder sizing; and water piping design.

An accurate estimate will provide the correct cooling and heating requirements, offer options for load reductions at the least incremental cost, provide properly sized equipment, and yield efficient air, water, and electrical distribution designs.

Load Estimate versus Load Calculation

One often hears load estimate and ***load calculation*** used synonymously. A load estimate is an approximation gained from methods that are not detailed, such as a rule of thumb, ***check figures***, or quick hand calculation based on assumptions. Check figures and rules of thumb have been derived for common applications by HVAC designers and are often complied for ease of use. Some common check figures represent cooling in ft^2/ton and people density as ft^2/person.

A load calculation is a more detailed analysis of load components based on actual building design knowledge and is usually performed by computer software spreadsheets and programs. Even though a load calculation is considered more accurate, not all the details of the inputs required by the software are known. The user must rely on good judgment, so the word "estimate" is still appropriate for the results.

Current calculation models have increased the accuracy of software programs. However, simplifying assumptions are a part of these methods too, so as far as trying to approach the reality of nature, it is still an estimate, but on increasingly higher levels.

Proper load estimating requires gathering enough details to obtain the level of accuracy required. Preliminary HVAC load estimates can be gained from minimal project knowledge by use of rules of thumb and check figures, since order of magnitude for preliminary budgets and rough schematics are the goals.

As a project progresses in the design process, more information becomes available and the load estimates must be refined. The methodologies used for the detailed sizing calculations or system modeling often will dictate the type and depth of information that is needed for input into various software programs. These more in-depth subjects will be covered in the two remaining load estimating TDPs.

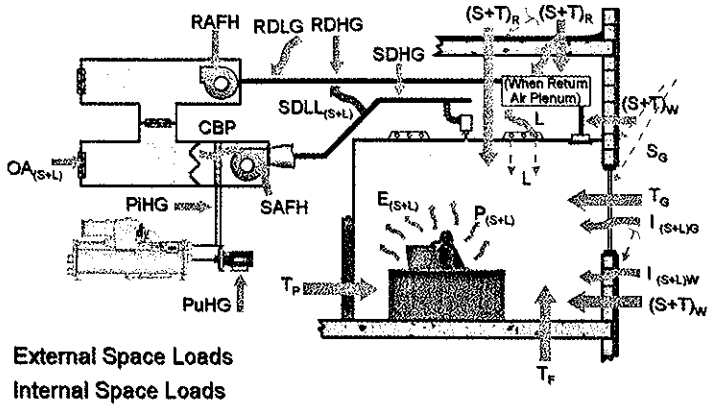


Figure 1

Cooling Estimate Load Components

Load estimates are the summation of heat transfer elements into (gains) or out of (losses) the spaces of a building. Each heat transfer element is called a load component. Load components represent sensible and latent heat transfers (in Btuh) from a multitude of sources, but all can be assembled into one of three basic groups, external space loads, internal space loads, and system loads. Each will be covered in detail in a separate section of this TDP. The Appendix contains a terms list for the abbreviations found in the figure.

To properly understand the workings of the various external, internal and ***system load components***, details regarding the following items will need to be gathered from a set of plans, existing building surveys, or occupant interviews:

- Occupancy of the entire building (office, health care, retail store, educational, etc.)
- Orientation of the building (sun effects on surfaces)
- Weather data (design conditions, heat transfer Δt)
- Use of the spaces within the building (offices, conference room, lab, data center)
- Hours of operation (occupied and unoccupied)
- Thermostat set points (main comfort parameter)
- Dimensions of walls, roofs, windows, and doors
- Construction materials (gather densities, external color, and ***U-factors***, or describe material type layer-by-layer (***R-value***s)
- Stairways and elevators (floor-to-floor openings)
- People occupancy and activity, and when they are present
- Lighting intensity and hours used
- Motor and appliance sizes or kW, and times they are used
- Ventilation needs (IAQ and exhaust makeup)

While many of these items will be explored as this module progresses, the Load Estimating TDP on Block and Zone Loads needs to be consulted for a full ***building load takeoff*** example.

Types of Load Estimating

There are many types of load estimating and calculation procedures. They range from fast and easy hand calculations to detailed descriptions for advanced computational procedures and methods. You might want to think of these like golf clubs or fishing rods. You may need to have more than one type on hand depending on what you need to do.

Rules of Thumb

There are tables of rules of thumb available from various industry sources. One such compilation is found in the Carrier *Commercial Systems Quick Reference* (CSQR). These are reproduced in the Appendix of this manual. The Cooling Check Figures allow for approximation of total cooling needs (building gross area * cooling ft^2/ton), electrical space load component (building gross area * lights + equipment Watts/ft^2), and outdoor air ventilation requirements (building gross area ÷ people ft^2/person * ventilation ***cfm***/person). The Approximate Design Air Distribution cfm (Cooling) table allows early approximation of the ***zone*** primary supply air needs for terminal selection and ductwork space evaluation.

Rules of Thumb

These ballpark numbers are very useful in the preliminary design phases and are good for comparing with the results of more detailed procedures to see if any gross input errors were made.

Manual vs. Software Generated

Going beyond the rules of thumb method, the designer now has to do a load takeoff and provide a level of detail on the project that allows individual computations for the major and hopefully important minor load components.

A manual method can use a tabular form like the Carrier E-20A, Air Conditioning Load Estimate form (see Appendix), or a computer spreadsheet like Excel or Lotus. In one, a pencil and a calculator are used to run a load estimate for a simple building or to study unusual space or zone requirements. In the other, input is still manual, but the computer spreadsheet tabulates the many calculations required for larger multi-room systems with extensive ***zoning*** or multiple central equipment selections to be sized.

Manual methods still have a place, especially in the early stages of design, and when a quick calculation is needed to check sizing requirements or the magnitude of load change created by a change in a takeoff value. Larger commercial jobs, especially ones with unusual load components, should have the cooling and heating loads estimated using one of the many software load estimating programs available to designers. This method proves to be efficient whenever a change in the takeoff at the later stages of design requires a recompilation of the load estimate.

The computer printed outputs generally organize the data well (Figure 2), and in some cases can provide graphical outputs of the detailed calculations, such as a psychrometric plot of the system design (Figure 3). Software-generated load estimates run the gambit from simple load estimating to a full computer ***energy simulation*** derived from detailed inputs that include full system and component descriptions. The energy calculation modeling is described in the energy estimating TDP.

Air System Design Load Summary for Top Floor

Project Name: Chicago Office Building (TDP-301)
09/09/2004 08:40AM
Prepared by:

	DESIGN COOLING			DESIGN HEATING		
	COOLING DATA AT Jul 1500 COOLING OA DB / WB 91.0 °F / 74.0 °F			HEATING DATA AT DES HTG HEATING OA DB / WB -6.0 °F / -7.2 °F		
ZONE LOADS	Details	Sensible (BTU/hr)	Latent (BTU/hr)	Details	Sensible (BTU/hr)	Latent (BTU/hr)
Window & Skylight Solar Loads	2200 ft²	69444	-	2200 ft²	-	-
Wall Transmission	4400 ft²	12367	-	4400 ft²	60557	-
Roof Transmission	14814 ft²	31416	-	14814 ft²	101100	-
Window Transmission	2200 ft²	14972	-	2200 ft²	91111	-
Skylight Transmission	0 ft²	0	-	0 ft²	0	-
Door Loads	0 ft²	0	-	0 ft²	0	-
Floor Transmission	0 ft²	0	-	0 ft²	0	-
Partitions	0 ft²	0	-	0 ft²	0	-
Ceiling	0 ft²	0	-	0 ft²	0	-
Overhead Lighting	23628 W	67014	-	0	0	-
Task Lighting	7876 W	24354	-	0	0	-
Electric Equipment	12759 W	40257	-	0	0	-
People	79	14557	16146	0	0	0
Infiltration	-	0	0	-	0	0
Miscellaneous	-	0	0	-	0	0
Safety Factor	0% / 0%	0	0	10%	25277	0
= Total Zone Loads	-	274381	16146	-	278045	0
Zone Conditioning	-	303189	16146	-	268466	0
Plenum Wall Load	0%	0	-	0	0	-
Plenum Roof Load	0%	0	-	0	0	-
Plenum Lighting Load	0%	0	-	0	0	-
Return Fan Load	12900 CFM	0	-	1213 CFM	0	-
Ventilation Load	1710 CFM	25596	53044	1607 CFM	124115	0
Supply Fan Load	13294 CFM	38601	-	1607 CFM	-2886	-
Space Fan Coil Fans	-	0	-	-	-1570	-
Duct Heat Gain / Loss	2%	5488	-	2%	5561	-
= Total System Loads	-	372874	69190	-	393685	0
Central Cooling Coil	-	372874	69198	-	0	0
Preheat Coil	-	0	-	-	98260	-
Terminal Reheat Coils	-	0	-	-	295386	-
= Total Conditioning	-	372874	69198	-	393645	0
Key:	Positive values are clg loads Negative values are htg loads			Positive values are htg loads Negative values are clg loads		

Figure 2

System Block Load

Space, Zone, and Block Loads for Equipment Sizing

Each of the load types, ***space loads***, ***zone loads*** and ***block loads*** are all used for equipment sizing, and can be obtained from almost any load estimating software or hand calculation method. The purpose is to get the Btuh and fluid quantities (cfm and gpm) for selecting diffusers, sizing fans, and determining the cooling and heating capacities for selecting the zone and system HVAC equipment. A total building or project block load is useful for sizing central cooling and heating plant equipment such as chillers and boilers. A system level block load (Figure 2) provides fan and coil sizing details for the central air-handling units.

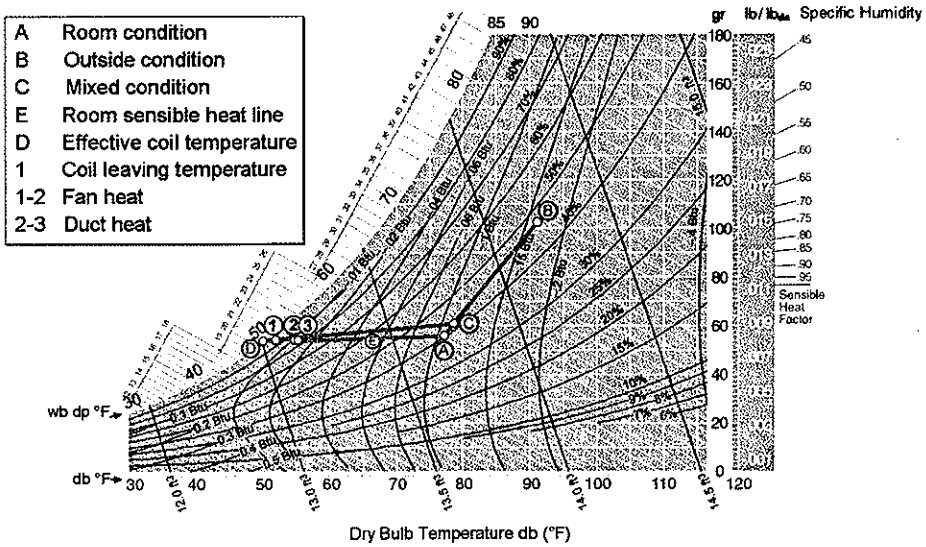


Figure 3

Psychrometric Plot of an HVAC System

A zone load provides the information to select such items as CV and VAV terminals and terminal heating coils (Figure 4), and room air diffusers (Figure 5). The peak block load and zone loads may occur at different times of the day and at different months, so each is usually run. The Carrier *Block Load* program and the *Hourly Analysis Program (HAP)* are examples of this type of software.

One item of importance not listed previously is the type of system equipment that is chosen for the project. Some procedures will have an input for cooling coil ***bypass factor*** or some other input regarding the efficiency of the cooling coil. A typical 20-ton rooftop HVAC unit will have a bypass factor of about 0.08. This will be discussed in later in the module.

Even more important is the cooling supply air temperature used in the calculations. This is needed to select the equipment that will meet the load requirements, and perform as expected on startup. Applied equipment has many more choices than packaged equipment for selecting coil rows, fins, and circuiting; so a 55° F supply air temperature for various system applications can be easily achieved. The lower supply air set point gives better part load dehumidification, very important in limiting room IAQ issues around moisture damage and mold growth.

Zone Sizing Summary for Top Floor

Project Name: Chicago Office Building (TOP-301)
09/09/2004
Prepared by: 08:40AM

Air System Information

Air System Name ..... Top Floor
Equipment Class ..... CW AHU
Air System Type ..... VAV
Number of zones 9
Floor Area ..... 15752.0 ft²
Location ..... Chicago IAP, Illinois

Sizing Calculation Information

Zone and Space Sizing Method:
Zone CFM ..... Peak zone sensible load
Space CFM ..... Individual peak space loads
Calculation Months ..... Jan to Dec
Sizing Data ..... Calculated

Zone Sizing Data

Zone Name	Maximum Cooling Sensible (MBH)	Design Air Flow (CFM)	Minimum Air Flow (CFM)	Time of Peak Load	Maximum Heating Load (MBH)	Zone Floor Area (ft²)	Zone CFM/ft²
North West Corner Top	8.1	385	16	Jul 1600	8.8	156.3	2.46
North East Corner Top	6.9	329	16	Jul 0800	8.8	156.3	2.10
South East Corner Top	8.1	384	16	Sep 0900	8.8	156.3	2.46
South West Corner Top	9.1	434	16	Sep 1400	8.8	156.3	2.78
North Side Top	39.0	1852	188	Jun 1700	59.6	1875.6	0.99
North West Corner Top	36.4	1728	94	Jul 0800	29.8	937.8	1.84
East Side Top	72.9	3456	188	Jul 0800	59.6	1875.6	1.84
West Side Top	39.4	1868	94	Jul 1600	22.8	937.8	1.99
Interior Top	112.0	5314	950	Jul1700	71.3	9500.0	0.56

Zone Terminal Sizing Data

Zone Name	Reheat Coil Load (MBH)	Reheat Coil Water gpm @ 40.0 °F	Zone Htg Coil Load (MBH)	Zone Htg Water gpm @ 40.0 °F	Mixing Box Fan Airflow (CFM)
North West Corner Top	9.0	0.45	0.0	0.0	166
North East Corner Top	9.0	0.45	0.0	0.0	166
South East Corner Top	9.0	0.45	0.0	0.0	166
South West Corner Top	9.0	0.45	0.0	0.0	166
North Side Top	62.5	3.13	0.0	0.0	1131
North West Corner Top	31.3	1.56	0.0	0.0	565
East Side Top	62.5	3.13	0.0	0.0	1131
West Side Top	24.2	1.21	0.0	0.0	432
Interior Top	86.3	4.32	0.0	0.0	0

Figure 4

Zone Sizing Summary

Zone Sizing Summary for Top Floor

Project Name: Chicago Office Building (TOP-301)
09/09/2004
Prepared by: 08:40AM

Air System Information

Air System Name ..... Top Floor
Equipment Class ..... CW AHU
Air System Type ..... VAV
Number of zones 9
Floor Area ..... 15752.0 ft²
Location ..... Chicago IAP, Illinois

Sizing Calculation Information

Zone and Space Sizing Method:
Zone CFM ..... Peak zone sensible load
Space CFM ..... Individual peak space loads
Calculation Months ..... Jan to Dec
Sizing Data ..... Calculated

Space Loads and Airflows

Zone Name / Space Name	Mult.	Cooling Sensible (MBH)	Time of Load	Air Flow (CFM)	Heating Load (MBH)	Floor Area (ft²)	Space CFM/ft²
North West Corner Top							
North West Corner Top	1	8.1	Jul 1600	385	8.8	156.3	2.46
North East Corner Top							
North East Corner Top	1	6.9	Jul 0800	329	8.8	156.3	2.10
South East Corner Top							
South West Corner Top	1	8.1	Sep 0900	384	8.8	156.3	2.46
South West Corner Top							
South West Corner Top	1	9.1	Sep 1400	434	8.8	156.3	2.78
North Side Top							
North Typical Top	12	3.3	Jun 1700	154	5.0	156.3	0.99
East Side Top							
East Typical Top	6	6.1	Jul 0800	288	5.0	156.3	1.84
South Side Top							
South Typical Top	12	6.1	Jul 0800	288	5.0	156.3	1.84
West Side Top							
West Typical Top	6	6.6	Jul 1600	311	3.8	156.3	1.99
Interior Top							
Interior Typical Top	4	28.0	Jul 1700	1329	17.8	2375.0	0.56

Figure 5

Space Sizing Summary

Note:

Most packaged equipment and room fan coils have limited coil circuitry options and as a general guideline, deliver a supply air temperature of about 58° F to 59° F.

Safety Factors

A ***safety factor*** should be a function of the designer's confidence in his or her load survey and the details gathered on the project's construction materials.

There is usually a built in creeping safety factor as the HVAC system is designed. A creeping safety factor occurs as additive items of capacity and safety are added on top of each other due to the nature of the estimating method and equipment sizing procedures. For instance, assume the room airflow requirement is 108 cfm of conditioned supply air. This value was then increased to 113 cfm based on a 5 percent sensible load safety factor. The equipment selected to deliver air to the room would probably then be rounded up and selected for 125 cfm. This is a 10 percent increase on top of the original 5 percent calculation safety factor.

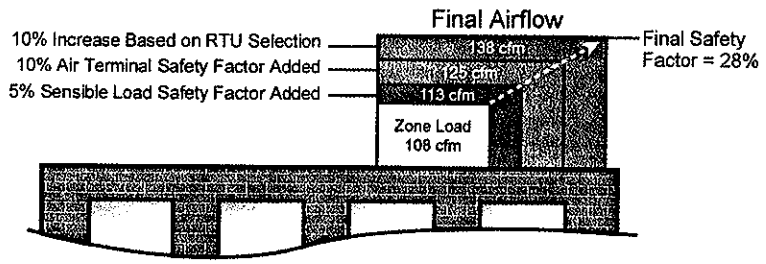


Figure 6

Creeping Safety Factor

Then the total load is calculated for the cooling equipment at 105,445 Btuh (8.79 tons), again based on a safety factor. However, the piece of equipment selected to do the job only comes in discreet sizes such as 5, 7½ and 10 tons. So a 10-ton unit is installed, which is over capacity by 10 percent on a load that already contained multiple safety factors. In fact, for more favorable outside conditions than the standard rating values (80° F db/67° F wb), the unit capacity can be even more oversized.

Minimal Safety Factors

The use of safety factors in the sizing of heating and cooling equipment is discouraged. Anyone who has taken off and double-checked a set of plans knows that there are many factors in addition to the mathematical calculations that affect the heating and cooling loads.

This over-conservative approach to cooling sizing causes over-sized equipment to be installed. This results in less efficient operation and difficulty in controlling room conditions, especially relative humidity and the associated IAQ (***indoor air quality***) problems.

Designers should examine the results of the calculations, understand where the numbers come from, and satisfy themselves that the resulting numbers make sense. Because beginners may be unsure of the calculation methods and output values, the tendency is to use safety factors as insurance. If one feels that this insurance is needed, a factor of a few percent, not to exceed 5 percent, is suggested for cooling. As you will see latter in the heating section, a single safety factor is recommended, often quite large (up to 25 percent) in cold climates.

Loads Run for HVAC System Modeling

There are software applications that can model and size different kinds of HVAC systems and components in more detail than basic block and zone load software. This modeling software allows the designer to do system-based design because the inputs describe schedules, set points and equipment operation unique to the selected HVAC system. For programs like Carrier's *HAP*, by building on the information already used for the detailed load modeling, an annual energy analysis simulation can also be done. Additional items needed beyond the load calculation inputs include hourly weather data for the entire year, comprised of 8760 instances of dry bulb, wet bulb, and solar data, electric and fuel rate costs, and equipment efficiencies. With this degree of input data and program computational methods, detailed ***life cycle costs*** for alternative HVAC designs can easily be compared.

Energy Code Requirements

Now that the various model building codes across the United States have been unified into the International Codes Council family of codes, it is easy to discuss energy code requirements by referring to the International Energy Conservation Code 2003 requirements. As the design progresses, the designer should make sure that the following items are completed:

- Read over the Administrative and Definitions chapters to coordinate those requirements with the specific project being worked on.
- Determine exterior design conditions (dry and wet bulb temperatures, and degree days) from ASHRAE (American Society of Heating, Refrigerating, and Air-Conditioning Engineers) and NOAA (National Oceanic and Atmospheric Administration) references.
- By direct reference, all commercial buildings follow *ASHRAE Standard 90.1-2001, Energy Standard for Buildings Except for Low-Rise Residential Buildings*. While most items in Section 6 (Heating, Ventilating, and Air-Conditioning) of ASHRAE 90.1 relate to system design and operating efficiency (see 90.1 summary in Appendix), paragraph 6.2.2 requires running heating and cooling load estimates before selecting equipment.
- Chapter 8, which provides an alternative to ASHRAE 90.1 for architects who want to follow "acceptable practice" tables for building exterior envelope heat transfer *U*-factors and *R*-values.
- Model Codes can always be superseded by the AHJ (local Authority Having Jurisdiction, or code official). The designer needs to review critical issues (e.g. will the simplified approach option, paragraph 6.1.3, be allowed for your project?) with local standards and/or ask for interpretation from the AHJ directly.

Load Estimating Methods

Over time, load estimating has become more sophisticated. The early methods, like instantaneous method and storage load factor/ETD method, use hand calculations that were not quick and easy, even with the simplifying assumptions and look-up tables. Doing an entire building makes for lengthy calculations. Working up to the current methods, which try to track actual heat transfer within a building, only the use of computer software allows executing the millions of calculations required to complete a load estimate. Load estimating today can still be done manually using hand calculations or software spreadsheets, but these methods are usually reserved for early design rough approximations, or studies of special zones or material options effects. Running a final load for a large multizone building requires doing away with as many simplifying assumptions as possible, and using the HVAC system modeling techniques and refinement of heat transfer calculations of the transfer function and radiant time series methods.

Figure 7, plots relative accuracy vs. complexity for many common methods. The two extreme methods on the ends are instantaneous and heat balance. Instantaneous is the fastest and easiest to use, however, it is now only used for heating load calculations; it is too simple in the assumptions used to be used for cooling loads. The method does not take into account the storage and release of heat over time within the building material and furnishings. The heat balance method at the other end of the chart uses the least amount of simplifying assumptions, making it the most accurate method, but it must be done using computer software. All the load components (solar, wall, roofs, people, lights etc) interact dynamically with each other. This method calculates these interactions simultaneously and yields very precise results. Since the load components are calculated interactively, the method cannot provide output on how much of the load comes from walls, windows, roofs, people, lights etc. Since one of the main uses of load calculations is for designers to determine the individual load component contributions, this method is not used very much for actual load estimates. The heat balance method is very good for doing energy simulations, which is a topic for the Energy Estimating TDP.

1. Instantaneous, q = U * A * Δt
2. Storage Load Factor/ETD, Carrier E20
3. CLTD/CLF, Cooling Load Temperature Differential/ Cooling Load Factor, ASHRAE
4. Radiant Time Series, ASHRAE
5. Transfer Function, ASHRAE
6. Heat Balance, ASHRAE

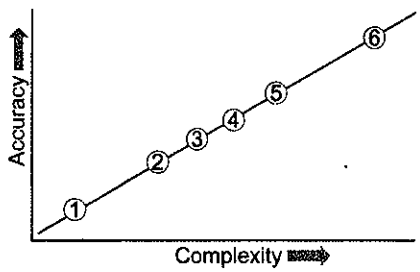


Figure 7

Accuracy vs. Complexity

Carrier's SDM-1 storage load factors/ETD manual load estimating method is still available, along with Block Load and Hourly Analysis Program computer software programs, both of which currently use the transfer function method. Plans are underway to convert these programs over to the current ASHRAE radiant time series method.

Storage Load Factor/ETD Method

The Carrier E20 method (storage load factor/ ***equivalent temperature difference*** method) was developed before ASHRAE became the industry source. In this method, a way for hand calculating the effects of storage and release of heat over time was put into tables to determine the appropriate storage factor. The method is covered in complete detail in Carrier's System Design Manual, Part 1, and two User Guides that cover the use of the E-10A (heating) and E-20A (cooling) forms for load estimating and coil performance.

TABLE 6—PEAK SOLAR HEAT GAIN THRU ORDINARY GLASS*
Btu/(hr)(sq ft)

NORTH LAT.	MONTH	EXPOSURE NORTH LATITUDE N†	NE	E	SE	S	SW	W	NW	Horiz	MONTH	SOUTH LAT.
0°	June	59	156	147	42	14	42	147	156	226	Dec	
	July & May	48	153	152	52	14	52	152	153	233	Nov & Jan	
	Aug & April	25	141	163	79	14	79	163	141	245	Oct & Feb	
	Sept & March	10	118	167	116	14	118	167	118	250	Sept & March	0°
	Oct & Feb	10	79	163	141	34	141	163	79	245	Aug & April	
	Nov & Jan	10	52	152	153	69	153	152	52	230	July & May	
	Dec	10	42	147	156	82	156	147	42	212	Oct & Feb	
10°	June	40	153	153				135	39	179	Sept & March	30°
	July & May	30			162	159	162	116	16	165	Aug & April	
	Aug & April	12	12	105	162	165	162	105	12	151	July & May	
											June	
40°	June	17	133	162	111	54	111	162	133	237	Dec	
	July & May	15	127	164	125	69	125	164	127	232	Nov & Jan	
	Aug & April	11	102	162	146	102	146	162	102	214	Oct & Feb	
	Sept & March	9	58	149	162	140	162	149	58	183	Sept & March	40°
	Oct & Feb	7	35	122	163	162	163	122	35	129	Aug & April	
	Nov & Jan	5	12	100	156	166	156	100	12	103	July & May	
	Dec	5	10	86	148	165	148	86	10	85	June	
50°	June	16	126	164	135	93	135	164	126	220	Dec	
	July & May	14	117	163	143	106	143	163	117	211	Nov & Jan	
	Aug & April	11	94	158	157	138	157	158	94	185	Oct & Feb	
	Sept & March	8	58	138	163	158	163	138	58	148	Sept & March	50°
	Oct & Feb	5	29	105	157	167	157	105	29	94	Aug & April	
	Nov & Jan	4	9	64	127	153	127	64	9	53	July & May	
	Dec	3	7	47	116	141	116	47	7	40	June	
		S	SE	E	NE	N	NW	W	SW	Horiz		
		EXPOSURE SOUTH LATITUDE										

Solar Gain Correction	Steel Sash or No Sash	Haze	Altitude	Dewpoint	Dewpoint	South Lat
	×1/.85 or 1.17	−15% (Max)	+07% per 1000 ft	Above 67 F −7% per 10 F	Below 67 F +7% per 10 F	Dec or Jan +7%

*Abstracted from Table 15, page 43.

†Solar heat gain on North exposure (in North latitudes) or on South exposure (in South latitudes) consists primarily of diffuse radiation which is essentially constant throughout the day. The solar heat gain values for this exposure are the average for the 12 hr period (6 a.m. to 6 p.m.). The storage factors in Tables 7 thru 11 assume that the solar heat gain on the North (or South) exposure is constant.

Figure 8

Peak Solar Gain through Glass (from Carrier SDM-1, Chapter 3)

When determining actual cooling loads, Table 6 from SDM-1 is used to determine the peak solar heat gain figures. This table represents the monthly maximums.

The variation of solar intensity and movement of heat into and out of the building mass is taken care of through the storage load factor tables. To demonstrate some of load estimating procedures we will begin creating an example building, and continue with its use throughout the remaining load estimating TDPs. The example building, the Chicago Office Building, has a 12-hour schedule of occupied operation. Figure 9 shows a portion of SDM-1 Table 11, which would be used to determine the appropriate storage factors. Other tables deal with different equipment operating hours.

TABLE 11—STORAGE LOAD FACTORS, SOLAR HEAT GAIN THRU GLASS
12 Hour Operation, Constant Space Temperature†

EXPOSURE (North Lat)	WEIGHT§ (lb per sq ft of floor area)	INTERNAL SHADE* SUN TIME AM 6	7	8	9	10	11	12	PM 1	2	3	4	5	BARE GLASS OR EXTERNAL SHADE‡ AM 6	7	8	9	10	11	12	PM 1	2	3	4	5	EXPOSURE (South Lat)
Northeast	150 & over	.59	.67	.62	.49	.33	.27	.25	.24	.22	.21	.20	.17	.34	.42	.47	.45	.42	.39	.36	.33	.30	.29	.26	.25	
	100	.59	.68	.64	.52	.35	.29	.24	.23	.20	.19	.17	.15	.35	.45	.50	.49	.45	.42	.34	.30	.27	.26	.23	.20	Southeast
	30	.62	.80	.75	.60	.37	.25	.19	.17	.15	.13	.12	.11	.40	.62	.69	.64	.48	.34	.27	.22	.18	.16	.14	.12	
East	150 & over	.51	.66	.71	.67	.57	.40	.29	.26	.25	.23	.21	.19	.36	.44	.50	.53	.53	.50	.44	.39	.36	.34	.30	.28	
	100	.52	.67	.73	.70	.58	.40	.29	.26	.24	.21	.19	.16	.34	.44	.54	.58	.57	.51	.44	.39	.34	.31	.28	.24	East
	30	.53	.74	.82	.81	.65	.43	.25	.19	.16	.14	.11	.09	.36	.56	.71	.76	.70	.54	.39	.28	.23	.18	.15	.12	
Southeast	150 & over	.20	.42	.59	.70	.74	.71	.61	.48	.33	.30	.26	.24	.34	.37	.43	.50	.54	.58	.57	.55	.50	.45	.41	.37	
	100	.18	.40	.57	.70	.75	.72	.63	.49	.34	.28	.25	.21	.29	.33	.41	.51	.58	.61	.61	.56					
	30	.09	.35	.61	.78	.86	.82	.69	.50	.30	.20	.17	.13	.14	.27			.64	.75	.79						
South	150 & over	.28	.25	.40	.53	.64																				
	100	.26	.22	.38	.51																					
	30																									

Figure 9

Solar Glazing Storage Load Factors (from Carrier SDM-1, Chapter 3)

As an example, lets look at a typical east-facing window. Chicago is approximately 40° N latitude, so looking at July, Table 6 (Figure 8) indicates a peak solar heat gain of 164 Btuh per ft^2. Going to Table 11 (Figure 9), and using the east exposure with a building weight of 100 pounds per ft^2 and internal shading, the peak storage load factor would be 0.73 at 8 a.m., reducing the solar heat gain to 164 * 0.73 = 120 Btuh/ft^2. By contrast, if there was no shading, the load factor would drop to a peak of 0.58 at 9 a.m., and the load profile is more leveled, demonstrating the heat storage effect.

Cooling Load Temperature Differential/Cooling Load Factor (CLTD/CLF) Methods

In the 1970s, ASHRAE published the CLTD/CLF Methods. These hand calculation methods were derived from the dynamic heat balance method using transfer functions on computer runs to derive location-specific tables that could be used in hand calculations.

As hand calculation procedures, both methods (CLTD and CLF) did a good job of balancing complexity (and therefore effort) with accuracy, however, both methods lacked flexibility. Building loads are affected by a wide variety of factors involving design, construction, environment, and building use. Table-based hand calculation methods typically deal with a fixed set of basic conditions (such as envelope loads for July 40 degrees North latitude) and then attempt to handle other conditions via correction factors. Ultimately, this approach introduces error and reduces accuracy when compared with methods that are more detailed. Some way of calculating loads specific to each design application was needed, which lead to the use of later methods that can best be done or only be done on a computer. Neither method is used currently since the transition from manual calculations to software-based load estimating has been successfully completed.

Transfer Function Method

The transfer function method is a simplification of the heat balance method that requires computer software to use. The transfer function method uses mathematical Laplace transformations to simplify the heat balance solution process, thereby yielding calculation times that are faster than those of the heat balance method without sacrificing too much of its accuracy.

The dynamics of calculating heat gain over time are retained, but individual load components can be calculated and then summed to get total loads. This allows the designer to gain element contribution values. For example, in Figure 10, the lights are turned on for 5 hours. The heat from the lights is split between convective (instantaneous) and radiant (heat storage) portions. The radiant portion is stored and released over time, just like solar load coming through a window. Therefore, as the lights remain on, the resultant load is

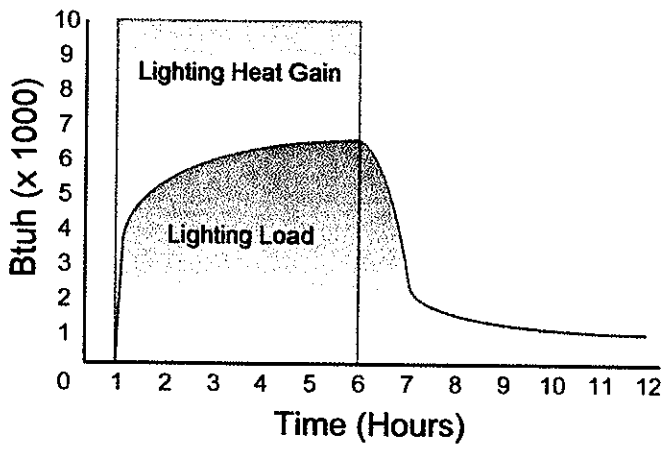


Figure 10

Lighting Load Transfer Function

a mixture of convective from instantaneous light heat and released radiant heat from storage. Note that after the lights are off, there is still the residual stored heat to be removed.

In the transfer function method, loads are calculated over a 24-hour period to allow for the heat storage dynamics. In that way, the transfer function method accounts for loads that occur during the unoccupied period which affect the loads during the occupied period.

The temperature of the room air is tracked for both occupied and unoccupied periods, and the effects of set point changes, and the magnitude of thermostat response to offset from set point are considered.

Looking at Figure 11, there are three items graphed: cooling loads, ***heat extraction rate***, and zone temperature. The first step of the transfer function procedure is to use the ***conduction*** and room transfer equations to calculate the cooling load, assuming a constant room air temperature over 24 hours. The next step uses the equipment sizing derived in the first step, along with the control system characteristics to simulate the way the room would be conditioned. From this, the true amount of heat that needs be extracted to maintain the room air temperature set point is determined. These calculations take into account the temperature during the unoccupied time. The graph shows that the system starts up early in the morning. The ***pulldown load*** on the air conditioning equipment can be readily seen. The system then operates during the course of the occupied period until it shuts off late in the evening. The zone air temperature achieved is also plotted. During the occupied time, the thermostat set point is traditionally about 75° F, and the unoccupied set point will often climb up to around 85° F.

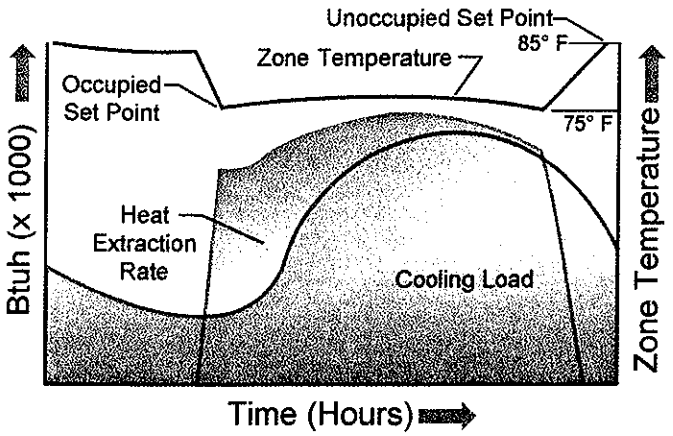


Figure 11

Transfer Function Elements

Because the transfer function method retains the dynamics of the heat balance method, it can be used for energy analysis as well.

Radiant Time Series Method

The radiant time series method, as you may expect by now, is a further simplification of the heat balance method, reducing the computation burden of the transfer function method. The term "radiant time series" refers to a set of factors that define the fraction of a heat gain converted to cooling load during a series of hours. The results from this method are similar to the first step of the transfer function method. It calculates a load based on a constant zone air temperature over 24-hours. It is not applicable for doing energy analysis calculations, but is easier to interpret pure load calculation results.

For walls and roofs, a solar air temperature is derived. This is a temperature that has the combined effects of solar intensity, external color, and actual dry bulb temperature. It represents the effective temperature on the outside of the wall. Then, each load element for each of 24-hours is split into convective (instantaneous) CTS (Conduction Time Series) and radiant RTS (Radiant Time Series) portions. A sample of the radiant and convective splits used by some components is listed in Figure 12. The items are then summed up for each hour to get a total for any hour of the day.

Table 19 Convective and Radiant Percentages of Total Sensible Heat Gain

Heat Gain Source	Radiant Heat %	Convective Heat %
Transmitted solar, no inside shade	100	0
Window solar, with inside shade	63	37
Absorbed (by fenestration) solar	63	37
Fluorescent lights, suspended, unvented	67	33
Fluorescent lights, recessed, vented to return air	59	41
Fluorescent lights, recessed, vented to return air and supply air	19	81
Incandescent lights	80	20
People		
Conduction, exterior walls	63	37
Conduction, exterior roofs	84	16
Infiltration and ventilation	0	100
Machinery and appliances (see Table 13)	20 to 80	80 to 20

Sources: Pederson et al. (1998), Hosni et al (1999).

Figure 12

Radiant/Convective Splits (from ASHRAE 2001 Fundamentals Handbook, Chapter 29)

Choosing Which Method to Use

No single load estimating method is best. Each method is appropriate for a specific application, depending on the point in the design process, time available to run the calculations, the sophistication of the user, and the desired level of accuracy in the results. The following recommendations are meant to guide the beginning designer as they become familiar with the various methods available.

- Preliminary design rough loads – rule of thumb check figures
- Schematic building loads – manual methods like storage load factor/ETD or simple ***block load*** software using transfer function or radiant time series
- Final loads for equipment selection – space, zone, and system loads using block load or system-based design software (HAP) using transfer function or radiant time series

With the release of the 2001 Fundamentals Handbook, ASHRAE began promoting the heat balance method for energy analysis and radiant time series method for pure load calculations.

External Space Loads

Earlier, when load components were first presented in Figure 1, external space loads were shown to include everything that directly caused the room temperature to vary, either above (heat gain), or below (heat loss) the set point. These loads can be further broken down into three subgroups:

- Weather-Related – made up of predominantly solar gain through glass, along with solar and transmission gains through walls and roof.
- Infiltration-Related – not directly weather-related, though outside wind speed affects the amount of airflow through the exterior envelope of the building. Infiltration (and exfiltration) occurs through any opening in the building envelope, whether it is planned like a window, door or HVAC unit sleeve, or unplanned gaps due to poor construction or building component deterioration.
- Adjacent-Space Related – made up of heat transfer across any interior partition with a temperature differential across the material assembly. Traditionally these have been thought of as walls, but horizontal assemblies that separate the floors in a building often have the required temperature differential to cause heat transfer.

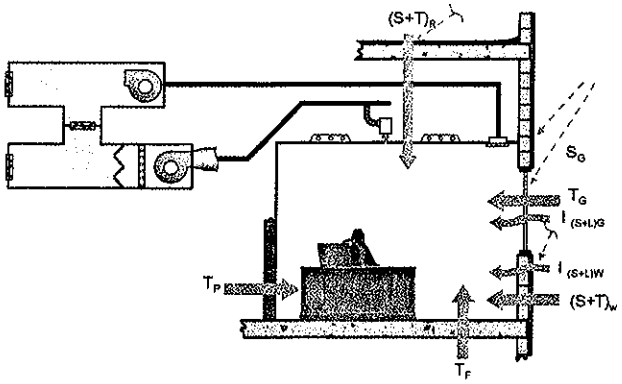


Figure 13

External Space Load Components

These heat gains and losses, which cause the space temperature to vary from set point, are the same no matter what kind of system is used, whether it is a window unit or a VAV terminal. Later in this TDP we will see that the system chosen will impact the load.

Design Conditions

As had been discussed, a ***temperature difference*** (Δt) is needed for heat to flow. So, some way to determine the peak or extreme outside air temperatures, along with the inside design temperatures, is needed to calculate the maximum heat flows through the building exterior envelope. The load estimating methods also need to calculate heat flow over time, so a means of determining a daily profile of hourly temperatures is needed.

Climatic Design Conditions

The designer is concerned about both cooling and heating; so two groups of climatic design data are needed. The most current ASHRAE Handbook of Fundamentals is the best source of outside weather data for design conditions across the globe. Tables are available for the United States, Canada, and other world locations.

The inside design temperatures are the comfort design conditions, generally 74 to 78 F design for cooling and 70 to 74 F design for heating.

Data is presented for dry-bulb dependent comfort cooling applications, wet-bulb dependent evaporation equipment (cooling towers, evaporative coolers, and fresh air ventilation systems) and dew point extreme conditions that often occur during months of moderate dry bulb temperatures.

Cooling

In the ASHRAE Climatic Design Data tables, there are three temperature variance choices, 0.4%, 1%, and 2%. These percent values indicate the number of hours in a year that the indicated temperatures may be exceeded. For instance, 0.4% db/mwb means that for 0.004 * 8760 hours/year = 35 hours per year the 0.4% cooling design dry bulb temperature for Chicago of 91° F may be exceeded. It is common for designers to use their best judgment when selecting which design conditions to use. However, beware of the creeping safety factor problem. Consult with your owner to see if being uncomfortable by only a couple degrees inside on those really hot days that do not occur every summer are worth the savings in both first cost and yearly cooling operating costs. Remember, going from the 0.4% column to the 1.0% column means only 53 additional hours per year, or only 7 to 10 occurrences a season, assuming approximately 6 hours of temperature overage on an above-design day. Therefore, a reduction of 3° F in the maximum outside temperature represents nearly a 20 percent drop in the Δt used to calculate the transmission heat transfer (88° F - 75° F, versus 91° F - 75° F).

TABLE 2–CORRECTIONS IN OUTDOOR DESIGN TEMPERATURES FOR TIME OF DAY
(For Cooling Load Estimates)

DAILY RANGE OF TEMPERATURE* (F)	DRY- OR WET-BULB	SUN TIME									
		AM			PM						
		8	10	12	2	3	4	6	8	10	12
10	Dry-Bulb	–9	–7	–5	–1	0	–1	–2	–5	–8	–9
	Wet-Bulb	–2	–2	–1	0	0	0	–1	–1	–2	–2
15	Dry-Bulb	–12	–9	–5	–1	0	–1	–2	–6	–10	–14
	Wet-Bulb	–3	–2	–1	0	0	0	–1	–1	–3	–4

TABLE 3–CORRECTIONS IN OUTDOOR DESIGN TEMPERATURES FOR TIME OF YEAR
(For Cooling Load Estimates)

YEARLY RANGE OF TEMPERATURE(F)*	DRY- OR WET-BULB	TIME OF YEAR								
		March	April	May	June	July	August	8	Oct.	Nov.
100	Dry-Bulb	–29	–19	–10	–3	0	0	–6	–16	–27
	Wet-Bulb	–14	–10	–5	–2	0	0	–3	–8	–14
95	Dry-Bulb	–29	–19	–10	–3	0	0	–6	–16	–27
	Wet-Bulb	–14	–10	–5	–2	0	0	–3	–8	–14

Figure 14

Outdoor Temperature Corrections (from Carrier SDM-1, Chapter 2)

Cooling is a time-dependant series of calculations. Many of the methods require a full day's time frame for the dynamics of the procedures. The practical reasons are that different thermal loads peak at different times. For example, the east peaks in the summer morning, while the west peaks in the summer afternoon (Figure 16 shows this graphically). The south side may peak in the fall because of the influence of the solar load created by the sun's lower position in the sky. However, when the system or coil load is determined, it may be in the summer mid-afternoon because of the effect of the high enthalpy (sensible and latent heat content) outdoor air. Figure 14 shows portions

of tables from SDM-1 that can be used to determine peak cooling temperature for various sunlit hours for a hot day in each month of the year. The tables are based on the ***daily range*** and the ***annual range*** (full tables are in the Appendix).

Figure 15 shows a graphical representation of a design day for Chicago in August.

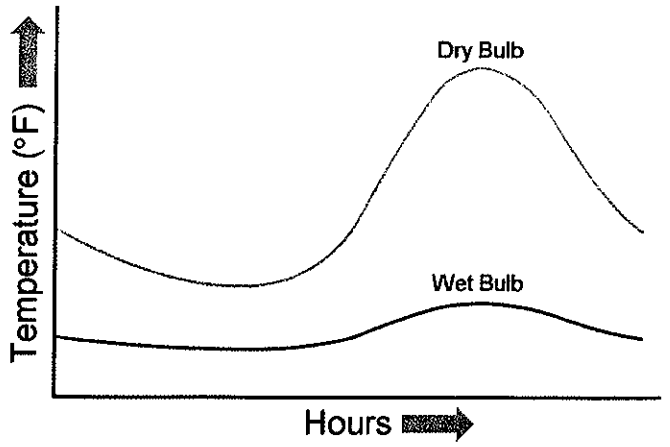


Figure 15

Design Temperature Profiles - August

Heating

Design heat loss is a simple calculation involving the design minimum outside air temperature. It is usually based on the ASHRAE 99.6 percent value or the judgment of the designer. For Chicago, the number is -6° F. When performing design heat loss calculations, there are no credits taken for any internal loads or solar heat gains. This is because the maximum heating losses usually occur in the morning hours, often before the building is occupied, meaning offsetting loads from lights, people, equipment, and even solar would not be present.

When selecting climatic design temperatures, the addition of a few degrees to the ASHRAE design data may be the first of many additive safety factors that creeps up on the designer.

Solar

To go along with the cooling design temperature profiles, a means of calculating peak effects from the sun is needed to figure out the solar loads through the glass. The solar load component represents the amount of radiant energy transmitted to the space through ordinary single pane glass. Hand calculation methods use tables of data, while computer software programs make this data ready to use for the site city. The solar data is in Btuh per ft^2 of glass area, comprised of both ***beam radiation*** (direct) and ***diffuse radiation*** (indirect, or scattered).

The compass points and number of exposures have an impact on the solar loads. The position of the sun not only varies with time of day, but significantly with time of year, which is determined by the latitude (angle of the sun) and longitude (sun time versus time of day) of the building site. For example, an east exposure has a morning peak in the summer. The south side has a peak around noon and it may be in the fall. The west glass peaks late afternoon in the summer, while the north glass receives only scattered diffuse sunrays in northern latitude locations like the United States. Occasionally, a cooling load study is done based on the building site orientation, especially if a large amount of glass is to be used.

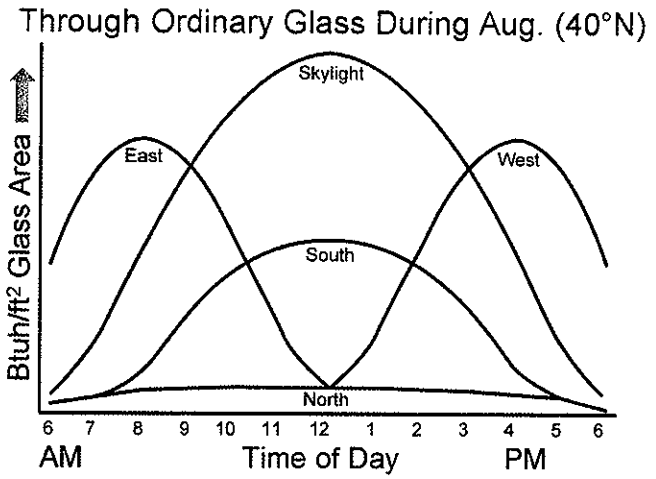


Figure 16

Solar Heat Gain

Since radiant energy is not a heat gain until it strikes some object and is absorbed, this does not represent the amount of heat to be removed by the air conditioning equipment. Glass types, special coatings, and internal and external shading all cut back on the amount of solar energy that comes into the room.

Coefficient Conversion

You can convert from one to another. SHGC = 0.87 * SC

The value that is eventually derived to modify the solar heat gain value is called either a shade coefficient (SC) or a ***solar heat gain coefficient*** (SHGC). The SHGC is the current rating method used for both the transfer function and radiant time series methods. These two coefficients are not the same. The ***shading coefficient*** should be used if the CLTD/CLF method is utilized.

Window and Glazing Types

The National Fenestration Rating Council (NFRC) is a good source for detailed information on windows and glazing types (simply called "glass"). Most manufacturers of windows and glass also offer technical information for computing solar heat gain when using their products.

Energy codes have all but done away with single pane 1/8-inch clear glass. Double-pane windows using two layers of glazing are most common, and on occasion in very cold climates triple-pane glazing assemblies are used. Glass comes in different thickness, 3/16 in., 1/4 in., and 1/8 in. Between the panes, there

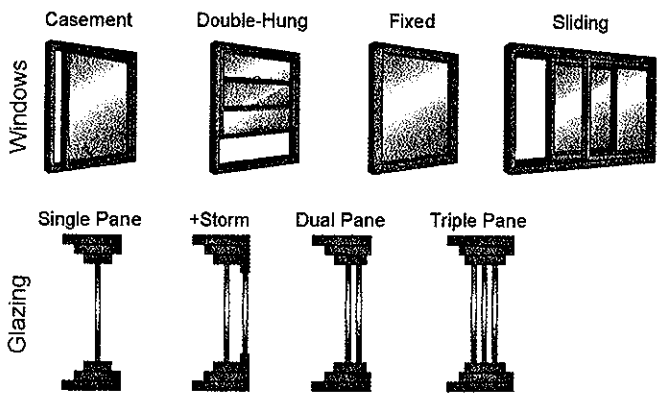


Figure 17

Window and Glazing Types

can be different gaps 1/4 in. and 1/2 in., filled with air, krypton, or argon. These refinements all reduce the SHGC and affect the overall U-factor.

The frame types, such as aluminum, wood, or vinyl, with or without thermal breaks, greatly affect the thermal performance and comfort for those seated near the windows.

Coatings

Further refinements in glazing performance come from inner and outer panes that have both tinting such as bronze, gray, or blue-green, and reflective coatings to reduce the transmission of solar energy with limited reduction of the visible light portion of the spectrum. Low-emission glass (low-e) is a clear glass that has a microscopically thin coating of metal oxide. This allows the sun's heat and light to pass trough the glass into the building. At the same time it blocks heat from leaving the room, reducing heat loss considerably. Buildings no longer need to excessively limit the glazing area to meet the energy conservation codes.

Shading - Internal, External, and Adjacent Buildings

One way to cut down on the amount of solar beam energy is to reduce the effective glass area and/or reflect some radiation back to the outside. Internal shade type describes the type of drapes, shades or Venetian blinds used with the window. The presence of interior shades, and the characteristics of the shading device, affects conductive and convective heat flows through the window assembly, as well as the transmission of solar radiation into the building. Shade properties therefore influence the calculation of the overall window U-factor, the shading coefficient, and the transmission and solar loads for the window.

Internal shades, depending on color, reflect a portion of the solar radiation back out the window, reducing the SC. Light colors reflect solar more than dark colors. For example, the windows in the Chicago example building have a U of 0.650 and an SC of 0.658, without internal shades. With an open weave light-colored fabric, the U is improved to 0.544 because of the additional insulation, and the SC is lowered to 0.496. With a medium colored fabric, the U is unchanged but the SC is larger at 0.547. Because this is radiant heat, the way the solar is accounted for is affected. If the shades are drawn, the radiant solar energy strikes the shades and is absorbed because the shade material is lightweight the heat is released to room almost immediately. If the shades are not drawn, the solar comes into the room that has much more mass and the solar heat is absorbed and released later and at a slower rate, lowering the value of the peak load and shifting it to later in the day. External shading is commonly done with fins, overhangs, and reveals that are described in relation to the window. They reduce the area through which the sun directly shines. Hand calculation methods do not provide

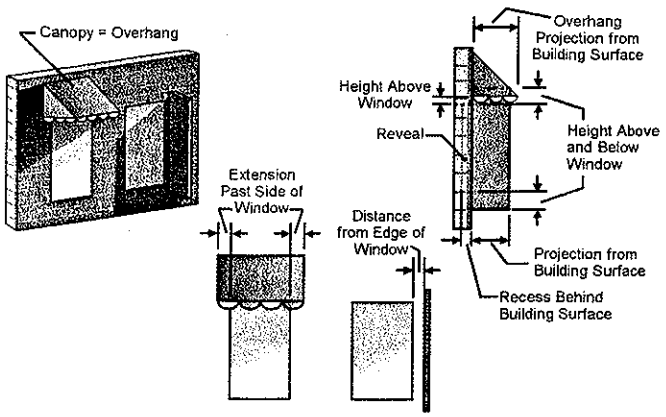


Figure 18

External Shading

accurate results for external shading. However, computer software can do the job with all the calculations needed to get hour-by-hour shading from exact solar angles, just as latitude and longitude are accounted for.

Shading (or even bounce light from reflective surfaces) from adjacent buildings is very difficult to calculate unless it shades surfaces all day. Then, just diffuse radiation can be assumed and a "north" exposure can be used to model this situation. Computer methods would need to reference the relationships of the adjacent building to walls and windows in the subject building in x-, y-, and z-coordinates. This would be very user-intensive and prohibitive from a practical standpoint. Therefore, designers will have to wait until these building interrelationships can be gained by CAD software communicating with load estimating software.

Transmission

If a temperature difference exists between the inside and outside surfaces, heat will flow by conduction to the cooler side. These are called transmission losses. There are varieties of surfaces that need to be accounted for in a cooling or heating load estimate to calculate all the conduction thermal loads. Remember, the equivalent temperature difference is used to account for the impact of solar loading on walls and roofs.

Wall and Door

Wall calculations can be performed manually using a U-factor for the wall materials assembly most closely matching the building design. There are tables of transmission coefficients (U-factor) in SDM-1 or older ASHRAE Fundamentals Handbooks that allows the user to select the most appropriate U-factor. The ASHRAE Fundamentals Handbook no longer has these tables since current methods like radiant time series use heat transfer coefficients layer-by-layer. The transfer function method uses the same method to calculate heat transfer as radiant time series.

However, quick comparisons of the overall U-factors will generally indicate relative resistance to heat flow of various assemblies. As discussed earlier, U-factors can be derived from tables of material R-values.

Door transmission gains or losses in commercial load estimates are sometimes ignored because they represent a very small fraction of the total load. If they are to be considered, a simple U-factor calculation is sufficient. This would not be the case if the doors were mostly glass, in which case they could be considered a separate window type.

Roof

The roof on a commercial building is usually horizontal and receives a lot of sunlight. Therefore, the effective temperatures are relatively high. If the roof is sloped, the area to be used in the transmission calculation is determined by measuring the area of the insulation layer at its location, either along the slope line or at the horizontal plane of the attic. The heat flow path starts with the ceiling material, then the air space above, all the way through the insulation and roof material assembly to the outside surface. Manual load calculations will require deriving the U-factor, or coefficients are internally computed for current software methods.

Windows and Skylights

Besides the solar load component, glass also has a transmission component. Similar to walls and roofs, the overall U-factor is used for this calculation.

On Grade and Below Grade

The temperature of the ground complicates calculation of the heat gain through the on grade floor or through basement walls below grade. The ground temperature near the surface is close to the outside air temperature, but further down the ground temperature approaches a rather constant value of from 45° F to 65° F, depending on the geographical area. There is very limited impact on the cooling load from the earth as most of the heat transfer is a heat loss. Heating loads, which will be discussed later, are much larger.

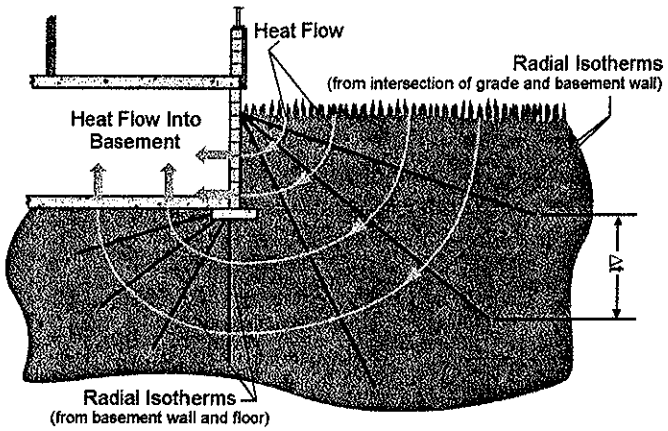


Figure 19

Transmission Below Grade

Infiltration

In addition to the other sources of heating or cooling loads, another tricky component is the effect of infiltration. These unplanned airflows must be heated or cooled to room conditions, often adversely affecting perimeter spaces. Experienced designers will say that estimating infiltration is like sticking a wet thumb in the air to estimate wind velocity.

Modern construction techniques result in tightly constructed buildings with minimal envelope infiltration. In low-rise buildings with fixed windows, and in those facilities operated under a slight positive pressure, infiltration can often be ignored. Most designers simply add a small safety factor to the load estimate to account for infiltration. The exceptions are high-rise buildings and those expected to be subjected to high wind loads.

Some old rules of thumb

that have been used for older buildings are 1 air change per hour in the summer and 2 air changes per hour in the winter.

Wind and Stack Effects

The pressure of the wind causes air to enter (infiltrate) the windward side of a building and leave (exfiltrate) the opposite (leeward) side. Stack effect, often called buoyancy, is caused by air density differences due to temperature and relative humidity differences between the indoor and outdoor air. Stack-effect induced airflow can be significant in tall buildings that are not sealed tightly at the floorline.

The relative importance of the wind and stack pressures in a building depends on building height, internal resistance to vertical airflow, location and flow resistance characteristics of exterior envelope openings, local terrain, and the immediate shielding of the building.

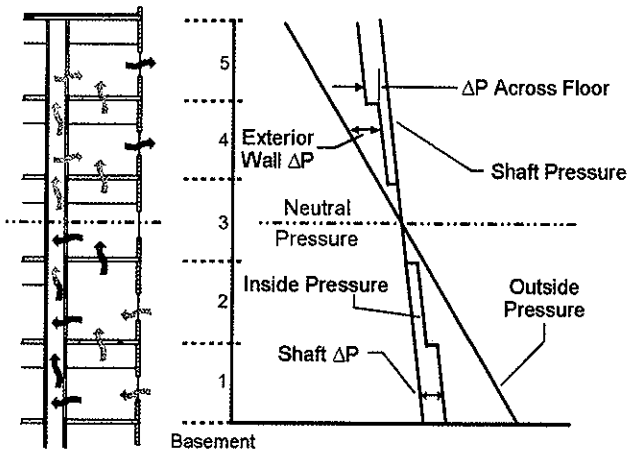


Figure 20

Stack Effect for an Idealized Building

Sensible and Latent Components

The infiltration air, when it is at a different temperature or moisture content, will impose additional space loads on the heating and cooling system.

Infiltration is treated as a space thermal load. The difference in air temperatures is a sensible load addition.

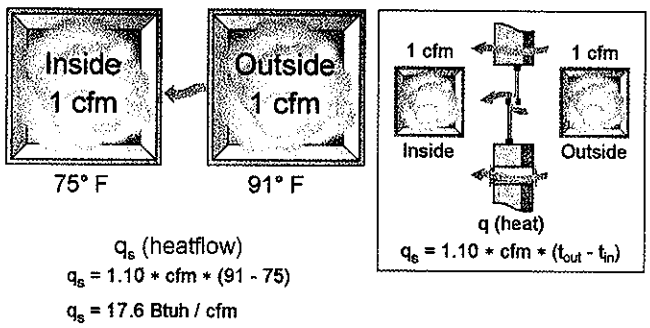


Figure 21

Infiltration, Sensible Component

The difference in moisture content is a latent load addition.

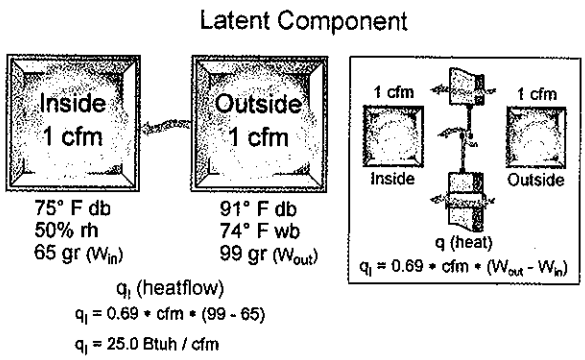


Figure 22

Infiltration, Latent Component

Partitions, Floors, and Ceilings

Some conditioned spaces in a building may be adjacent to unconditioned or partially conditioned spaces. When this occurs, heat is conducted by the temperature differences across the dividing material assemblies. It may be an interior wall (partition) next to a minimally heated warehouse. On the other hand, it may be a floor above an unconditioned parking garage or a ceiling that has a hot attic above it. All that is needed, of course, for a hand calculation is the U-factor, Δt, and area. Computer software may offer ways to change the temperature on the unconditioned side in some relationship with outside air to account for the dynamics over time.

Internal Space Loads

In non-residential buildings, the internal loads are a significant item in the load calculations. The designer must estimate the peak usage of lights, number of people, electric equipment, and plug load. The other item of judgment, besides how much, is when. The current dynamic methods used in computer software need the amount during the unoccupied hours as well to estimate the magnitude of pulldown loads. In some industrial applications items such as electric motors, exposed piping, tanks, and other process items need to be included in the calculation.

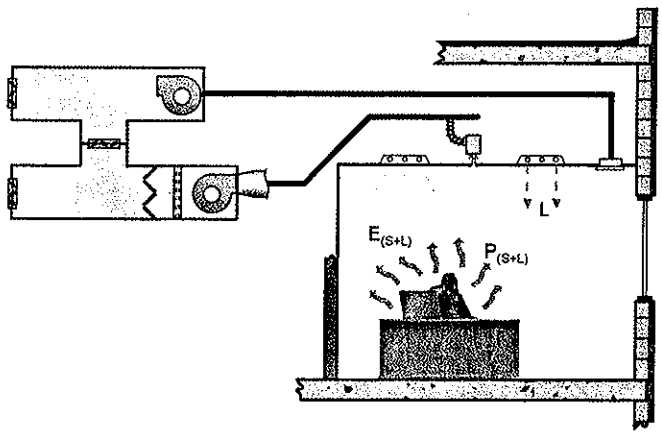


Figure 23

Internal Space Load Components

People

Heat is generated within the human body by oxidation – commonly called metabolic rate. This heat is carried to the surface of the body and dissipated by:

1. Conduction from the body to the objects the person touches and to the air surrounding them.

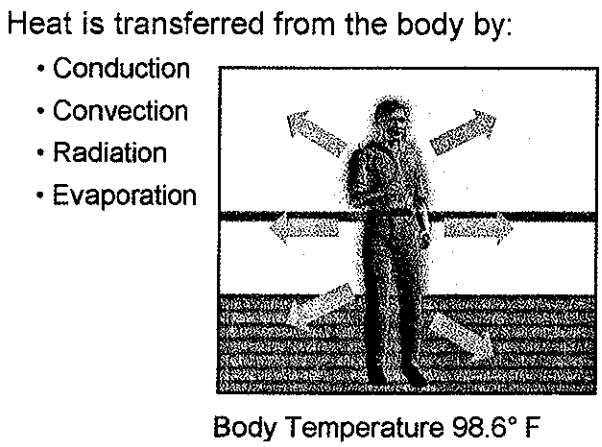


Figure 24

Heat Transfer from People

2. ***Convection*** from the body and respiratory tract into the air close to the person. Since warm air is lighter, or less dense, than cool air, the warm air floats upward. The warm air is replaced by cooler air, and the process continues.
3. Radiation from the body to the surrounding colder surfaces.
4. Evaporation of moisture from the body surface and in the respiratory tract to the surrounding air.

This chart shows representative values for latent and sensible heat gain from people. The values depend on room temperature and size (male, female or child). ASHRAE has tables and correction factors and computer software may have additional activity levels available for the designer.

A quick rule of thumb

Is 100 people represent 3 to 4 tons of total cooling load.

	Sensible Btuh	Latent Btuh	Total Btuh
Theatre	230	120	350
Office	245	205	450
Dancing	325	525	850
Bowling	525	925	1450

Figure 25

Heat Gain from People

Lighting

Lights generate sensible heat by the conversion of electrical power, or Watts (W), into light and heat. The energy is dissipated by radiation to the surrounding surfaces, by conduction into adjacent materials, and by convection to the surrounding air. Lights as a load component are typically referred to as lamp Watts per ft^2 of floor area. The radiant energy from lights is stored in the mass of the building and appears later as a load in the space. This is the same process that occurs with solar radiant energy, and current software methods factor this into the calculations.

Incandescent lamps convert approximately 10 percent of W into light. The remaining 90 percent of W turns into heat within the lamp. This heat then transfers into the space, 80 percent by radiation and 10 percent by conduction plus convection. This kind of light is used for task lighting, where its application reduces the need for higher overall lighting levels.

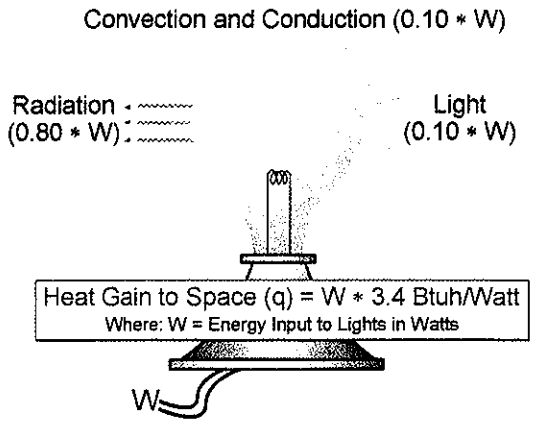


Figure 26

Lighting, Incandescent

Fluorescent light fixtures are more commonly used in commercial buildings for general lighting. Since about 23 percent of W is converted into light by fluorescent lamps, they are more efficient than incandescent lamps. Even though more efficient, a full 45 percent of W dissipates by conduction and convection to the space. Additionally, the final 10 percent of the input power turns into heat in the electronic ballast feeding the lamps.

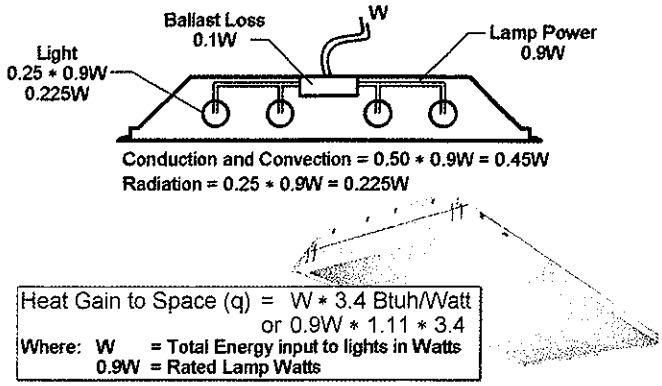


Figure 27

Lighting, Fluorescent

Electric Equipment (Plug Load)

Other devices such as computers, printers, copy machines, and appliances use electrical energy. Sometimes they are all lumped into a category of miscellaneous electrical equipment or plug load. Carrier SDM-1 or ASHRAE has tables of information for many kinds of appliances. For instance, a 12-cup coffee maker with two burners is 3750 Btuh sensible and 1910 Btuh latent.

Just taking the nameplate data and adding them up gives too large a plug load. In an article in the ASHRAE Journal (June 2000, *Heat Gain from Office Equipment*), it was reported that the average nameplate Watts for tested computers was 391, while the average heat dissipation was only 55 Watts. It was also found that heat gain was between 0.44 Watts per ft^2 to 1.05 Watts per ft^2, with an average of about 0.81 Watts per ft^2. The more unique the equipment or specialized the usage, i.e. medical laboratory, it is best to use published heat dissipation values for the actual equipment from the manufacturer (or installation and service manuals).

Electric Motors

Electric motors contribute sensible heat to a space by converting the electric power input (Watts or horsepower) to heat. Heat gain from electric motors depends on many factors, including motor horsepower, motor efficiency, motor hours used within a time period, how heavily the motor is loaded, and whether or not the motor and driven equipment is located inside or outside of the conditioned space or airstream.

Motor Nameplate or Rated Horsepower	Standard Efficiency	EPACT Min. Efficiency	Premium Efficiency	Heat Gain From Standard Eff. Motor in Location Btu/hr		
				A	B	C
1/20	35.0	-	-	364	127	236
1/12	35.0	-	-	606	212	394
1/8	35.0	-	-	909	318	591
1/6	35.0	-	-	1,212	424	788
1/4	54.0	-	-	1,178	636	542
1/3	56.0	-	-	1,515	848	667
1/2	60.0	-	-	2,121	1,273	848
3/4	72.0	-	-	2,651	1,909	742
1	75.0	82.5	-	3,393	2,545	848
1 1/2	77.0	84.0	-	4,958	3,818	1,140
2	79.0	84.0	-	6,443	5,090	1,353
3	81.0	86.5	89.5	9,426	7,635	1,791
5	82.0	87.5	89.5	15,518	12,725	2,793
7 1/2	84.0	88.5	91.7	22,723	19,088	3,636
10	85.0	89.5	91.7	29,941	25,450	4,491
15	86.0	91.0	93.0	44,390	38,175	6,215
20	87.0	91.0	93.0	58,506	50,900	7,606
25	88.0	91.7	93.6	72,301	63,625	8,676
30	89.0	92.4	93.6	85,787	76,350	9,437
40	89.0	93.0	94.5	114,382	101,800	12,582
50	89.0	93.0	94.5	142,978	127,250	15,728
60	89.0	93.6	95.0	171,573	152,700	18,873
75	90.0	94.1	95.4	212,083	190,875	21,208
100	90.0	94.1	95.4	282,778	254,500	28,278
125	90.0	94.5	95.4	353,472	318,125	35,347
150	91.0	95.0	95.8	419,505	381,750	37,755
200	91.0	95.0	96.2	559,341	509,000	50,341
250	91.0	95.4	96.2	699,176	636,250	62,926

NOTES: Ratings for open, drip-proof type, 4-pole, 1800 rpm motors
Totally-enclosed, fan-cooled (TEFC) motor ratings slightly higher
Consult motor manufacturer's ratings for specific data

Figure 28

Electric Motor Efficiencies

If the building is an industrial application, then the motors driving the process machinery also are a source of internal heat gain. There are sources beyond Figure 28 for data used in calculating motor heat loads, including manufacturers, ASHRAE 90.1 Energy Standard, and the Federal Energy Policy Act (EPACT). Minimum efficiency levels continue to climb because of the substantial use of electricity by motors.

Gas-Fired Equipment

Equipment within the space that is fired by natural gas is normally limited to cooking appliances and miscellaneous laboratory items like Bunsen burners. To limit the large amounts of both sensible and latent heat liberated by the natural gas combustion and the process itself, most of these devices are located under hoods directly exhausted to the outside. Even though, a certain percentage of the heat makes it into the space. Tables 51 and 52 in SDM-1 give values on an extensive list of gas-fired restaurant appliances and lab equipment. When high densities of equipment are present, like in medical laboratories and industrial pilot plants, data should be taken directly from the manufacturer's literature.

Piping, Tanks, and Evaporation

In most commercial applications, the level of insulation provided on both hot and cold piping and tanks keeps this ***internal load component*** at a negligible level. In industrial plants, such heat gains to the space are common, along with common items like furnaces and dryers. These contribute sensible heat to the space by convection and radiation from the outside surfaces, and frequently dryers contribute sensible and latent heat from the drying process.

Heat transmission tables (No. 54 through No. 58) can be found in SDM-1, and most manufacturers publish similar performance data to assist with either manual or software load estimating. Most estimating methods have an entry for miscellaneous loads where both sensible and latent load values may be entered. The user must sum up the sensible and latent gains for the piping lengths, equipment, or moisture transfer and then enter them into the manual form or computer software.

Bare and Insulated Piping

Both hot and cold piping exposed to the conditioned air in a room will have an effect on the load calculations. Hot water and steam piping heat transfer amounts depend on both the insulation type and thickness, and the temperature of the hot water or the pressure of the steam. Tables 54 and 55 in SDM-1 can be used in those cases where this may be a factor. Heat transfer coefficients for piping conveying cold fluids can be found in SDM-1, Table 56.

Closed and Open Tanks

Tanks that contain liquids at temperatures different than the room cooling or heating set points are a source of heat gain or loss. SDM-1 Table 57 may be used for sensible heat transfer from uninsulated closed tanks, and Table 58 gives similar data for free water surfaces for open tanks.

Steam Leaks, Absorption, and Evaporation

Steam leaking into the room has both a sensible and latent component. The sensible portion of the steam flow is calculated at

$$q_S = lb/hr * \Delta t * 0.45 Btu/lb$$

The temperature difference is between the room air dry bulb and steam temperature.

The latent portion is calculated at

$$q_L = lb/hr * 1050 Btu/lb$$

If moisture-absorbent materials (hygroscopic) are brought into the room, like large quantities of paper in a copy center, the moisture absorbed releases sensible heat (latent heat of vaporization at 1050 Btuh/lb of absorbed moisture) that needs to be included in the load calculation. If the hygroscopic material is removed from the space, an equivalent latent credit is included in the calculation.

Conversely, moisture evaporating at the room wet bulb (not heated or cooled from external sources), like a room full of plants or a reflecting pool, uses the room sensible heat for heat of evaporation. This form of latent gain should be deducted from the room sensible heat and added to the room latent heat. The total room heat gain does not change, but the effect on the sensible heat factor may be significant.

System Loads

The last group of load components to be considered when estimating a cooling load is those due to operating characteristics of the system serving the building or zone. These include loads associated with ventilation air, coil bypass, fan heat, duct heat transfer, duct airflow leakage, return plenum heat pickup, and in cases like underfloor air distribution, the supply plenum heat loss. Some of these items can only be accurately evaluated after the system is designed. Those computerized methods that perform system-based design loads are well suited to evaluate these items.

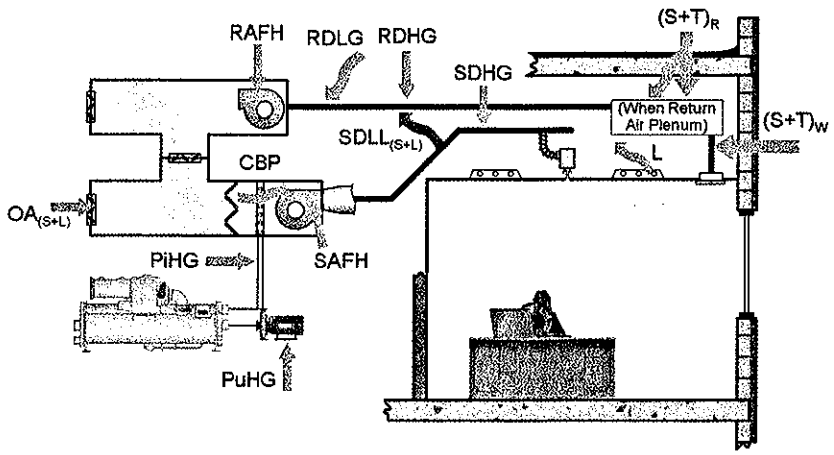


Figure 29

System Load Components

Ventilation Outdoor Air

Ventilation outdoor air normally creates a significant load for both heating and cooling estimates because of the large design Δt and/or moisture content of most locations. The first step is to determine the quantity of outdoor air needed. Air brought in from outside the building must be at least equal to the direct exhaust needs. Direct exhaust is that airflow which is directly exhausted from the building, not entering the main return air system. Toilet, laboratory, and kitchen hood exhausts are examples of direct exhaust.

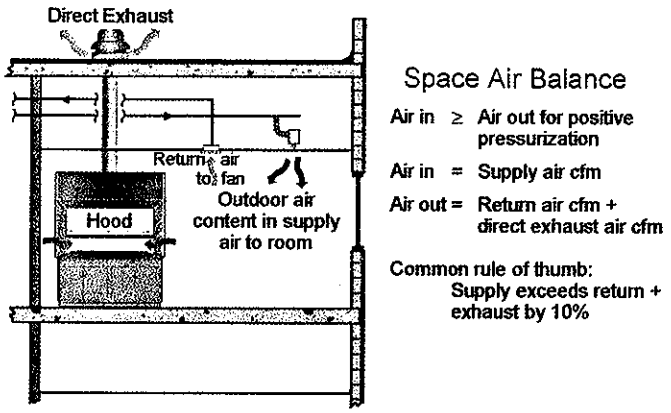


Figure 30

Ventilation, Space Air Balance

Codes and standards, notably the ASHRAE IAQ Standard 62, indicate the minimum amount of outdoor air to be used in different applications to provide an acceptable indoor air quality. Most of the time, these quantities are in excess of the direct exhaust requirements, especially in low-density occupancies like offices.

Maintain proper space pressure,

usually positive relative to the building exterior or adjacent circulation space, by supplying air in excess of the combined direct exhaust plus return air, regardless of the amount of outdoor air brought in at the main air handling unit.

Sensible and Latent Loads

Outdoor air brought in for ventilation has both sensible and latent load components. In simple calculations, the entering outdoor air is "assumed" the same as the outside conditions. Conditions like a hot rooftop alter the design climatic conditions and need to be accounted for in the load calculations. Since ventilation outdoor air is the same as infiltration air, the formulas shown in Figure 21 and Figure 22 can be used. In a few systems, like packaged terminal air conditioners (PTACs) and fan coils without dedicated makeup air systems, the ventilation air directly enters the space untreated by the cooling coil, so it is treated as a space load.

Duct Heat Transfer and Airflow Leakage

In transferring air from the system cooling coil to the conditioned spaces and back to the air-handling unit there are load components that must be considered. An accurate evaluation of these parameters can only be made when the air-conditioning system has been designed. Since load estimating is the first step in system design, a reasonable approximation of these parameters is necessary at this point.

Duct Heat Gain

Heat gains or losses in the supply and return ductwork will affect system performance and increase cooling and heating coil loads.

With supply air from 50° F to 60° F flowing through the ductwork, and upwards of 90° F air surrounding the duct in an unconditioned space, a potential for heat gain to both the supply and return air exists. Typically, this heat source has a negligible affect on comfort air-conditioning systems because of the use of higher R-value insulation on the ducts has kept heat transfer low, somewhere under 2 percent. A conservative approach, therefore, would be to use 2 percent of the room sensible heat, assuming that for a specific hour, the sum of zone sensible loads is 120,000 Btuh. Therefore, with the duct heat gain at 2 percent of 120,000 Btuh, the supply duct heat gain would be 2,400 Btuh. With a lower Δt in the return ductwork, only 1 percent heat gain is normal. The first of these heat gains will be used to determine the change in temperature of the supply air between the time it leaves the air handler and the time it reaches the zone supply terminals/room diffusers. The second heat gain will be used to figure the rise in the return air before it reaches the air handler.

Duct Leakage Loss

Air leakage in supply ductwork increases the airflow requirement for the central supply fan, and in turn, increases the fan heat gain and power use. Air leakage in the return duct adds to the airflow, but since it usually compensates for the supply duct losses, it does not add to the fan cfm.

The greatest potential for lost capacity exists in the supply air duct and depends on the duct shape, the duct pressure, and the workmanship used in constructing the duct system. Of the above three parameters, the one that is the key toward approximating the leakage loss is duct pressure.

Lower pressure class ductwork (±0 to 2-in. static pressure) tends to be installed using rectangular ductwork with taped joint connections. Due to variations in workmanship, leakage tends to be an average of 5 percent of the supply air quantity.

Mid-level pressure class ductwork (±2 to +6-in. static pressure) is made from medium quality spiral duct with tight joints. Loose joints are not permissible due to the whistling noise that would result from a leak. Leakage in this case is around 2 to 3 percent of the supply air.

Higher pressure class ductwork (+6.0-in. and higher static pressure) is made from high quality spiral duct with positive leak-tight joints. Leakage tends to be 1 percent or less of the supply air quantity.

A general conservative approach would be to add 5 percent of the room sensible heat to cover the leakage loss in manual calculations.

Computer software may model this as an increase in supply air cfm, and the quantity that leaks from the supply duct is assumed to flow directly or indirectly into the return airstream.

Fan Horsepower and Motor Heat Gains

Draw-thru air handling units have the supply fan located downstream of the cooling coil, adding heat to the supply airstream due to the work done on the air to raise its pressure within the ductwork. This heat gain diminishes the capacity of the supply air to cool the room, requiring a greater airflow for the space load. A blow-thru supply fan arrangement increases the coil load without impacting the supply airflow to the space. A similar heat gain situation occurs if a return air fan is used in the air-handling system, but in this case it only affects the coil load.

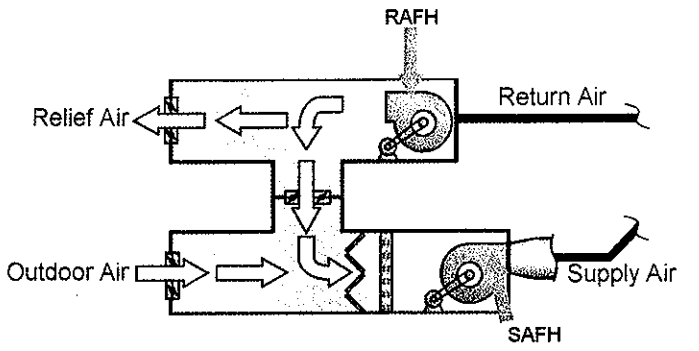
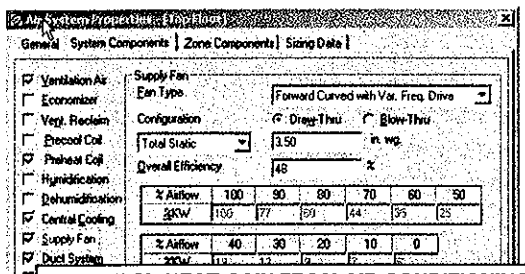


Figure 31

Fan Heat Gain Load Components

Should the motor driving the fan be located in the airstream, the electrical losses of the motor also show up as a heat gain. Figure 32 is a portion of Table 59 from SDM-1 (full table in Appendix), which allows an accurate estimate of this parameter. The table reveals that there are a number of considerations necessary to properly evaluate this parameter. A reasonable approach would be to add 5 percent of the room sensible heat to cover fan heat gain for draw-thru units. Blow-thru air-handling units have the fan located upstream of the cooling coil. Therefore, fan and motor heat gain would not be part of the system supply losses. Instead, they would be considered system return losses.



MANUAL
OR
SOFTWARE
METHOD

TABLE 59–HEAT GAIN FROM AIR CONDITIONING FAN HORSEPOWER, DRAW

FAN TOTAL PRESSURE† (In. of Water)	CENTRAL STATION SYSTEMS‡ Temp Diff Room to Supply Air					APPLIE Temp Room
	10 F	15 F	20 F	25 F	10F	10 F
	PERCENT OF ROOM SENSIBLE					
0.50	1.2	0.8	0.6	0.5	0.4	2.2
0.75	1.9	1.3	1.0	0.8	0.6	3.5
1.00	2.7	1.8	1.4	1.1	0.9	4.8

Figure 32

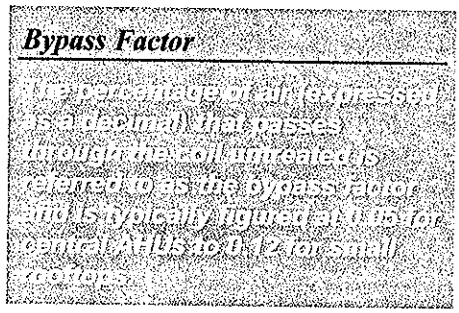
Fan Heat Gain

Computerized methods allow the designer to account for fan heat in a number of ways such as horsepower, kW, or static pressure. An estimate of total fan pressures is often the easiest way, as is shown in the figure.

Bypassed Outdoor Air

Since the cooling coil is not a perfect heat exchange device, some of the air entering the coil passes through the coil untreated. This represents a loss similar to a draw-thru fan heat gain that diminishes the supply air capacity to cool the room. Another way to look at it is a mixing of air-streams (conditioned and bypassed) that raises the coil leaving air temperatures above theoretically ideal conditions.

The coil bypass factor can be obtained from product literature for the specific equipment used in the load estimate. In most system design applications, however, the HVAC equipment has not yet been selected. In such a situation, specify a bypass factor that is generally representative of the type of equipment you intend to select (e.g., small, medium, or large rooftop units, central station air handlers of a particular type, etc.).



Plenums

Instead of running ductwork to convey return air from the spaces back to the air-handling unit, the building cavity between the suspended ceiling and floor or roof above can be used as a plenum. Now, with it becoming more common for building interior design to have a raised access flooring system, this underfloor plenum space, which is used for many building services, can also serve as a zone supply air plenum.

The distribution and accounting for some of the load components are affected as the plenums are at a different temperature than the room cooling and heating set points. The temperature of the return air in a plenum rises as heat is picked up, and likewise, the temperature of the supply air rises as cooling capacity is lost across the access floor and structural slab. Use of plenums influences the assignment of the heat gains, transferring portions of space loads over as coil loads.

Ceiling Plenums

When air is returned through the ceiling plenum, the cooling load caused by lights can be broken into two components. One component is that portion of the light heat, which flows directly into the space. The other component is the amount of heat absorbed as the return air passes over or through the fixture.

An accurate estimate of the two load components requires a detailed heat balance calculation and varies with the type of light fixture, type of ceiling, and the velocity of the return air over or through the fixture. However, for ordinary load estimating calculations, it is sufficient to calculate the total cooling load component for the light fixtures, and then distribute the load between the space and the return air as simple percentages of this total.

In a similar manner, a ceiling plenum under a roof can pick up even more heat from the roof load component. In this situation, the return air picks approximately 70 percent of the roof heat while 30 percent makes its way into the space. Again, 70 percent of the light heat reaches the space and the return air absorbs 30 percent of the light heat. The figure represent the composite effect of the various heat exchange processes that take place in this situation.

The return air heat transfer from the lights and roof are no longer space load components but are system load components directly affecting the cooling coil.

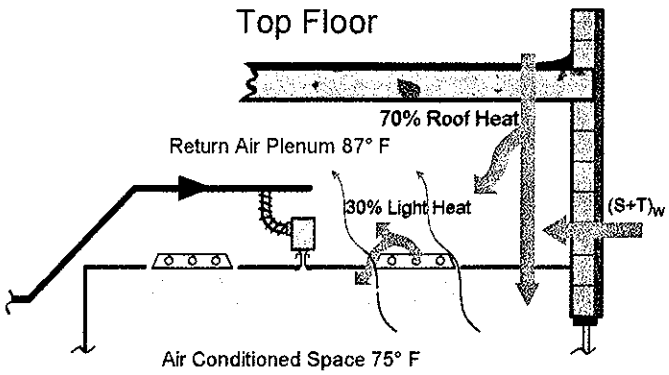


Figure 33

Ceiling Plenum, Top Floor

Underfloor Plenum

When an access flooring system is used as a supply plenum, the conditioned supply air is provided to the underfloor cavity and boosted by zone mixing fans before entering the conditioned area. The cool air in the plenum creates temperature differences across the plenum boundaries, both the perimeter walls and the floors (raised and structural). However, for the most part, the sensible load picked up comes from within the area to be conditioned, so nothing needs to be done to account properly for the heat transfer in the load calculation. The structural slab is usually not insulated, but the heat transferred to the floor below helps cool the material. A similar situation is occurring on the floor above, helping to cool the return air plenum below it, so at the main air handler the two load components cancel each other out.

Load Component Profile

Load component profiles are a visual representation of the dynamics of the major load components in a building or a zone over the course of a year. They are created by making some simplifying assumptions and plotting the results of peak cooling and heating load calculations on simple graph paper. The profiles plot the major thermal loads, such as transmission, internal loads and solar against outside dry bulb temperature. The loads are usually represented as Btuh ÷ 1000 or MBtuh.

The profiles let the designer visualize the relative size of each component and the relative change in the cooling and/or heating loads when changes are made to any individual component characteristic. This can be particularly helpful during schematic design when insulating values and areas of glass are being determined. Profiles are also an educational tool to begin the process of translating the fundamentals of load estimating to a system-based approach where the building overall loads can be related to the outside temperature and solar loading, much in the same way the HVAC system capacity responds to the changing outdoor air load conditions.

Peak Load Data for the Chicago Office Building:

Cooling (91° F/Jul 1500)

Transmission	58,755 Btuh
Lights	91,368 Btuh
People	14,557 Btuh
Electric	40,257 Btuh
Solar	69,444 Btuh

Heating (-6° F)

Transmission	252,768 Btuh

Figure 34

Example Building Load Data (Top Floor)

Building the Profile

The first step is to set up a grid of the outside dry bulb temperatures that cover the annual range for the subject building or zone on the horizontal axis. Sensible load values are plotted vertically in Btuh, both as positive heat gains (cooling), and negative loads or losses (heating).

The transmission peak loads are first plotted on the design heating and cooling temperature lines, using the data from the completed load estimate, in this case the top floor of the Chicago Office Building (Figure 34). Then a line is drawn between the design conditions to establish the slope of the load profile (Figure 35). The line crosses zero Btuh at the room set point. This uses the simplifying assumption of steady

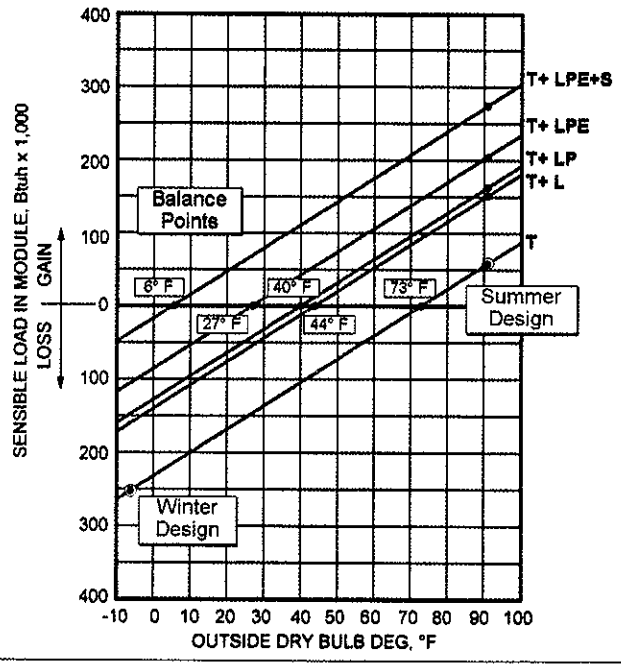


Figure 35

Completed Space Load Component Profiles

state heat flow. The slope of this line will change with different glass ratios, becoming steeper as the amount of glass increases. It can be seen that the net heating is less when it is cold outside and net cooling is greater when it warm outside.

Next, design cooling condition points should be established for lights, people, and miscellaneous electric (plug load). These internal loads are always cooling loads. Lines are drawn through each point parallel to the base profile line established by the transmission load. Without the addition of the transmission sloping line, these items would be straight horizontal lines as they are cooling loads at all temperatures. Adding the solar data completes the load profile (Figure 35). Here, a straight line based on the design cooling condition represents the solar data. In actuality the solar load varies significantly, both by time of day and time of year, requiring more complexity than a manual profile warrants.

Balance Points

Load profile balance points are those temperatures where the load lines cross over from cooling to heating. Looking at the completed profile, heat gain from lights will be offset by transmission losses when it is 44° F or lower outside. Adding people load causes the balance point to decrease to 40° F. When miscellaneous electrical loads are added, balance point drops even further to 27° F. As noted, the solar loads are highly variable, but it is easy to see how a building might require cooling even below freezing conditions outside.

The Chicago Office Building example shows the internal loads offsetting the transmission losses all the way down to 6° F. This extreme condition should be tempered with the infiltration load and the fact that not every light will be turned on or every person occupying the building. A more reasonable statement would be that our example building would most likely be in cooling down to the 30s, possibly even to below freezing. The eventual system selected for the building needs to accommodate this need for cooling during lower outside temperatures, either with an economizer cycle or through equipment selected to operate at the reduced temperatures.

Plotting on the Psychrometric Chart

A psychrometric plot of the system under design is necessary for a thorough understanding of the process and perhaps troubleshooting potential problems. If hand calculations are done, the process must be plotted manually on a psychrometric chart to assist in the design of the system. Some software load estimating programs can create these as part of the output.

State Points

When any two properties of an air-water mixture are known, the exact condition of the air can be located on the psychrometric chart, then all other properties can then be found from this one point. Such a point is referred to as a ***state point***. There are a number of state points that can be determined from running a cooling load estimate for an HVAC system. The simplest two are the room cooling set point and outside air design climatic condition. A third state point is the supply air temperature entering the space. This number is not needed for a basic manual load estimate, but is a common input for load estimating software. Efforts to model the HVAC system to refine the load require it as an input or it is a system data output when the designer uses other defining parameters such as airflows for either the air handler or the zone terminals/diffusers.

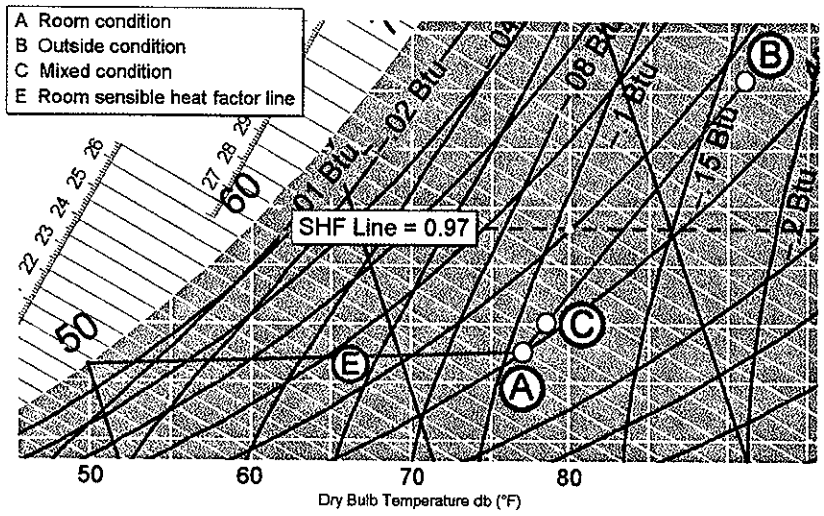


Figure 36

State Points and RSHF Line

Room Sensible Heat Factor (RSHF) Line

The room sensible heat factor is the ratio of the room sensible heat to the summation of room sensible and room latent heat. This ratio for a typical Chicago Office Building space is calculated as follows:

$$RSHF = \frac{RSH}{RSH + RLH} = \frac{6072\,Btuh}{6072\,Btuh + 160\,Btuh} = 0.97$$

The RSHF line is drawn through the room set point on Figure 36 as a line parallel to a sensible heat factor line of the same value (using the reference point on the chart). When adequate conditioned air is supplied to offset the room loads, the room set point will be satisfied, provided both the dry bulb and wet bulb temperatures of the supply air fall on the RSHF line. Colder air would need less airflow and warmer air more airflow.

System Lines

The psychrometric chart shown here has completed the circle by showing the effects of system loads and the performance of the cooling coil. The effect of fan heat (point 1 to 2), supply duct heat gain (point 2 to 3) are shown, along with point C, which is the effect of mixing outdoor air with the return air to arrive at the coil entering conditions. These advanced load-estimating concepts for system-based design will be covered in both the third and fourth TDPs in the Load Estimating series.

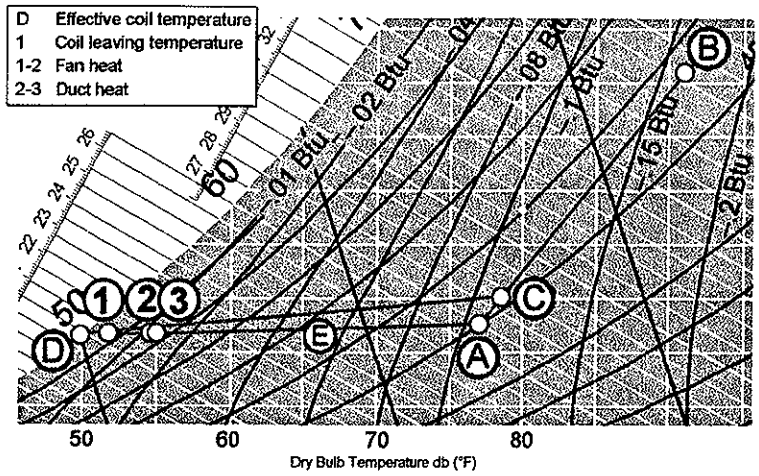


Figure 37

System Lines

Heating Load Estimate

Heating loads are a simplified version of cooling loads. Figure 38 shows that the envelope heating load losses are derived from steady state transmission and infiltration losses, while loads like ventilation outdoor air are most often system load components, which also apply to heating loads. The methods involved in calculating the loads have already been discussed in previous sections of this TDP. Determining the heat loss through basements and slab on grade floors becomes a little more complex than the simple $q = U * A * \Delta t$ transmission equation. The manual method, presented in detail in SDM-1, utilizes Form E-10A (found in Appendix). Most computer software load estimating methods automatically perform a heating load when calculating a cooling load.

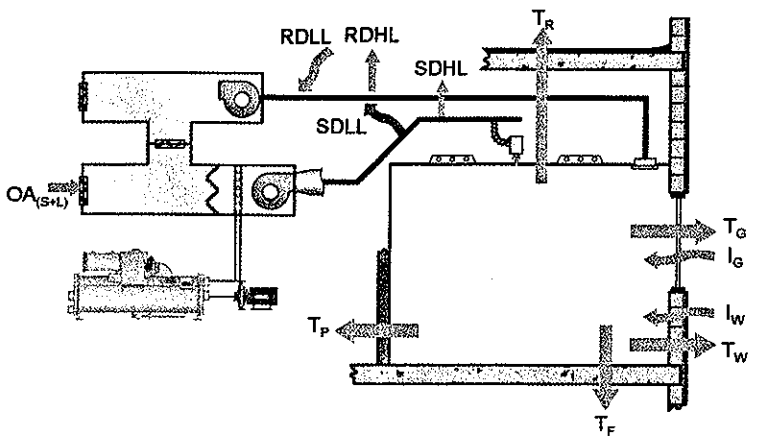


Figure 38

Heating Load Components

Design Conditions

To determine the temperature difference to be used across the exterior walls, the climatic design conditions need to be established first. From the ASHRAE Fundamentals Handbook, the heating 99.6 percent column value (reversal of 0.4 percent column used for cooling) for Chicago is -6° F. The room set point depends on the application. Comfort for an office building may be 70° F, while a warehouse may be only 45° F. Healthcare projects are applications that may require a wide range of set points in some areas, so you may need to design for 68° F cooling and 80° F heating.

Transmission

Transmission losses are normally the major heat transfer values in a space heating load estimate. The building walls, roof, glass, and foundation / basement comprise the load components. The temperature differences used are the differences between the room set point and the temperatures on the other side of the appropriate structural barrier (heating design climatic data, adjacent room temperature, or soil temperature). Heat loss through walls above grade is determined in the normal steady state transmission equation fashion.

Floor and Basement Heat Losses

For slab on grade and partially below-ground construction, the major heat loss takes place around the perimeter. The International Energy Conservation Code and ASHRAE 90.1 Energy Code require good levels of insulation along the full perimeter, extending down into the upper level of the soil, usually the first couple of feet, but no greater than the frost depth. The standard transmission equation is used to calculate the heat loss. While the floor loss is a minor percentage of the total heat loss, it is quite important for comfort, since cold floors mean an improperly heated building.

The heat loss from a full-depth basement is the result of a complex heat transfer process. The complexity is due to the variable ground temperature surrounding the basement. Below 8-ft, the ground is an excellent heat sink and can absorb large amounts of heat with little appreciable change in soil temperature. Above the 8-ft level, a temperature gradient exists due to the influence of the outside air temperature. The greatest variation exists at the surface and decreases down to the 8-ft level. In order to simplify the heat loss calculation, Figure 39 shows four possible calculations that could be involved.

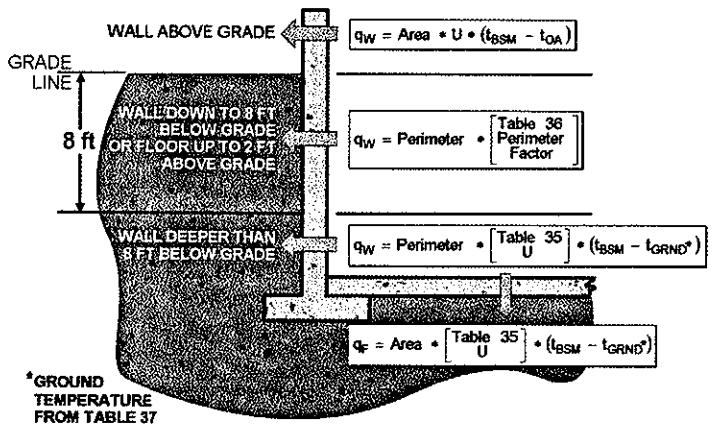


Figure 39

Basement and Floor Heat Losses

Basement Floor

The heat loss through a basement floor is normally small and relatively constant year round because the ground temperature under the floor varies little throughout the year.

SDM-1 Table 37 shows the ground temperature to vary only 25° F while the outside air temperature varies 50° F (from -30° F to +20° F). SDM-1 Table 35 also shows the transmission coefficient (U) for an uninsulated masonry floor of any thickness to be approximately $0.05\,\text{Btuh}\cdot\text{ft}^2\cdot°\text{F}$. Heat losses through the basement floor is therefore determined by the following equation.

$$q = U * A * (t_B - t_G)$$

Where:

U is $0.05\,\text{Btuh}\cdot\text{ft}^2\cdot°\text{F}$

A is the net floor area in ft^2

t_B is the basement temperature

t_G is the ground temperature

TABLE 37—GROUND TEMPERATURES

FOR ESTIMATING HEAT LOSS THROUGH BASEMENT LOSSES

Outdoor Design Temp (F)	–30	–20	–10	0	+10	+10
Ground Temp (F)	40	45	50	55	60	65

Figure 40

Ground Temperatures (from SDM-1, Chapter 5)

Walls 8-ft Below Grade

Like the basement floor, the heat loss through a masonry wall of any thickness may be approximated in the same manner as that done for a basement floor. The U-factor for masonry walls from Table 35 should be 0.08 instead of $0.05\,\text{Btuh}\cdot\text{ft}^2\cdot°\text{F}$.

TABLE 35—TRANSMISSION COEFFICIENT U–MASONRY FLOORS AND WALLS IN GROUND

(Use only in conjunction with *Table 35*)

Floor or Wall		Transmission Coefficient U (Btuh) (sq ft) (deg F)
* Basement Floor		0.05
Portion of Wall Exceeding 8 feet ground level	Below	0.08

*Some additional floor loss is included in perimeter factor, See *Table 36.*

Equations:

Heat loss through floor, Btuh = (area of floor, sq ft) * (U value) * (basement – ground temperature).

Heat loss through wall below 8 foot line, Btuh = (area of wall below 8 ft line, sq ft) * (U value) * (basement – ground temp).

NOTE: The factors in Tables 35 and 36 may be used for any thickness of uninsulated masonry wall or floor, but there must be a good contact (no air space which may connect to the outdoors) between the ground and the floor or wall. Where the ground is dry and sandy, or where there is cinder fill along wall or where the wall has a low heat transmission coefficient, the perimeter factor may be reduced slightly.

Figure 41

Transmission Coefficients (from SDM-1, Chapter 5)

Grade to 8-ft Below

In order to account for the variable temperature gradient, SDM-1 Table 36, provides a perimeter factor based on the distance of the floor (wall height) below grade. Heat loss is determined by the following equation:

$$q = L_p * Q * \Delta t$$

Where:

Q is the perimeter factor from Table 36

L_p is the lineal feet of basement wall (perimeter)

Δt is the difference between outside air and basement temperatures.

TABLE 36—PERIMETER FACTORS

FOR ESTIMATING HEAT LOSS THROUGH BASEMENT WALLS AND OUTSIDE STRIP OF BASEMENT FLOOR

(Use only in conjunction with *Table 35*)

Distance of Floor From Ground Level	Perimeter Factor (Q)
2 Feet above	0.90
At ground level	0.60
2 Feet below	0.75
4 Feet below	0.90
6 Feet below	1.05
8 Feet below	1.20

Equation:

Heat loss about perimeter, Btuh = (perimeter of wall, ft) * (perimeter factor) * (basement – outdoor temperature).

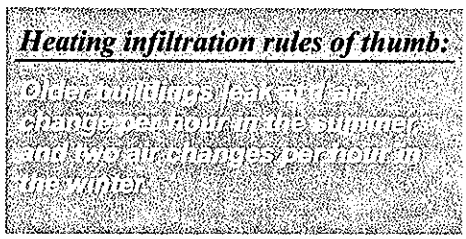
Figure 42

Perimeter Factors (from SDM-1, Chapter 5)

Infiltration

Infiltration air quantity in the winter can be more than in the summer because of the increased wind velocities, from an average of 7.5 mph in summer to 15 mph in winter. In addition, a much larger temperature difference exists between the outside and inside air temperatures.

Predicting infiltration rates before a building is constructed is not an easy task. Modern building construction techniques result in tightly constructed buildings with minimal infiltration. Infiltration can often be ignored for low-rise buildings with inoperable windows and those operated under a slight positive pressure. Most designers simply add a small safety factor to their heating load estimate to account for infiltration. The exceptions are high-rise buildings and those expected to be subjected to high wind loads. Unoccupied heating should be checked to see if including infiltration raises the total load because most buildings go to intermittent fan operation at night, allowing infiltration to occur.



No Credits

The heating load is estimated for the winter design temperatures, which usually occur at night. Consequently, no credit is taken for the heat given off by internal sources (people, lights, etc.). Some buildings, such as telephone exchanges, which have a continuous reliable source of heat, could use their heat source as a credit. Typically, however, a conservative approach to sizing heating equipment is taken, i.e., sizing for the worst-case condition. Due to the large difference between outside and inside air temperatures, space comfort could be adversely affected in worst-case conditions. Worse yet, frozen pipes, tanks, etc. could lead to very costly damage if the heating capacity was not available.

Warm-Up Safety Factor

If night setback temperature control is used to conserve energy, then some warm-up safety factor is needed as equipment reserve capacity to be available at the start of the occupied cycle to bring the space up to set point in a reasonable amount of time. A warm-up safety factor of 25 percent is often used in colder climates.

Summary

Load estimating is very important to HVAC system design because it not only calculates the many heat transfer load components that make up cooling and heating loads, it can model the dynamics of the building, along with the unique response of a specific HVAC system.

Heat transfer theory is relatively straightforward, utilizing simple formulas and computer software that models the complex dynamics of building heat storage and time-dependant heat balance equations.

Like many other aspects of HVAC design, simple check figures and rules of thumb allow the load estimating process to begin early in the design cycle when rough approximations are needed by the design team.

Load estimate output data is useful to the designer in many ways. Load profiles give the designer a visual representation of the load estimate relationship to outside air temperature and permit analysis of key load component impacts on building performance. Plotting state points and the RSHF line on the psychrometric chart begin the process of representing the interaction between the building load estimate and HVAC system response to the load.

Always make sure the building data is accurate, that the load estimate output is consistent with the building/HVAC system design, and do not forget to use the simplest method of estimating that fits the current need. Towards the middle of the design process, once most of the building component and system equipment details have been firmed up, use a software load estimating process that will be easy to update and keep current as the designs are finalized.

Work Session 1

1. Why is it important to understand the sources of the various building load components and how to combine them? _____

   _____

2. What three categories can all load components be classified into?

   _____

   _____

   _____

3. Explain what infiltration is. _____

   _____

4. Explain why it is difficult to accurately estimate this load component _____

   _____

5. Under what conditions would ventilation air be considered a room load? _____

   _____

6. Name the four supply air losses that exist between the coil of a central air handler and the room supply air terminal.

   _____ _____

   _____ _____

7. Would a blow through fan arrangement in a central air handler cause you to calculate the supply air fan heat (SAFH) as a loss between the coil and supply air terminal? (Yes/No)? Why? _____

   _____

8. Why must exhaust air be considered in a typical air conditioning system?

   _____

   _____

9. Define Grand Total Heat (GTH). _____

   _____

10. Define refrigeration load. _____

_____

11. What is the meaning behind the ASHRAE 0.4%, 1%, and 2% cooling db/mwb temperature columns? _____

_____

12. Of what use is the maximum design summer wet bulb temperature? _____

_____

13. In what project locations might one expect to use the maximum wb/mdb temperature columns? _____

14. Why is the peak of the horizontal glass curve so much greater than that of the south glass?

_____

_____

15. Why is the solar gain through the west glass nearly zero before noon? Why isn't it zero?

_____

_____

16. Explain why the actual cooling load caused by the sun's rays penetrating a window is smaller than and occurs at a peak time later than the instantaneous solar heat. _____

_____

_____

_____

17. Can the simple steady state equation ($q = U * A * \Delta t$) be used to determine the wall and roof transmission loads? _____

_____

18. Explain what Equivalent Temperature Difference (ETD) represents.

_____

_____

_____

19. Determine the sensible and latent load created by 100 people dancing (moderately) in a room whose temperature is maintained at 75° F. _____

_____

_____

20. Explain why stored energy from lights is generally neglected in calculating the light load component for a building. _____

_____

21. Explain why an accurate estimate of supply duct losses is not possible at the load-estimating phase of an air conditioning system design. _____

_____

_____

22. What is a typical percent of RSH used to estimate supply duct heat gain? _____

_____

23. Indicate where you would estimate the fan heat losses for the following:

	Supply Duct	Return Duct
A. Draw through fan	_____	_____
B. Blow through fan	_____	_____

24. Explain what bypassed outdoor air load is. _____

_____

25. When the ceiling plenum is located under a roof, _____ % of the roof load is absorbed by the return air, and _____ % is transmitted to the space. _____ % of the light heat is transmitted to the space, while _____ % is picked up by the return air.

26. Name the load components that make up a building heating load.

_____

_____

_____

27. What load components could be used as a heating load credit? Explain why. _____

_____

_____

_____

Appendix

Glossary

Cooling Check Figures, Commercial System Quick Reference

Approximate Design Air Distribution cfm (Cooling), Commercial Systems Quick Reference

Air Conditioning Load Estimate Form E-20A, Catalog No. 797-006

ASHRAE Standard 90.1 References for HVAC System Designers

Table 2 – Corrections in Outdoor Design Temperature for Time of Day, SDM-1

Table 3 – Corrections in Outdoor Design Conditions for Time of Year, SDM-1

Table 59 –Heat Gain Form Air Conditioning Fan Horsepower, Draw-Thru System, SDM-1

Heating Load Estimate Form E10A, Cat. No. 797-005

References

Work Session Answers

Load Components Terms List

	External Load Components
I_{LG}	infiltration, latent, glass
I_{SG}	infiltration, sensible, glass
I_{LW}	infiltration, latent, wall
I_{SW}	infiltration, sensible, wall
S_G	solar, glass
S_R	solar, roof
S_W	solar, wall
T_C	transmission, ceiling
T_F	transmission, floor
T_G	transmission, glass
T_P	transmission, partition
T_R	transmission, roof
T_W	transmission, wall
	Internal Load Components
E_L	equipment, latent
E_S	equipment, sensible
L	lighting
P_L	people, latent
P_S	people, sensible

	System Load Components
CBP	coil bypass
L	lighting
OA_L	outdoor air, latent
OA_S	outdoor air, sensible
PiHG	piping heat gain
PuHG	pump heat gain
RAFH	return air fan heat
RDHG	return duct heat gain
$RDLG_L$	return duct leakage gain, latent
$RDLG_S$	return duct leakage gain, sensible
SAFH	supply air fan heat
SDHG	supply duct heat gain
$SDLL_L$	supply duct leakage loss, latent
$SDLL_S$	supply duct leakage loss, sensible
S_R	solar, roof
S_W	solar, wall
T_R	transmission, roof
T_W	transmission, wall

Glossary

Word	**Definition**
airflow	movement of air, usually within boundaries such as ducts: commonly expressed as cfm.
applied equipment	field-assembled and commissioned equipment that has a range of selectable performance capacities that can be used either singly or in combinations to create an HVAC system or subsystem.
block load	maximum building or HVAC system heating or cooling load that occurs within the hours of a day, or days of a year; not necessarily as large as the sum of the individual zone loads within the block load area.
Btuh (Btu per hour)	the basic unit for measuring the rate of heat transfer within an HVAC system, British thermal units per hour
bypass factor	percentage of airflow that passes through a coil or filter untreated by the heat transfer surface or media.
cfm	cubic feet per minute. The unit of measure of the volume rate of air flow, as in a heating system.
check figure	a general sizing guideline used to evaluate preliminary equipment selections, capacities and fluid flows (i.e. cfm, gpm)
conduction	heat transfer by which heat is moved from molecule to molecule of a substance by a chain collision of those molecules
convection	heat transfer within a fluid by the movement of heated molecules from one place to another
energy simulation	software-based modeling of equipment or system performance for obtaining energy usage over time.
equivalent temperature difference (ETD)	alternative dry bulb temperature difference that creates the same transmission heat transfer as the actual dry bulb temperature difference, plus the transmission associated with the solar load component.
external load component	load component originating outside the space, zone, or building, but not associated with the HVAC equipment or distribution systems; normally external to the building, like solar and transmission loads.
heat	a form of energy that can be transferred by conduction, convection, or radiation; only transferred from a warmer substance to a colder substance.
heat extraction rate	rate at which heat is removed from the conditioned space.
indoor air quality (IAQ)	controlling the types and levels of indoor contaminants; involves material selections and moisture control within buildings, adequate ventilation with outdoor air, proper filtration, and the design, operation, and maintenance of comfort air conditioning systems for occupant benefit
internal load component	load component originating within the space, zone, or building, such as lighting, people and equipment.
life cycle cost	cost of equipment over its entire life, including operating and maintenance cost.
load calculation	mathematical calculation of the heat transfer associated with any of the various load components.
load component	heat transfer associated with a building or system element
load estimate	approximation of the heating and cooling requirements of a space, zone, system or plant; made up of individual load calculations for the associated load components.

packaged equipment	factory-assembled and tested equipment that has published performance capacities that can be used either singly or in combinations to create an HVAC system or subsystem.
pulldown load	cooling load at the initiation of a system on cycle; often made up of residual or stored load left over at the initiation of the off cycle.
R-value (thermal resistance)	quantity determined by the temperature difference, at steady state, between two defined surfaces of a material or construction that induces a unit heat flow rate through a unit area. $R = DT/q$.
room	an area within a building bounded on all sides with walls, a ceiling/roof and floor; rooms are considered spaces.
rule of thumb	a useful design guideline having wide application but not intended to be strictly accurate or reliable in every situation.
set point	point at which the desired value of the controlled variable is set.
safety factor	additional or reserve capacity or capability built into a system or component to account for unexpected situations or guard against failure or loss of set point.
shading coefficient (SC)	ratio of absorbed and transmitted solar heat relative to fenestration fitted with shading devices to that occurring with unshaded single strength glass.
solar temperature	alternative outside air temperature which, in the absence of all radiation exchanges, would give the same rate of heat entry into the surface as would exist with the actual combination of incident solar radiation, radiant energy exchange with the sky and other outdoor surroundings, and convective heat exchange with the outdoor air; used to determine equivalent temperature difference for sunlit building assemblies.
solar heat gain coefficient (SHGC)	fraction of incident irradiance that enters through the glazing and becomes heat gain; includes both the directly transmitted portion and the absorbed and re-emitted portion.
space	An area within a building not necessarily bounded on all sides by walls or a ceiling, such as a lobby or hall; rooms are considered spaces.
state point	A point or location on a psychrometric chart, pressure-enthalpy diagram, etc. that indicates the physical conditions of a mixture or process.
system load component	load component originating outside the conditioned space, zone, or building, and associated with the HVAC equipment or distribution systems, such as fan heat gain and coil bypass.
temperature difference (Δt)	difference between the temperatures of two substances, surfaces, or environments involving transfer of heat.
ton, ton of refrigeration	value used for cooling equipment capacity; equivalent to 12,000 Btuh.
transmission	transport of substances, energy, or indicated values from one place to another with or without impedances.
U-factor (thermal transmittance)	heat transmission in unit time through unit area of a material or construction and the boundary air films, induced by unit temperature difference between the environment on each side. *Note* - This heat transmission rate has been called the overall coefficient of heat transfer.
zone	a space, room or rooms, or building that is treated as a single comfort control area by an HVAC system.
zone load	peak cooling or heating load for the zone, occurring at a particular time of day and day of year; most likely, zones within an HVAC system peak at different times and dates.

Cooling Check Figures

Building Type	Cooling (ft^2/ton)	Lights/Equip. (Watts/ft^2)[1]	People (ft^2/person)	Ventilation[2]
Apartments				
Rental Suites	450	1.0	600	0.35 air changes per hour, but not
Hallways	550	2.0	-	less than 15 cfm/person
Auditoriums/Assembly	250 (20 seats/ton)	2.0	15	15 cfm/person
Bakeries[3]	225	6.0[1]	80	20 cfm/person
Banks	230	4.0[1]	30	15 cfm/person
Barber Shops	250	3.5[1]	45	15 cfm/person
Bars/Taverns/Pubs	100	1.0	20	30 cfm/person
Bathrooms (public)				
with Blow Driers[2]	200	8.0[1]	50	50 cfm/water closet
with Towel Dry	400	2.0	50	50 cfm/water closet
Beauty Shops	180	5.0[1]	45	25 cfm/person
Bowling Alleys	250 (1.5 tons/alley)	2.5	40	25 cfm/person
Clothing Shops (retail)	280	2.0	50	0.30 cfm/ft^2
Computer Rooms	80	6.0[1]	-	15 cfm/person
Department Stores				
Main Floors	300	2.0	35	0.30 cfm/ft^2
Upper Floors	380	2.0	50	0.20 cfm/ft^2
Doctor's Office				
Waiting Rooms	330	2.0	30	15 cfm/person
Examination Rooms	380	2.0	50	15 cfm/person
Drug Stores	250	3.0[1]	50	15 cfm/person
Educational				
Classrooms	280	2.0	20	15 cfm/person
Hallways	350	2.0	-	0.10 cfm/ft^2
Factories				
Light Manufacturing	240	3.0[1]	100	15 cfm/person
Medium Manufacturing	150	10.0[1]	200	15 cfm/person
Heavy Manufacturing	80	45.0[1]	300	15 cfm/person
Florists	250	3.0[1]	125	15 cfm/person
Gambling Casinos[4]	100	15.0[1]	25	30 cfm/person
Gift Shops	300	2.0	50	15 cfm/person
Grocery Stores	350	2.0	125	15 cfm/person
Hospital/Nursing				
Patient Rooms	300	1.5	100	25 cfm/person
Hallways	400	2.0	-	0.05 cfm/ft^2
Exam/Procedure Rooms	300	3.0[1]	50	15 cfm/person
Waiting Rooms	330	2.0[1]	30	15 cfm/person
Hotel, Motel, Dorm				
Guest Rooms	500	1.0	150	30/room
Hallways	560	2.0	-	0.05 cfm/ft^2
Lobby/Public	220	2.0	30	15 cfm/person
Offices				
One-Story	350	3.0[1]	150	20 cfm/person
Mid-Rise	400	3.0[1]	150	20 cfm/person
High-Rise	450	3.0[1]	150	20 cfm/person
Reception	350	2.0	50	15 cfm/person
Post Offices	280	2.0	30	15 cfm/person
Residences				
15% Glass[5]	700[6]	1.5[1]	400	0.35 air changes
25% Glass[5]	550[6]	1.5[1]	400	not<15 cfm/person
Restaurants				
Kitchen	180	12.0[1]	100	15 cfm/person
Fast Food Service, Dining	150	3.25[1]	30	20 cfm/person
Formal Dining	250	3.25[1]	30	20 cfm/person
Shoes Stores	300	2.0	50	15 cfm/person
Smoking Lounges	100	1.0	20	60 cfm/person

NOTES: There is no substitute for an accurate load analysis of a building, but these figures will help you to get a ballpark idea of the occupancy, lighting, ventilation and cooling requirements of a building in a timely manner. All figures assume the outdoor air ventilation rates shown.

[1] Where indicated, the lighting load includes other equipment such as driers, stoves, computers, etc.

[2] Ventilation rates are based on ASHRAE Standard 62-1999.

[3] Bakery figures assume that half of the floor space is dedicated to the kitchen and half to seating/serving/display area.

[4] The figures for the casino include some are allocated for a modest restaurant and for public bathrooms.

[5] In residences, the percent shown is the ratio of glass square footage to total outside wall square footage, expressed as a percent. A typical standard is 15%. A custom-built standard is 25%.

[6] Do not count unexposed, below-grade areas. They have no cooling load.

Approximate Design Air Distribution cfm (Cooling)

Building Type	Zone Primary Air cfm/ft² (by exposure)		
	East/South/West	North	Core
Apartments			
Rental suites	1.0	0.6	--
Hallways	--	--	0.7
Auditoriums and places of assembly	1.6		
Bakeries	1.8		
Banks	1.75		
Barber shops	1.6		
Bars/Taverns	4.0		
Bathrooms (public)[1]			
w/blow driers (2)	--	--	2.0
w/towel try	--	--	1.0
Beauty shops	2.2		
Bowling alleys	1.6		
Clothing shops (retail)	1.6	1.4	1.2
Computer rooms	7.5		
Department and discount stores			
Main floor	1.6	1.45	1.2
Upper floors	1.3	1.1	0.9
Doctor's offices			
Waiting rooms	1.2		
Examination rooms	1.0		
Drug stores	1.6		
Educational facilities			
Classrooms	1.6	1.3	1.2
Hallways	--	--	1.2
Factories			
Light manufacturing	1.7		
Medium manufacturing	2.7		
Heavy manufacturing	5.0		
Florists	1.6		
Gambling casinos	4.0		
Gift shops	1.3		
Grocery stores	1.3		1.1
Hospitals/nursing homes			
Patient rooms	1.4	1.2	--
Hallways	--	--	1.0
Exam/procedure rooms	--	--	1.3
Waiting rooms	1.5	1.3	1.0
Hotels/motels/dormitories			
Guest rooms	1.0	0.7	--
Hallways	--	--	0.7
Lobby/public areas	2.0	1.8	1.7
Jewelry stores	1.6		
Kitchens	2.2		
Libraries and museums	1.7	1.3	1.2
Offices			
One story	1.5	1.2	1.0
Mid-rise	1.3	1.0	0.8
High-rise	1.2	1.0	0.8
Reception	--	--	1.15
Post offices	1.4		
Residences[2]			
15% glass	0.6		
25% glass	0.75		
Shoe stores	1.3		
Smoking lounges	4.0		

Air quantities vary according to internal load assumptions, local code, designer preference, and air system selection criteria.

[1] Be sure to observe general building code for minimum exhaust. Bathroom figures shown above assume that the bathrooms are maintained at building design temperature.

[2] Percent shown is percent glass area of total outside wall area. 15% is typical; 25% is generous (as used for custom-built homes).

Air Conditioning Load Estimate

Carrier® Turn to the Experts.

Sheet ____ of ____ Date: _____
Prepared by: _____
Name of job: _____
Location: _____ Latitude: _____

Office: _____
System Name/Type: _____ Job No. _____
Approved: _____ Date: _____

Space used for: _____

Size: ____ x ____ = ____ ft² x ____ ft = ____ ft³

Item	Area or Quantity	Sun Gain or Temp. Diff.	Factor	Btuh Millions	Btuh 1,000's	Btuh 100's
SOLAR GAIN — GLASS						
Glass ( )	ft² x		x			
Glass ( )	ft² x		x			
Glass ( )	ft² x		x			
Glass ( )	ft² x		x			
Skylight	ft² x		x			
SOLAR & TRANSMISSION GAIN — WALLS & ROOF						
Wall ( )	ft² x		x			
Wall ( )	ft² x		x			
Wall ( )	ft² x		x			
Wall ( )	ft² x		x			
Roof–sun	ft² x		x			
Roof–shaded	ft² x		x			
TRANSMISSION GAIN — EXCEPT WALLS & ROOF						
All Glass	ft² x		x			
Partition	ft² x		x			
Ceiling	ft² x		x			
Floor	ft² x		x			
Infiltration	cfm x		x 1.10			
INTERNAL HEAT						
People			x			
Power	hp or kw x					
Lights	Watts x		3.4			
Appliances	Watts x		3.4			
Additional Heat Gains			x			
			subtotal			
Storage	ft² x		x (-)			
			subtotal			
Safety Factor			%			
Room Sensible Heat (RSH) ■						
Supply Duct Heat Gain	%+ Supply Duct Leak Loss		%+ Fan hp	%		
Outdoor Air	cfm x	° F x	BF x 1.10			
Effective Room Sensible Heat (ERSH) ■						
LATENT HEAT						
Infiltration	cfm x		gr/lb x 0.68			
People	x					
Steam	lb/hr x 1050					
Appliances, etc.						
Additional Heat Gains						
Vapor Trans.	ft² x 1/100 x		gr/lb x			
			subtotal			
Safety Factor			%			
Room Latent Heat (RLH)						
Supply Duct Leakage Loss			%			
Outdoor Air	cfm x	gr/lb x	BF x 0.68			
Effective Room Latent Heat (ERLH)						
Effective Room Total Heat (ERTH) ■						
OUTDOOR AIR HEAT						
Sensible	cfm x	° F x (1-	BF) x 1.10			
Latent	cfm x	gr/lb x (1-	BF) x 0.68			
Return Duct Heat Gain	%RSH + Return Duct Leakage Gain		% RSH			
Return Air or Blow-thru Fan			hp x 2545			
Grand Total Heat (GTH) or Dehumidifier Load) ■						
Pump	% GTH + Piping Heat Gain		% GTH			
REFRIGERATION LOAD ■						

Estimate for: Local Time Sun Time | Peak Load: Local Time Sun Time

Equipment Operation Hours/Day

Location	Conditions db	wb	%rh	Dp	gr/lb
Outdoor (OA)					
Room (RM)					
Difference		xxx	xxx	xxx	

OUTDOOR AIR

Ventilation: ____ people x ____ cfm/person = ____
____ ft² x ____ cfm/ ft² = ____
cfm ventilation ■ ____

Infiltration:
Swinging, revolving doors ____ people x ____ cfm/person = ____
Open doors ____ doors x ____ cfm/door = ____
Exhaust fan ____
Crack ____ feet x ____ cfm/ft = ____
cfm infiltration ■ ____
cfm outdoor air thru apparatus ■ ____ cfm_{OA}

APPARATUS DEWPOINT & DEHUMIDIFIED AIR QUANTITY

ESHF: Effective sensible heat factor = ERSH / ERTH = ____

ADP: Indicated ADP = ____ ° F Selected ADP = ____ ° F

Dehum. Rise: (1 - ____ BF) x (T_{RM} ____ ° F – T_{ADP} ____ ° F) = ____ ° F

Dehum. cfm: ERSH / (1.10 x ° $F_{Dehum.\ Rise}$) = ____ cfm_{DA}

SUPPLY AIR QUANTITY

Outlet Temp. Diff.: RSH / (1.10 x cfm_{DA}) = ____ ° F (RM – Outlet Air)*
*If this ΔT is too high, determine supply cfm for desired difference by supply air quantity formula.

Supply cfm: RSH / (1.10 x ° $F_{Desired\ Difference}$) = ____ cfm_{SA}

Bypass cfm: ____ cfm_{SA} - ____ cfm_{DA} = ____ cfm_{BA}

RESULTING ENT & LVG CONDITIONS AT APPARATUS

T_{RM} ____ ° F + (cfm_{OA} / cfm**) x (T_{OA}) ° F - T_{RM} ____ ° F = T_{EDB} ____ ° F

T_{ADP} ____ ° F + ____ BF x (T_{EDB}) - T_{ADP} ____ ° F = T_{LDB} ____ ° F

From Psych. Chart: T_{EWB} ____ ° F, T_{LWB} ____

**When bypassing a mixture of outdoor and return air, use supply cfm. When bypassing return air only, use dehumidified cfm.

NOTES

Form No. E-20A Cat. No. 797-006

ASHRAE Standard 90.1-2001
References for HVAC System Designers

This standard, *Energy Standard for Buildings Except Low-Rise Residential Buildings,* addresses many aspects relating to design of the system that should be considered during the selection process. Before listing those items, a quick review of what the standard is trying to address and accomplish is in order.

- Since it's beginning in 1975, this standard has had the purpose of providing minimum requirements for the energy-efficient design of buildings.
- The provisions of the standard apply to all building energy use related components and systems, with Chapter 6 addressing Heating, Ventilating and Air-Conditioning.
- Three compliance paths are available.

Item	Reference #	Notes
Simplified Approach Option for HVAC Systems (Option 1)	6.1.3	Works for smaller systems. Many pre-qualifiers
Mandatory Provisions (used with both of the next two items)	6.2	Deals with equipment efficiency, controls, construction, insulation, and balancing.
Prescriptive Path	6.3	Option 2 additive to Mandatory Provisions. Deals with economizers, simultaneous heating and cooling limitations, air system design and control, hydronic system design and control, heat rejection equipment, energy recovery, exhaust hoods, radiant heating system, and hot gas bypass limitations.
Energy Cost Budget Method	11.	Option 3 additive to Mandatory Provisions. Requires simulation of HVAC system design and *design energy cost* cannot exceed *energy cost budget.*
Mechanical Equipment Efficiency	6.2.1	Selected equipment must meet tabulated values.
Load Calculations	6.2.2	Heating and cooling load estimates must be run before equipment selection occurs.
Thermostatic Controls	6.2.3.1	Each zone shall be individually controlled.
Dead Band	6.2.3.1.2	$\geq$ 5° F required, unless special application.

Zone Isolation	6.2.3.2.5	HVAC systems serving zones operating non-simultaneously shall be divided into isolation areas.
Ventilation Control for High-Occupancy Areas	6.2.3.9	Controls shall automatically reduce outdoor air when spaces are partially occupied.
System Balancing	6.2.5.3	Combined with required VFDs, throttling losses in ductwork and piping are minimized.
Economizers	6.3.1	Required on most systems, per Table 6.3.1
Integrated Economize Controls	6.3.1.3	Economizers must be integrated with mechanical cooling system, with limited exceptions
Simultaneous Heating and Cooling Limitations	6.3.2	Prevents reheating, recooling or mixing of heated and cooled airstreams, with limited exceptions that permit VAV terminals.
Dehumidification	6.3.2.3	Does not allow humidistatic controls that violate 6.3.2, unless strict exceptions apply.
Humidification	6.3.2.4	If hydronic cooling is used and humidification set point exceeds 35° F, then a hydronic economizer is required.
Fan Power Limitation	6.3.3.1	Ratio of fan system power to supply fan airflow controlled by Table 6.3.3.1
VAV Fan Control	6.3.3.2	Guidelines for proper design (i.e. VFDs on motors $\geq$ 30hp)
Hydronic System Design and Control	6.3.4	Guidelines for proper design (i.e. VFDs on motors $\geq$ 50 hp and 100ft head)
Heat Rejection Equipment	6.3.5.2	Fan speed control on motors $\geq$ 7½ hp.
Exhaust Air Energy Recovery	6.3.6.1	Required on individual systems $\geq$ 5000 cfm and have minimum outdoor air supply $\geq$ 70% design airflow.
Exhaust Hoods	6.3.7	Special requirements for kitchen hoods and fume hood systems

TABLE 2—CORRECTIONS IN OUTDOOR DESIGN TEMPERATURES FOR TIME OF DAY

(For Cooling Load Estimates)

DAILY RANGE OF TEMPERATURE* (F)	DRY- OR WET-BULB	SUN TIME									
		AM			PM						
		8	10	12	2	3	4	6	8	10	12
10	Dry-Bulb	− 9	− 7	−5	−1	0	−1	−2	− 5	− 8	− 9
	Wet-Bulb	− 2	− 2	−1	0	0	0	−1	− 1	− 2	− 2
15	Dry-Bulb	−12	− 9	−5	−1	0	−1	−2	− 6	−10	−14
	Wet-Bulb	− 3	− 2	−1	0	0	0	−1	− 1	− 3	− 4
20	Dry-Bulb	−14	−10	−5	−1	0	−1	−3	− 7	−11	−16
	Wet-Bulb	− 4	− 3	−1	0	0	0	−1	− 2	− 3	− 4
25	Dry-Bulb	−16	−10	−5	−1	0	−1	−3	− 8	−13	−18
	Wet-Bulb	− 4	− 3	−1	0	0	0	−1	− 2	− 3	− 5
30	Dry-Bulb	−18	−12	−6	−1	0	−1	−4	−10	−15	−21
	Wet-Bulb	− 5	− 3	−1	0	0	0	−1	− 3	− 4	− 6
35	Dry-Bulb	−21	−14	−7	−1	0	−1	−6	−12	−18	−24
	Wet-Bulb	− 6	− 4	−2	0	0	0	−1	− 3	− 5	− 7
40	Dry-Bulb	−24	−16	−8	−1	0	−1	−7	−14	−21	−28
	Wet-Bulb	− 7	− 4	−2	0	0	0	−2	− 4	− 6	− 9
45	Dry-Bulb	−26	−17	−8	−2	0	−2	−8	−16	−24	−31
	Wet-Bulb	− 7	− 5	−2	0	0	−1	−2	− 4	− 8	−10

*The daily range of dry-bulb temperature is the difference between the highest and lowest dry-bulb temperature during a 24-hour period on a typical design day. (See *Table 1* for the value of daily range for a particular city).

Equation: Outdoor design temperature at any time = Outdoor design temperature from *Table 1* + Correction from above table.

TABLE 3—CORRECTIONS IN OUTDOOR DESIGN CONDITIONS FOR TIME OF YEAR

(For Cooling Load Estimates)

YEARLY RANGE OF TEMPERATURE(F)*	DRY- OR WET-BULB	TIME OF YEAR								
		March	April	May	June	July	August	Sept.	Oct.	Nov.
120	Dry-Bulb	−39	−22	−11	−4	0	0	−9	−24	−44
	Wet-Bulb	−23	−12	− 5	−2	0	0	−4	−13	−27
115	Dry-Bulb	−33	−22	−11	−4	0	0	−8	−20	−36
	Wet-Bulb	−18	−11	− 5	−2	0	0	−4	−10	−21
110	Dry-Bulb	−30	−20	−11	−4	0	0	−6	−17	−31
	Wet-Bulb	−15	−10	− 5	−2	0	0	−3	− 8	−16
105	Dry-Bulb	−30	−20	−11	−4	0	0	−6	−17	−29
	Wet-Bulb	−15	−10	− 5	−2	0	0	−3	− 8	−14
100	Dry-Bulb	−29	−19	−10	−3	0	0	−6	−16	−27
	Wet-Bulb	−14	−10	− 5	−2	0	0	−3	− 8	−14
95	Dry-Bulb	−29	−19	−10	−3	0	0	−6	−16	−27
	Wet-Bulb	−14	−10	− 5	−2	0	0	−3	− 8	−14
90	Dry-Bulb	−29	−19	−10	−3	0	0	−6	−16	−26
	Wet-Bulb	−14	−10	− 5	−2	0	0	−3	− 8	−14
85	Dry-Bulb	−29	−19	− 9	−3	0	0	−5	−16	−25
	Wet-Bulb	−14	−10	− 5	−2	0	0	−3	− 8	−14
80	Dry-Bulb	−24	−16	− 8	−3	0	0	−4	−12	−20
	Wet-Bulb	−13	− 9	− 4	−2	0	0	−2	− 6	−11
75	Dry-Bulb	−14	− 9	− 4	−1	0	0	−3	− 7	−15
	Wet-Bulb	− 7	− 5	− 2	0	0	0	−2	− 4	− 8
70	Dry-Bulb	−13	− 9	− 4	−1	0	0	−2	− 7	−14
	Wet-Bulb	− 6	− 4	− 2	0	0	0	−1	− 4	− 6
65	Dry-Bulb	−11	− 8	− 4	−1	0	0	−2	− 6	−12
	Wet-Bulb	− 6	− 4	− 2	0	0	0	−1	− 3	− 6
60	Dry-Bulb	− 9	− 7	− 3	−1	0	0	−2	− 5	−10
	Wet-Bulb	− 4	− 3	− 2	0	0	0	−1	− 3	− 5
55	Dry-Bulb	− 6	− 5	− 3	−1	0	0	−2	− 4	− 8
	Wet-Bulb	− 3	− 3	− 2	0	0	0	−1	− 2	− 4
50	Dry-Bulb	− 5	− 4	− 3	−1	0	0	−2	− 4	− 7
	Wet-Bulb	− 3	− 2	− 1	0	0	0	−1	− 2	− 3

*Yearly range of temperature is the difference between the summer and winter normal design dry-bulb temperatures (*Table 1*).

Equation: Outdoor design temperature = Outdoor design temperature from *Table 1* + Corrections from above table.

TABLE 59—HEAT GAIN FROM AIR CONDITIONING FAN HORSEPOWER, DRAW-THRU SYSTEM‡‡

	FAN TOTAL PRESSURE† (In. of Water)	CENTRAL STATION SYSTEMS‡ Temp Diff Room to Supply Air					APPLIED OR UNITARY SYSTEM** Temp Diff Room to Supply Air				
		10 F	15 F	20 F	25 F	30 F	10 F	15 F	20 F	25 F	30 F
		PERCENT OF ROOM SENSIBLE HEAT*									
Fan Motor Not in Conditioned Space or Air Stream	0.50	1.2	0.8	0.6	0.5	0.4	2.2	1.5	1.1	0.9	0.7
	0.75	1.9	1.3	1.0	0.8	0.6	3.5	2.4	1.8	1.4	1.2
	1.00	2.7	1.8	1.4	1.1	0.9	4.8	3.2	2.4	1.9	1.6
	1.25	3.9	2.6	1.9	1.6	1.3	6.5	4.3	3.2	2.6	2.2
	1.50	4.6	3.1	2.3	1.9	1.6	7.8	5.2	3.9	3.1	2.6
	1.75	5.4	3.6	2.7	2.2	1.8	9.1	6.1	4.6	3.6	3.0
	2.00	6.2	4.1	3.1	2.5	2.1	10.4	6.9	5.2	4.2	3.5
	3.00	10.4	6.9	5.2	4.2	3.5	16.7	11.2	8.4	6.7	5.6
	4.00	15.3	10.2	7.7	6.1	5.1					
	5.00	19.2	12.8	9.6	7.7	6.4					
	6.00	24.4	16.3	12.2	9.9	8.2					
	8.00	38.0	25.4	19.0	15.2	12.7					
Fan Motor†† in Conditioned Space or Air Stream	0.50	1.6	1.1	0.8	0.6	0.5	2.7	1.8	1.4	1.1	0.9
	0.75	2.6	1.8	1.3	1.1	0.9	4.2	2.8	2.1	1.7	1.4
	1.00	3.6	2.4	1.8	1.5	1.2	5.8	3.8	2.9	2.3	1.9
	1.25	5.0	3.4	2.5	2.0	1.7	7.6	5.1	3.8	3.1	2.6
	1.50	6.0	4.0	3.0	2.4	2.0	9.2	6.1	4.6	3.7	3.1
	1.75	7.0	4.7	3.5	2.8	2.4	10.7	7.2	5.4	4.3	3.6
	2.00	8.0	5.4	4.0	3.2	2.7	12.2	8.2	6.1	4.9	4.1
	3.00	13.2	8.8	6.6	5.3	4.4	19.5	13.1	9.8	7.8	6.5
	4.00	19.0	12.7	9.5	7.6	6.4					
	5.00	23.8	15.9	11.9	9.5	8.0					
	6.00	30.0	20.0	15.0	12.0	10.0					
	8.00	45.5	30.3	22.8	18.2	15.2					

*Excludes from heat gain, typical values for bearing losses, etc. which are dissipated in apparatus room.

†Fan Total Pressure equals fan static pressure plus velocity pressure at fan discharge. Below 1200 fpm the fan total pressure is approximately equal to the fan static. Above 1200 fpm the total pressure should be figured.

‡70% fan efficiency assumed.

**50% fan efficiency assumed.

††80% motor and drive efficiency assumed.

‡‡For draw-thru systems, this heat is an addition to the supply air heat gain and is added to the room sensible heat. For blow-thru systems this fan heat is added to the grand total heat; use the RSH times the percent listed and add to the GTH.

Heating Load Estimate

Carrier
Turn to the Experts.

Sheet ____ of ____ Date: _____
Prepared by: _____
Name of job: _____
Location: _____ Latitude: _____

Office: _____
System Name/Type: _____ Job No. _____
Approved: _____ Date: _____

HEATING CONDITIONS

Room _____db _____wb _____% rh _____gr/lb
Outside _____db _____wb _____% rh _____gr/lb
Diff. _____° F Diff. _____gr/lb

TEMPERATURE OF AIR ENTERING UNIT

____ % outdoor air x _____° F = _____° F
____ % recirculated air x _____° F* = _____° F
Total (average entering air temp.) = _____° F

*Room temperature plus temperature correction for heat of unit

TRANSMISSION

Surface	ft²	Trans. Coeff.	Btuh per °F Diff.	Total Btuh per °F Diff.	Temp. Diff.	Btuh

Transmission Loss Subtotal ■

INFILTRATION/VENTILATION

	CRACK METHOD	AREA METHOD	cfm
Window	_____ lin ft x _____	_____ ft² x 0.8	
Skylight	_____ lin ft x _____	_____ ft² x 0.5	
Door	_____ lin ft x _____	_____ ft² x _____	
Door Usage	_____ ft² x _____	_____ ft² x _____	
		SUBTOTAL	

subtotal infiltration _____ cfm x wind velocity/direction factor _____ = _____ cfm*
ventilation air _____ cfm x _____° F x 1.10
total infiltration* _____ cfm x _____° F x 1.10
Outdoor Air Subtotal ■
OVERALL HEAT LOSS SUBTOTAL ■
Warm-up Safety Factor @ _____%
GRAND TOTAL HEAT LOSS ■

AIR CHANGE METHOD

_____ ft³ / 60 x { _____ Max / _____ Min changes/hr

	cfm
Max	
Min	

HUMDIFICATION LOAD (Water Evaporated)

(_____ cfm infil + _____ cfm_{OA}) x (_____ gr/lb Diff.) / 1580 = _____ lb/hr

EQUIPMENT SELECTION

Quantity	Unit Size	rpm	cfm	Fin. Temp °F	Basic Rating	Btu Constant	Installed Capacity	Steam Condensate lb/hr	Hot Water gpm	ft² gas/hr

Form No. E-10A Cat. No. 797-005

References

Carrier Corp.

Commercial Systems Quick Reference (CSQR), Catalog No. 795-205.

System Design Manual – Load Estimating, SDM-1, Catalog No. 510-304.

User Guide – Load Estimating Using Storage Load Factors and Equivalent Temperatures, Catalog No. 791-090.

ASHRAE

ASHRAE Handbook – 2001 Fundamentals:

Chapter 27, Climatic Design Information

Chapter 29, Nonresidential Cooling and Heating Load Calculation Procedures

ASHRAE Standard 62 – Ventilation for Acceptable Indoor Air Quality.

ASHRAE Standard 90.1 – Energy Efficient Design of New Buildings (Except Low Rise Residential Buildings).

Work Session Answers

1. To perform a load estimate for the selection of the heating and cooling equipment that will condition the various building spaces.
2. External, internal, and system
3. The passage of outdoor air into the conditioned space due to positive outside wind pressure.
4. Due to the tremendous variability of building geometry, building construction quality and wind patterns.
5. When introduced untreated directly into the space and/or when bypassed through or around an air handling unit coil.
6. Supply Duct Heat Gain, Supply Duct Leakage Loss, Supply Air Fan Heat, and Bypassed Outdoor Air.
7. No. The load would be calculated on the return side of the coil since this is physically where the fan would be located.
8. Since ventilation air is continually being drawn into the system, a similar quantity of return air must be exhausted (or relieved) in order to maintain balanced airflow through the system. With no exhaust, air pressure would build in the system, preventing the influx of any new ventilation air.
9. The total load seen by the coil consists of room loads, supply duct losses and outdoor air loads.
10. In a chilled water system, GTH plus pumping and piping losses between the coil and the chiller. In a DX system refrigeration load equals GTH.

11. They mean that the dry bulb and mean coincident wet bulb design temperature values shown are exceeded on average 0.4% (35 hours), 1% (88 hours), or 2% (175 hours) of the 8760 hours in a year.

12. For selecting wet bulb performance related equipment such as cooling towers.

13. In warm maritime areas that are close to the ocean.

14. In August the sun is relatively high in the sky. Thus, a great deal of solar energy striking the south glass is reflected off the glass. The horizontal glass on the other hand has the sun's rays striking it more directly with little reflection taking place. Most of the energy penetrates the glass.

15. The west is in shade up to noon. Even though in the shade the glass is still subject to diffuse radiation.

16. The sun's rays penetrating the glass are in the form of radiant energy must first strike a surface to be converted into heat. Not all of this heat shows up immediately in the space because some of it flows into the mass of the building by conduction. Over time, this stored heat makes its way into the space. Thus, the peak load caused by the sun's rays occurs later in the day and has a lower magnitude then the instantaneous heat gain.

17. Even though the storage effect and solar loading of the external wall or roof structure makes the flow of heat time-related and thus not a steady-state heat flow situation, the use of ETD allows the load allocation to be completed using the steady state equation.

18. ETD is a fictitious temperature difference that allows one to stop the flow of heat (like a frame in a camera) at one point in time and use the steady state heat transfer equation to calculate the load. It is the equivalent of the complex storage related heat transfer process that is actually taking place through the wall or roof.

19. 100 people * 325 Btuh/person = 32,500 Btuh sensible
    100 people * 525 Btuh/person = 52,500 Btuh latent

20. Carpeting is widely used. The primary target for radiant energy from the lights is the floor. Carpet insulates the floor from radiant light rays. The radiant energy strikes the carpet and is converted into heat. The heat is then dissipated almost immediately into the space.

    Unless the air conditioning equipment is run for an hour or two before lights are turned on, the mass of the building will already be saturated with stored heat from the night before. As the light heat is absorbed in the mass, the equivalent amount of energy is almost simultaneously released to the space.

21. At the load estimating phase of the project, the designer has not designed the ductwork or piping systems and in many instances has not as yet decided on the type of system to be used. Thus estimates must be made.

22. 2%-3% for mid-level pressure class ductwork (±2-in. to +6-in. static pressure). Lower pressure class ductwork would likely have 5% leakage.

23. A. Supply Duct

    B. Return Duct

24. This represents a loss that air leaving the coil must overcome before entering the room. Some air passing through the cooling coil untreated due to the coil bypass factor phenomenon causes it.

25. 70%, 30%, 70%, 30%

26. Transmission, infiltration, and ventilation

27. Only load sources that are reliable and can be counted on as being there. Solar and lights are typically not used for this reason. Generally, the peak heating load occurs in the early morning hours when the lights are not normally on and the sun has not yet risen.

Prerequisites:

An understanding of air conditioning equipment and systems along with psychrometric properties of air-water mixtures. This knowledge can be gained from TDP-103, Concepts of Air Conditioning, and TDP-201, Psychrometrics, Level 1: Introduction.

Learning Objectives:

In this module, participants will learn the skills and knowledge necessary to:

- Understand the importance of load estimating and its role in HVAC design
- Understand the heat transfer theory behind conduction, convection, radiation and thermal storage
- Choose the appropriate load estimating method for each project, from manual check figures to computerized heat balances
- Categorize load components as to external, internal, and system groups, and calculate each
- Evaluate building loads using a load profile
- Plot state points and the room sensible heat factor line on the psychrometric chart

Supplemental Material:

Carrier Commercial Systems Quick Reference, ENG-05A, Catalog No. 795-205

Carrier Block Load, version 3.05. HVAC load estimating program for commercial buildings

Carrier Hourly Analysis Program (HAP), version 4.20a. HVAC system-based design and energy analysis program for commercial buildings

Air Conditioning Load Estimate Form No. E-20A, Catalog No. 797-006

Heating Load Estimate Form No. E-10A, Catalog No. 797-005

System Design Manual, Part 1, Load Estimating, Catalog No. 510-304

Load Estimating, Level 3, Application, Form No. TDP-302, Catalog No. 796-035

Load Estimating, Level 4, Theory, Form No. TDP-304, Catalog No. 796-036 (in development)

Instructor Information

Each TDP topic is supported with a number of different items to meet the specific needs of the user. Instructor materials consist of a CD-ROM disk that includes a PowerPoint™ presentation with convenient links to all required support materials required for the topic. This always includes: slides, presenter notes, text file including work sessions and work session solutions, quiz and quiz answers. Depending upon the topic, the instructor CD may also include sound, video, spreadsheets, forms, or other material required to present a complete class. Self-study or student material consists of a text including work sessions and work session answers, and may also include forms, worksheets, calculators, etc.



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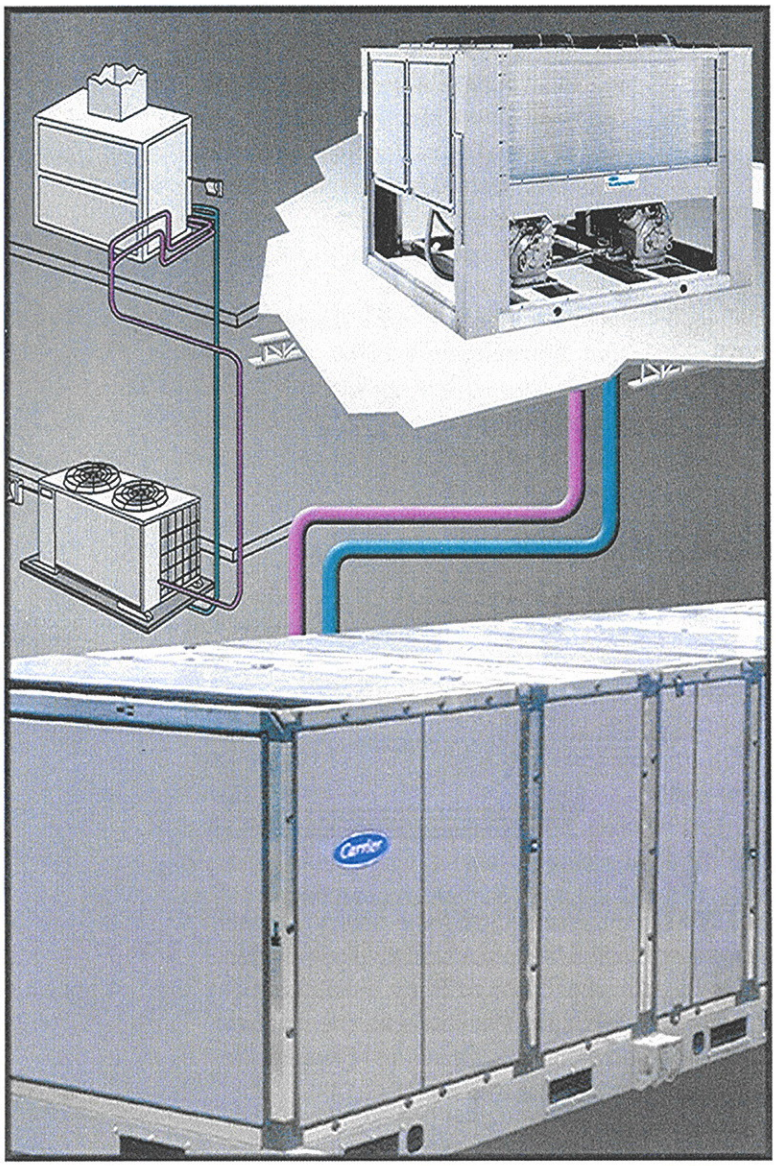


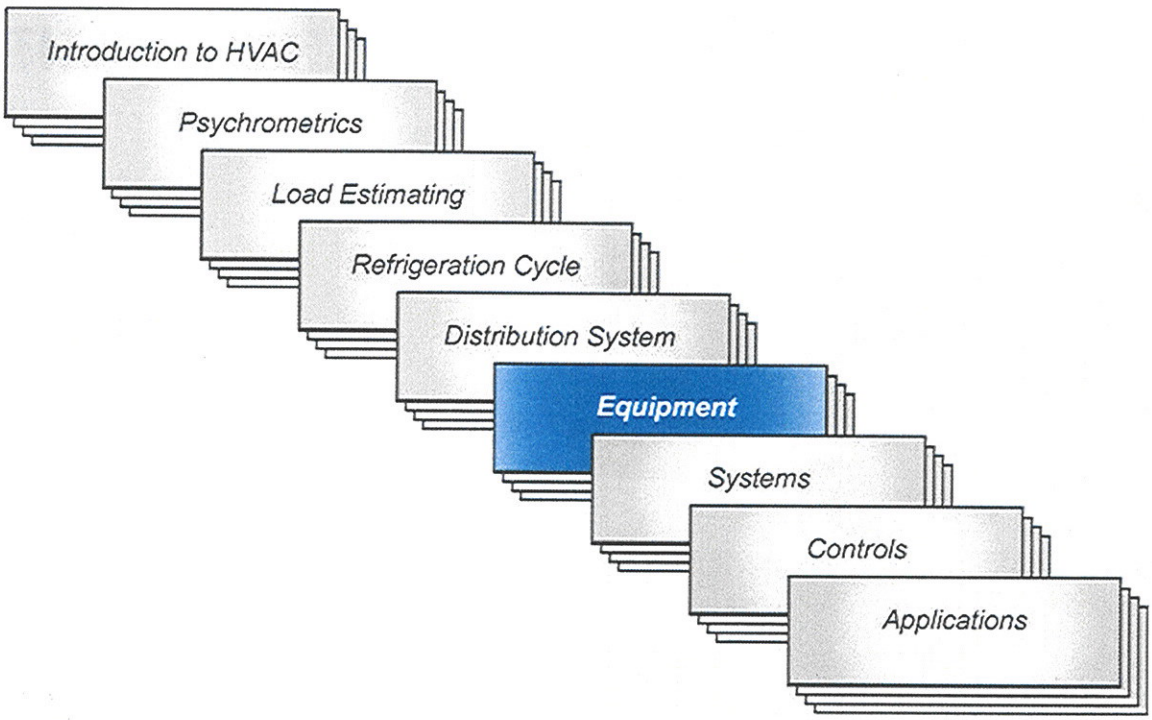
COMMERCIAL HVAC PACKAGED EQUIPMENT

Split Systems

Technical Development Program

Technical Development Programs (TDP) are modules of technical training on HVAC theory, system design, equipment selection and application topics. They are targeted at engineers and designers who wish to develop their knowledge in this field to effectively design, specify, sell or apply HVAC equipment in commercial applications.

Although TDP topics have been developed as stand-alone modules, there are logical groupings of topics. The modules within each group begin at an introductory level and progress to advanced levels. The breadth of this offering allows for customization into a complete HVAC curriculum – from a complete HVAC design course at an introductory-level or to an advanced-level design course. Advanced-level modules assume prerequisite knowledge and do not review basic concepts.



Spilt systems are one of the major categories of HVAC equipment, and the primary system type used in residential air conditioning. Split systems are classified as a unitary, or packaged unit; and, as such, have many of the benefits of packaged equipment while offering the flexibility associated with applied products. This module will describe what split systems are, the components of the system and accessories frequently used. It will show the designer how systems are applied, explain common installation issues, and describe how to select a system.

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The information in this manual is offered as a general guide for the use of industry and consulting engineers in designing systems. Judgment is required for application of this information to specific installations and design applications. Carrier is not responsible for any uses made of this information and assumes no responsibility for the performance or desirability of any resulting system design.

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Introduction

A system designer must be able to choose the system that will best fit the application. To do this, the designer must thoroughly understand each system, its benefits, and the components that make up the system.

A split system is a direct expansion (DX) air conditioning or heat pump system that has an evaporator, fan, compressor, and condenser section where one or more of the components are separated and connected by refrigerant piping. In most residential and commercial applications, the compressor and condenser are combined into a single piece of equipment called a condensing unit. Refrigerant piping and control wiring connects the system components and is field-installed to meet the physical requirements of each individual application.

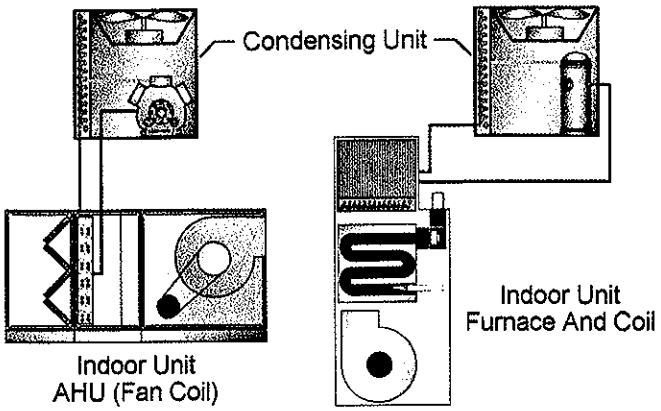


Figure 1

Split System Components

Split Systems

Split systems are a popular way to cool buildings, from residential and small commercial applications to large commercial applications. Split systems range in size from less than one ton in small applications to above 120 tons in larger applications. When utilized in a multi-unit design, very large commercial buildings can be handled with split systems. Split systems include cooling only applications, air source heat pumps, and process applications. They may be equipped with electric heat, hydronic heat, or steam heat. Split systems may also be combined with furnace systems to provide cooling and heating.

Split systems provide the opportunity to utilize packaged products in an applied manner. This means that factory-assembled products may be applied in factory-approved combinations to provide an engineered system that most closely meets the need of the application. There are many benefits to split systems, including this flexibility, and they will be discussed in detail.

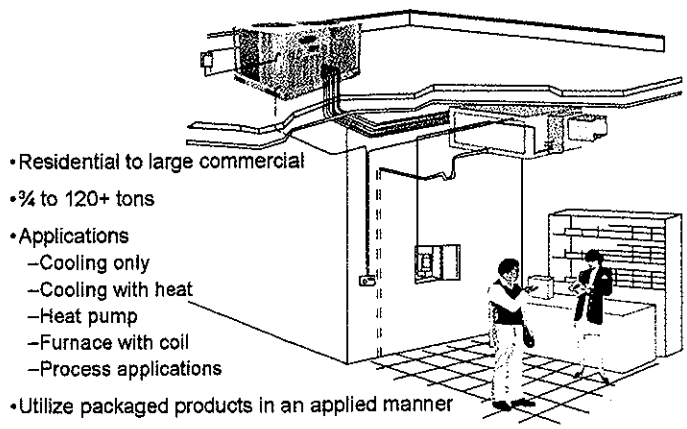


Figure 2

Split Systems

Definitions and Descriptions

The term "packaged" covers a wide range of factory-assembled products from room air conditioners to large tonnage water chillers. For purposes of this TDP, packaged is defined as those products that fall within the unitary air conditioner category. The Air Conditioning and Refrigeration Institute (ARI) defines the unitary air conditioner as one or more factory made assemblies that normally include an evaporator or cooling coil, an air moving device or fan, a compressor, and a condenser.

Split systems are defined as those systems that have more than one factory-made assembly, such as a packaged air handler and a condensing unit. These separate units may be placed indoors or outdoors, depending on the requirements of the application.

ARI has five basic categories of split systems. For split systems, there are options for air-cooled, water-cooled, and evaporative-cooled systems. As shown here, there are many different ways of separating the four unit components to develop a split system. As you can see, split systems have a wide variety of combinations, which provide a high degree of flexibility.

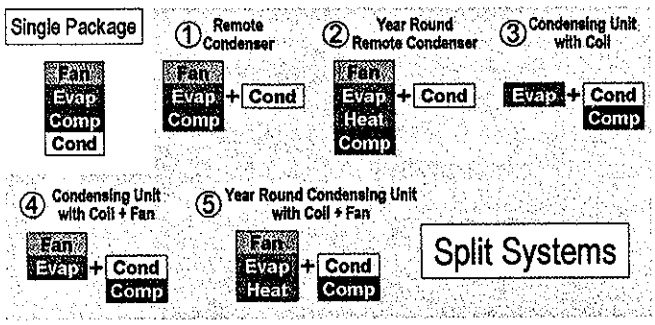


Figure 3

ARI Definition of Packages

Common Use of Split Systems

The split system industry is a mature market that has been relatively stable for many years, with typical year-after-year variations in volume being quite small. The exception to this has been the heat pump segment of the market. This segment has grown significantly in recent years as more attention is given to energy costs and comparisons are made to more traditional fossil fuel heating methods.

The split system industry is more often used in the replacement market than in new construction. It is generally accepted that at least 50 percent of the split system business is replacement, and some markets say it may be as high as 80 percent. Rooftop units are used more often in new construction because of their low first cost in comparison to split systems; only one unit needs to be installed and only one electrical service needs to be provided.

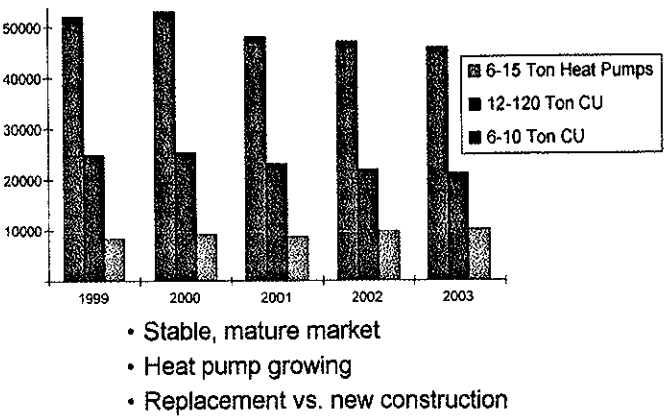


Figure 4

Recent Market Statistics

Advantages of Split Systems

The key advantage in using split systems is their flexibility. This flexibility allows many possible solutions to application challenges. Typically, splits are applied when one or more specific needs must be addressed. These needs include aesthetics, space utilization, duct requirements, and performance and zoning needs.



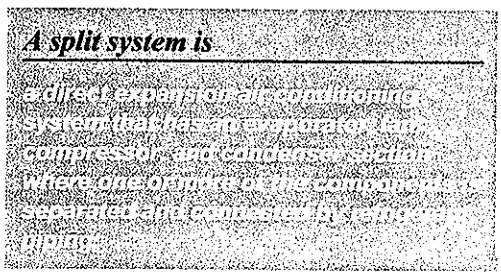
Figure 5

The key advantage of split systems is their flexibility.

Aesthetics is a significant factor in choosing split systems for an application. For example, a restaurant with a large skylight in the dining area would not be an appropriate application for a rooftop unit, but a split system condensing unit could be hidden behind the building. Splits are popular with churches for the same reason. The air handler may be located anywhere in the building, within refrigerant line limitations. The condensing unit may be located outdoors where it may be concealed, thereby contributing to the building's aesthetics, rather than detracting from it. For structures greater than two stories in height, the cost of ductwork may override the initial first cost advantage of a rooftop unit. With a split system, you may place the evaporator very close to or in the conditioned space, thereby greatly reducing ductwork cost. This also allows a building to be zoned on a floor-by-floor basis, eliminating the need for a large vertical duct chase. The split system also eliminates the need for large penetrations in the roof or exterior walls that are required with other packaged products. The performance aspect relates to the ability to mix and match components in order to engineer a system that is exactly right for the application. For example, a split system using an up-sized indoor unit can more closely match the requirements of an application that has a higher sensible load than a typical rooftop. Conversely, up-sizing the outdoor unit provides a system with greater latent performance.

Split System Basics

There are many types of systems available for a project, so why are split systems selected for a given application? With the various ways of dividing split system components, when is one selected over another? To answer these questions, a system designer should understand the components of a split system and the limits of their application.



As discussed previously, a split system is comprised of two or more packaged assemblies. These assemblies are interconnected with refrigerant piping and wiring, and they comprise the air conditioning system. The most common split system is made up of two assemblies, the outdoor unit, and the indoor unit. The outdoor unit is a condensing unit or heat pump and the indoor unit is a coil/fan combination, for example a packaged air handler. Another type of split system is the "triple split" in which the compressor and condenser are separated components. In this presentation, we will concentrate on the two-unit style split system.

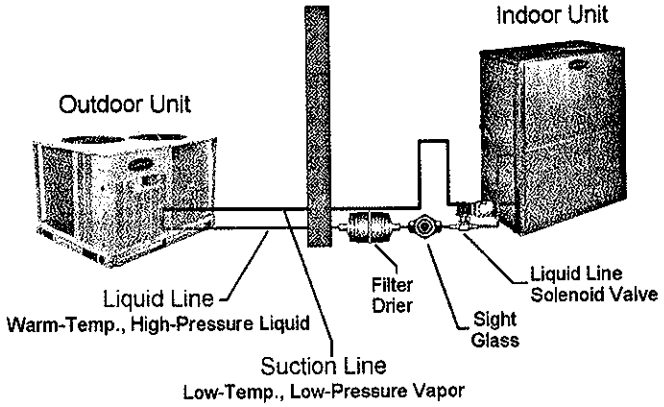
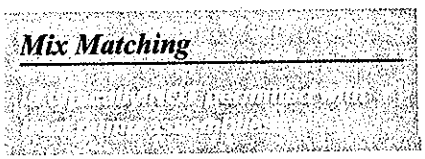


Figure 6

Basic Split System

Mix and Match Components

The flexibility advantage of the split system is a result of the designer's ability to mix and match assemblies, within manufacturer's guidelines. The most common combination of outdoor and indoor units would be assemblies that have the same capacity, e.g., a 10-ton outdoor unit combined with a 10-ton indoor unit. However, the designer may be able to match a 10-ton outdoor unit with the next size larger indoor unit, e.g., a 12½-ton indoor unit. This combination will typically provide higher airflows and higher sensible heat ratios. Alternatively, the designer may be able to match a 7½-ton outdoor unit with a 6-ton indoor unit. This combination will typically provide better latent performance. Always consult the manufacturer's recommendations regarding the limitations on mix-matching indoor and outdoor assemblies. In most cases, mix matching of heat pump assemblies is NOT allowed.



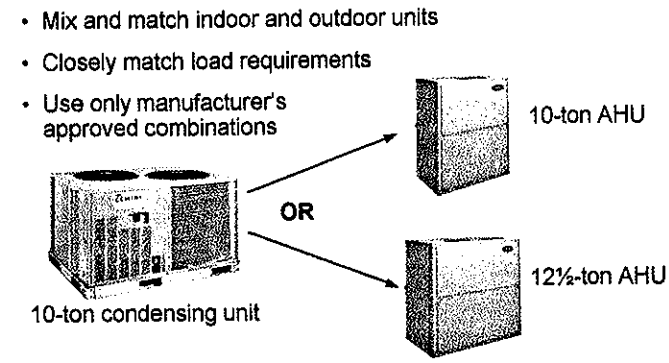


Figure 7

Split systems provide the flexibility to mix and match assemblies.

Residential and Duct Free Systems

Two additional variations of the split system concept are the residential style and the duct-free type. Residential split systems typically utilize an air-cooled condensing unit or heat pump matched with either a fan coil or an indoor coil assembly. In general, residential systems are defined as systems less than five tons. However, this does not mean that residential systems are less sophisticated. Some residential products use variable speed and highly-refined control technology

Duct-free systems, as their name implies, utilize indoor units that are placed in the conditioned space, thereby eliminating the need for ducts. Again, these systems can be sophisticated air conditioning units.

Both types of systems are frequently used in many commercial applications for smaller spaces and special application requirements.

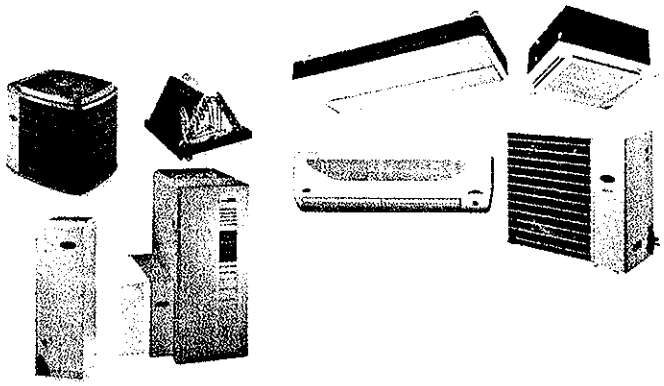


Figure 8

Residential and Duct-Free Split Systems

Typical Split System – Outdoor Unit

As mentioned previously, the outdoor unit of a two-assembly style split system is a condensing unit. A condensing unit is comprised of a compressor, a condenser, and a control system. The control system for a condensing unit includes an interface with space temperature controls and safety circuits, as well as to the control of the indoor unit. The controls for a condensing unit may be as simple as single-stage thermostat or a more complex programmable controller.

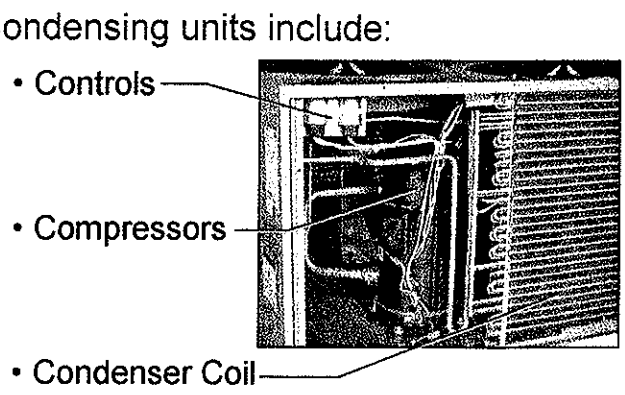


Figure 9

Condensing Unit

Condensing units smaller than 10 tons will typically have only one compressor. Larger tonnage condensing units may have one or more compressors with 40 tons generally being the largest single compressor unit. The condenser in most condensing units is air-cooled. However, water-cooled condensing units are also available.

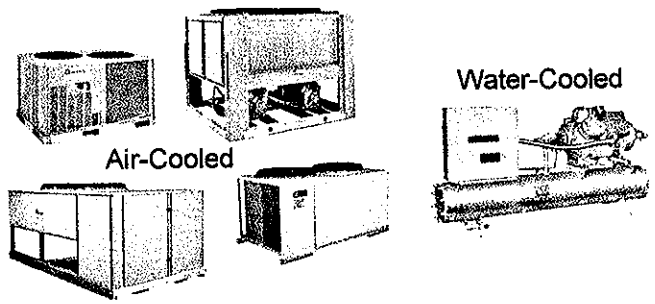


Figure 10

Typical Condensing Units

Typical Split System – Indoor Unit

The indoor unit in most commercial applications will be an air handler. This air handler may be a packaged air handler or it may be a built-up type, also known as a central station air handler. Central station air handlers can be further classified into three types: factory-assembled, custom air handlers, and field-erected air handlers. In factory-assembled air handlers, a wide range of pre-engineered components is available for selection. They are factory-assembled in a number of defined configurations. With custom air handlers, within certain limits, the components are selected and factory assembled for a specific project. The components of field-erected air handlers are selected for the project, and the air handler is field-constructed around the components. All three types of air handlers are used with split systems.

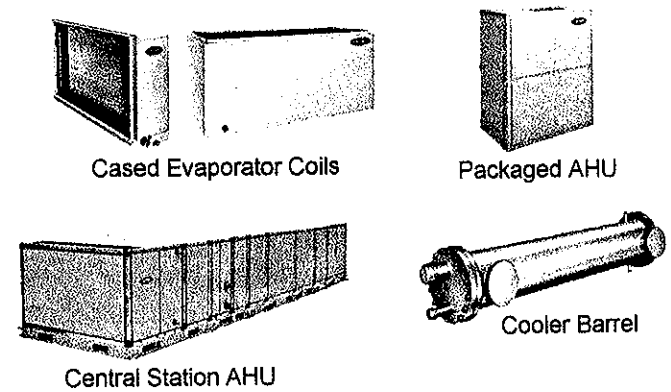


Figure 11

Indoor Units

Residential split systems and some commercial systems will use a cased evaporator coil as the indoor unit. In these applications, some other device, such as the fan in a furnace, provides the air movement.

An air-cooled chiller may also be constructed by matching a split-system condensing unit with a cooler barrel (i.e. evaporator). However, a packaged air-cooled chiller may be a better choice when available as the cooler and condensing sections are already pre-selected. The cooler barrel can be remote mounted in some cases.

Heat Pump Systems

The typical systems described previously may be defined as cooling-only systems. Split systems may also be heat pump systems. The most common heat pump system is an air-to-air heat pump arrangement.

These heat pump systems employ special indoor and outdoor units that are designed to function as either an evaporator or condenser. Typically, the coils used are larger than a comparably sized cooling-only unit. In addition, the metering devices are different in order to accomplish both heating and cooling. When a heat pump unit is in cooling mode, it functions in the same manner as a cooling-only unit; the outdoor coil is the condenser and the indoor coil is the evaporator. However, when the unit is in heating mode, a 4-way valve is used to reverse the cycle; the outdoor coil is now the evaporator and the indoor coil is the condenser. In this way, heat is removed from the outdoor air and transferred to the indoor air.

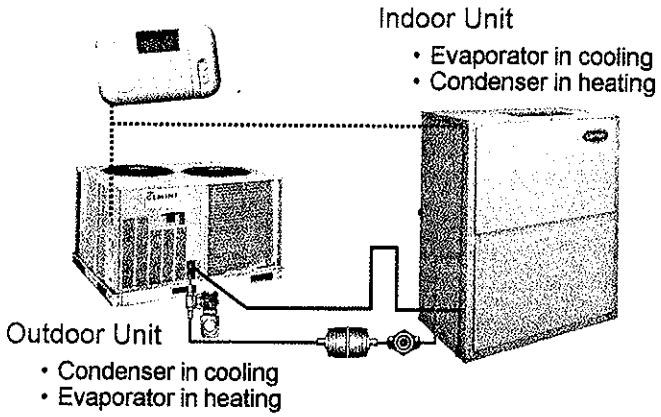


Figure 12

Heat Pump Split System

Heat pump system components are designed and tested as matched pairs and must only be applied according to the manufacturer's recommendations.

Refrigerant Circuits

The number of refrigerant circuits, single or dual, may also classify split systems. This definition is most often applied to the condensing unit. 10-ton and smaller condensing units are typically single circuit. Most single-circuit condensing units have only one compressor, however, specially designed dual-compressor single circuit systems are available. A single circuit system may be identified by the single liquid line and single suction line connecting the outdoor unit to the indoor unit. Single circuit systems are the simplest systems and in many cases are the least costly to install. Dual circuit condensing units have two independent refrigerant circuits and at least two compressors. Dual circuit systems utilize two liquid lines and two suction lines between the indoor and outdoor units. The primary advantage of dual circuit systems is redundancy. If one compressor fails, the other circuit will continue to operate and provide 50 percent of the nominal capacity.

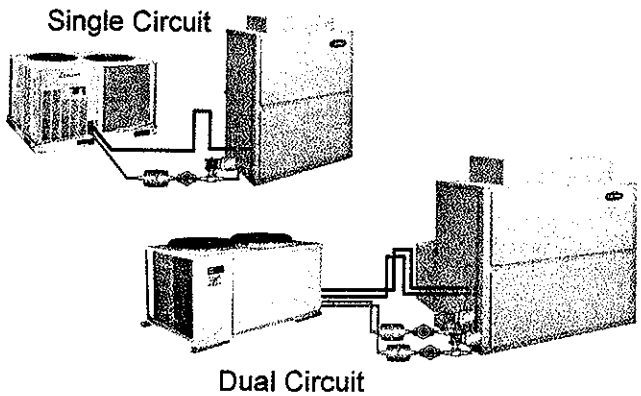


Figure 13

Refrigerant Circuits

Refrigerant Circuits – Indoor Unit

Indoor units may also be referred to as single or dual circuit, meaning the refrigerant either flows through the coil in a single path or splits into two paths. Single circuit coils typically have one TXV/distributor assembly and dual circuit coils will have two TXV/distributor assemblies. A single-circuit condensing unit may be connected to a single or dual-circuit indoor unit. However, a dual-circuit condensing unit must only be connected to a dual-circuit indoor unit due to compressor oil management. Dual-circuit condensing units have at least two compressors.

In any properly operating refrigeration system, a small portion of the compressor oil is constantly moving throughout the system. The key to compressor oil management is that the oil leaving the compressor through the discharge side must be continually replaced by oil returning on the suction side. Dual independent refrigerant circuits ensure that the oil that leaves compressor A of a dual-circuit condensing unit may only return to compressor A. If a dual circuit-condensing unit were applied to a single-circuit indoor unit by manifolding the refrigerant lines, the ability to manage the compressor oil would be lost.

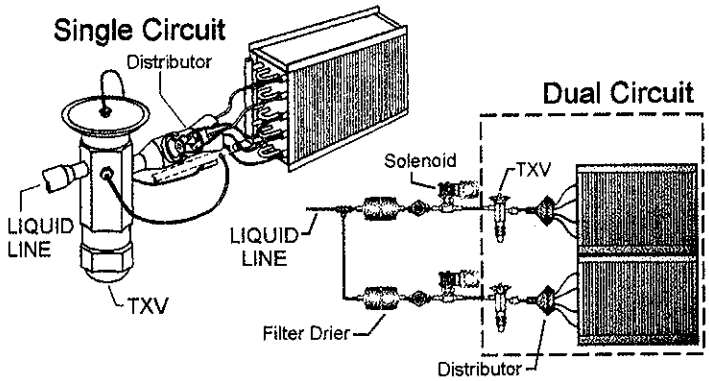


Figure 14

Indoor Unit, Refrigerant Circuits

Codes and Standards

System designers should be aware of a number of codes and standards. These include ARI and ASHRAE standards that have been incorporated into building codes. The Air Conditioning and Refrigeration Institute (ARI) standards primarily define performance-testing methods.

The standard applicable to split systems depends upon the capacity of the system, expressed in Btuh. For example, ARI Standard 340/360 applies to air-cooled split systems with a capacity greater than 65,000 Btuh and less than 250,000 Btuh. This standard defines that the equipment will be tested at 80° F db/67° F wb return air, 95° F outdoor air. These conditions are known as ARI conditions. Since performance is a function of both the indoor and outdoor performance, Standard 340/360 applies to a system, such as a combination of an indoor and an outdoor unit.



Standard #	Applies to	Capacity Range
210/240	Unitary Air Conditioners	<65,000 Btuh
	Air Source Unitary Heat Pumps (Air-Cooled)	<65,000 Btuh
340/360	Unitary Air Conditioners	65,000 to <250,000 Btuh
	Air Source Unitary Heat Pumps (Air-Cooled)	65,000 to <250,000 Btuh
365	Air Conditioning Condensing Units	>135,000 to <250,000

Figure 15

ARI Standards influencing Split Systems

Typically, manufacturers submit data to ARI stating that a given split system has been tested according to the applicable ARI standard and they verify the performance value in Btuh and the Energy Efficiency Ratio (EER). This data is listed by ARI and is available to the system designers through ARI. In the case of Standard 340/360, listed systems are subject to performance verification by ARI. To verify performance, ARI may, at any time, randomly select from a manufacturer's inventory listed units or combinations. These units are sent to an independent laboratory for performance testing. The equipment performance must match the listed values within 5 percent. Figure 15 lists the ARI standards applicable to split systems.

Calculating EER

Since EER is used to comply with standards, it is important to understand how it is calculated. The formula is: EER equals capacity (expressed in Btuh) divided by the total power input (expressed in Watts). EER is expressed as a pure number with the units of measure (Btuh/Watts) are normally left off. A higher EER number represents a higher efficiency. The simple formula noted here is suitable for a stand-alone condensing unit and listed combinations of indoor units. For example, the published capacity of a 25-ton condensing unit operating at 95° F outdoor air and 45° F saturated suction temperature (ARI conditions) is 290 MBtuh. The power input equals compressor power plus the total power required by the condenser fan motors. The published compressor power at the conditions noted is 22.8 kW. The condenser fan motors require a total of 3.1 kW.

$$EER = \frac{\text{Capacity (Btuh)}}{\text{Total Power Input (Watts)}}$$

Example 25-ton condensing unit @ ARI conditions

$$EER = \frac{\text{Capacity (Btuh)}}{\text{Total Power Input (Watts)}}$$

$$EER = \frac{290\ \text{MBtuh}}{(22.8 + 3.1)\ \text{kW}}$$

$$EER = \frac{290}{25.9}$$

$$EER = 11.2$$

Figure 16

Calculating EER

Therefore, the EER of this 25-ton condensing unit operating at ARI conditions is:

EER= 290 MBtuh / (22.8 + 3.1) kW

EER = 290 / 25.9

EER = 11.2

Net vs. Gross Capacity

It is slightly more complicated to calculate EER for a system, which is not a listed combination. ARI published data is for a system combination at the specific ARI rating conditions. The calculation procedure is different when a condensing unit rating or a point other than the ARI rating is used.

The formula for a system is: EER equals net cooling capacity (Btuh)/ total power input. Remember that the operating conditions will affect capacity; therefore, they also affect the EER. The ARI condition for a commercial split system is defined as 80° F db and 67° F wb return air temperature, 95° F outdoor air. First, the difference in gross capacity versus net capacity must be addressed. The capacity value published by most manufacturers is the gross capacity; that is, the amount of heat removed by the evaporator coil. However, the indoor fan motor (IFM) adds heat to the system which means the actual cooling to the space is less. Net capacity during cooling mode is defined as the gross capacity minus the indoor fan heat. The first step in determining the system EER is to calculate the net cooling capacity. To do this, you need to know the heat added by the IFM. Typically, manufacturer's data will provide the brake horsepower (bhp) requirements of the IFM operating at given airflow (cfm) and resistance (static pressure). The ARI standard defines the minimum external resistance based on the size of the unit.

$$\text{System EER} = \frac{\text{Net Capacity (Btuh)}}{\text{Total Power Input}}$$

$$\text{Net Capacity} = \text{Gross Capacity} - \text{IFM Heat}$$

ARI Minimum External Resistance Table	
Standard Ratings MBtuh	**Minimum External Resistance Inches of Water**
135 - 210	0.35
211 - 280	0.40
281 - 350	0.45
351 - 400	0.55
401 - 500	0.65
501 and over	0.75

Figure 17

Net vs. Gross Capacity

Example of bhp

As an example, lets look at a 25-ton packaged air handler operating at 10,000 cfm with 0.44 in. wg of external static. Interpolating from the published data, between 0.4 and 0.6 in. wg external pressure, the bhp requirement is 4.0 bhp.

AHU Size	Airflow cfm	External Static Pressure (in. wg)							
		0.0		0.2		0.4		0.6	
		rpm	bhp	rpm	bhp	rpm	bhp	rpm	bhp
028	10,000	615	3.12	641	3.36	692	3.87	743	4.41

Interpolate to derive bhp for 0.44 in. wg

bhp @ 0.44 in. wg = 4.0 bhp

Figure 18

Demonstration of the bhp required for specific levels of external static pressure.

Indoor Fan Motor Heat

IFM heat equals the bhp (from the published data) multiplied by 746 (Watts/hp), divided by the motor efficiency. If motor efficiency is not known, 0.83 is a good assumption. This equation will provide the IFM heat expressed in Watts. To convert to Btuh, multiply the result by 3.414 (Btuh/ Watt) to express the IFM heat in Btuh.

$$\text{IFM Heat} = \frac{(\text{bhp} * 746)}{\text{Motor Efficiency}}$$

$$\text{IFM Heat} = \frac{(4.00 * 746)}{0.83}$$

$$\text{IFM Heat} = 3{,}595 \text{ Watts}$$

$$3{,}595 \text{ Watts} * \frac{3.414 \text{ Btuh}}{\text{Watts}} = 12{,}274 \text{ Btuh}$$

Figure 19

Indoor Fan Motor Heat

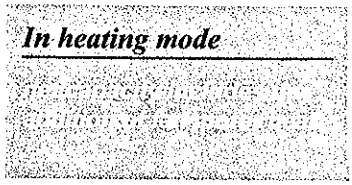
Net Capacity

Now we can determine the net capacity during the cooling mode. The manufacturer's data indicates that the gross capacity of the example air handler at the conditions noted is 294 MBtuh. To calculate net capacity:

Net Capacity = Gross Capacity – IFM Heat

Net Capacity = 294 MBtuh – 12.274 MBtuh

Net Capacity = 282 MBtuh



Total Power Input

To calculate the total power input, add all of the electrical inputs of the system, the compressor(s) plus the IFM, plus the outdoor fan motor(s) (OFM). If you do not have the power value for the OFM, it may be calculated if OFM motor horsepower is known.

$$\text{OFM power per motor} = \frac{(\text{bhp} * 746)}{\text{Motor Efficiency}}$$

Now calculate Total Power Input using data from previous slides

Total Power Input = Compressor power + IFM power + OFM power

Total Power Input = 22.8 kW + 3.6 kW + 3.1 kW

Total Power Input = 29.5 kW

Figure 20

Total Power Input

System EER

Now you can calculate system EER of our example 25-ton system.

System EER for the 25-ton example system:

$$\text{System EER} = \frac{\text{Net Cooling Capacity}}{\text{Total Power Input}}$$

$$\text{System EER} = \frac{282\ \text{MBtuh}}{29.5\ \text{kW}}$$

$$\text{System EER} = 9.6$$

Figure 21

System EER

SEER

The Seasonal Energy Efficiency Ratio (SEER) is similar to EER in that it defines the energy efficiency of a unit or system in the cooling mode. SEER only applies to units that operate on single-phase power and have a capacity of 5 tons or less. SEER differs from EER in a couple of ways. First, SEER considers the fact that the fan motor(s) and compressor cycle, therefore, the

energy usage is not constant. Secondly, SEER is calculated using three operating conditions plus a cycle test. Net capacity is determined at the ARI rating point, 80° F db, 67° F wb and 95° F outdoor air. Then ratings at two points: 80° F db, 67° F wb return air temperature, 82° F outdoor air; and 80° F db 57° F wb return air temperature, 82° F outdoor air. The later condition is used with a cyclic test to determine seasonal energy efficiency. SEER provides a means to evaluate performance at two seasonally different conditions, one high humidity and one low humidity. Calculating SEER involves laboratory testing to record the power and capacity measurements. Therefore, SEER information is provided by the manufacturer and cannot be calculated in the field.

Applies to:
- Single phase power only
- Capacity less than 60 MBtuh

Calculated at three conditions and cycle test:
- 80/67° F return air, 95° F outdoor air
- 80/67° F return air, 82° F outdoor air
- 80/57° F return air, 82° F outdoor air
- 80/57° F cycle test, 82° F outdoor air

Requires laboratory testing and is not calculated in the field.

Figure 22

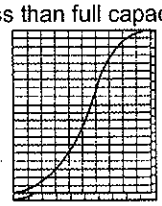
Calculating Seasonal Energy Efficiency Ratio (SEER)

IPLV

Integrated Part Load Value (IPLV) is used to evaluate the efficiency of a unit or system operating in the cooling mode at less than full capacity. IPLV is only applicable to equipment that has more than one stage of capacity, for example, equipment with multiple compressors or a single compressor unit with unloading. IPLV is a weighted average of the EER calculated at each stage of capacity of the unit. A unit that has a small number of steps of capacity will have a higher IPLV than one with many steps of capacity, all other factors being equal. It is important to understand that a unit with a higher number of steps of capacity will have the ability to more closely match the cooling load of the application and, therefore, is more efficient. Unless the unit will be operating at 100 percent capacity at all times, a unit with a higher IPLV is preferred.

FOR ALL 3 Ø AND WATER-COOLED UNITS AND AIR-COOLED UNITS ABOVE 60 MBH CAPACITY

- Evaluate equipment efficiency at less than full capacity
- Applicable only to equipment with more than one stage of capacity
- Weighted average of EER at each capacity step



- Equipment with greater number of capacity steps can more closely match the load requirements of the space
- Unless equipment is always operated at 100% capacity, a higher IPLV is preferred

Figure 23

Integrated Part Load Value (IPLV)

Note, IPLV is commonly expressed as EER (Btuh/Watt) for packaged equipment and as kW/ton for chillers. There is a fixed relationship between kW/ton and EER (EER = 12/(kW/ton)). This relationship shows that EER increases as kW/ton decreases, and vice versa. Therefore, a "better" IPLV is shown as a lower value when the units are kW/ton, and, a "better" IPLV is a higher value when the units are expressed in terms of EER.

COP

Coefficient of Performance (COP) is a value used to measure a unit's efficiency while operating in the heating mode and applies to heat pumps that operate on three-phase power. Since the compressor and indoor fan motor heat provide a positive benefit in heat pumps, their power is included in the heating calculation as a benefit. A higher COP value represents a more efficient heat pump.

- Applies to heat pumps that operate on 3-phase power only
- Measures efficiency while operating in the heating mode
- A higher COP indicates a more efficient heat pump

$$COP = \frac{\text{Net Capacity (Watts)}}{\text{Total Power Input (Watts)}}$$

Figure 24

Coefficient of Performance (COP)

COP = net capacity (Watts)/total power input (Watts)

Net capacity now includes the supply fan heat

Net capacity = gross compressor capacity + supply fan heat

Total Power Input = supply fan (Watts) + compressor(s) (Watts) + OFM motor(s) (Watts)

Heating performance varies as the outdoor temperature drops and when the temperature is below freezing and defrost is required. Defrost energy decreases the usable energy for space heating. To account for this, heat pump ratings are calculated at two points: high temperature at 70° F db and 60° F wb indoor and 47° F db and 43° F wb outdoor, and low temperature at 70° F db and 60° F wb indoor and 17° F db and 15° F wb outdoor.

HSPF

Heating Seasonal Performance Factor (HSPF) is used to measure the efficiency of heat pumps that operate on single-phase power and have a cooling capacity of less than 5.5 tons. HSPF is similar to SEER in that it represents the seasonally adjusted heating efficiency of a heat pump. A higher HSPF value represents a higher efficiency heat pump. Also, like the SEER, the measurement and calculation technique dictates that the testing can only be done in a laboratory. The impacts of defrost and supplemental heaters are factored into these calculations as well.

HSPF:

- Applies to heat pumps that operate on single phase power and have a cooling capacity of < 5.5 tons only
- Is similar to SEER in that it measures the seasonally adjusted efficiency of a heat pump
- Accounts for defrost and required electric heat
- A higher HSPF is a more efficient heat pump

Figure 25

Heating Seasonal Performance Factor (HSPF)

Building Energy Codes

Building codes regulate the building and the products used in them. Their primary purpose is to assure the safety of the building occupants. However, after the energy crunch of the 1970s, building performance standards started to become a provision of building codes. This activity has continued and today, energy requirements are a part of nearly every building code. One important point about building codes is that they establish minimum levels. Buildings may be built to levels that are more stringent but not less. Several ASHRAE standards have become incorporated into code requirements.

Many codes rely on ASHRAE 90.1 that sets minimum efficiency requirements.

Air-Cooled Split System Requirements

- Performance requirements

< 65,000 Btuh	10.0 SEER 1Ø
≥ 65,000 – < 135,000 Btuh	10.3 EER
≥ 135,000 – < 240,000 Btuh	9.7 EER
≥ 240,000 – < 760,000 Btuh	9.5 EER / 9.7 IPLV
≥ 760,000 Btuh	9.2 EER / 9.4 IPLV

- Control requirements
- Motor hp limits
- Economizer requirements
- Heat pump requirements

Figure 26

Energy codes establish air-cooled split system minimum performance requirements.

ASHRAE 90.1,Energy Standard for Buildings except Low-Rise Residential Buildings, has become the benchmark for energy codes. At the very basic level, you may consider ASHRAE Standard 90.1 as defining minimum energy efficiency standards for a variety of devices, including air conditioning equipment. As this standard applies to split systems, it defines the minimum EER, IPLV, and COP of systems, or, in some cases individual units, such as large condensing units. It also has a number of other provisions that affect the design of split systems. These provisions include requirements on the control system, limits on the indoor fan motor horsepower, requirements on the use of an economizer and requirements on heat pumps. Additional information on these requirements can be found in Sections 6.2 and 6.3 of the Standard.

A key factor when comparing efficiency values of split systems is to ensure that you are comparing apples to apples. For example, when comparing two brands, do the values reflect the total of the indoor and outdoor units? If both values represent the system efficiency, are the airflow and static pressure values for the indoor unit the same? If not, the comparison is not valid.

Indoor Air Quality and Sustainable Design

As was the case with energy, requirements have been written into building codes that set minimum standards for ventilation and control of conditions that can lead to poor indoor air quality. ASHRAE Standard 62, *Ventilation for Acceptable Indoor Air Quality*, is the industry guideline defining ventilation requirements for a variety of commercial applications.

- IAQ – ASHRAE 62
  - Limits maximum humidity to less than 65%
  - Indoor unit condensate control
  - Indoor unit ventilation capability
- Sustainable Design – LEED™
  - Require meeting ASHRAE 90.1 efficiency and ASHRAE 62 IAQ features
  - Optimized energy performance and IAQ

Split System mix and match provides better humidity control and flexibility to meet these requirements

Figure 27

IAQ and Sustainable Design

This standard has tables that set minimum ventilation airflows based on the type of building, the usage of the space, the number of people, and the space area. It also contains a number of provisions that influence the use of split systems. One of these requirements is to control the chance of mold growth. Humidity in the space must be kept below 65 percent. ASHRAE Standard 62 addresses moisture by limiting the allowable relative humidity in an occupied space to 65 percent or less at either of the following two design conditions:

- Peak outdoor dew point design conditions and peak indoor design latent load, or
- Lowest space sensible heat ratio expected to occur and the concurrent (simultaneous) outdoor conditions.

ASHRAE Standard 62 also notes that the load on a space may be significantly different at outdoor dew point design conditions than at outdoor dry bulb design conditions. It is important to design the system to handle the worst-case scenario, which may be the dew point design condition. The Standard also requires the design minimum outdoor air intake airflow to be greater than the design maximum exhaust airflow. In other words, the total building must be pressurized, understanding that certain spaces within the building may be at a negative pressure condition.

Ventilation requirements in split system applications may be handled in a variety of ways. The ventilation may be addressed directly in the split system by equipping the indoor section with a mixing box or economizer section. The ventilation needs may also be addressed by dedicated outdoor air system that is independent of the split system.

Split systems can offer a distinct advantage in dealing with these requirements. When spaces have high latent requirements because of the activity in the space or large amounts of humid outdoor air, humidity control can be a challenge. As indicated before, split systems allow a variety of system matches and the use of DX allows lower coil temperatures, which can result in much better humidity control.

Provisions must also be made for ventilation air ducted to each unit, which can impact the location of the indoor air handler. In addition, requirements for control of condensate within the air handler dictate the use of condensate pans with no standing water, double-wall construction, surfaces downstream of the coil protected from condensate damage and other IAQ protection measures. These measures may influence the air handler selected or the options required.

While energy efficiency and IAQ have dealt with setting a minimum performance standard for units, there is interest today in programs that promote achieving a superior level of energy performance and IAQ. These efforts are commonly called sustainable design, green buildings, or by the most common certifier of these buildings, LEED™ (Leadership in Energy and Environmental Design). These programs are aimed at driving building design to achieve the maximum economical performance and minimal environmental impact. The LEED™ program requires meeting all the requirements of the ASHRAE 90.1 Energy Standard and the ASHRAE 62 requirements of the Ventilation Standard. It then uses these standards as a benchmark to measure how much performance has been improved. Split systems, with the ability to closely match the load requirements and offer superior part load control, are worthy of consideration for projects seeking high levels of indoor air quality and LEED™ certification.

Systems and Components

Rules of Thumb

There are a number of "rules of thumb" regarding split systems. These rules should be considered a guide, not always the final authority. For example, historically, the nominal condition, rule of thumb regarding airflow has been defined as 400 cfm/ton. The range is typically considered to be 300 cfm/ton on the low side up to a maximum of 500 cfm/ton. Therefore, the data for a 10-ton packaged air handler will include performance and fan information across a range of 3000 to 5000 cfm. As guidelines change regarding the amount of outdoor air required in many applications, it is causing the "rule of thumb" to shift downward. It is not uncommon today to see systems designed at 350 cfm/ton.

As addressed before, split systems offer the flexibility of matching different air handlers to a condensing unit. A good rule of thumb to follow is one size up and one size down is acceptable in the air handler match. Other options may be available but would need investigation by the manufacturer.

Rules of Thumb are considered to be guidelines only

- Airflow:
  - 400 cfm per nominal ton
  - Range of 300 to 500 cfm per ton
  - Today, 350 cfm per ton may be more appropriate
- Mix and Match:
  - Nominal and one size up, sometimes one size down, others depend (consult the manufacturer)
- Line Length:
  - Keep them at 100 ft or less

Figure 28

Rules of Thumb

One other quick rule to keep in mind is to limit the measured line length between the indoor and the outdoor units to 100 feet or less. While units are often capable of much greater distances, this is a good guideline in terms of selecting locations for the indoor and outdoor units.

Operating Limits

There are a number of parameters that define the proper operating envelope for a split system. These include:

- Maximum outdoor air temperature 115° F
- Minimum return air temperature 55° F
- Maximum return air temperature 95° F
- Saturated suction temperature range 25 - 55° F
- Maximum discharge temperature 275° F
- Minimum discharge superheat 60° F

Saturated suction temperature

If the equipment is a heat pump, two additional parameters are considered:

- Maximum outdoor air operating temperature in heating 75° F
- Minimum outdoor air operating temperature in heating -20° F

Although a heat pump can safely operate at very low temperatures, it should be understood that a heat pump does not operate efficiently at low temperatures. Therefore, heat pump systems may employ supplemental heating systems, most commonly electric heaters in the indoor units. In some applications, building codes set the need for heaters and the size of the heaters.

It is incumbent upon the designer to make sure that the equipment selected will operate within these limitations throughout the operating envelope of the application.

Outdoor Units

Let's discuss some of the variables found in outdoor units, or more generically, condensing units. Obviously, one variable is size, or capacity. As described earlier, residential condensing units typically have a nominal capacity range of 1½ tons to 5 tons. Commercial condensing units range in size from a nominal 6 tons to 120 tons and greater. Another variable is the type of compressor. Typically, condensing units with a nominal capacity of 10 tons or less use hermetic type compressors, with scroll compressors being the most common today. This choice provides a reasonably priced compressor that meets the relatively simple need of a small split system.

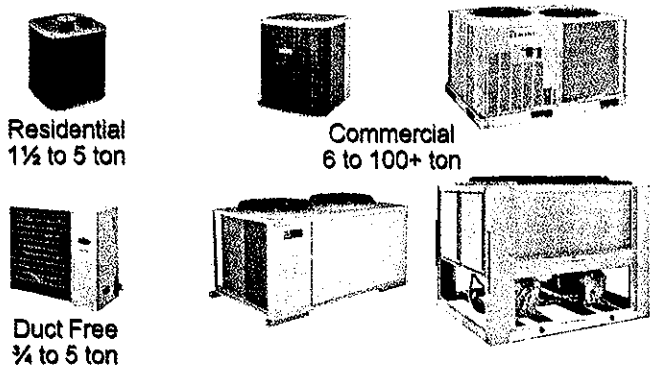


Figure 29

Outdoor Unit

Semi-Hermetic Compressors

10-ton and larger condensing units may be equipped with reciprocating semi-hermetic compressors. The semi-hermetic compressor offers the flexibility of a repairable compressor vs. replacement being the only option with a failed hermetic compressor. More importantly, reciprocating semi-hermetic compressors offer the capability of capacity control through cylinder unloading. This provides a means for a relatively large single-compressor condensing unit to adjust its capacity to meet the load requirements of the application. For example, a 40-ton semi-hermetic compressor may have 3 stages of capacity, 100 percent, 67 percent, and 33 percent. In other words, this 40-ton compressor may operate at 40 tons, 27 tons, or 13 tons, depending on the needs of the application.

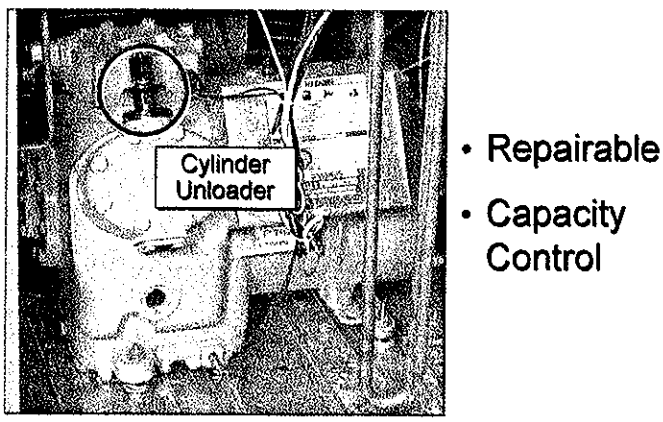


Figure 30

Semi-Hermetic Compressor

Multiple Compressors

Condensing units may also be equipped with more than one compressor. Typically, multiple compressor condensing units have a nominal capacity of 10 tons or larger. Many multiple compressor condensing units are dual circuit units. The use of multiple compressors provides another means of capacity control, i.e., by turning compressors on and off, the total capacity of the condensing unit may be changed. It is possible to have multiple compressors manifolded together on a single circuit, however this requires special consideration by the equipment designer in the area of compressor oil management.

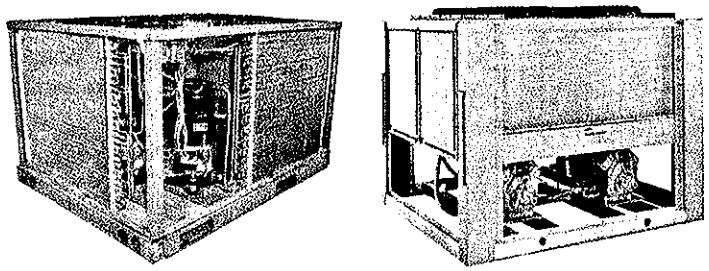


Figure 31

Multiple Compressors

Multiple Condensing Units

Another variation of the multiple compressor concept is the use of multiple condensing units. It is possible to use two, single-circuit, condensing units connected to a dual circuit air handler. This method provides a means of capacity control by staging the condensing units. It also provides a system in which the outdoor sections are completely independent, which in some applications may be an important additional level of redundancy. There is also an advantage in that one unit may be serviced while the other is operating. For critical applications, this provides a means of having at least 50 percent capacity while maintenance is performed on the other outdoor units. The disadvantages include: dual electrical services must be installed, two units must be rigged, two pads (mountings) must be provided, etc.

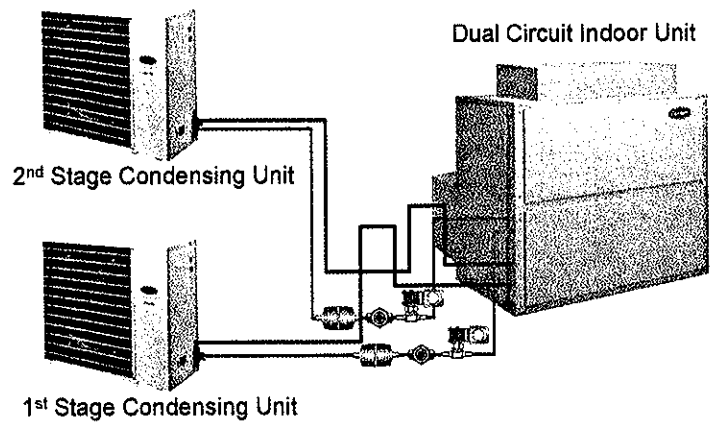


Figure 32

Multiple Condensing Units

Hot Gas Bypass

Hot gas bypass (HGBP) is a piping arrangement that is designed to protect the system in low load conditions. Specifically, HGBP will limit the minimum evaporator temperature in low load conditions to prevent coil icing. A HGBP system is not a form of capacity control, however, it is sometimes applied in that manner. For example, a condensing unit that is equipped with a single scroll compressor does not have any means of capacity control. Therefore, the designer, to protect the system in low load conditions, may specify HGBP. An HGBP system is composed of a hot gas valve, a solenoid valve, a connection point to inject the hot gas, and interconnecting piping and control wiring. The hot gas must be injected at the indoor unit evaporator coil, between the TXV and the distributor. If the indoor unit does not have a hot gas connection, an auxiliary side connection must be installed. Do not inject hot gas directly into the suction line because compressor overheating may result. If the system is equipped with a multi-step thermostat, the hot gas solenoid should be active only in the minimum stage of cooling.

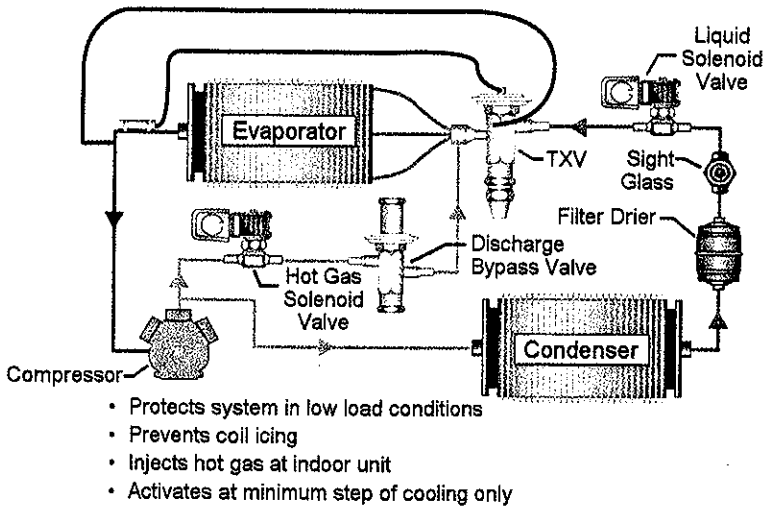


Figure 33

Hot Gas Bypass

Alternative Condensing Unit Solutions

There may be applications where it is desired to install the outdoor unit, the condensing unit, indoors. In these cases, it is necessary to make special provisions to remove heat from the space in which the condensing unit is located. The typical air-cooled condensing unit utilizes propeller type fans, which are designed to operate against very low static. Therefore, ducting the condenser air is not a viable option.

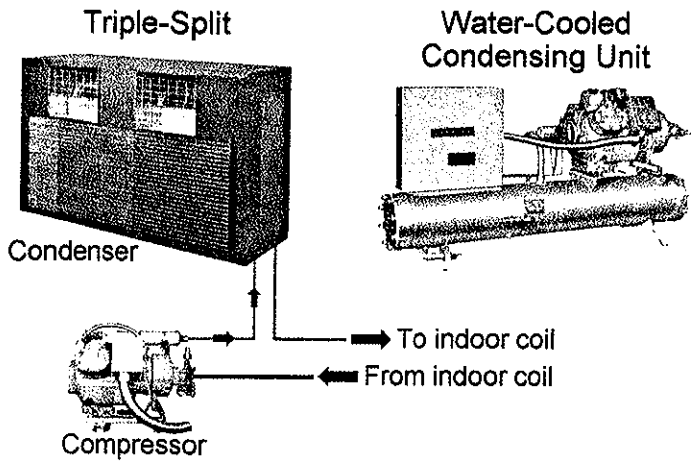


Figure 34

Alternative Condensing Unit Solutions

The potential options include use of:

- a specialty condensing unit equipped with fans capable of being ducted
- a water-cooled condensing unit
- an air-cooled indoor self-contained unit
- a triple split in which the separate compressor and fan coil are indoors and the air-cooled condenser is outdoors using propeller fans or indoors using centrifugal fans.

Heat Pump Outdoor Unit

The outdoor unit in an air-to-air heat pump system is a special adaptation of an air-cooled condensing unit. In addition to the components found in a condensing unit, the heat pump will also have a reversing valve and normally will include a suction accumulator. The reversing valve, or 4-way valve, provides the means to reconfigure the refrigerant flow path in order for the outdoor unit to be the condenser in cooling and the evaporator in heating. The accumulator is a protective device that prevents liquid refrigerant from reaching the compressor, thereby preventing damage that could result.

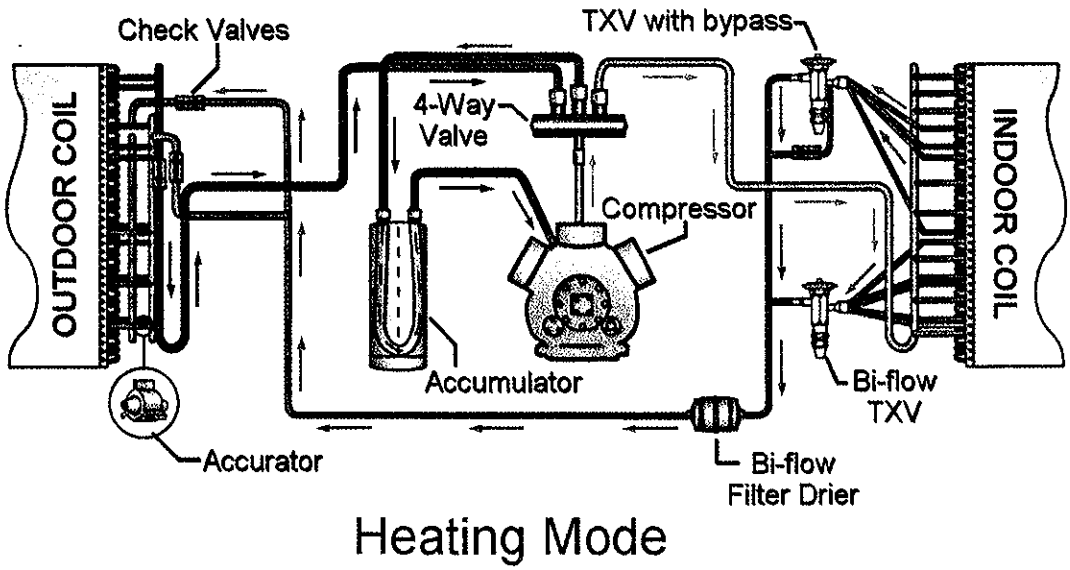


Figure 35

Heat Pump in Heating Mode

The design of the outdoor coil in a heat pump also receives special attention. In order for the coil to operate effectively as both a condenser and an evaporator, the coil must be designed and tested to work in conjunction with a particular indoor unit (coil). For this reason, heat pumps are provided as a system only, an outdoor unit matched with an indoor unit. It is not possible to mix and match indoor and outdoor units in a heat pump application unless the combination has been tested.

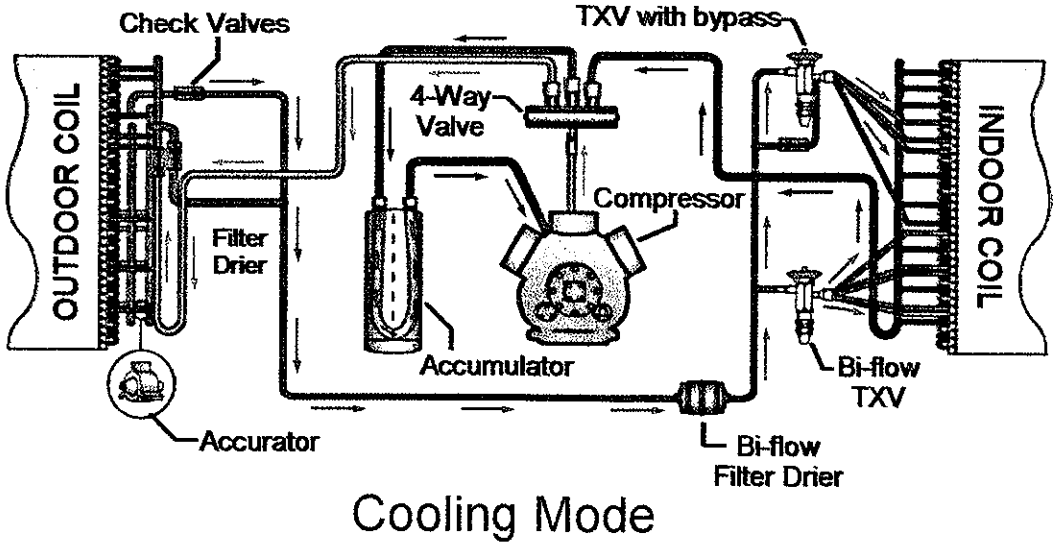


Figure 36

Heat Pump in Cooling Mode

Indoor Units

In most commercial applications, the indoor unit will be an air-handling unit (AHU), also known as an air handler. The AHU may be a simple packaged air handler. Packaged AHUs are typically available in capacities from 6 to 30-ton with the term "packaged" indicating that the product offering is available in a limited number of predefined sizes. The advantage of the packaged air handler is that the TXV(s) and nozzle(s) are factory installed. The other end of the spectrum for commercial AHUs is the applied air handler or central station air handler. The term "applied" is an appropriate description because air handlers of this type are designed and constructed in modules, based on the needs of the application. For example, the designer chooses the fan section, the coil, the filter section, etc.

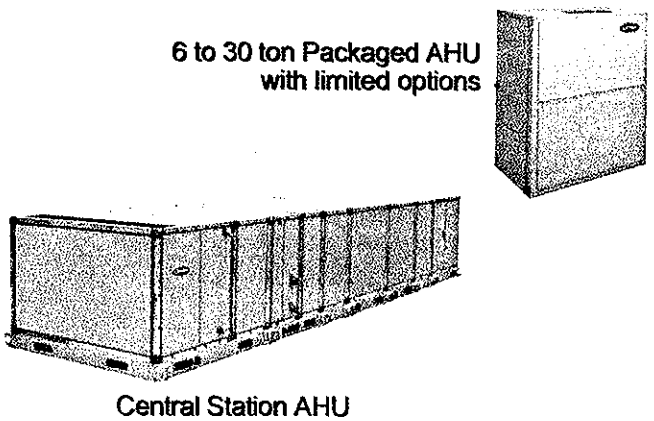


Figure 37

Indoor Unit – Air Handler

Classification of Air Handlers:

IAQ Features

Indoor Air Quality (IAQ) features are an important consideration when selecting an AHU. The type of construction is a very basic choice. An AHU with double-wall construction sandwiches the insulation between the outer casing and an interior metal liner. This design prevents exposure of the insulation to the moving airstream, thereby eliminating any possibility that insulation particles may be carried into the space. Double-wall construction is common on built-up style AHUs, but typically is not available on packaged AHUs. Packaged air handlers typically use a dual-density, coated insulation, which is designed for exposure to the moving airstream, yet will not shed particles at velocities encountered inside the AHU. This type of insulation may also be treated with an anti-microbial coating to inhibit the growth of bio-aerosols inside the AHU. Foil-faced insulation is also common in packaged air handlers. The double-wall system or the insulations described, all offer an AHU interior that may be cleaned. Ultra-violet UV-c lights mounted inside the AHU may also be utilized to limit the growth of bio-aerosols on the coil or in the drain pan.

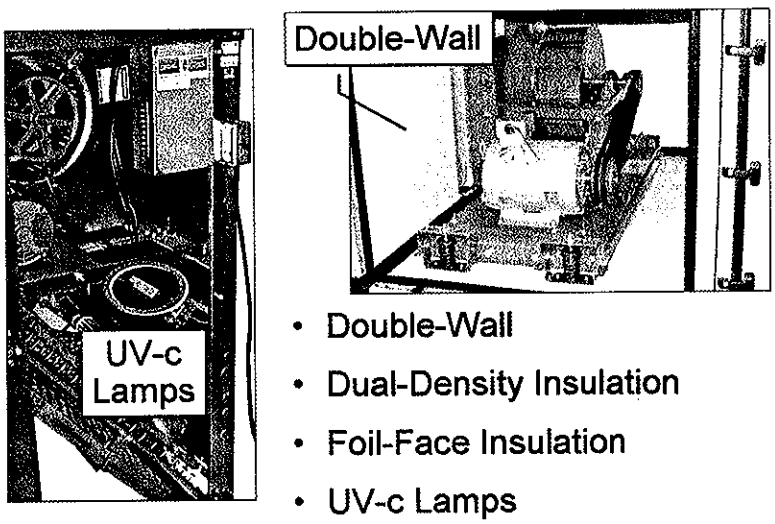


Figure 38

IAQ Features

Constant Volume AHU

Another facet in the choice of an indoor unit, specifically an air handler, is whether the system will be constant volume (CV) or variable air volume (VAV). In a CV system, the AHU fan operates at a constant speed and the external static in the system is constant. Therefore, the volume of air moving in the system is constant. Most units less than 30 tons use a variable pitch pulley on the air handler so the airflow can be adjusted during commissioning to meet the job requirements. After the unit is set up, the unit runs at a constant fan speed. A typical commercial condensing unit/packaged air handler combination, as supplied by the manufacturer, is designed for a CV application.

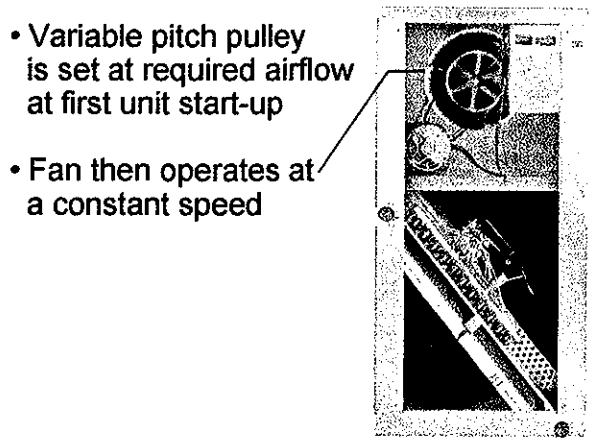


Figure 39

Constant Volume Unit

VAV Application

As the name implies, the volume of air moving through the VAV system is variable. The air handler fan must be capable of changing its airflow to respond to load changes in the space. This may be accomplished in a number of ways. The speed of the fan in the AHU may be variable, perhaps controlled by a variable frequency drive (VFD). The fan may be equipped with inlet guide vanes that mechanically change the inlet flow conditions to the fan, thereby varying the airflow. The air volume may also be controlled at the end of the ductwork, at the terminal devices. VAV terminals effectively throttle the airflow into the space, thereby varying the airflow in the system.

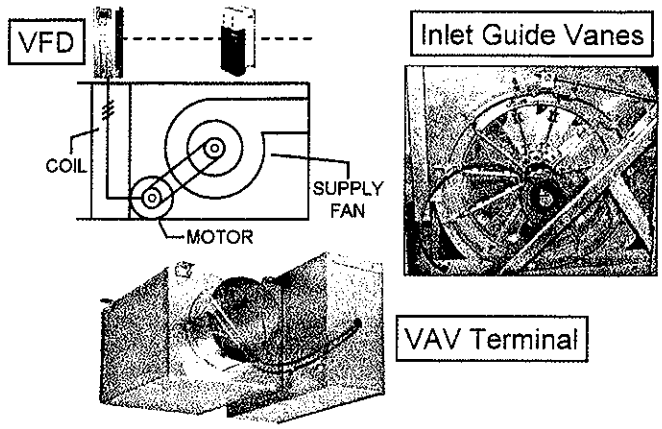


Figure 40

Variable Volume Units

VFDs

Split System VAV Indoor Requirements

When selecting equipment for a VAV application, a number of issues must be addressed. It must be understood that as the air volume varies, so will the load on the system. Therefore, the indoor and outdoor units must equipped to vary the capacity of the system. In the case of the indoor unit, multiple coil sections or circuits typically accomplish this. At the very least, the indoor coil will require two circuits, preferably more. The system must be equipped with capacity control solenoid valves that may be used to stage the number of active coil circuits based on the load.

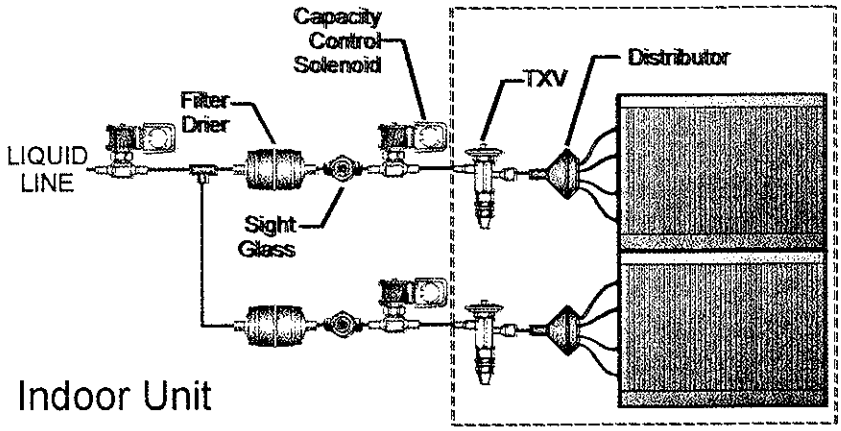


Figure 41

VAV System Requirements

VAV Outdoor Unit

Special consideration must also be given to the outdoor unit. Multiple stages of capacity are required, typically four or more. Commercial condensing units with four or more stages of capacity typically have capacities of 20 tons or more. Therefore, most VAV systems will be at least 20 tons in size. A condensing unit designed for VAV duty will differ from its CV counterpart in a number of ways:

Split system VAV requirements:

- VAV condensing unit will have additional stages of capacity.
- Stages of capacity will be electrically controlled.
- Suction line accumulators will protect compressors.
- Condensing unit will interface with a VAV controller.

VAV Control

VAV split systems must be equipped with a control device or controller. This VAV controller must be capable of starting and stopping the compressors, staging the steps of capacity of both the indoor and outdoor unit, and controlling the fan. A typical VAV controller is a discharge air controller. The controller utilizes a sensor in the ductwork, downstream of the AHU. Based on the sensed supply air temperature and the offset from set point, the controller will vary the stages of capacity to maintain a reasonably constant discharge air temperature. The VAV controller may be as simple as a self-contained device or it may be part of building automation system.

- **Typically controls discharge air temperature**
- **Controls:**
  - **Compressors**
  - **Stages of capacity**
  - **Fans**



Figure 42

VAV Controller

Indoor Coil Loading — Tons per Circuit

In a properly operating air conditioning system, the compressor oil is continuously circulating throughout the system. The oil is returning to the compressor at the same rate at which it leaves, thereby maintaining an adequate amount of oil in the compressor for lubrication. Compressor oil is fully miscible (mixes) with liquid refrigerant and readily moves with the liquid refrigerant. However, where the refrigerant is in a vapor state, for example in the evaporator, the refrigerant velocity must be high enough for the compressor oil droplets to be entrained with the refrigerant vapor. As long as the velocity of the refrigerant vapor remains high enough, the compressor oil droplets are carried by the refrigerant vapor and proper compressor oil management is achieved.

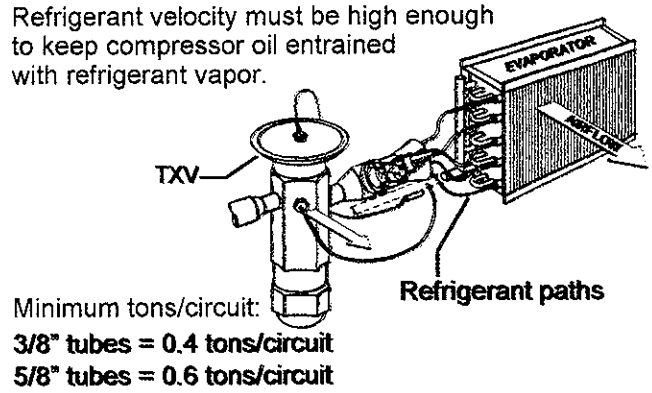


Figure 43

Tons per Circuit

The velocity of the refrigerant vapor in the evaporator is quantified by the term: tons per circuit, or stated another way, tons per refrigerant pathway. The circuits or pathways in an evaporator are the tubes that carry the refrigerant through the fins of the coil. The minimum tons per circuit (or path) for 3/8-in. tubing is 0.4 tons per circuit. In other words, if the capacity of the system using 3/8-in. tubes is equal to or greater than 0.4 tons per circuit (path), the velocity of the vapor will be high enough to insure that the compressor oil remains entrained with the refrigerant vapor. The minimum tons per circuit value for larger tubes is higher, for example, for 5/8-in.

tubes, the minimum tons per circuit is 0.6 tons per circuit. The refrigerant velocity will be lowest when the compressor is unloaded. Whenever you wish to add an unloader to a system, you must consider the refrigerant velocity at the minimum capacity step, when the compressor is fully unloaded.

Tons per Circuit Example

Let's consider a system using a 38ARS012 and a 40RM012. The 38ARS012 is equipped with a single pressure operated unloader as standard equipment. Therefore, the standard 38ARS012 when unloaded has a capacity of approximately 7 tons. The 40RM series uses 3/8-in. tubes and the size 012 has 18 refrigerant circuits (paths) in total, 9 circuits per split.

To determine the tons per circuit when the 38ARS012 is unloaded, simply divide the capacity of the *unloaded* condensing unit by the number of circuits. For the standard 38ARS012 with a 40RM012 system, the capacity of 7 tons is divided by 18 circuits and equals 0.4 tons per circuit.

40RM Model	# of coil splits	# of circuits/splits	# of circuits total
007	1	12	12
008	1	15	15
012	2	9	18
014	2	9	18
016	2	12	24
024	2	13	26
028	2	15	30
034	2	18	36

38ARS012 Standard – Unloaded capacity, 7 tons
ACCEPTABLE 7 tons/18 circuits = 0.4 tons/circuit

38ARS012 with additional unloader – Unloaded capacity, 3.3 tons
TOO LOW! 3.3 tons / 18 circuits = 0.2 tons/circuit

Add capacity control solenoid valve to 40RM012
ACCEPTABLE Now 3.3 tons / 9 circuits = 0.4 tons/circuit

Figure 44

Tons per Circuit Example

This meets the requirement for minimum tons per circuit. The 38ARS012 uses a six-cylinder reciprocating compressor so it is possible to add an additional unloader in the field. If an additional unloader is added to the 38ARS012, the condensing unit could then unload to approximately 3.3 tons. With a capacity of 3.3 tons divided by 18 circuits, the result equals 0.2 tons per circuit. That is much too low for adequate oil return. However, the 40RM012 coil uses a coil that is split into two sections. Therefore, the 40RM012 could be equipped with a capacity control solenoid valve to limit the flow of refrigerant to only one half of the coil when the condensing unit is unloaded to 3.3 tons. Then the equation becomes 3.3 tons divided by 9 circuits equals 0.4 tons per circuit. Therefore, if the application requires unloading to 33 percent with a 38ARS012 with 40RM012 combination, the 40RM must be equipped with a capacity control solenoid valve to effectively reduce the size of the coil when the compressor is unloaded to 33 percent.

In summary, when you are considering additional unloading for an application you must address two areas of concern:

- First, is it possible to add an additional unloader to the condensing unit, and
- Is the ton per circuit value high enough for adequate oil return when the compressor is fully unloaded?

It will only be possible to add additional unloading if you can satisfy both areas of concern.

Cased Evaporator Coils

Another type of indoor unit is the cased evaporator coil. These products are used where the air movement is accomplished by another component in the system, perhaps a furnace. Two types are shown here: an "A" coil design, which is used when two furnaces are twinned, and a simple cased evaporator coil that are installed in the ductwork. These coils are available in a variety of capacities; the most common are 7½ and 10 ton.

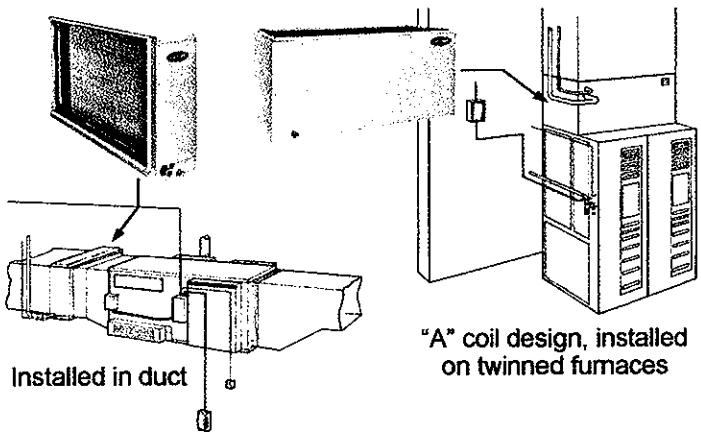


Figure 45

Cased Evaporator Coils

Residential Coils

Residential evaporator coils are similar to the cased evaporator coils described above, yet in smaller tonnage ranges. The coils are traditionally installed on the discharge side of a furnace. The coils are available in a number of configurations, "A," "N," slab, and in cased or uncased designs. The "A," "N," and slab refer to the shape the evaporator coil resembles.

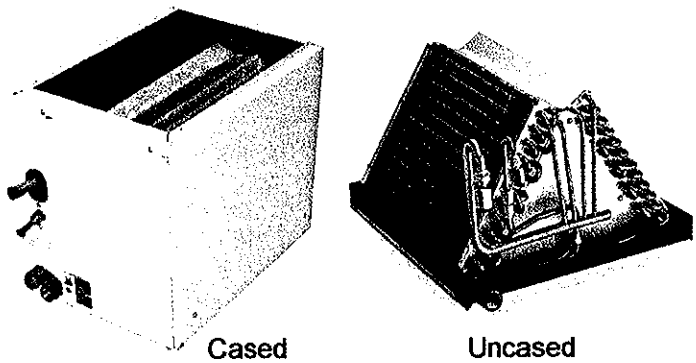


Figure 46

Residential Evaporator Coils

Remote Chiller Barrel

As mentioned previously, another type of indoor split system unit is a chiller (cooler) barrel or evaporator. This device is used in applications where an air-cooled system is desired yet it is necessary to locate the cooler barrel indoors. One reason for this choice is to provide freeze protection for the chilled water loop without the disadvantages of using glycol in the loop. Applications of this type require a condensing unit with multiple stages of capacity. A water temperature controller must control the compressor and stages of capacity electrically.

An alternative to this, once again, would be to use a factory-designed, air-cooled chiller and relocate the cooler barrel in the field if the manufacturer allows that configuration.

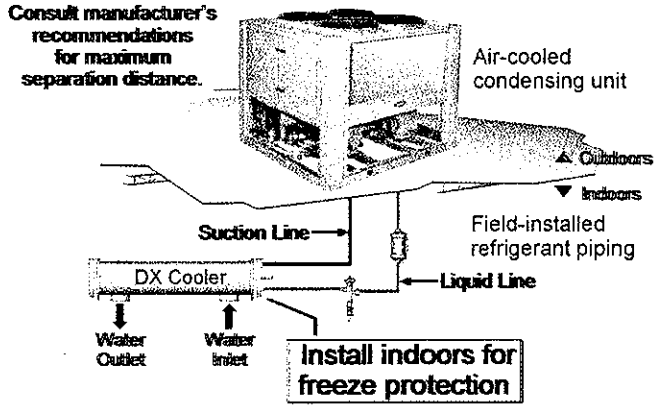


Figure 47

Remote Cooler Barrel

Accessories

Economizer

An important consideration in any split system is the introduction of outdoor air for ventilation purposes. One way to accomplish this is by using an economizer that also provides the benefit of "free cooling" when ambient conditions are appropriate. Historically, economizer control types included dry bulb control, enthalpy control, and differential enthalpy control. Today, CO_2 sensing is also a popular control method. The use of a CO_2 sensor-controlled economizer provides an effective method of demand controlled ventilation (DCV) for split systems.

As noted earlier, energy codes like ASHRAE 90.1 may require the use of an economizer and may dictate which type of control is to be used.

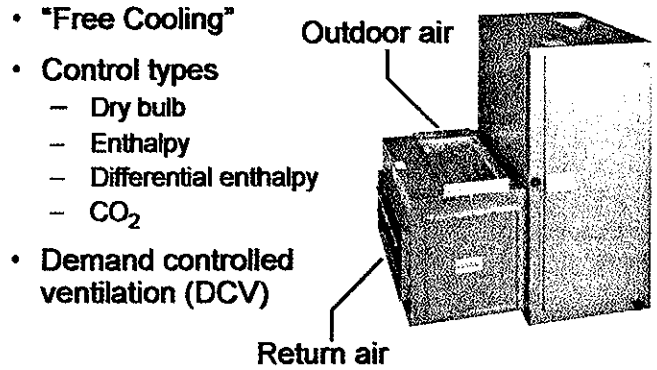


Figure 48

Economizer

Heating Accessories

The indoor units discussed previously focused on cooling only. Of course, many split systems also incorporate heating components. Heating may be accomplished in a variety of ways. Heating accessories for packaged air handlers include: electric, hot water, and steam heating options. These accessories are typically installed on the leaving airside of the packaged air handler. If the system is a heat pump, the coil in the indoor unit will provide heating when the system is operating in the heating mode. This type of heat may be referred to as "mechanical heating." Heat pump indoor units may also be equipped with accessory heating devices when the application requires more heat than the heat pump system can provide and to provide heating during defrost conditions.

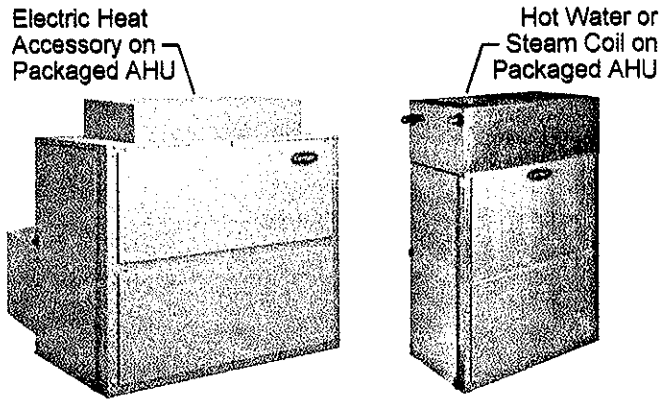


Figure 49

Heating Accessories

Furnaces

Heating may be supplied by a furnace. This furnace may be of the typical design with a cooling coil on the leaving side of the furnace. The furnace may also be a duct type furnace (not shown) placed downstream of the air handler. Furnaces can also be used in pre-selected pairs as shown, called twinned furnaces.

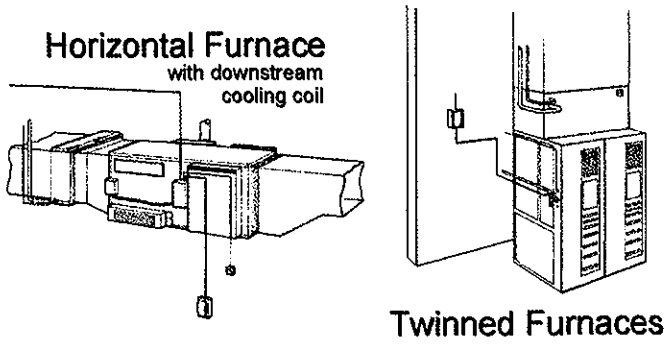


Figure 50

Furnace Applications

Other Accessories

A variety of other accessories may be available to complete the split system installation. These include:

- Plenum, used for free discharge applications.
- Return air grille, also used on free return to prevent larger debris from entering the unit.
- Subbase, used to hold the unit off the floor, typically to allow for installation of the condensate drain.
- Condensate drain kit, to provide the condensate trap.
- Overflow detection switch, to shut down the unit if condensate backs up.
- Suspension kit, provides the necessary brackets and in some cases isolation when the units are to be suspended from the structure above.

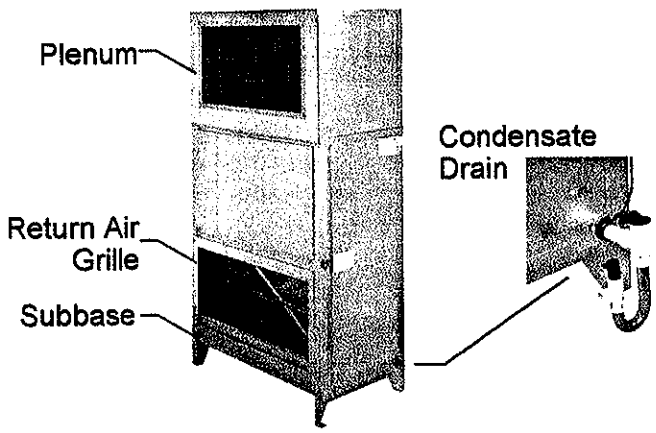


Figure 51

Accessories

Controls

Thermostat

From a control perspective, the typical split system is very simple. For this reason, the control is quite frequently a simple thermostat. The devices to be controlled include: indoor fan, outdoor fan, compressor, and liquid line solenoid (if equipped). On a very simple, small tonnage system, when the thermostat calls for cooling, the indoor fan is started, the liquid line solenoid opens, and the outdoor fan and compressor are started. When the thermostat is satisfied, the liquid line solenoid is closed, the compressor and condenser fan are cycled off, and the indoor fan stops. This type of control is known as solenoid drop control.

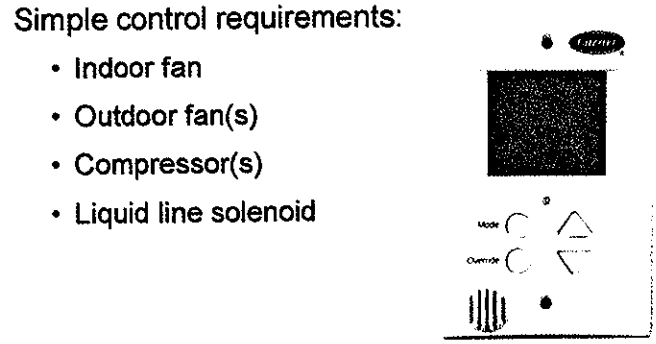


Figure 52

Control Thermostat

Two-Stage Thermostat

Dual circuit systems may be controlled similarly with a two-stage thermostat. In these applications, the first stage cooling (Y1) initiates the first circuit of the condensing unit. If the first circuit cannot satisfy the load demand on the space, the second stage cooling (Y2) function of the thermostat will initiate the second stage of the condensing unit.

- Y1 initiates 1st stage cooling
- Y2 initiates 2nd stage cooling

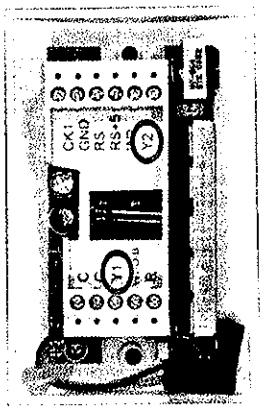


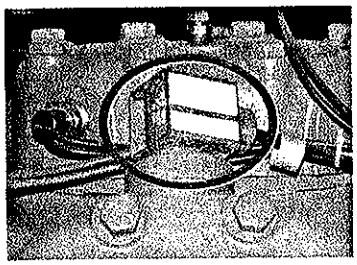
Figure 53

Two-Stage Thermostat

Electric Unloading

A two-stage thermostat may also be used to control a single reciprocating compressor equipped with an electric unloader. Y1 will start the cooling sequence as described previously and unload the compressor. In this way, the compressor will be operating at less than full capacity when the load is light. If the load cannot be satisfied with the compressor operating unloaded, Y2 will initiate and cause the compressor to load, thereby providing full coil capacity.

- Y1 initiates 1st stage cooling AND unloads the compressor
- Y2 loads the compressor



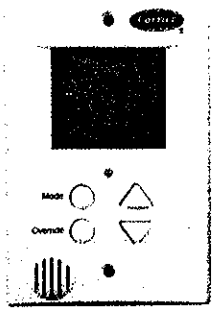


Figure 54

Electric Unloading

Capacity Control Valve

The two-stage thermostat may also be used to control a liquid line solenoid valve in conjunction with compressor unloading. First-stage cooling will start the cooling sequence, but only half of the indoor coil will be open to refrigerant flow. If the load cannot be satisfied with only half of the indoor coil active, Y2 will initiate and cause the second-stage liquid line solenoid valve to open, thereby providing full coil capacity.

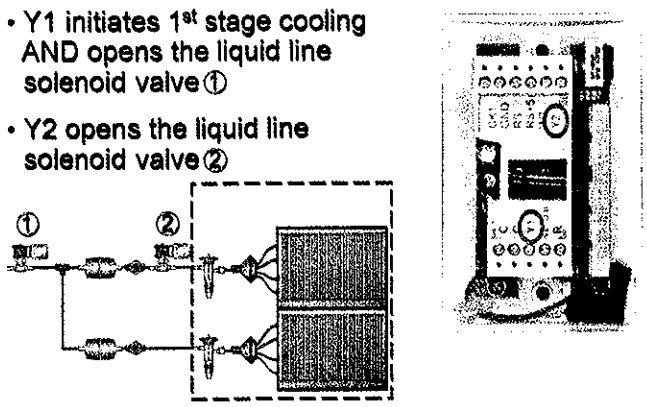


Figure 55

Capacity Control Solenoid Valve Control

DDC Control

Of course, digital controls or a building automation system may be required. Here again, the simplicity of the control needs of the typical split system allows interfacing with a variety of control types. An example may be a VAV controller, which not only controls the indoor unit, but also stages the capacity steps of the condensing unit to meet the load requirements of the system.

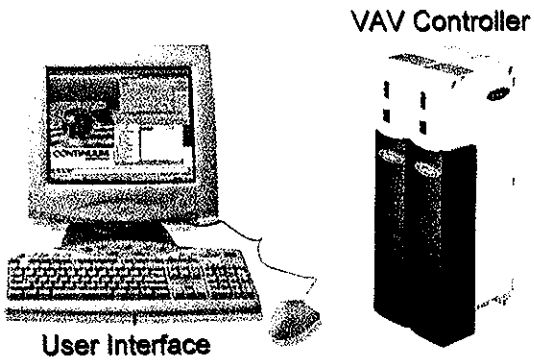


Figure 56

DDC Control System

Safety Controls

On a typical split system, the condensing unit is equipped with several safety controls. These may include:

- High-pressure switch, which protects the system from excessive discharge pressure.
- Low-pressure switch, to limit the minimum suction pressure and protect against loss of charge.
- Discharge gas thermostat, used on some units, which protects the compressor from overheating due to high condensing temperature or low return gas flow.

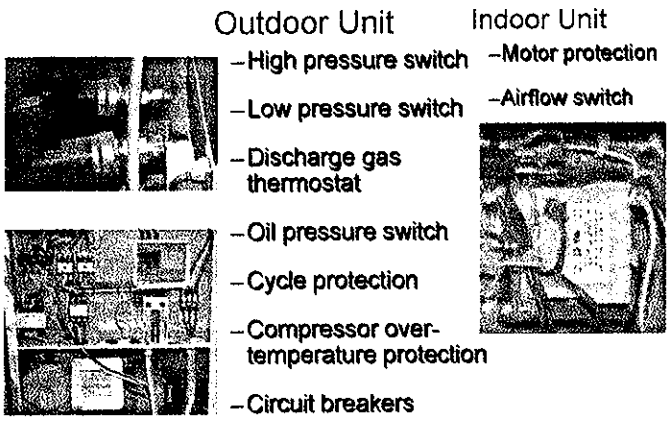


Figure 57

Safety Devices

- Oil pressure switch, on some units, that protects against a lack of lubrication.
- Compressor over-temperature switch, used on some units and internal to the compressor, to protect against compressor overheating.
- Circuit breakers, used on some units, others have internal protection, which protect against electrical motor overload.

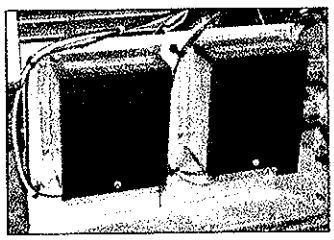
The indoor unit is typically equipped with indoor fan motor protection (internal protector or circuit breaker). Additionally, a common field-supplied safety is a proof-of-airflow switch. The proof-of-airflow switch is interlocked with the outdoor unit controls to prevent compressor operation if there is no airflow, in the event of indoor fan motor or belt failure. The primary control circuit is usually located in the condensing unit control box and the indoor and outdoor circuits need to be interlocked with field-installed control wiring.

Low Ambient Control

Another control issue that must be considered is the outdoor air temperature range at which the split system will be expected to operate. In order to ensure proper operation of the expansion device in the indoor unit, it is necessary to maintain a significant pressure differential across the expansion device. As the outdoor air temperature decreases, the saturated condensing temperature (SCT) of the system also decreases. The minimum outdoor air operating temperature is defined in the condensing unit application data. You will notice that the minimum outdoor temperature with standard outdoor fan (OFM) control is 35° F. If the system will be operated when the outdoor temperature is less than the standard value, it is necessary to apply a low-ambient control device to the condensing unit. The low-ambient control device is a speed control device that will vary the speed of the OFM motor(s) to maintain the SCT at a reasonable level, approximately 100° F. On split systems, DO NOT use a low-ambient control device that controls by cycling the fan motor off and on; it must be a variable speed motor control device. Notice in the table that these condensing units with low ambient control may be operated down to -20° F.

MINIMUM OUTDOOR-AIR OPERATING TEMPERATURE

UNIT 38AR	COND TEMP (F)	MINIMUM OUTDOOR TEMP (F)	
		Std.	With Motormaster® Control
Z007	80	35	-20
Z008	80	35	
Z012	80	35	
S012	80	35	
	80	35	
D012	80	35	



Motormaster® Control

- **Proper operation of expansion device at low outdoor temps**
- **Minimum outdoor air operating temperature**
- **Maintain minimum SCT**
- **Low ambient controller – OFM motor controller**

Figure 58

Low-Ambient Control

Fan-Cycling Pressure Switch

You may also encounter some condensing units that employ an intermediate season SCT control device, a fan-cycling pressure switch (FCPS). The FCPS is a pressure switch that senses pressure in the condenser coil. On condensing units that have multiple OFM motors, a FCPS may be used to cycle on or off one or more of the OFM motors. For example, on a condensing unit that has two OFM motors, a FCPS may control the #2 OFM motor. Once the FCPS has turned the #2 OFM motor off, if SCT temperature continues to fall, the low ambient control device must vary the speed of the #1 OFM motor to maintain a stable SCT. The important fact to remember is that the last operating OFM motor must be controlled by a variable speed device. Do not cycle the last operating motor.

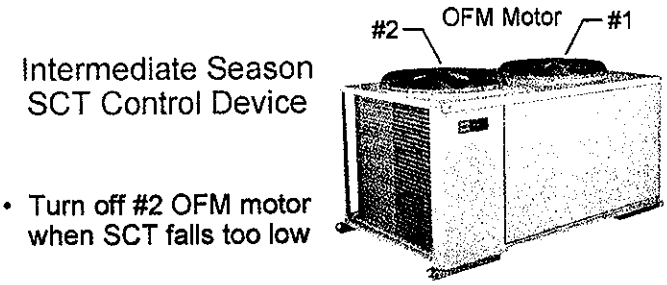


Figure 59

Fan-Cycling Pressure Switch

Wind Baffles

An additional element of the low ambient control system is the wind baffle. If the condenser coil is exposed to sustained winds, controlling the number of operating fans and/or, fan speed, may not maintain SCT at a reasonable level. The force of the wind alone may provide more air movement across the coil than is desired. In these applications, it is necessary to install wind baffles, at least on the windward side of the unit. Condensing units that employ horizontally-mounted coils do not require wind baffles.

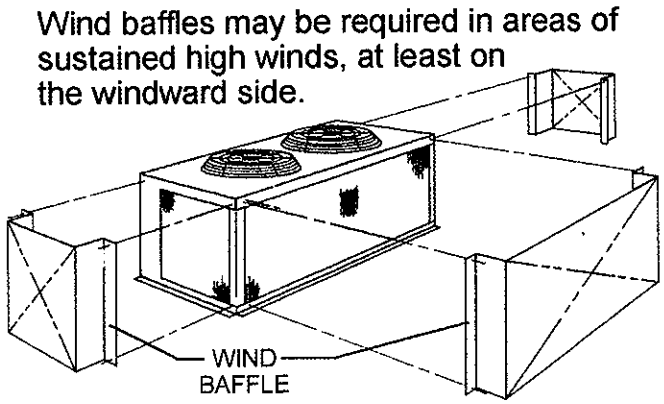


Figure 60

Wind Baffles

Installation

Designers should understand several issues related to installation in order to do a better job in system design. Understanding the requirements for electrical service, location, refrigerant piping, and control interfacing will result in more satisfactory split system designs.

Electrical

A split system has four electrical service requirements that need to be meet. First, the size of the wiring that needs to be run to the indoor and outdoor sections must be determined. Then the size of the fuse or circuit breaker that will protect each of the two sections from electrical overload needs to be determined. Third, the disconnect requirements for both the indoor and outdoor sections need to be specified. Finally, the requirements that interlock the two sections must be determined.

Power Supply

Another important part of the system designer's task is to define the power supply needed for the split system. Typically, this will involve at least two power circuits, one for the indoor unit, and one for the outdoor unit. If the indoor unit is equipped with electric heat and requires only one power supply, this is called a "single point" connection. If the electric heat is a duct heater or an add-on to the air handler, it may require separate power supplies for the air handler and the electric heater. The key terms to understand in defining the power supply requirements are Minimum Circuit Ampacity (MCA) and Maximum Over-current Protection (MOCP).

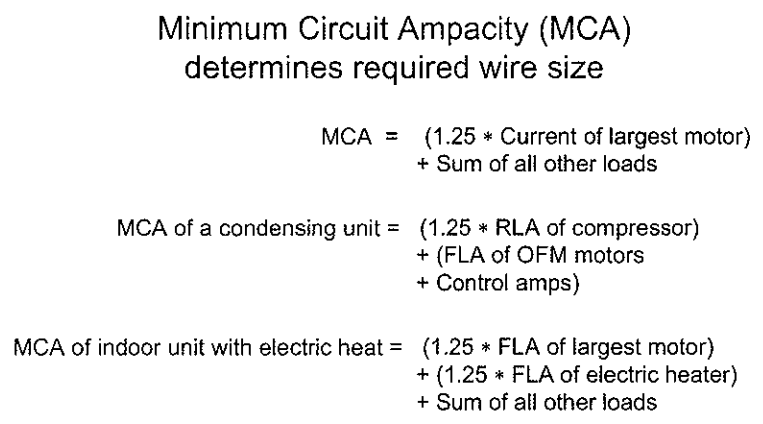


Figure 61

Power Supply MCA

MCA

The value of the MCA determines the wire size required for the circuit. MCA is calculated:

MCA = (1.25 * current of the largest motor) + sum of all other loads

The amperage drawn by a compressor depends on the operating point; the industry has agreed to determine this current draw at a selected set of operating conditions indicative of normal maximum current draw. This value is referred to as run load amps (RLA). Other motor amperage is listed based on the motor operating at fully-loaded conditions without going into the service factor, referred to as the full load amps (FLA).

Therefore, the MCA of a condensing unit would be:

MCA = (1.25 * RLA of the compressor) + (FLA of the OFM motors) + Control Amps

The MCA of an indoor unit is calculated similarly unless it is equipped with electric heat. If equipped with electric heat, the MCA is:

MCA = (1.25 * FLA of the largest motor) + (1.25[†] * FLA of the electric heater) + sum of all other loads

[†]1.00 if heater is 50 kW or larger

MOCP

The MOCP value defines the maximum overcurrent protective device. The key word is "maximum." If the MOCP for a condensing unit is 60 amps, this means the largest overprotection device (fuse or circuit breaker) allowed by UL or the NEC (National Electric Code) is 60 amps. If a 50-amp device is used, that is not a problem from the perspective of UL or NEC. The risk in using a smaller fuse or circuit breaker is that the unit could trip the protective device on start-up or in times of high current draw, for example, in high ambient conditions. The designer must consider the benefit of a smaller protective device (less cost) compared to the potential for nuisance tripping of the protective device. To calculate MOCP:

MOCP = (2.25 * current of the largest motor) + sum of all the other loads

If the value derived does not equal a standard current rating of an over current protection device, the MOCP is to be the next lower standard rating, but not lower than the MCA.

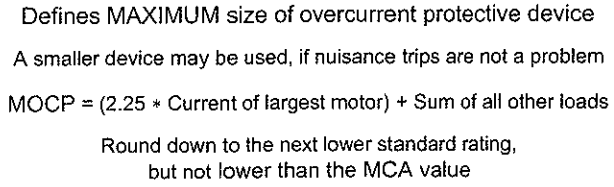


Figure 62

MOCP

ROCP

There is an alternate method of calculating overcurrent protection known as recommended overcurrent protection (ROCP). To calculate ROCP:

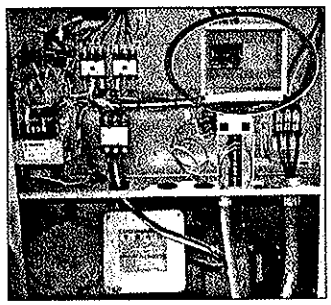
ROCP = (1.5 * current of the largest motor) + sum of all the other loads

UL1995 states that a value smaller than the MOCP, i.e., ROCP, may be published, if the unit is tested at the lower value and does not trip the over current protection device. The key point is that the unit must be tested at the lower value to confirm that it will function without nuisance trips of the overcurrent device.

Protective Device

The type of protective devices used in the HVAC industry may be fuses or circuit breakers, depending on the application and locale. If circuit breakers are used, they must be a type specifically designed for the HVAC industry, known as HACR breakers (heating and air conditioning rated). Generally speaking, HACR breakers will be used whenever acceptable by code and when available in the size required. Fuses will be used if required by code or if the MOCP value is greater than the largest HACR breaker available. Be sure to check the manufacturer's installation information since some units will be rated for use with fuses only.

- Fuse or Circuit Breaker
- HACR Breaker



Usage defined by code and availability

Figure 63

Protective Devices

Disconnects

For safety reasons, electrical codes such as the NEC require that a "disconnecting device" be located within line of sight of the unit. This disconnect may be installed in the field by an electrician or it may in some cases be provided as a factory-installed option. Disconnects may be fused or non-fused. If a non-fused disconnect is used to meet the "disconnecting device" requirement of the NEC, the circuit must still be protected by fuses or HACR breakers. These protective devices would then be located between the non-fused disconnect and the electrical power service to the building.

- Located within "line of sight"
- Field-installed or factory-supplied
- Fused or non-fused

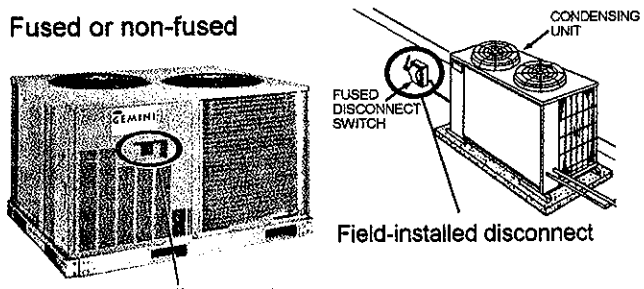


Figure 64

Disconnects

Installation Instructions

For specific information regarding installation, it is imperative to consult the manufacturer's installation instructions. Different units and different manufacturers for the same tonnage may have very different requirements for clearances, electrical service and refrigerant piping requirements. However, there are a number of general considerations that apply to most installations.

Installation cautions:

The indoor and outdoor unit should be located as close to one another as is practical. This will keep the length of the refrigerant piping to a minimum. Long refrigerant lines increase the potential for performance problems and increase job installation cost. Be careful to check for the maximum separation between the indoor and outdoor sections, both for run (total line length) and lift (the height the liquid refrigerant must travel up to the evaporator). Different compressors can have very different requirements for run and lift. Use the manufacturer's piping recommendations to size refrigerant piping whenever they are provided. Piping recommendations are often the result of considerable testing for oil return and refrigerant charge limitations.

Unit location must also take into consideration condenser coil airflow (outdoor unit) and maintenance accessibility. Always refer to the manufacturer's recommended clearances when deciding where and how to locate the equipment. For the outdoor unit, be sure to consider the potential for vegetation growth blocking the coil in the future. Service pads are required for units mounted on the ground. It is sometimes desirable to hide the condensing unit for sound or aesthetic reasons. When this is done be sure that the barrier allows for adequate airflow and that it does not cause recirculation of hot condenser air.

Sound

The sound produced by air-conditioning equipment is becoming increasingly important as designers, owners, and occupants seek quieter and quieter environments. It must be recognized that split system components will produce sound, and steps must be taken to insure that the sound produced is not objectionable. The sound level produced by the equipment is typically identified in the unit product data. Manufacturers may also provide sound reduction accessories for the equipment to reduce the sound produced. It is equally important that designers consider the impact of sound when locating equipment. For example, it would not be wise to locate a large condensing unit on the roof of an office building if the space directly below is the company president's office. It would not be wise to locate a packaged air handler in a closet with louvered return air doors at the back of a classroom. When in doubt regarding sound issues, utilize the services of a skilled acoustical consulting engineer.

SOUND LEVELS, dB

UNIT 38		SOUND RATING (60 Hz) dB (A)	OCTAVE BANDS							
			63	125	250	500	1000	2000	4000	8000
ARZ	007	80.0	71.5	76.8	75.3	74.0	73.9	74.1	73.5	64.0
	008	84.0	84.1	84.5	80.4	78.6	77.2	76.9	78.8	72.5
	012	85.0	88.7	85.4	81.7	80.9	82.1	76.2	73.6	67.1
ARS	012	83.0	78.0	82.6	84.9	78.6	79.5	72.1	69.9	65.8
AKS	014	86.2	—	93.0	86.0	83.0	80.0	78.0	73.0	71.0
	016	86.2	—	93.0	86.0	83.0	80.0	78.0	73.0	71.0
	024	90.0	83.5	81.5	88.5	86.5	85.5	82.5	76.5	61.5
ARD	012	85.0	87.3	84.1	85.9	79.2	77.6	75.3	75.0	68.2
	014	86.9	—	90.9	86.1	83.1	84.0	73.5	71.7	66.7
	016	87.5	—	90.9	86.1	83.4	84.5	76.6	73.2	63.5
	024	88.0	—	90.9	86.1	83.8	84.5	79.2	74.3	65.5

- Understand sound power (both indoor and outdoor)
- Utilize sound reduction accessories when available
- Consider the effect of sound when recommending unit location
- Isolate and support refrigerant tubing

Figure 65

Typical manufacturer's sound ratings.

Some units have options available that can be used to assist in controlling the sound levels. These should be used whenever the job is acoustically sensitive.

One often-overlooked sound issue with split systems is the refrigerant piping. It is good design practice to allow for some movement in the piping, and use piping supports to isolate the units and prevent the piping from transmitting vibration and noise.

Elevation

The indoor and outdoor sections may be located at the same elevation or they may be located on different elevations. If the indoor unit is located above the outdoor unit, the outdoor unit must "lift" the liquid refrigerant up to the indoor unit. In this case, the designer must confirm that the vertical distance between the indoor and outdoor unit does not exceed the liquid lift capability of a condensing unit. If the separation is too great, one or both of the components must be relocated. The lift impacts the pressure drop in the liquid line. Excess pressure drop can result in the liquid flashing to vapor in the line and result in hunting problems with the TXV.

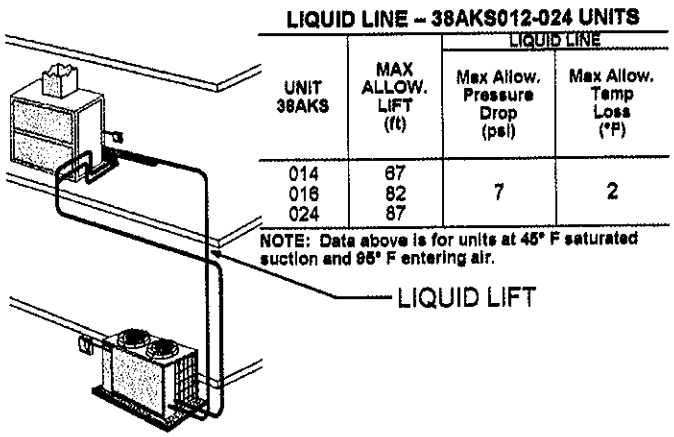


Figure 66

Elevation

Suction Riser

If the outdoor unit is located above the indoor unit, consideration must be given to the vertical section of the suction piping known as the suction riser. In order to ensure proper compressor oil management, the velocity of the refrigerant must be high enough to entrain compressor oil with suction vapor in the suction line. The manufacturer's data may also define limitations on the maximum length of suction risers where applicable. In some cases, it may be necessary to use two refrigerant lines in the suction line to assure adequate velocity for oil return. This arrangement is referred to as a double suction riser.

Double suction risers

Refrigerant Piping

Proper sizing and installation of refrigerant piping is imperative for proper operation and long component life in a split system. Information regarding refrigerant piping practices and methods may be found in TDP-501, Refrigerant Piping.

Caution: DO NOT bury refrigerant piping underground! Buried refrigerant lines can result in refrigerant condensing taking place in the lines and liquid refrigerant slugging back to the compressor.

Remember to select the proper size refrigerant lines (tubing) for split system applications. There are a number of sources of this information; always use the manufacturer's data when it is provided. If this data is not available from the manufacturer, then alternate sources such as the Carrier Refrigerant Piping Software and System Design Manual may be used. Refrigerant lines should always be sized for no more than a 2° F line loss.

UNIT 38AR	LINEAR LENGTH OF PIPING — FT							
	0-25		25-50		50-75		75-100	
	Line Size (in. OD)							
	L	S	L	S	L	S	L	S
Z007	3/8	1 1/8	3/8	1 1/8	3/8	1 1/8	3/8	1 1/8
Z008	3/8	1 1/8	1/2	1 1/8	1/2	1 1/8	1/2	1 1/8
Z012	1/2	1 3/8	1/2	1 3/8	1/2	1 3/8	1/2	1 3/8
S012	1/2	1 3/8	1/2	1 3/8	1/2	1 3/8	1/2	1 3/8
D012	(2) 3/8	(2) 1 1/8	(2) 3/8	(2) 1 1/8	(2) 3/8	(2) 1 1/8	(2) 3/8	(2) 1 1/8

LEGEND
L — Liquid Line S — Suction Line

NOTES:
1. Pipe sizes are based on a 2° F loss for liquid and suction lines.
2. Pipe sizes are based on the maximum linear length, shown for each column, plus a 50% allowance for fittings.
3. Charge units with R-22 in accordance with unit installation instructions.

Refer to manufacturer's recommendations

***DO NOT** bury refrigerant piping underground!*

Figure 67

Refrigerant Piping Sizes (6-10 Ton, R-22)

Maximum Length of Refrigerant Piping

A common question is: What is the maximum allowable length of the refrigerant piping system? Unfortunately, the best answer is, "It depends." For example, the maximum allowable linear length of piping for a commercial heat pump application is 100 ft. Lift and suction riser limitations determine what portion of the 100 ft can be vertical. The maximum allowable length will vary based on the manufacturer and the size of the unit. Typically, larger condensing units will allow larger maximum line lengths. Refer to the manufacturer's guidelines when a long line application is being considered.

Note:

Long Line Applications

A long line application is defined as one that has a total linear length of 75 ft or greater. Long line applications require special considerations to reduce the risk of equipment failures. For example, all long line applications must have a liquid line solenoid valve(s) and a suction line accumulator(s) to provide an additional degree of protection for the compressor.

On long line applications:

System Selection

Selection of split system units can be more complicated than the selection of packaged rooftop equipment for two reasons. First, it is possible to match a condensing unit to several different combinations of evaporators, and second, a considerable distance may separate the evaporator and condensing unit. Both of these can influence the selection.

- System Balance Point:
  - Coil performance plotted
  - Condensing unit performance plotted
  - Suction line losses:
    Suction Line: 2.0° F loss
    Accumulator: 0.5° F loss
- Balanced SST = 45.6
- Balanced SCT = 126.6
- Balanced Total Capacity:

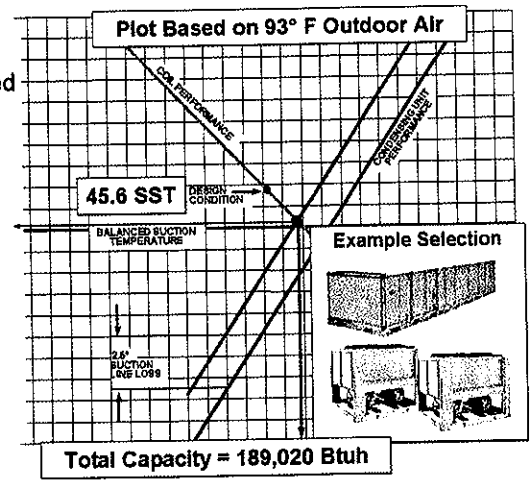


Figure 68

Balance Diagram

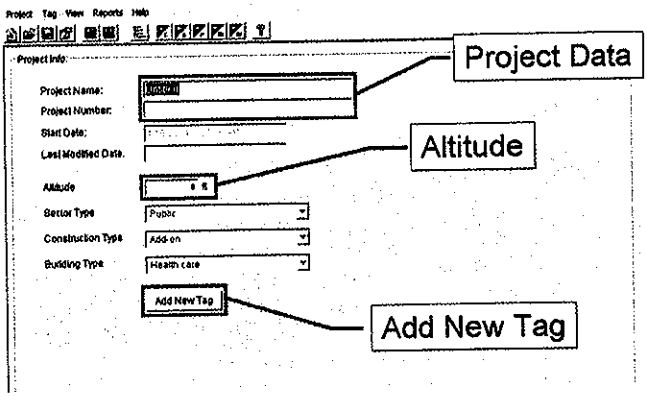


Figure 69

Sample Input of Carrier's E20 Input Screen

In the past, it was necessary to make a selection by graphically plotting the performance of a condensing unit against the performance of an evaporator using a balance diagram. Since electronic selection programs have become available, computers can easily perform the balance.

Manufacturers now provide split system selection software tools that evaluate both the indoor and outdoor unit as a system. As an example, the Carrier selection program is used in the next few sections to explain the procedure and the required inputs. The programs result in outputs that provide the designer with all the information normally required to complete the schedule for the design drawings.

The first data input screen is primarily for project management, with the exception of the altitude input. Inputting an altitude above sea level, will automatically be compensated for air density at the altitude specified. A "Tag" is just a name for the unit, for example "SSU-1 Office area."

Input

Next, select the type of equipment, DX Cooling, DX Heat Pump, or Chilled Water. If one of the DX choices is selected, the program will select a system using an indoor and outdoor unit. If "Chilled Water" is selected, the program will select a chilled water indoor unit only. Electrical service is the next important consideration. Select the electric service for the indoor and outdoor units. The values do not have to be the same. The only other data that is required to make a selection is indoor unit airflow.

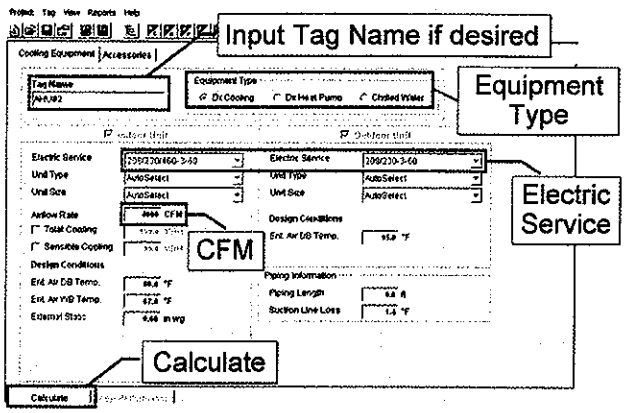


Figure 70

Input Screen

If you input 4000 cfm and press the "Calculate" button, the software will return a list of all of the available combinations that will safely operate at 4000 cfm at the default ARI rating conditions

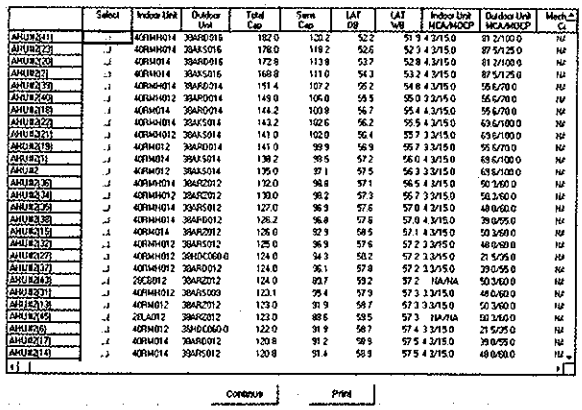


Figure 71

E-Cat Initial Results

Notice that the list is rather lengthy. Therefore, you would typically make additional selections in each of the drop down boxes to narrow the number of choices. For example, you may know that the indoor "Unit Type" desired is a packaged AHU. You may desire a single circuit, scroll compressor, condensing unit. Input the design conditions for your application. Press the calculate button now and see all of the combinations that meet your criteria and the performance of each, or take one more step to further reduce the choices.

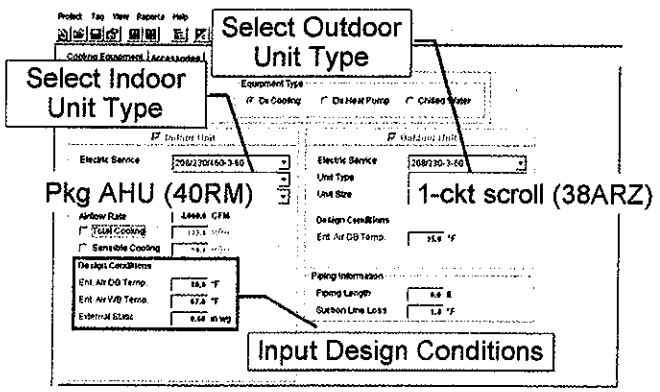


Figure 72

Narrow the Choices

Specify Total or Sensible Cooling

Input the amount of Total Cooling or Sensible Cooling required, and then press the calculate button. The system will now return only those combinations that fit the physical criteria and meet your performance requirements. Note the fields for Piping. Some programs will calculate the impacts of pipe size and lengths on the performance.

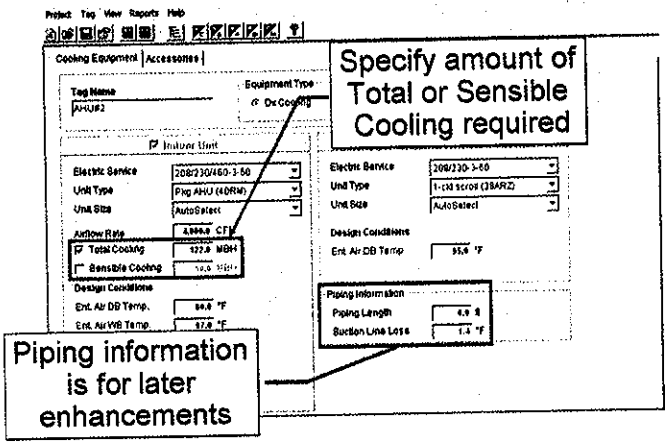


Figure 73

Specify Cooling Needs

Input Accessories

If the indoor unit will be equipped with accessories that will affect the unit's performance, you should click on the accessory tab. Select the appropriate accessories and then recalculate.

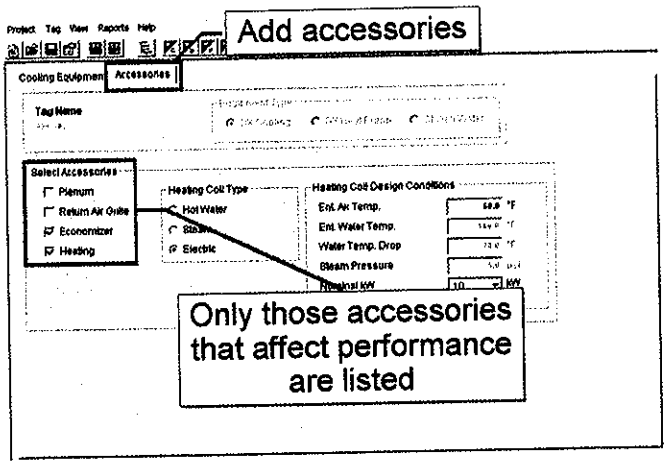


Figure 74

Add Accessories

Performance with Accessories

The performance will now take into account the effect of the accessories.

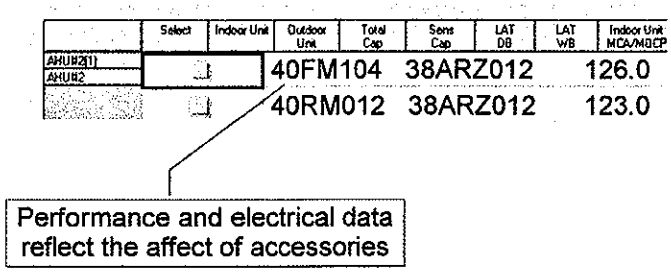


Figure 75

Performance with Accessories

Select the System

To see the complete data on the system, select the system desired, and then click "Print" at the bottom of the screen. The result will be a screen print of the Performance Data Report.

Performance Data Report

The report generated will provide all of the pertinent data required for a schedule. Typical data include performance, temperatures, electrical data, fan motor requirements, sound power, etc.

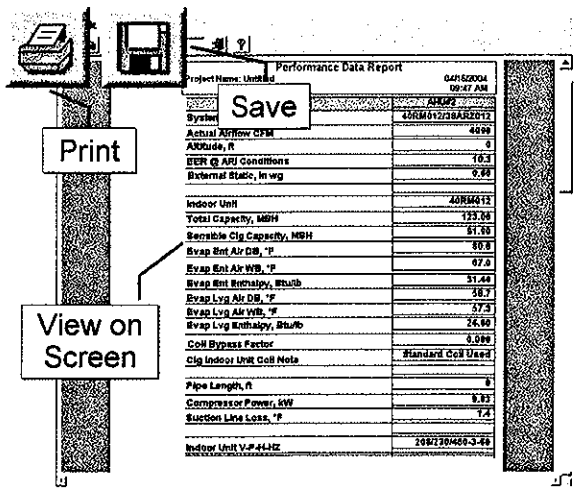


Figure 76

Performance Report

Summary

The objective of this module has been to familiarize the participants with split system equipment, the nature of the business, and the technical aspects of selection and application of split systems. Specific attention has been given to the flexibility of the system, issues which are specific to split systems and the tools available to the designer.

Split systems

Work Session 1

Multiple-choice questions may have more than one correct answer; identify all correct selections.

1. A typical commercial split system includes _____.

   a) an indoor unit only
   b) a compressor, an indoor fan, an evaporator and condenser as one or more sections
   c) an indoor and an outdoor unit
   d) a compressor, an indoor fan, an evaporator and condenser as a package

2. True or False? All commercial split systems are at least dual circuit. _____

3. True or False? Net capacity will always be greater than gross capacity. _____

4. SEER applies to _____

   a. units which are cooling and heat
   b. all units under 65,000 Btuh
   c. single phase units under 65,000 Btuh
   d. all units under 135,000 Btuh

5. The standard ARI rating condition used to calculate EER for commercial splits is _____

   a. 80° F db/57° F wb indoor , 82° F outdoor
   b. 70° F db/57° F wb indoor, 47° F outdoor
   c. 80° F db/67° F wb indoor, 95° F outdoor
   d. 80° F db/67° F wb indoor, 82° F outdoor

6. True or False? Semi-hermetic compressors may be equipped with unloaders for capacity control. _____

7. True or False? Heat pump systems can always match an indoor unit one size above the nominal capacity of the outdoor unit. _____

8. The condensing unit used in a VAV application will differ from a condensing unit used in a CV application in which of the following ways? _____

   a. They are the same only the indoor section is different.
   b. VAV units have a suction line accumulator.
   c. VAV units have additional steps of capacity.
   d. VAV units have a VFD on the compressor.

9. Calculate the MOCP and MCA for a condensing unit that has one compressor rated at 75 amps, one indoor fan rated 21 amps, and two outdoor fans rated at 6 amps each.

_____

10. Find the system EER for a unit with a gross capacity of 137,000 Btuh, two OFM motors at 1500 Watts, and a 5 hp indoor fan motor operating at 4.1 bhp. The compressor draws 6 kW.

_____
_____
_____
_____

11. Refrigerant lines with a length greater than 75 feet require _____.

a. a compressor which unloads to 33 percent capacity
b. a suction line accumulator
c. a receiver
d. a double suction riser and a solenoid valve

12. A condensing unit can be unloaded from 10 tons to 4 tons, the indoor coil has 20 circuits of 3/8 in. tube. Is this an acceptable combination? Why? _____

_____
_____
_____

13. Split systems are a good selection for applications_____.

a. requiring high latent capacity.
b. requiring high sensible capacity.
c. that are not concerned about building aesthetics.
d. such as a shop in a downtown district located on the first floor of a 5-story building

14. The following are appropriate accessories for a condensing unit. _____

a. Hot water coil mounted on the discharge
b. Enthalpy-controlled economizer
c. Condensate trap
d. Unit mounted disconnect switch

15. Residential systems differ from commercial systems in the following ways. _____

a. Because they do not have any options
b. They have fixed metering devices on all models
c. They are units of 5 tons or less
d. They can only be used on residential homes

Notes

Notes

Prerequisites:

This module assumes the participant has an understanding of industry terminology, basic concepts of the air conditioning, and the mechanical refrigeration process. The following TDPs are good reference for this material:

Form No.	Color Book	Instructor Presentation	Title
TDP-102	796-026	797-026	ABCs of Comfort
TDP-103	796-027	797-027	Concepts of Air Conditioning
TDP-105	796-029	797-029	Comfort Design Steps
TDP-401	796-037	797-037	Principles of Mechanical Refrigeration
TDP-404	796-040	797-040	Compressor Types

Learning Objectives:

At the conclusion of this module, participants will be able to:

- Identify applications that utilize the strengths of split systems.
- Demonstrate an understanding of the various components of split systems.
- Explain how codes and rating requirements affect selection of a split system.
- Describe the common types of outdoor units and the differences in each.
- Describe the common types of indoor units and the differences in each.
- Describe the options and application limits when applying CV or VAV type systems.
- Calculate the minimum circuiting requirements.
- Select the appropriate control system for a split system application.
- Identify the key installation issues when applying a split system.
- Describe how to size refrigerant piping for split systems.
- Describe how to select a split system unit and what precautions are needed.

Supplemental Material:

Additional information on subject covered in this module may be found in:

Form No.	Color Book	Instructor Presentation	Title
TDP-501	796-042	797-042	Refrigerant Piping
TDP-403	796-039	797-039	Expansion Devices and Refrigeration Specialties
TDP-701	796-066	797-066	System Features and Selection Criteria
TDP-702	796-067	797-067	Comfort Control Principles

Instructor Information

Each TDP topic is supported with a number of different items to meet the specific needs of the user. Instructor materials consist of a CD-ROM disk that includes a PowerPoint™ presentation with convenient links to all required support materials required for the topic. This always includes: slides, presenter notes, text file including work sessions and work session solutions, quiz and quiz answers. Depending upon the topic, the instructor CD may also include sound, video, spreadsheets, forms, or other material required to present a complete class. Self-study or student material consists of a text including work sessions and work session answers, and may also include forms, worksheets, calculators, etc.



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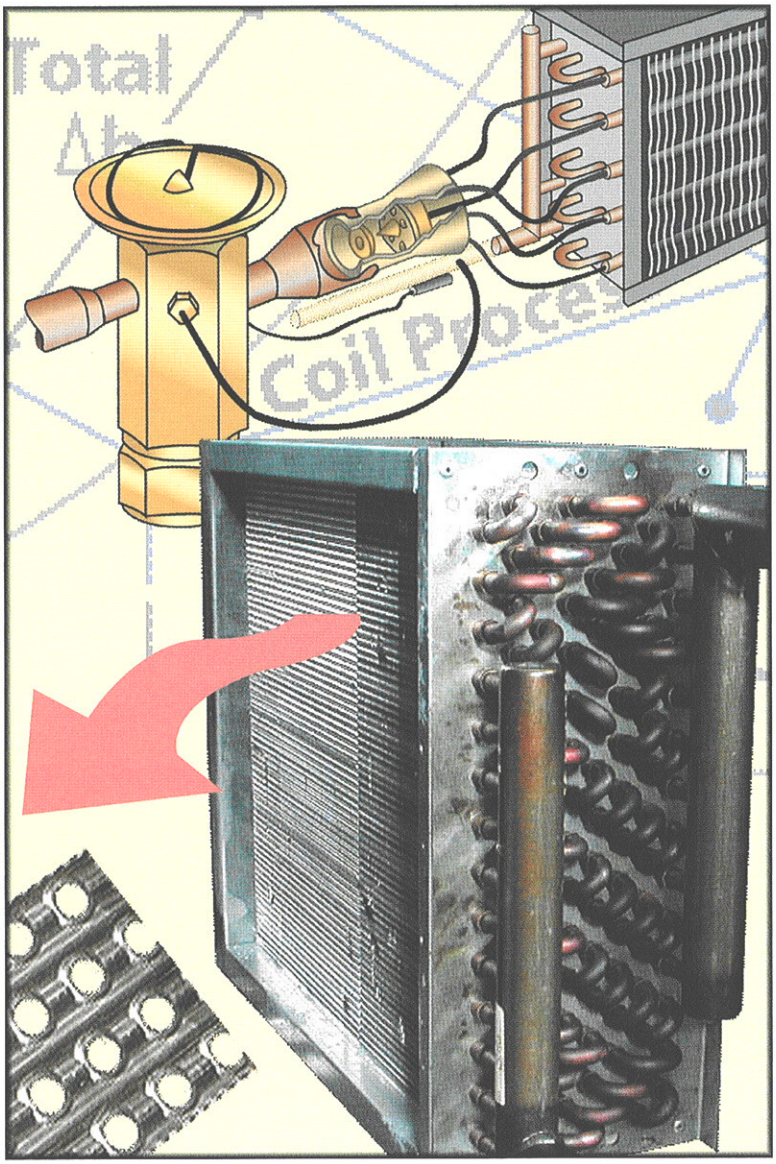


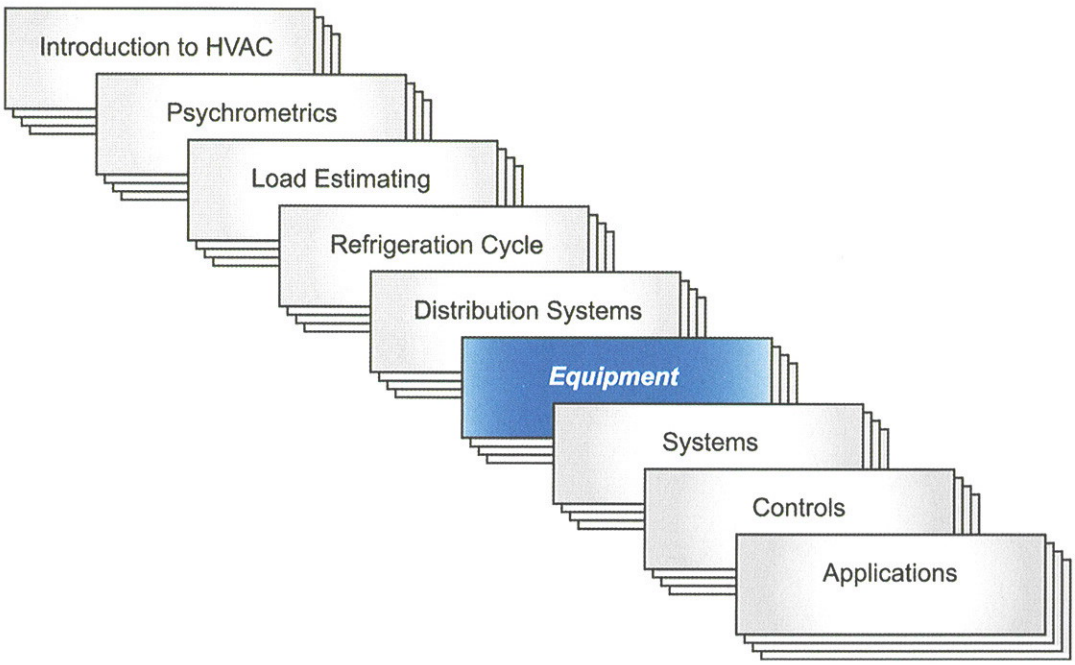
COMMERCIAL HVAC EQUIPMENT

Coils: Direct Expansion, Chilled Water, and Heating

Technical Development Program

Technical Development Programs (TDP) are modules of technical training on HVAC theory, system design, equipment selection and application topics. They are targeted at engineers and designers who wish to develop their knowledge in this field to effectively design, specify, sell or apply HVAC equipment in commercial applications.

Although TDP topics have been developed as stand-alone modules, there are logical groupings of topics. The modules within each group begin at an introductory level and progress to advanced levels. The breadth of this offering allows for customization into a complete HVAC curriculum – from a complete HVAC design course at an introductory-level or to an advanced-level design course. Advanced-level modules assume prerequisite knowledge and do not review basic concepts.



There are many different coil applications used in HVAC design. They range from small residential sizes to large built-up coil banks in custom air-handling units. Regardless of their size, all coils serve the important function of changing the temperature of the air to satisfy comfort or process requirements. There are two main categories of coils, heating or cooling. Heating coils use electricity, hot water, or refrigerant hot gas as a heating medium. Cooling coils use direct expansion (cold refrigerant) or chilled water. In this TDP, a design engineer will learn about the components, features, and applications for direct expansion and chilled water cooling, and hot water, steam, and electric heating coils. With an understanding of these items, the design engineer can proceed with confidence to perform a proper coil selection and prepare a specification.

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The information in this manual is offered as a general guide for the use of industry and consulting engineers in designing systems. Judgment is required for application of this information to specific installations and design applications. Carrier is not responsible for any uses made of this information and assumes no responsibility for the performance or desirability of any resulting system design.

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Introduction

This TDP module reviews the terms, construction features, heat transfer characteristics, performance, and applications of the various types of heating and cooling coils. Heating coils use electricity, hot water, steam, hot gas reheat, or the reverse cycle of a heat pump unit to raise the temperature of the air flowing through the coil. Cooling coils use direct expansion (refrigerant) or chilled water to lower the temperature of the air flowing through the coil.

The term "coil" refers to a fluid-to-air heat exchanger. The fluid used in the coil may be water, steam, antifreeze solution, or refrigerant. The exception is electric heat coils, which do not use fluids. Coils are used for heating and cooling in air-handling units, packaged air conditioning units, and VAV terminals and can also be mounted in a duct or on a furnace. Figure 1 shows an example of a water coil.

The primary emphasis in this TDP will be placed on coils used in air-handling units operating in comfort air-conditioning applications because the design engineer for those products has the widest variety of coil types to choose from. In packaged equipment, the coil is already included as part of the unit design; however, some coil options may be available. The technical principles are the same for coils in packaged equipment and air-handling units.

"Cooling coil" is a generic term for coils that use chilled fluid or refrigerant as the cooling medium. The term "evaporator coil" has been used in the past for cooling coils that use refrigerant since refrigerant evaporates at a low temperature and pressure to extract heat from the airstream. "Direct expansion" or DX coil is the term that will be used in this TDP for coils that use refrigerant for cooling. If the heating or cooling coil application requires a fluid other than fresh water for purposes of freeze protection, that fluid will be referred to as antifreeze.

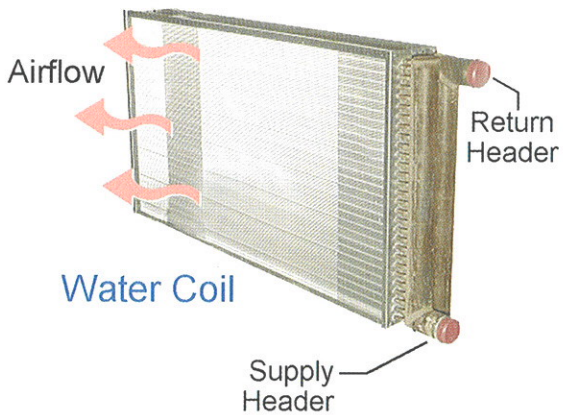


Figure 1

What is a coil?

Outdoor refrigerant condenser coils that are part of packaged equipment designs, such as condensing units and rooftop units, are not covered in this module because their design is normally determined by the manufacturer. For information on condenser coils in packaged equipment refer to TDP-634, Split Systems.

Before starting this module, the reader should have knowledge of the following topics: cooling load estimation, psychrometric theory, refrigeration principles, and air-handling equipment. The Carrier Technical Development Program for each of these topics is listed in the Prerequisite List on the inside back cover of this book.

Typical Coil Applications in HVAC Systems

In comfort cooling applications, there are five general application categories that use coils: residential systems, commercial packaged systems, duct mounted systems, air terminals, and air-handling units. We will discuss each and examine the coil designs that each of them use. Later in the TDP we will examine the construction and materials used in each coil type discussed below.

Residential Systems

Residential systems usually have less than five tons of cooling capacity. Residential cooling coils are usually a direct expansion (DX) design. Residential heating coils are available for heat pump units or electric heat. Hot water, steam, and chilled water coils are uncommon for residential applications so will not be discussed here

A residential split system is comprised of a separate indoor coil (fan required) or coil and fan combination unit, coupled to an outdoor cooling-only or heat pump condensing unit.

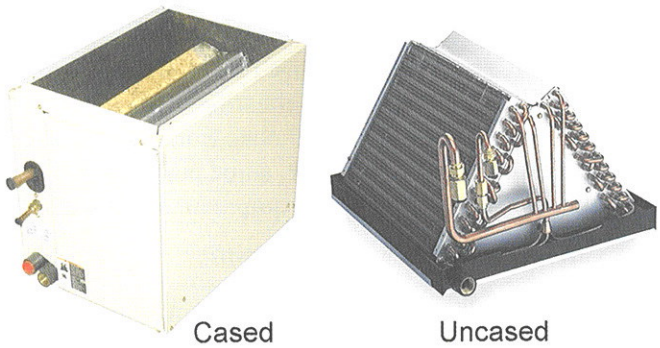


Figure 2

Residential Coils

The indoor DX cooling coil is often mounted on top of a residential furnace or fan unit. Residential cooling coils are similar to the larger commercial packaged unit cooling coils, but are available in smaller tonnage ranges. The coils are traditionally installed on the discharge side of the fan. Cooling coils are available in a number of configurations, "A," (shown here) "N," and slab. The coil can be a cased (factory enclosed) or uncased design. When an uncased coil is used, the field fabricated ductwork forms the casing around it when it is installed. See Figure 2 for an example of cased and uncased coils.

Commercial Packaged Units

Packaged commercial units are typically available in capacities from 7½ tons to over 100 tons. Packaged units are available in a limited number of pre-defined sizes. The advantage of the packaged air handler DX cooling coil is that the components, such as thermostatic expansion valves (TXVs) and nozzles, are normally factory selected and may be mounted at the factory. Nozzles and TXVs are discussed on pages 22 and 23 of the TDP.

Direct expansion cooling coils are used in small commercial applications. These coils depend on the airflow provided by a furnace or small commercial fan unit to circulate the air through the coil. Two types of coils can be used: an "A" coil design, which is used when two furnaces are twinned (used together as one), or a cased evaporator coil that is installed in the ductwork. See Figure 3 for an example. These coils are available in a variety of capacities. The most common capacities are 7½ and 10 tons.

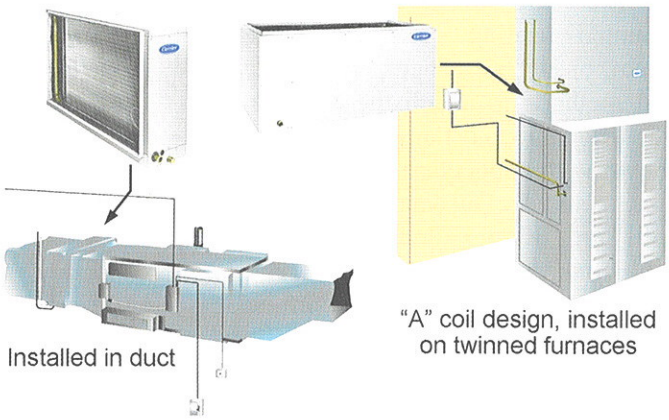


Figure 3

Small Commercial Packaged Unit Coils

Larger commercial packaged units include indoor vertical packaged products and outdoor rooftop units. These types of units will also utilize a direct expansion cooling coil. See Figure 4 for an example. However, chilled water coils are also available for use in many indoor packaged fan coils. Chilled-water cooling coils tend to be used in larger central station air-handling units which are discussed on page 26.

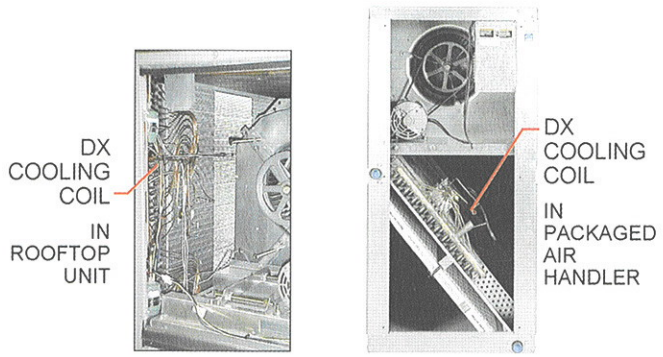


Figure 4

Large Commercial Packaged Unit Coils

Indoor commercial packaged air handlers often utilize hot water or electricity as the source for heating coils. Typically the air handler is available standard with a DX cooling coil, and a heating coil is a field-installed accessory.

Duct-Mounted Coils

Duct-mounted coils are usually heating type. Cooling coils are not typically used because a duct-mounted cooling coil would require an insulated condensate pan.

There are several types of duct-mounted heating coils: hot water, steam, or electric. There are also several methods to attach the ductwork. The drive slip and flanged casings are shown with connection details in Figures 5 and 6. Duct-mounted heating coils are often called reheat coils. Multizone systems that use a reheat coil in each zone supply duct are limited in their application by ASHRAE Standard 90.1 because of potential excessive energy usage.

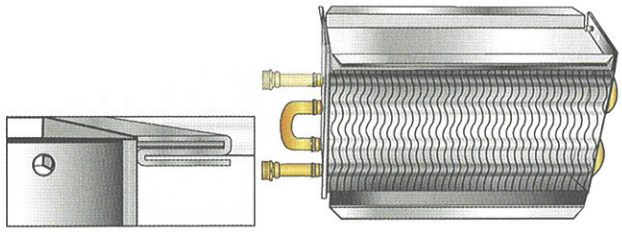


Figure 5

Duct-Mounted Coils with Drive Slip Casing

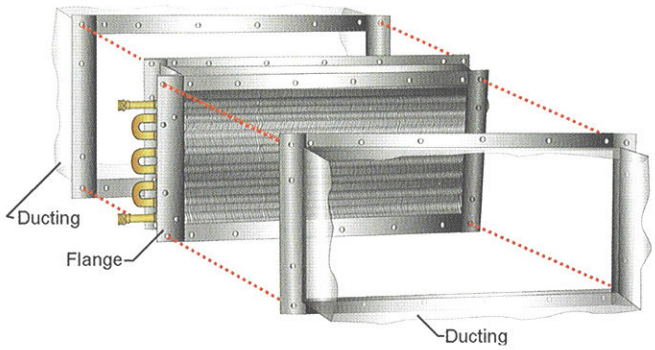


Figure 6

Duct-Mounted Coil with Flanged Casing

Air Terminals

Air terminals are used in variable air volume systems and dual-duct systems and often incorporate small hot water or electric heating coils. These coils are available factory mounted or ready to install as an integral part of the air terminal as an accessory. See Figure 7. The industry also classifies unit ventilators and fan coils as air terminals.

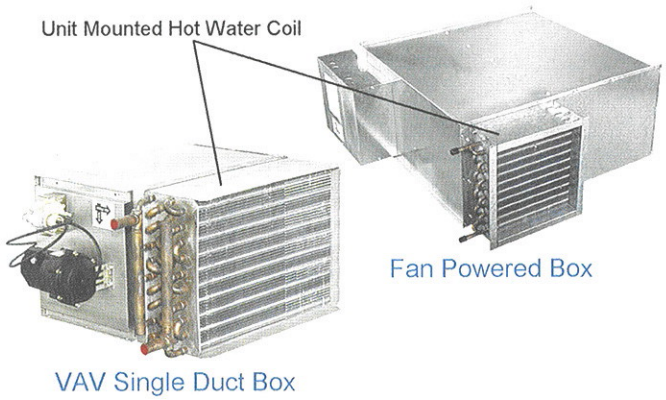


Figure 7

Air Terminal Mounted Heating Coils

Field Built-Up Coil Banks

Coils may also be stacked together to form a coil bank for larger custom applications. The built-up coil banks are stacked coils but can be comprised of more than 2 coils. The coils are arranged and supported by the contractor to fit a special configuration. Field built-up coil banks tend to be used for special large projects or industrial designs.

Air-Handling Units

The coils are important components in air-handling units. Central station air handlers, in particular, offer a wide selection of coil sizes, materials, row and fin options, and features to meet a broad range of applications. The cooling coil face velocity directly affects which cabinet size is chosen for the central station air handler. The cabinet size can be determined by the calculation shown in Figure 13. While packaged air-handling equipment typically offers only a draw-thru arrangement for the cooling coils, central station air-handling units also provide the option to locate the cooling coil in a blow-thru position downstream of the fan.

Coils in packaged products are most often DX and are matched to split system condensing units and heat pumps, although optional chilled water coils may be available. Manufacturers provide matched performance ratings with their condensing units and certify these ratings per ARI. As a result, packaged air handler coil capacity is classified by nominal cooling tonnage, rather than airflow.

Draw-Thru Versus Blow-Thru Coils

The position of the cooling coil within the air-handling unit affects the configuration. A draw-thru arrangement positions the cooling coil upstream of the air handler fan, motor, and drive such that the air is drawn through the coil by the fan. A blow-thru arrangement positions the cooling coil downstream of the fan such that the air is blown through the coil by the fan. Figure 8 shows a typical draw-thru air-handling unit arrangement, which is most common.

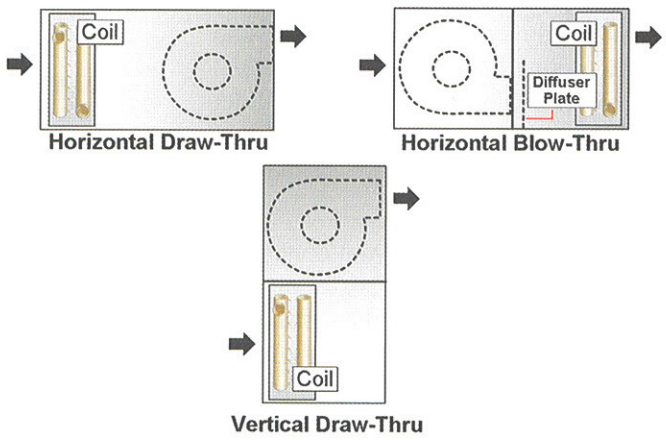


Figure 8

Draw-Thru and Blow-Thru Coils

Draw-Thru Arrangement

A draw-thru arrangement positions the cooling coil upstream of the air handler fan, motor, and drive such that the airflow is drawn through the coil by the fan.

Blow-thru arrangement requires a diffuser plate between the fan discharge and the cooling coil to achieve uniform air distribution. This can add to the length and cost of the air handler, therefore most designs are a draw-thru arrangement. Blow-thru is used on certain air-handling configurations which are discussed in TDP-611, Central Station Air-Handling Units.

Basic Coil Terminology and Construction

This section examines the terms that are used for both heating and cooling coils and also their different construction features.

Tubes

This section applies to steam, water, and refrigerant coils. Electric heating coils do not have tubes but are constructed from elements that heat the air. Electric heating coil construction is examined separately in the electric heat section on page 29.

The tube is a small-diameter conduit through which the heating or cooling medium passes as it rejects or absorbs heat. See Figure 9. Tubes are constructed from materials that have high thermal conductivity. Copper is the most common tube material due to its excellent heat transfer properties, reasonable cost, and durability. Steel is also used in coil tube designs, however it has a slightly lower heat transfer efficiency than copper and the labor costs for steel coils is much higher. Aluminum has also been used successfully on small residential and some small-capacity commercial equipment, like rooftop units. Other tube materials such as red brass and stainless steel are used for special applications as well.

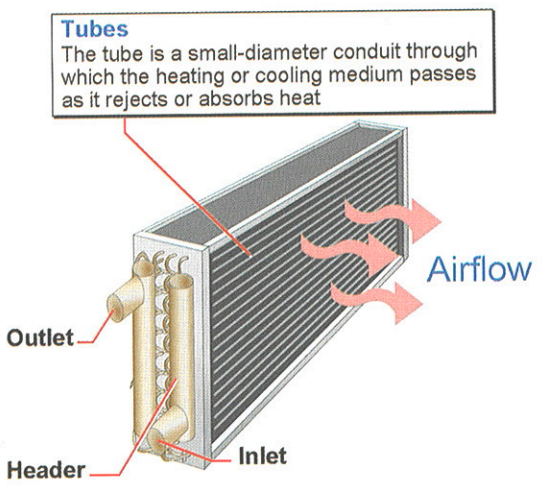


Figure 9

Tubes

The tube itself does not contribute very much to the heat transfer process (relative to the fins) other than distributing the heating or cooling medium. The fins, which are mounted on the tube, contribute most to the heat transfer because the fins constitute most of the coil surface area exposed to the airstream.

Although tubes with smooth inner walls are most common in coils, some manufacturers offer "enhanced" or rifled surface tubing or even internal devices intended to promote fluid turbulence in order to increase heat transfer. When these devices are used, the recommended maximum fluid velocity is usually decreased.

A prime surface tube has no finning or enhancements. Because of their limited heat transfer design, they are typically only used with steam coils.

Tube Diameters

Coils are made in several outside diameters (OD) such as 5/16, 3/8, ½, 5/8, ¾ and 1 inch. The three most widely used tube diameters in HVAC coils are ½-inch, 3/8-inch and 5/8-inch. The 3/8-inch OD is used in DX coils.

With larger tube diameters, lower waterside coil pressure drops may be achievable. It is possible to achieve similar waterside pressure drops with a smaller diameter tube by changing the circuiting of the coil. Circuiting will be discussed later in the TDP on page 16. Cross-sectional coil volume comparisons between ½-inch and 5/8-inch coils show that the 5/8-inch coil has around 25% more volume, which means that similarly circuited 5/8-inch coils will have a lower waterside pressure drop but less heat transfer. It is recommended that the smallest diameter tube with allowable pressure drop be used.

There are fewer larger OD tubes in a fixed face area coil than smaller OD tubes, therefore the 5/8-inch coil does not have a larger airside pressure drop.

Tube Wall Thickness

The coil's tube wall thickness is determined by the required working pressure of the coil. The coil's factory burst pressure testing also helps determine the wall thickness. The maximum allowable working pressure is determined by the manufacturer according to the ASME (American Society of Mechanical Engineers) rating requirements. The materials and construction chosen for other parts of the coil such as headers, also affect the tube wall thickness.

Tube Sheets and Support

On long tube lengths, the tubes may require additional support. Intermediate tube support sheets may be necessary to achieve proper support along the total length. The coil manufacturer typically determines the tube support sheet spacing and material required to support the tubes correctly.

Tube Face

Tube face is the number of tubes in the first row of the coil. The tube face is eight in the coil shown in Figure 10.

Rows

The tubes are arranged in rows within the coil. See Figure 10 for a depiction of coil rows. The more rows (if the number of fins per inch is constant), the greater the heat transfer capability of the coil. Typically, cooling coils will have more rows than heating coils. The primary reason a coiling coil requires more surface area than a heating coil is that coiling coils have a much smaller heat transfer coefficient. Coils must reduce their surface temperature below the dew point of the air passing over them in order to condense out the moisture, so greater surface area is required. More rows, however, result in increased airside pressure drop. Coil selection software will typically pick a coil with the minimum amount of rows necessary to do the duty requested.

Heating coils are often 1 or 2 rows deep. Cooling coils are often 3 rows deep or greater, especially in central station air handlers where even 10-row coils are available. The number of rows affects the amount of air that can pass through the coil untreated. For example, an 8-row coil will have a smaller amount of untreated or bypassed air than a 4-row coil.

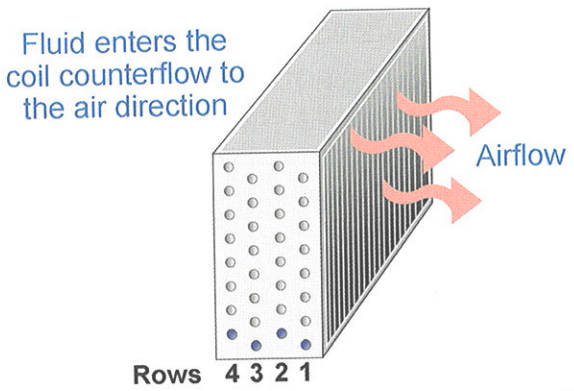


Figure 10

Coil Rows

Fins

The coil fin is a thin metal plate mechanically bonded to the tube to improve the heat transfer efficiency by increasing the surface area in contact with the air. The fins are stamped from sheets and contain holes where the coil tubes are attached. They generally have "enhanced" surfaces, which create turbulence in the airstream to reduce air bypassing the coil surfaces and improve heat transfer. See Figure 11.

The fins are stacked on the tubes and spaced at specific intervals. The spacing is generally expressed as the number of "fins per inch" which is the number of fins present in a one-inch length of the tube. Fin spacing can also be represented in fins per foot, which is used by some manufacturers. Fin spacing on coils ranges from 4 to 20 fins per inch. For typical comfort cooling applications, ranges from 8 to 14 fins per inch are common. This range provides a reasonable balance between heat transfer performance

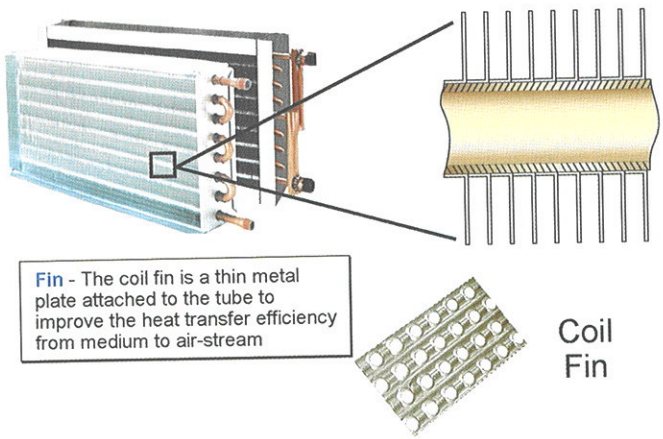


Figure 11

Rows and Fins
(Photo courtesy of Heatcraft USA)

and air friction, which relates to fan energy required to pass the air through the coil. The more rows and fins per inch, the higher the heat transfer capability, but the higher the airside pressure drop. The fin spacing should take into account possible lint and dirt accumulation which is a function of the level of filtration involved.

When fins are stamped, the holes are extruded. When the holes are extruded, they are shaped with flat edges so the tube does not rest on a sharp cut edge.

Fin Material

Aluminum and copper are the most popular materials for fins on cooling coils used for comfort applications. See Figure 12 for an example of bonding fins and tubes. The fins are bonded to the tubes by expanding the soft copper tube with a device called a mandrel. This creates a secure bond between the tube and fin with excellent heat transfer properties. Aluminum tubes with aluminum fins have been used successfully for residential cooling coils and for some small capacity comfort commercial equipment. Copper tubes with aluminum fins, however, dominate the commercial equipment market as the most popular material combination. Copper finning on copper tubes, while significantly more expensive, is a material combination which offers corrosion resistance on appropriate applications. Other tube and fin materials and coil coatings are available where standard coil offerings are not suited to the air content.

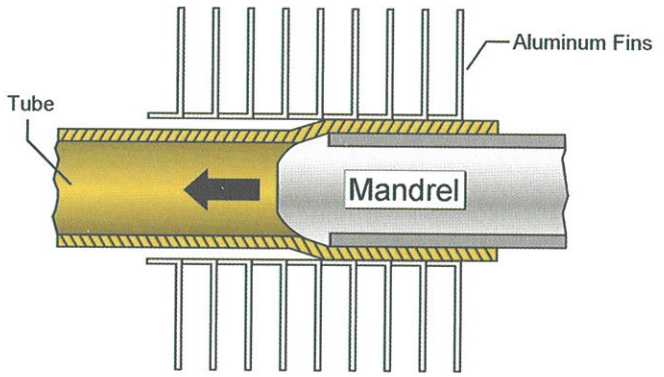


Figure 12

Fins and Tube Binding

Face Area

The actual effective area of the coil is defined as the width times the length of the finned area through which air passes. This is called the finned or face area. This area does not include the extra dimensions for the casing. It is generally expressed in square feet and is also used by many air handler manufacturers to define the model size of the air handler.

Face Velocity and Required Face Area

This is the air velocity in fpm across the finned or face area of a coil. It is determined by dividing the air volume in cfm by the coil face area in square feet. See Figure 13.

This is the air velocity in fpm across the finned or face area of a coil. It is determined by dividing the air volume in cfm by the coil face area in square feet. See Figure 13.

The relationship between airflow volume (cfm), velocity (V) and area (A) is:

$V = cfm / A$

From this, we can determine the required face area of the coil.

For example, if the airflow is 25,000 cfm and the maximum face velocity is 500 fpm then:

$A = 25{,}000 \text{ cfm} / 500 \text{ fpm}$

Minimum coil area (A) = 50 ft^2

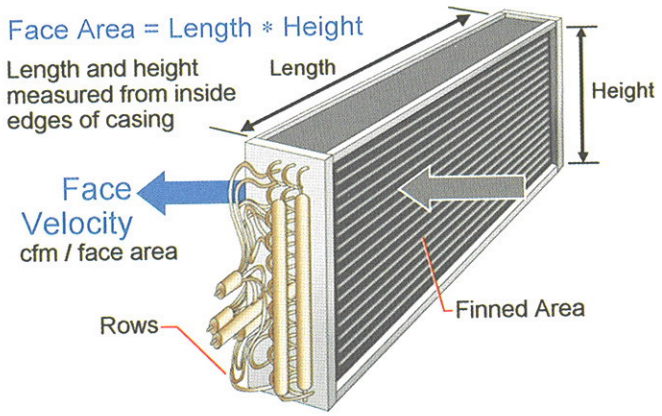


Figure 13

Face Area Calculation

A cooling coil with at least 50 square feet of face area will be required. The height of active surface times the length should be 50 square feet or greater.

Bypass

The number of rows of tubes and fins will change the coil performance. The amount of air, expressed as a portion of the total airflow, that passes through the coil untreated is called the bypass. The bypass for any coil depends upon the coil construction (the number of tubes, size [face area], number of fins, and the tube and fin spacing). The bypass is also affected by the velocity of the air passing through the coil. The bypass factor is the ratio of untreated (bypassed) air to the total air. Bypass factors are shown in Tables 1 - 3.

Table 1
Bypass Factors and Rows

ROWS	BYPASS FACTOR
2	0.31
3	0.18
4	0.10
5	0.06
6	0.03

Table 2
Bypass Factors and Fins

FINS PER INCH	BYPASS FACTOR
8	0.31
12	0.18
14	0.03

Table 3
Bypass Factors and Velocity

AIR VELOCITY	BYPASS FACTOR
300 fpm	0.11
400 fpm	0.14
500 fpm	0.18
600 fpm	0.20

Casing

The supporting metal structure or frame for tubes and header is called a casing. It is usually made of galvanized steel but can be stainless steel or other materials. The end panels that the tubes pass through are called tube sheets. In addition to structural support, they also contain the airflow as it moves across the coil. The assembly of tubes, fins, and coil casing is sometimes referred to as the coil core.

Header

The header is a large diameter pipe to which the tubes are connected. It serves as a distributor of the heating or cooling medium to the coil tubes. See Figure 14. Headers are typically of non-ferrous metal (copper) or steel.

Non-ferrous material

Non-ferrous headers are typically made of copper and contain no iron or steel.

When copper is used for the header and for the tubes, the connecting joint between the header and the tubes will also be copper. This is considered desirable from a corrosion standpoint since the use of dissimilar metals should be avoided.

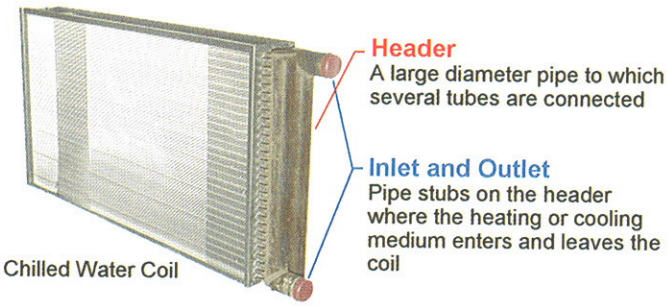


Figure 14

Header

Some manufacturers offer a removable plug for each return bend to access the tubes for cleaning. Another method for mechanical tube cleaning is to make the header a removable assembly. This is not a standard practice as most water in a closed loop is filtered and maintained free of sand and other foreign materials. See Figure 15.

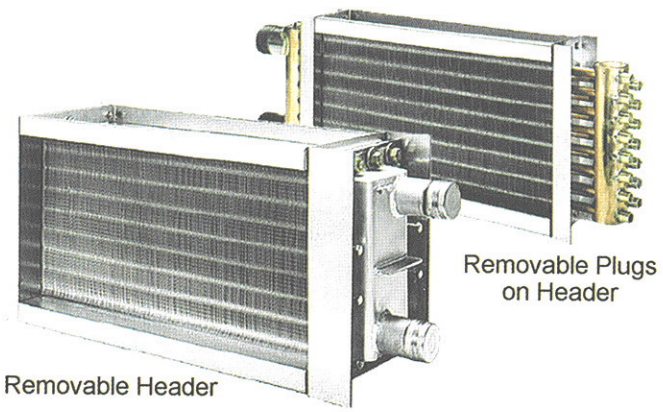


Figure 15

Coils with Removable Tubes
(Photos courtesy of Heatcraft USA)

Inlet and Outlet Connections

The inlets and outlets are on the header where the heating or cooling medium enters and leaves the coil. For water coils the inlets and outlets are sometimes referred to as the supply and return connections. The connections may be sweat/brazed/welded, flanged, pipe thread, or grooved type.

Piping the Coil

In water coils, the inlet (supply) is the pipe stub located on the downstream side (where the air leaves the coil). The outlet (return) is the stub on the entering air-side of the coil. Such an arrangement is known as counterflow, which maximizes heat transfer. See the Heat Transfer section of this TDP for more information.

Copper connections are either brazed (DX) or sweated (room or fan terminals). Brazing and sweating are often looked on as the same thing, but they are different. Brazing results in a stronger joint. Both involve sealing the joint with a high temperature and a joint material. New refrigerants require the use of brazing to avoid using acid flux with the POE (polyol ester) oil used in the newer refrigerants.

Pipe thread connections provide a seal within the threaded joint. A lubricant is used on the pipe threads when screwing the connections together. Male pipe thread (MPT) is the most commonly supplied coil connection. Pipe thread connections are used on water and steam coils generally up to 2½ inches of size.

Grooved connections use a clamp mechanism to attach the piping to the coil pipe stubs. Grooved connections might be used in a facility where no welding sparks are desired. Grooved connections are used mostly on water coils and never used on DX coils.

In steam coils, the inlet is always positioned higher than the outlet so that condensate will drain out of the lower connection.

Direct expansion coils also follow the counterflow arrangement. However, some DX coil designs will route the last pass of the refrigerant circuit back to the warm entering airside of the coil to accomplish superheating of the refrigerant suction gas.

Coil Hand

When you stand in front of a coil, the connections will either be on the right side or the left side. This is what is meant by "hand" connections – right or left. Face the entering air side of the coil to determine its hand connection. There are exceptions which vary by coil manufacturer. See Figures 16 and 17.

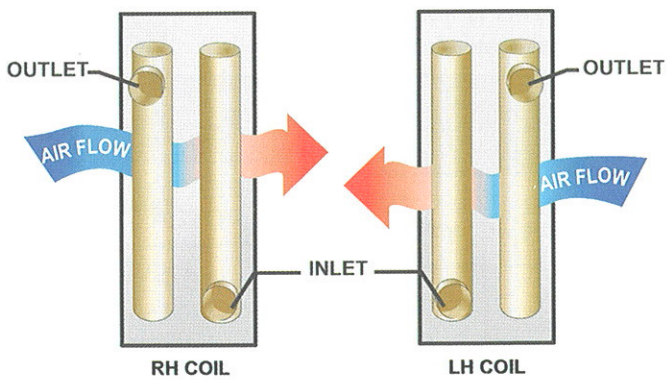


Figure 16

Chilled or Hot Water Coil Hand

Correct coil hand is determined by side of the connections which maintain maximum heat transfer within the coil. In water coils, the inlet will be closest to the air exiting the coils. In DX coils, the refrigerant distributor will be normally closest to the air exiting the coil and the suction connection is closest to the entering air.

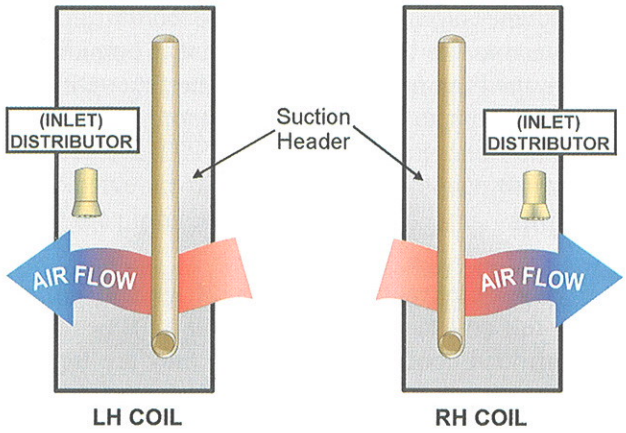


Figure 17

DX Coil Hand

Coil Splits

Face Split

The term face split applies to DX coils only. Face-split coils divide the face of the coil horizontally, on a plane parallel to the airflow (as shown by the dotted line in Figure 18). Each split can be piped to a separate condensing unit. Each split is controlled independently of the other by a liquid line solenoid valve placed in the liquid line upstream of the TXV. Because of their design, face split coils can create uneven leaving air temperatures when one split is deactivated.

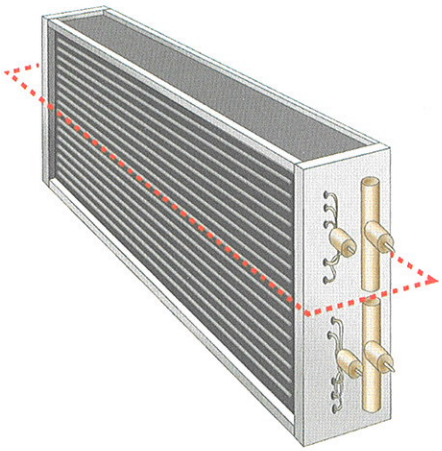


Figure 18

Face-Split Coil

During part load, a face-split coil deactivates the top face. This creates an inactive top coil, which basically becomes a large bypass for untreated air. This untreated air (slightly more than 50% of total airflow) mixes with the treated air from the bottom face and results in a mixture temperature that is often too high to maintain proper room relative humidity control.

NOTE: Water coils use either flow modulation or water temperature adjustment as a capacity control technique.

Face-split coils are often used on constant volume systems. They are not generally applied on variable air volume (VAV) systems or others requiring uniform coil leaving air temperature. When staging a face-split coil, the top split should never remain on with the bottom split off where latent load is involved. This causes the water condensed on the top split to run down the

fins onto the surface of the inactive bottom split. A portion of the condensate will then re-evaporate into the airstream and cause a humidity rise in the conditioned space. To avoid this problem, the bottom split should be the first split on with load increase and the last split off with load reduction.

Row Split

In row split arrangement, each refrigerant split covers the entire face of the coil. This ensures that the full coil face will be active at all times to provide uniform air temperatures leaving the coil at all load conditions. Although they are frequently called row split coils, intertwined coils differ from true row splits in that each split passes refrigerant through all rows of the coil. The circuits of each split weave in and out, or intertwine, throughout the coil to ensure equal load on each split at full-load operation when both are active. See Figure 19.

Intertwined circuited coils are historically preferred for variable air volume VAV systems and others that utilize discharge air control because the full face active feature delivers a slightly more consistent leaving air temperature off the coil. Examples include multizone and double-duct systems.

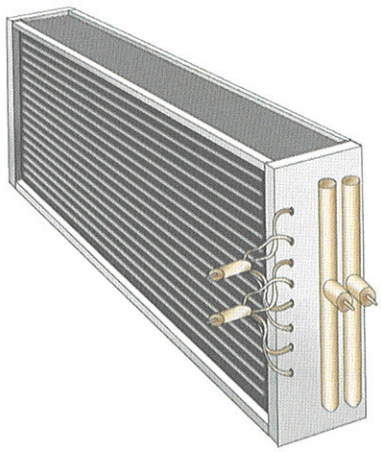


Figure 19

Row Split Coil

Vent and Drain Connections

Vent connections are located on the top of water coils and are used to allow purging of air from the coil, primarily when filling the system with water, but also allow periodic venting to maintain performance or if a vapor lock is suspected in the coil. Vent connections are frequently located on the inlet and outlet stubs or the top of the headers. See Figure 20.

Air bubbles tend to separate out of water or an antifreeze solution in areas where pressure drop is experienced. The flow of air bubbles back

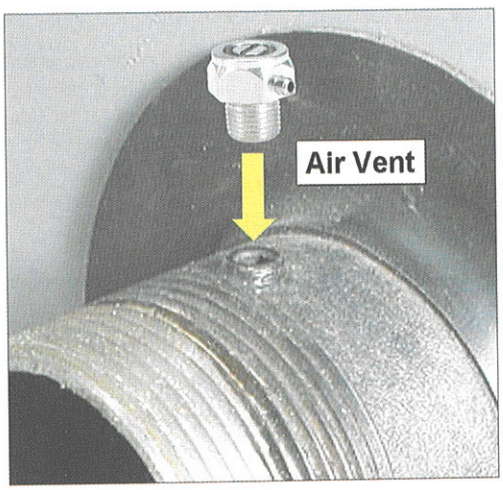


Figure 20

Vent Connection

out of the coil is enhanced by a proper water piping connection to the coil. Air is normally vented at the air separator and not at the coil in a chilled water or hot water system. Elimination of the air from the water system when it is drained is necessary to prevent excessive air build-up within the water system over time. From a systems standpoint, proper application of expansion tanks or compression tanks and air separators are required. See TDP-502, Water Piping and Pumps for a discussion on hydronic system accessories. Excessive air in the system can reduce thermal performance and lead to noise and even vibration of the piping.

Drain connections are used to drain coils for service, or for freeze protection of coils that are not in service during cold weather, such as a cooling coil during the winter. The drain connections are located at the lowest point in the coil to ensure complete drainage.

Return Bends and Hairpins

Return bends are 180° elbows soldered to the coil tubes to reverse the flow back through an adjacent tube in the coil. To minimize the number of return bends and associated solder joints, many manufacturers use "hairpins" which are simply tubes bent at an angle of 180°. A hairpin then results in two tubes within the coil.

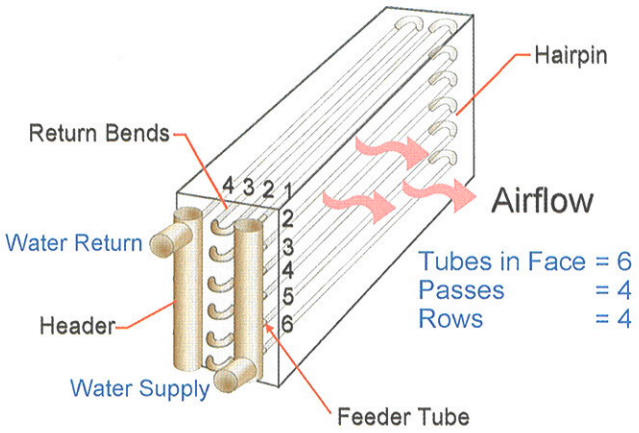


Figure 21

Return Bends and Hairpins

Coil Passes

The number of times the fluid traverses from one end of the coil to the other across the airstream in a given circuit is the number of passes. Shown in Figure 21 is an illustration of a 4-row coil core. The tubes may be connected in different ways to vary the flow pattern of individual circuits, which, in turn will change the pressure drop and heat characteristics of the coil to meet specific job requirements.

Refrigerant Distributor

Direct expansion (refrigerant) coils use a return header similar to a water coil. See Figure 22. It is generally called a suction header, since it is connected to the inlet of a compressor, rather than a water loop. On a DX coil, the supply header is replaced by a refrigerant distributor, nozzle and feeder tubes. Their functions are explained in more detail on pages 22 and 23.

Many distributors contain a nozzle, which has a small orifice designed to add turbulence to the refrigerant and improve mixing of the flash gas and liquid refrigerant to ensure that all circuits within the coil receive equal quality flow.

Quality, as it refers to refrigerant coming out of a distributor and nozzle assembly, means that it should normally be about 20% superheated refrigerant gas and 80% liquid prior to entering the coil.

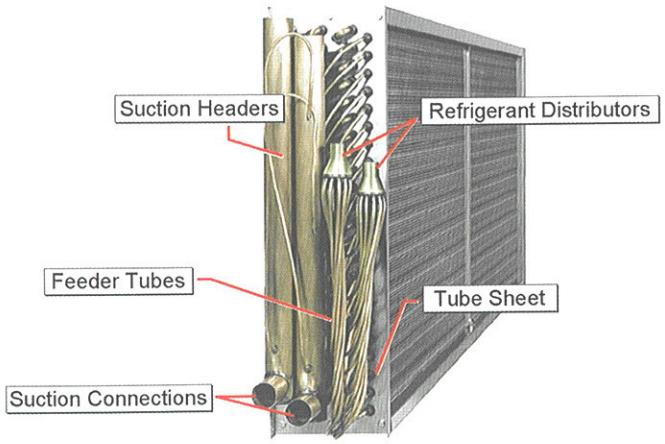


Figure 22

Refrigerant Distributor
(Photo courtesy of Heatcraft USA)

Coil Circuiting

Circuiting applies to both water coils and to DX coils. The path of travel that the cooling or heating medium takes as it enters, travels through the coil, and leaves the coil is a complete circuit. See Figure 23. The refrigerant is introduced into each coil tube typically through a small feeder tube on a DX coil. This is different from the short straight piece of piping from a water coil header that is used to introduce the water into the coil tube. Each coil type offers multiple circuiting arrangements to meet the capacity required within the pressure drop constraints that have been defined. For a water coil, the maximum waterside pressure drop is defined by the user and is job specific. A circuiting arrangement is then selected by the computer to stay within that pressure drop.

For a DX coil, the coil manufacturer has designed the coil to stay within a workable refrigerant pressure drop in the tubes. The user does not need to define the refrigerant pressure drop limitations as with a water coil. The DX selection program will use those circuiting arrangements that are able to deliver the required heat transfer.

The design of the headers and the arrangement of the hairpins and return bends divides the coil core into several independent passages called circuits. Each circuit connects to the supply and return headers (or distributor and suction header in the case of a DX coil). In any coil, these circuits can operate simultaneously to provide the coil's heat transfer capacity. By varying the coil return bend and hairpin turn arrangements, several circuiting types may be provided for the same coil core.

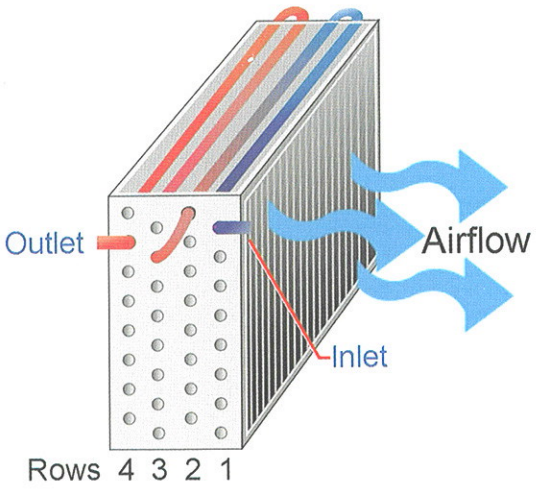


Figure 23

Coil Circuiting

This coil core illustrates the concepts related to various circuiting types in Figures 24 to 27. In each figure the coil will be viewed from the coil connection end. In each figure, solid lines interconnecting tubes represent return bends or hairpin turns on the near end while the light dotted lines represent those on the far end of the coil.

The purpose of circuiting is to provide several different path lengths of water or refrigerant flow within the coil in order to achieve the heat transfer and Δt in the fluid as required by the specific application.

The term non-trapping circuit applies to water coils only. If the coil vent is open and the coil is isolated from the system, then the tubes will not trap water inside of them when flow has stopped. The water from each circuit drains into the header and can be removed by the drain in the header. This feature may be desirable to prevent freezing of standing water in the coil.

Full, half, quarter, and double circuiting normally only applies to water coils.

Tube Fluid Velocity

Part-load operation requires additional design and selection considerations for coils. Chilled water coils use either flow modulation or water temperature adjustment as a capacity control technique. Where water temperature adjustment is used, flow is usually relatively constant. Where water flow modulation is used, the water velocity within the tubes of the coil can vary through a rather broad range.

As an example, one manufacturer suggests a range of coil tube velocity from slightly greater than 1 foot per second up to about 12 feet per second. A coil selected for a design load flow of 8 feet per second tube velocity would allow smooth sensible capacity modulation down to about one eighth (12.5%) of the coil design flow rate. Flow modulation alone, or in conjunction with water temperature reset, is generally used to provide adequate range for part-load capacity control when using water coils.

Fluid Velocity Design Range

The typical design range for velocity is approximately 1 to 12 feet per second.

A selection of circuiting arrangements is available to increase or decrease pressure drop. Greater pumping energy is required to offset the higher coil pressure drop. At excessive velocities, noise and tube erosion can also result. Factors that increase turbulence such as internal tube enhancement increase the temperature difference across the tube wall. Internal enhancements or a device mounted the tube are used to reduce the heat transfer film coefficient. However these devices tend to increase the fluid-side pressure drop.

The waterside pressure drop for water and antifreeze mixtures in coils can range from less than a 1 ft wg head pressure to upwards of several dozen ft wg. If the pressure drop rises too far for economical pumping, it can often be reduced by decreasing the path length of the fluid through the coil.

Full Circuiting

For full circuiting, each circuit passes through each row once and then exits into the return header. Shown in Figure 24 is a circuit flow pattern referred to as "full" circuiting. Each tube in the last row is fed with heat exchange fluid from the supply header or refrigerant distributor. The fluid traverses the 4-row coil shown in this example a total of four times in each circuit (i.e., four passes) and exits into a return header at the same end of the coil as the supply connections. Each set of tubes interconnected in this manner represents a separate, enclosed circuit with a source of supply and a means of return.

In this example, there are eight circuits for this 8-tube face coil. The number of passes will vary with coil row depth. A full circuit coil is the most commonly used arrangement. A full circuit coil has a good balance between heat transfer and waterside refrigerant pressure drop.

Full Circuiting

In a full circuiting arrangement, the number of circuits equals the tube face count of the coil.

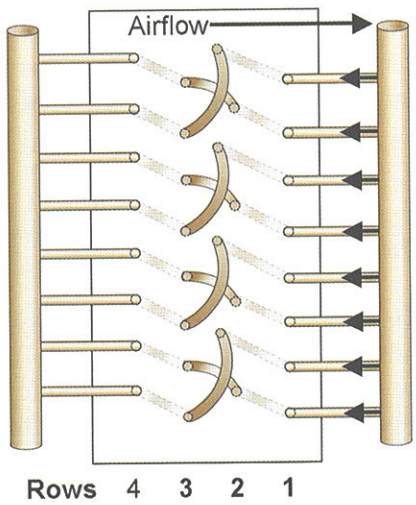


Figure 24

Full Circuiting

Half Circuiting

For half circuiting, each circuit passes through each row twice and then exits into the return header. Therefore there are half the number of circuits as full circuiting and each circuit travels twice as long. Shown in Figure 25 is an illustration of "half" circuiting. Every other tube of the last row, or half the tubes as the name implies, is fed with heat exchange fluid. The number of circuits will be half that of the full-circuited coil, or 4 in this case. For the same 4-row coil core, the fluid now makes 8 passes, which means that it travels twice as far through the coil before being returned as in the full-circuited coil. A half circuit coil would be selected over a full circuit coil when a lower flow rate is required.

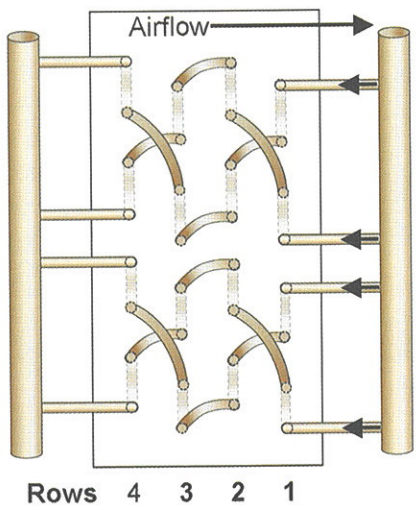


Figure 25

Half Circuiting

Quarter Circuiting

For quarter circuiting, each circuit passes through each row four times and then exits into the return header. Therefore there are one quarter the number of circuits as full circuiting and each circuit travels four times as long. This type of circuiting has the highest pressure drop and the lowest flow rate. Quarter circuiting is shown in Figure 26. In this case each fourth tube (one quarter of the tube face) of the last row is fed with heat exchange fluid. The number of circuits is ¼ of the tube face and for this coil, this results in 16 passes per circuit or twice that of half circuiting. Quarter circuiting accommodates the lowest possible flow rate through the coil. Quarter circuiting is also called high rise circuiting.

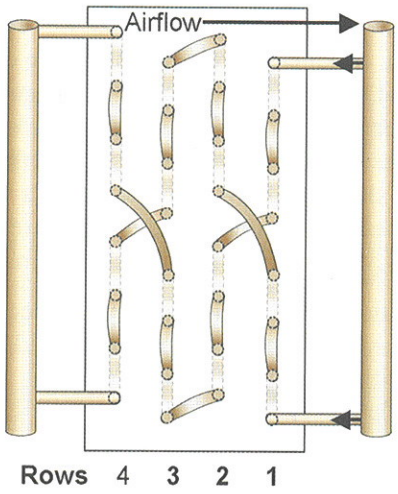


Figure 26

Quarter Circuiting

Double Circuiting

For double circuiting, each circuit passes through every other row and then exits into the return header. Therefore there are twice the number of circuits as full circuiting and each circuit travels half as long. This type of circuiting has the lowest pressure drop and the highest flow rate. Shown in Figure 27 is double circuiting. In this case, every tube in each of the last two rows (double the tube face) is fed with heat transfer fluid. This represents the circuiting arrangement with the greatest number of circuits (twice the tube face) and the fewest passes (two in this case). It handles the highest fluid flow rates through the coil.

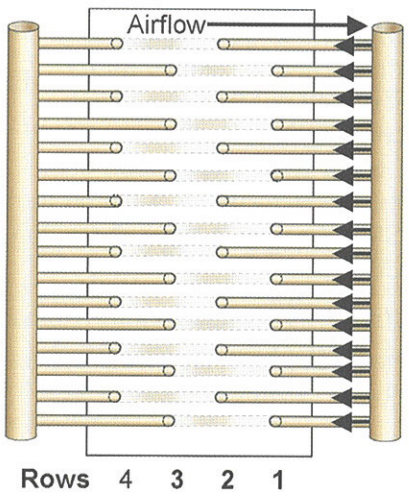


Figure 27

Double Circuiting

Coil Cost Factors

Coil circuiting permits a given coil core to accommodate a wide range of fluid flows and pressure drop requirements to meet the designer's needs. For a given heat removal capacity, the greater the distance traveled through the coil by the fluid in one circuit (i.e., the less the number of circuits), the lower the required flow rate of the heat transfer fluid will be. This is true because the fluid inside the tube has a longer time to exchange heat with the air on the outside of the tube. This means that each pound of fluid circulated through the tubes of the coil will absorb more heat than in a shorter circuit. Longer circuits also produce higher heat transfer fluid tube velocities, with the accompanying higher circuit pressure drop. See Figure 28. There is a practical limit to the length of the circuit. Manufacturers may not offer all circuit types for every length coil.

The rate of heat exchange is influenced by the thermal resistance to heat transfer created by a stagnant layer of water immediately inside the tube wall. Higher tube velocities make this boundary layer thinner, thereby enhancing heat exchange. However, since pressure drop varies as the square of the velocity (or flow), pumping energy can quickly exceed the small gains in heat transfer. Maximum velocities are limited by pressure drop, tube erosion, and noise constraints.

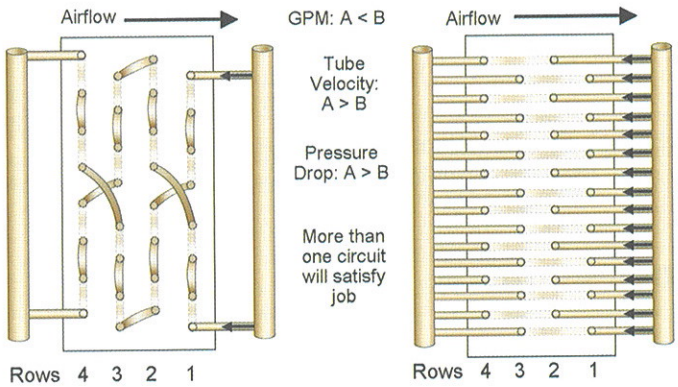


Figure 28

Comparing Circuits

Frequently more than one circuiting type will satisfy the job at hand. There will typically be a combination of coil size, row depth, fin spacing and circuiting which is optimal for the system. This may not be synonymous with the coil that has the lowest cost. However, when two or more coil arrangements satisfactorily meet the performance requirements, the least expensive coil should be selected. Shown below is the cost ranking for the various coil parameters from highest to lowest cost impact.

Cost Ranking Factors for Coils (from highest cost to lowest)

1. Face area (this is frequently related to the size of the air handler cabinet)
2. Rows
3. Fin spacing (fins per inch)
4. Circuiting

It is less expensive to increase the number of fins per inch than to increase the number of rows in the coil.

Types of Coils

Direct Expansion

A direct expansion coil transfers heat through the evaporation of the refrigerant. The refrigerant in the coil evaporates (boils) and absorbs heat from the air passing over the fins through the coil. This is in contrast to a chilled water coil where the cold water cools the warmer air passing through the coil.

Shown in Figure 29 is a schematic representation of a system with a direct expansion cooling coil and air- cooled condensing unit. This type of system requires 2 steps of heat transfer to move heat from indoors to outdoors.

The first step is an air-to-refrigerant heat transfer from the air entering the cooling coil through the coil tube wall into the refrigerant within each coil circuit. The next step is a refrigerant-to-air process as heat within the refrigerant is rejected, after compression, through the condenser tube wall to the outside air. Direct expansion refrigeration systems can make use of water-cooled condensing units in addition to the air cooled variety shown. In that case three steps of heat exchange are involved instead of two.

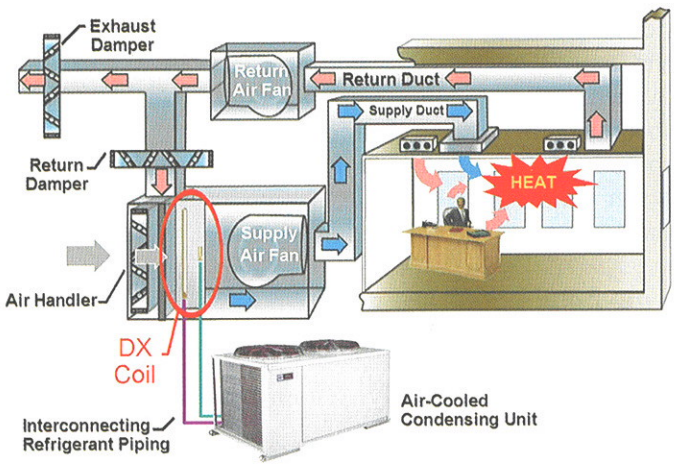


Figure 29

DX Coil System

Direct expansion systems are most frequently applied with one or two condensing units for each air-handling unit that contains a DX coil. This tends to limit the maximum refrigeration system capacity to about 150 tons. By contrast, a chilled water system typically feeds several chilled water cooling coils from a single chiller. This enables a designer to treat physically separated air-handling units and systems from a common refrigeration system without the complications which result from extensive refrigeration piping systems. As a result, the maximum capacity of central water chiller systems is almost unlimited. The benefits of chilled water systems tend to grow the larger the system becomes.

How DX Coils Work

A TXV is a device that is used to meter the flow of liquid refrigerant into the distributor. As the warm temperature liquid refrigerant passes through the TXV orifice, the resulting pressure drop causes a small percentage of the liquid to evaporate, producing flash gas. This flash gas, in turn, cools the refrigerant liquid/vapor mixture to a level below the air temperature entering the coil to allow heat transfer from the air to the refrigerant. See Figure 30. Typically the saturated refrigerant temperature inside the DX coil is between 40 and 45° F.

From the TXV, the low temperature mixture of gas and liquid refrigerant flows into a distributor that, with the help of the nozzle, evenly distributes it to all circuits within the evaporator. Within each circuit, the liquid evaporates as it passes through the coil, absorbing heat from the airstream. The TXV has a sensing bulb mounted on the suction line leaving the coil. It controls the valve position to maintain a constant superheat at the coil outlet to ensure that no liquid refrigerant returns to the compressor.

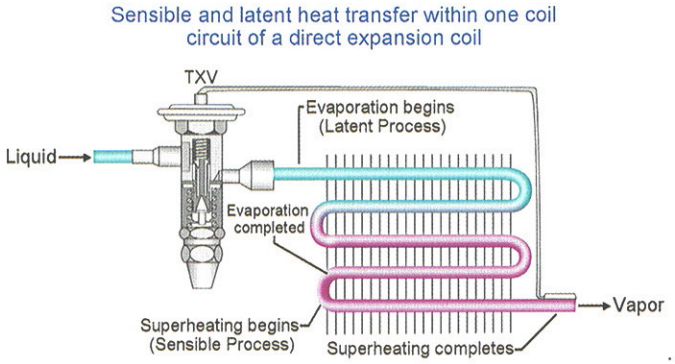


Figure 30

DX Coil Operation

In comfort cooling applications, the cooling system will run at peak design load only a small percentage of the time. The majority of time, the system will operate at part load. Low load with a direct expansion coil is more restrictive than that for chilled water coils. It is imperative that the designer fully understand how the DX coil, thermostatic expansion valve, and distributor function, and what their limitations are.

Low-Load Limiting Factors

Shown in Figure 31 is an evaporator with thermostatic expansion valve, distributor/nozzle, and feeder tube assembly. Within this group, three devices define the minimum load limit. They are:

- Thermostatic expansion valve (TXV) low-load limit
- Distributor nozzle low-load limit
- Evaporator circuit low-load limit

Thermostatic Expansion Valve (TXV)

A thermostatic expansion valve should not operate below the published percentage of the nominal capacity. Below a certain percentage of the nominal valve capacity, the valve's clearance between its needle and seat alternates between being too far open (over capacity with too little superheat) and too far closed (under capacity with too much superheat). This "hunting" action causes loss of system capacity control and poses the threat of liquid flood-back to the compressor. For these reasons, this unstable range of operation should be avoided. The stable operating range for a typical thermostatic expansion valve used on a comfort system is from slightly more than 100% to about 50% of nominal rated capacity. There are,

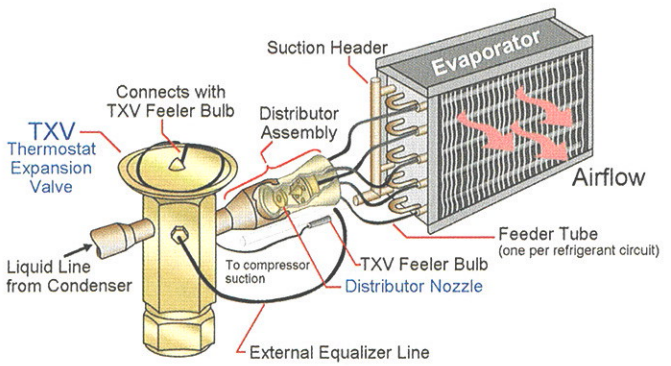


Figure 31

DX Coil with TXV

however, valves that can operate with stability below 40%. For example, one valve manufacturer publishes a minimum capacity of 35% for their standard comfort air-conditioning valves, and 25% for their "broad range" model. Specific manufacturer's data should be checked when selecting specific valves. For a complete discussion of TXVs, see TDP-403, Expansion Devices and Refrigerant Specialties.

Distributor Nozzle

Because of the wide application range possible with central station air handlers, the system designer, aided by manufacturer's selection software, usually selects the nozzles for these product types. The nozzle performs the function of equally distributing the mixture of gas and liquid refrigerant leaving the TXV to each of the feeder tubes. See Figure 32. The nozzle is rated for optimal performance with a flow capacity in tons equal to 25 psi pressure drop across its orifice. The nozzle should be selected to operate from 50% to 200% of rated capacity. Shown in Figure 33 is an excerpt from the nozzle capacity table published by Sporlan Valve Company. Below 50%, the nozzle does not add enough turbulence to the refrigerant stream to produce a homogenous mixture of gas and liquid on its leaving side. The liquid refrigerant tends to settle out and enter the bottom feeder tubes and circuits while the upper tubes get only gas. This results in inefficient evaporation, incorrect superheat sensing by the TXV feeler bulb and possible liquid refrigerant flood-back to the compressor. Operation below 50% of design capacity should therefore be avoided. Nozzles may be easily changed during installation. There are a number of nozzles which fit a given distributor body. Each nozzle contains both a letter and number code. The letter code refers to the outside diameter of the nozzle and is determined by the inlet connection size of the distributor. The number code on the nozzle refers to the orifice size and is determined by the operating capacity of the system.

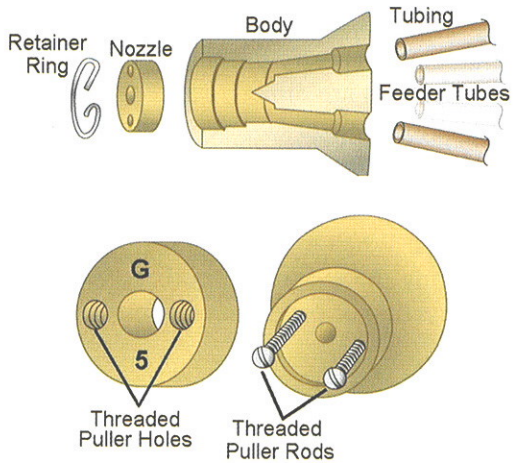


Figure 32

Distributor and Nozzle

- Nozzle creates pressure drop to provide turbulence and mixing of liquid-vapor mixture
- Acceptable nozzle load range 50 to 200% of nominal

Example:
Nozzle Size 5
- Nominal (optimum) cap = 6.1 tons
- Acceptable range: 3.1 to 12.2 tons

- Check full and minimum load

	R-22 Nozzle Capacity (tons)		
Nozzle Load – Percent of Nominal	50	100	200
Nozzle Pressure Drop (psi)			
Nozzle Orifice Size			
1.5	0.9	1.8	3.6
2	1.2	2.4	4.8
25	14.8	29.6	59.2
30	17.1	34.2	68.4

Source: Sporlan Valve Co.

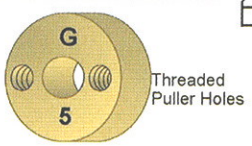


Figure 33

Distributor Nozzle Load Limit

Evaporator Coil Circuit

Oil has an affinity for refrigerant and oil and refrigerant mix in any proportion, so oil continually escapes from an operating compressor and is pumped out into the system. This oil flow is normal and acceptable if it returns to the compressor crankcase at roughly the same rate at which it leaves. The system, including refrigerant piping and coils, must be properly designed to ensure that the oil will return to the compressor.

Although the liquid refrigerant evaporates in the evaporator coil, the oil remains and clings to the evaporator tube walls because oil mixes well with liquid refrigerant and poorly with refrigerant vapor. Therefore, the velocity of the refrigerant flowing within the coil tubes must be maintained above a minimum level to keep the oil entrained (the oil is carried along with the refrigerant). The proper coil circuiting must be selected to ensure adequate design velocity. Manufacturers will specify the design velocity range for their coils. Since velocity and refrigerant flow (and capacity) are directly related, some manufacturers express velocity in terms of tons of cooling capacity per coil circuit, or simply "tons per circuit." This expression makes it easy to select and evaluate full and part-load performance.

With adequate design velocity at full design load, low gas velocity in the evaporator can occur at part load. When the load on the system drops, the compressor capacity must be reduced to prevent freezing the evaporator coil. Compressor capacity control is generally achieved through compressor cycling or compressor displacement reduction (cylinder unloading).

Carrier recommends an optimum design range of 0.8 to 2.0 tons per circuit for its ½-inch tube coils, with a minimum flow rate of 0.6 tons per circuit. For 5/8-inch tube DX coils, 0.9 tons per circuit is the recommended minimum. For 3/8-inch tube coils, 0.4 tons per circuit is the recommended minimum.

Each of the three limiting factors (TXV, distributor nozzle, and evaporator circuit) can become problems where compressor cycling or unloading reduces the flow rate within the system. It is important to select a coil, expansion valve, and nozzle that will satisfy both full and part-load requirements.

If the reduced flow at minimum system load is passed through only part of the coil, controllable and safe system operation can be sustained. Coil splits make partial deactivation of the evaporator possible by maintaining a minimum tons per circuit loading within the active split.

The following example illustrates how coil split deactivation, when coordinated with compressor capacity reduction, can keep the system in a safe, stable operating range.

Split Coil Control Example

A direct expansion coil with split arrangement is used on a job that requires 15 tons of capacity on design day. The direct expansion coil has two liquid and suction connections as shown on the coil in Figure 34.

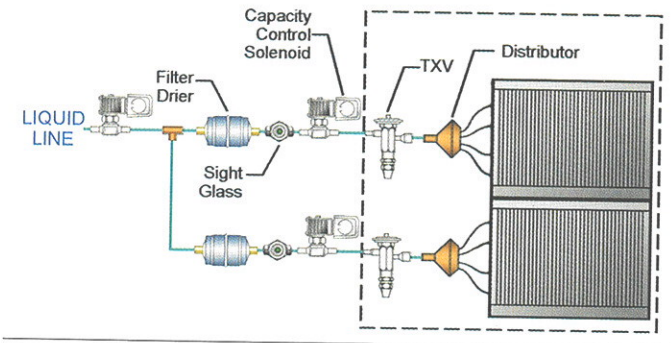


Figure 34

Split Coil Control Example

The compression/condensing equipment is a factory-assembled air-cooled condensing unit. The reciprocating compressor is equipped with cylinder unloaders, which allow it to unload to a minimum of 33% of design refrigerant flow through the system.

Table 4 shows the relationship between compressor unloading at each step of control and its affect on the TXV, nozzle, and coil circuit loading. The center columns represent the loading with the full coil active. As you can see, at 67% compressor capacity, the components are within their acceptable load range with the full coil active. If one split were deactivated at this point, the columns at the right show that the TXV load and coil circuit load would increase above their recommended ranges. Therefore, the full coil should remain active when the compressor is operating at 67% capacity.

Table 4
DX Coil Part Load System Analysis

% Full Load Capacity	Tons	Full Coil Active		½ Coil Active	
		Tons per Coil Circuit	TXV & Nozzle Loading & Design	Tons Per Coil Circuit	TXV & Nozzle Loading Design
100	15.0	1.875	100		
67	10.0	1.25	67	→2.50*	133*
33	5.0	6.25	33*	1.25	67

* Problem area

However, when the compressor unloads to 33% capacity, the coil circuit loading falls to 0.625 tons/circuit, which is near the minimum limit. Also, the TXV and nozzle fall to 33% load, below their minimum acceptable limits. Therefore, one split of the coil should be deactivated when the compressor is operating at 33% capacity. The right hand column shows that this brings the loading back into the acceptable operating range.

DX coil split deactivation, when coordinated with compressor capacity reduction, can keep the system in a safe, stable operating range.

Many direct expansion systems, especially larger systems over 30 to 40 tons, utilize multiple condensing units or "dual circuit" condensing units that have two independent refrigerant systems. Although they require two sets of refrigerant piping, valves, etc., the installed cost is frequently lower, due to the fact that smaller sizes are less expensive and easier to work with. Multiple condensing units generally provide more steps of capacity control, allowing the system to more closely match load changes. One other primary benefit that separate refrigerant systems provide is redundancy. They allow partial system operation in the event of a refrigerant leak or component failure in one of the systems.

Chilled Water Coils

A chilled water coil removes heat from the system through the use of cold water flowing through the coil. When the air passes through the coil, heat is transferred from the warmer indoor air to the cold water.

Shown in Figure 35 is the position of the cooling coil (circled) within a typical air-conditioning system. It also shows the relationship of a chilled water cooling coil to the refrigeration system components. This figure illustrates a four-step heat transfer process required to move heat from indoor air to outdoor air. The first of these steps is an air-to-water heat exchange that occurs as the water in the coil absorbs heat from the air coming back from the conditioned space as well as from the outdoor air used for ventilation. The second step is from water to refrigerant, which occurs in the cooler portion of the central water chiller. Here heat flows from the water coming back from the cooling coil, across the tubes of the cooler into the refrigerant inside the tubes. The third step is a refrigerant-to-water heat transfer as refrigerant from the cooler, after compression, passes its heat through the wall of the tubes in the condenser into the condenser water. Finally, a water-to-air heat transfer occurs as water leaving the condenser rejects its heat to the outdoor air by action of the cooling tower.

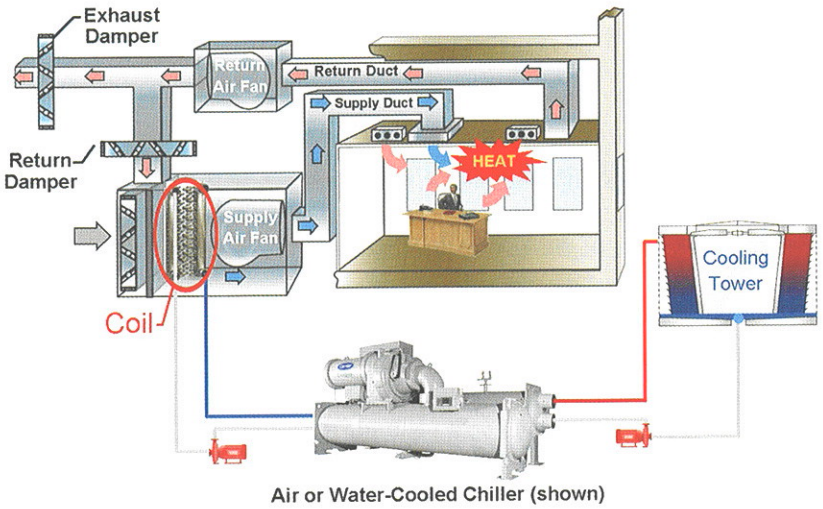


Figure 35

Chilled Water Coil System

Chilled water systems tend to be matched with larger capacity refrigeration machinery than direct expansion systems. Larger capacity central machinery tends to achieve better full load and part load efficiencies than does smaller equipment. This helps to offset the heat transfer disadvantage of chilled water systems as discussed above. In addition, water-cooled condensing systems tend to be used on larger tonnage systems. These, in turn, tend to be primarily chilled water systems. Lower compression ratios that result from water-cooled condensing again tend to enhance chilled water system efficiencies. Finally, the larger capacities delivered by chilled water systems make the use of centrifugal compressors feasible. The centrifugal compressor tends to be more efficient than an equivalent capacity reciprocating compressor, particularly at part load. This tends to facilitate energy efficiency in the chilled water system.

Chilled water coils are made up of a number of heat transfer circuits. These circuits are grouped together to make up the entire coil construction. The existence of splits or subdivisions within a coil offers no operating benefit for chilled water coils. Consequently, chilled water coils are not subdivided into splits. They can, however, be stacked as shown on page 48 of the application section of this TDP.

Shown in Figure 36 is a typical chilled water coil. Chilled water coils normally contain 6 to 8 rows. Chilled water coils often are asked to produce air in the 50 to 55° F range as part of a central system in a large air-handling unit, therefore 6 to 8 rows are usually needed. Other numbers of rows are less common and depend upon the application. Popular fin spacings are 8 or 14 fins per inch.

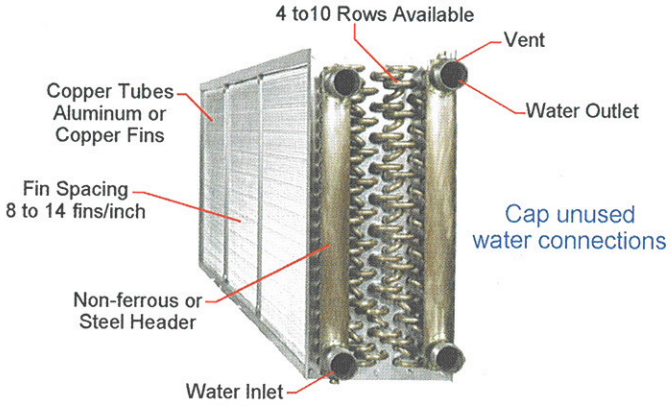


Figure 36

Chilled Water Coil
(Photo courtesy of Heatcraft USA)

Heating Coils

Heating coils are commonly used directly with air-conditioning systems and are designed to heat air under forced convection. Such coils are usually located within the air conditioning apparatus and/or ductwork. The media used in heating coils includes steam, hot water, and electricity. These coils are basically used for preheating, and for tempering or reheating. The size of the coils is determined by the required heating capacity, space, coil face velocity and air friction limitation. The coil air velocity is determined by the air quantity and the coil size. The number of rows and fin spacing is determined by the required temperature rise.

There are preheat and reheat types of coils. A preheat coil tempers the mixture of return and outdoor air in an air-handling unit. A reheat coil supplies added heating capacity during cold weather applications. A preheat coil can also be used to temper or heat the free cooling (economizer) air when the outside temperature is too low.

There are different types of heating media:

- Hot water – water temperature is typically 120 to 200° F.
- High temperature hot water – water temperature is typically above 212° F but still in a liquid state due to the pressure of the system. The maximum temperature is usually 250° F.
- Steam – Steam pressures range from 2 to 250 psig at the coil supply connection, with 5 psig being the most common.
- Electric – typical voltages are 208/230, 460 and 575 V.

Hot Water

Hot water heating coils are similar in construction, size and appearance to chilled water coils other than row depth. Although comfort heating systems seldom require hot water coils with more than two rows, greater depth of surface is available. Fins are usually spaced at either 8 or 14 fins per inch of tubing. See Figure 37.

In order to provide optimum combinations of capacity and water-side pressure drop, various circuiting arrangements are employed. Some manufacturers use turbulators to produce the turbulent flow necessary for efficient heat transfer at the expense of pressure drop.

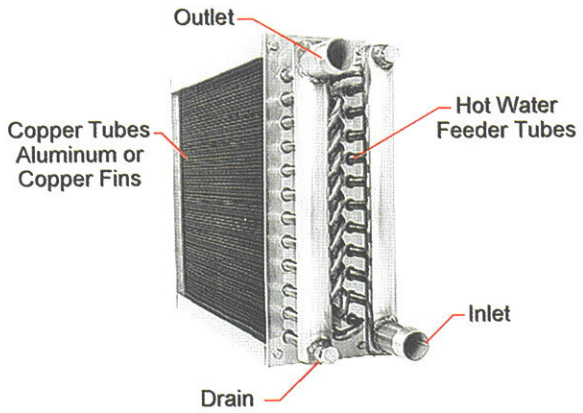


Figure 37

Hot Water Coil
(Photo courtesy of Heatcraft USA)

Hot water heating coils are used on low, medium and high temperature hot water systems. Normally the standard application calls for a hot water temperature of 150 to 200° F. In some larger systems however, particularly those which use a central plant, high temperature hot water may be supplied to the heating coil. Such high temperature hot water usually has a temperature of about 250° F or higher but is still in a liquid state due to the pressurized system in which it is circulated. The purpose of the higher temperature is to reduce the quantity of water required to perform the specified heating function, and thereby reduce the pumping energy. While hot water temperatures in this type of system may reach 300° F or more, the previously mentioned 250° F is used for normal applications. Applications involving temperatures in excess of 300° F are less common.

Selecting of hot water heating coils is based on producing the desired leaving air temperature (capacity) while attempting to maintain the desired Δt in the hot water flow. For instance many systems are based on a 20 to 40° F Δt for the heating circuit. The cooling flow is often based on a 10° F Δt, that is why the system heating gpm is usually less than the cooling gpm.

Steam

Steam heating coils consist of a series of tubes connected to common headers and mounted within a metal casing. To ensure efficient heat transfer, either plate type or spiral type fins are bonded to the tubes mechanically or with solder. Fins are mostly of aluminum with standard spacing of 8 or 14 fins per inch. One-row and two-row coils are available with many selectable tube faces, depending on size and use.

Since the proper performance of steam coils depends on the uniform distribution and condensation of steam in the tube, several methods have been devised to ensure this uniformity. Individual orifices may be built into the supply end of each tube or distributing plates may be installed within the steam header.

Shown in Figure 38 is a "Non-Freeze" type steam-distributing coil that provides uniform steam distribution and leaving air temperature as well as a minimum possibility of freeze-up. This design features a tube within a tube, with the inner tube perforated along its entire length. Steam is supplied to the inner tube and admitted through the orifices to the outer tube where condensation takes place. The condensate is then collected in the return header.

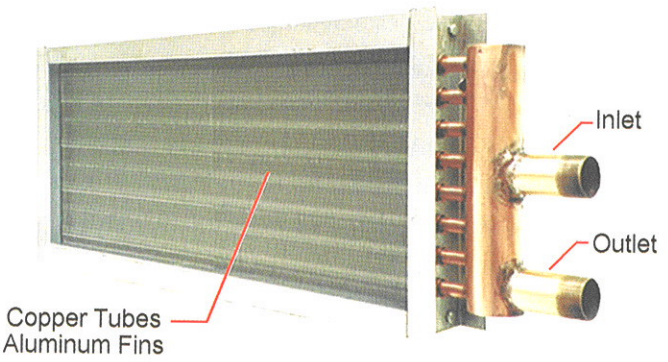


Figure 38

Steam Coil
(Photo courtesy of Heatcraft USA)

Are non-freeze steam coils really non-freeze?

The term "non-freeze" is somewhat misleading and implies that the coil cannot freeze up. This is true as long as the piping and the steam traps are sized and installed to assure rapid and complete condensate removal. It is recommended that full steam pressure be supplied to the coil without throttling when air temperatures entering the coil are below freezing. This will ensure that full design pressure is available to force the condensate out of the coil through the trap. When steam is throttled, it is possible for the pressure in the coil to reduce to a point that will result in condensate hang-up in the coil. In some instances, the pressure may even drop into the vacuum range. At reduced steam flow, the condensate can be cooled so rapidly that coil freeze-up is virtually unavoidable.

Steam coils are also available as a simple U-bend design. This design is not considered non-freeze and depends on complete drainage of the condensate as it uses a single tube as opposed to the tube-within-a-tube design of non-freeze designs.

Electric

Electric heating coils provide heat through the resistance of electricity through the wires. See Figure 39. There is no flow of heating medium as in a hot water or steam coil. Electric heating coils are commonly available in either the finned tubular type or the open type as illustrated. The finned tubular type heater is made of finned steel sheaths containing resistance wire surrounded by refractory material, while the open type consists of a series of electrical resistance coils framed in a metal casing and exposed directly to the airstream.

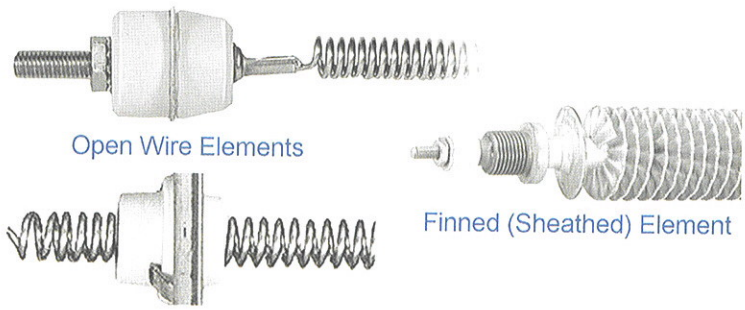


Figure 39

Electric Coil Elements
(Photo courtesy of Brasch Manufacturing Company, Inc.)

An important thing to remember when selecting and specifying an electric heater is that NEC limits the amp draw per stage to 48 amps. Based on the voltage of the heater, this defines the maximum wattage output per stage.

Example: 50 kW heater at 460 volts must be at least 3 stages.

Watts = VA (watts = voltage * amps)

Watts = 460 (48)

22,080 is the maximum number of watts per stage allowed, so this must be a 3-stage heater.

Electric Heaters

With electric heaters, all of the electrical engery used is converted to heat. Thus, heat capacities in Btuh are determined by multiplying the kW rating of the heater by 3,413 Btuh/kW.

Standard voltages include 208 and 230 volts in single or three phase but heaters are also available for operation on 460 and 575-volt service.

In addition to size and capacity, an electric heating coil selection should specify these electrical characteristics and the number of circuits required.

Electric heating coils are usually chosen to fit a branch duct of given dimensions without requiring entering and leaving duct transformations. Therefore, face velocity is not the usual determinant of coil size. However, for Underwriters' Laboratories' approval, a minimum face velocity must be maintained and uniform airflow provided. This minimum velocity is a function of entering air temperature and the total watts per square foot of duct area. Airside pressure drops are usually quite small, compared to steam and water coil pressure drops, seldom exceeding 0.10 in. wg for an open-type coil.

Each electric heater must have a means of disconnect (contactor), a primary and a secondary limit switch for protection, and an air flow proving device to shut down the heater on a loss of airflow.

Electric Duct Heaters

Electric duct heaters can be slip-in or flanged. A slip-in heater is placed into the duct through a hole that is cut in the ductwork. A flanged heater is supported by flanged sections of the duct that are attached to both sides of the heater.

Electric heaters are often installed in ductwork, but may also be installed in the air conditioner or air-handling unit. When the heaters are installed in the air conditioners, the combination that results must be tested by Underwriters' Laboratories (UL) to pass certain requirements. When heaters are provided separately, they must be installed at least 4 feet from the HVAC equipment.

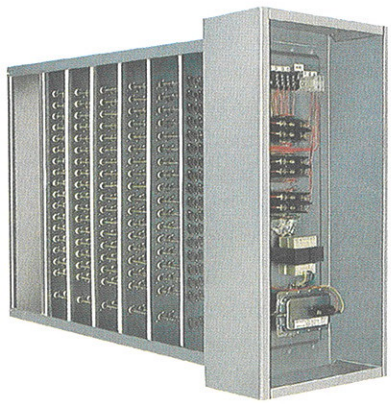


Figure 40

Electric Heating Coil
(Photo courtesy of Brasch Manufacturing Company, Inc.)

Electric Heat Components

An automatic reset thermal cutout is a thermally operated safety device that is required to deenergize the entire heater to prevent overheating.

Heat limiters or replaceable thermal cutoffs are secondary thermally operated safety devices used to protect the system against overheating in the event of failure of the primary safety device or the automatic reset thermal cutout and its circuit. Heat limiters are set to operate at higher temperatures than the automatic reset thermal cutout.

A manual reset thermal cutout is an alternate, secondary thermally operated safety device. It is set to operate at a higher temperature than the automatic reset thermal cutout. If any of the manual resets are tripped, then it is necessary to correct the unsafe condition and reenergize the heater by pressing the reset button.

A linear thermal cutout is a safety device that extends across the entire width of the heater and is used to provide additional protection for long heaters. The linear thermal cutout may be either automatic or manual reset. The safety device will open if the temperature exceeds the setting of the linear thermal cutout.

A contactor, also referred to as a relay, is an electrical device that magnetically pulls in to close an electric power circuit when the holding coil in the contactor is energized. Contactors are used when either the control thermostat or the safety cutouts or an airflow switch do not have sufficient electrical rating to carry the resistive load. There are operating contactors and safety contactors. Operating contactors are required to switch the electric load of each heater step and are operated by the control device, the automatic reset thermal cutout, or the airflow switch. Safety contactors are required with silicon control rectifier (SCR) control, pneumatic-electric (PE) switches, or for step controllers. When operating contactors are not required for each step and the total heater electrical load exceeds the resistive rating of the automatic reset thermal output and/or airflow switch, a safety contactor is required.

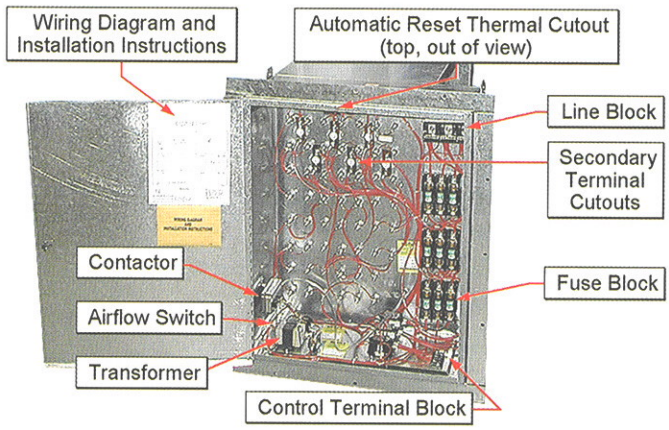


Figure 41

Electric Heat Coil Control Box
(Photo courtesy of Brasch Manufacturing Company, Inc.)

The room thermostat is a temperature control device that is mounted in the area it serves. Thermostats are available in either one step, two steps, three steps, or modulating type, and are also classified as line voltage (to 277 v) or low voltage (24 v). The voltage ratings of the thermostats must conform to the control voltage of the heater.

The duct thermostat is a control device with a remote bulb mounted in the duct or plenum of an air-handling unit to measure air temperature. The bulb should be mounted in a location that does not contact radiant heat directly from the heater. The duct thermostat is available in single, two-step, three-step, or modulating type for line or low voltage.

The transformer is a device that provides control voltage from the line voltage when two voltages differ. Transformers may be built into the terminal box of the heater or in a separately wired remote control panel.

An airflow switch is designed to prevent the heater from operating when there is no airflow in the duct. The most commonly recommended type is a pressure differential airflow switch. The airflow switch is connected in the control circuit in series with the control thermostat and automatic reset thermostat.

The fan interlock relay closes the control circuit to the heater when energized by the fan supply voltage. If it is impractical to interlock the fan and heater electrically by using the fan interlock relay, then an airflow switch is recommended.

The National Electrical Code requires overcurrent protective devices for any circuit which is over 48 amps. This protection is normally in the form of supplementary one-time fuses, but may also be in the form of resettable circuit breakers. The overcurrent protection devices may be built into the terminal box of the heater or in a remote control panel.

The safety disconnect switch may be interlocking or non-interlocking with the control box or panel door.

Heat Transfer and Coil Formulas

Airside Heat Transfer

The heat transfer in comfort air cooling can be determined by the use of formulas for each of the heat transfer steps: airside, coil, and waterside.

Airside heat exchange in a coil can be found with the following three formulas.

Formula 1:

$q_t = 4.5 * cfm * \Delta h$

Where:

q_t = Total heat exchanged as either capacity from an air cooling coil or load from a space. This is expressed in Btuh.

4.5 = Air total heat constant. This converts standard air (70° F and sea level) volumetric flow rate (cfm) to mass flow rate (pounds/hour).

cfm = Cubic feet per minute of standard air.

Δh = Air enthalpy difference between air entering and leaving the coil.

Formula 2:

$q_s = 1.10 * cfm * \Delta t$

Where:

q_s = Sensible heat exchanged as either capacity from an air cooling coil or load from a space. This is expressed in Btuh.

1.10 = Air sensible heat constant. This converts standard air volumetric flow rate (cfm) to mass flow rate (pounds per hour) and incorporates the specific heat of moist air per pound per degree F temperature change.

cfm = Cubic feet per minute of standard air.

Δt = Air dry bulb temperature difference.

Formula 3:

$q_l = 0.69 * cfm * \Delta gr$

Where:

q_l = Latent heat exchanged as either capacity from an air cooling and dehumidifying coil or load from a space, process or outdoor air.

0.69 = Air latent heat constant. This converts standard air volumetric flow rate (cfm) to mass flow rate (pounds per hour), converts grains of moisture per pound of dry air to pounds of moisture per pound of dry air and incorporates the latent heat of vaporization of one pound of water.

Δgr = Air specific humidity difference (grains per pound of dry air).

Overall Coil Heat Transfer

Coil heat exchange can be found through the following equation. This is the heat transfer taking place from the air to the coil.

$q_t = U * A * LMTD$

Each component in this formula will now be discussed.

The formula above is somewhat oversimplified because it ignores the fact that part of the coil is wet while the remainder is dry. When water covers the finned tube surface, heat exchange progresses more rapidly than on a dry coil. To accurately reflect the impact of condensate on cooling coil surfaces the wet and dry portions must be rated separately or the overall heat transfer coefficient for the coil (U) must be adjusted to reflect the net result of wet and dry portions combined. For more detail see ARI standard 410. The simplified equation above will suffice for establishing a working understanding of air cooling coils.

q_t = Coil heat transfer capacity in Btuh.

U = Overall conductance factor (also called coefficient of heat transfer) expressed in Btu/hr/ft^2 coil outside surface area per degree Fahrenheit difference. The overall conductance factor is the reciprocal of the sum of all the resistances to heat flow through the coil and surrounding surface films (i.e., $U = 1/R_T$). This total resistance can be thought of as resulting from a series of layers from outside the coil to inside the coil tube as follows.

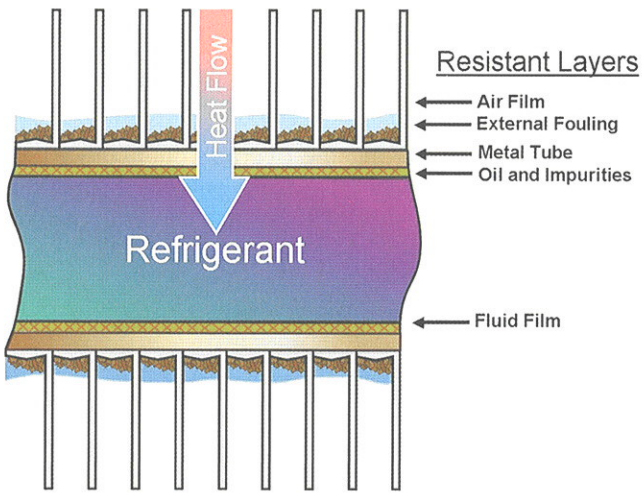


Figure 42

DX Coil Heat Transfer

$$T = R_A + R_D + R_M + R_F + R_{HTF}$$

Where:

R_T = Conductive resistance to all layers combined.

R_A = Resistance of air film adhering to fins and tubes.

R_D = Resistance of dirt on outside of tubes and fins (external fouling).

R_M = Resistance of metal of fins and tubes of coil.

R_F = Resistance of fouling layer inside tubes.

R_{HTF} = Resistance of heat transfer fluid film inside tube adhering to tube wall.

A = Total effective coil surface area. This includes all airside exposure of the finned coil over which the airstream flows expressed in square feet.

LMTD = The logarithmic mean temperature difference (°F) between the air temperature outside the coil and the heat transfer fluid temperature within the coil tubes.

Coil Selection

A design goal in coil selection is to achieve a reasonable match of airside and waterside capacity.

The general format of the coil heat transfer equation follows that of any conductive heat transfer process ($q = U * A * \Delta t$). While heat moves to the coil and away from the coil by means of conductive, convective and radiant modes of heat transfer, the heat moving from the air outside the coil into the media within it does so primarily through conductive heat transfer. For a system, which maintains a stable space condition, the coil absorbs airside load and provides equal airside capacity. Likewise, the refrigeration machinery absorbs waterside load and provides an equal amount of waterside capacity. These segments of the system need to work in harmony. Therefore, when dealing with air handlers that have a selectable coil, a design goal in coil selection is to achieve a reasonable match of airside and waterside capacity. Computer selection software determines the best balance of airside and waterside capacity in making a coil selection.

Factors Affecting Coil Heat Transfer Capacity

There are factors that have implications for coil selection, application and maintenance which will now be discussed.

As mentioned before, the overall conductance factor (U) is the reciprocal of the sum of the resistances to heat flow (R_T) from outside to inside the coil. Anything that effects a resistance component which is part of R_T in one direction effects the U and, therefore, the coil capacity in the opposite direction.

R_A is used to represent resistance to conductive heat transfer caused by the air film clinging to the exterior of the coil tubes and fins. This represents a substantial portion of the total resistance to heat flow through the coil. In fact, the U value of the coil is determined primarily by the resistances to heat flow of the fluid surface films inside and outside the coil. Factors which reduce the thickness of the outside film of air reduce its R and increase U, thereby increasing capacity. Increases in both air velocity and turbulence enhance the rate of coil heat transfer by reducing this film. For a given cfm of air, coil face area reductions increase coil face velocity and velocity over the coil fins and tubes. Factors which increase air turbulence over the coil fins and tubes are air velocity over the coil tubes, surface roughness of tubes and fins (an increase in roughness initially increases turbulence), tube spacing (the closer, the more turbulent) and fin shape. As air velocity and turbulence over the coil surface increases, the tendency for condensate to be carried off the coil surface into the airstream also increases.

R_D is used to represent the resistance to conductive heat flow caused by the dirt or external fouling on the coil. Any build-up of foreign matter on the coil surface detracts from heat transfer efficiency. Factors which increase the external fouling layer build-up over a fixed time period are: reduced air velocity, increased particle content in the airstream, wetting of the external coil surface, and alternating wetting and drying of the coil surface. During the design of the system, proper coil face area selection will ensure that air velocities over the coil do not drop below minimum values, which helps to prevent airside fouling. Proper selection of air filtration devices can also helps to avoid airside fouling in addition to improving space air quality. Coil surface wetness is a function of dehumidification capacity (and load) which the coil encounters. This is not a controllable variable for the designer if space air quality control is a top priority. Periodic cooling coil cleaning is a service procedure that may be required to restore a coil to its original heat exchange capacity when airside fouling becomes severe. Coils do not achieve full capacity at start-up because of the oils present on the coil from construction. In a period of time, the oils will be removed from operation and the coil will achieve full capacity.

R_M represents the resistance to heat flow caused by the metal of which the coil tubes and fins are constructed. Metals inherently possess a low resistance to conductive heat flow (R). Metal provides the lowest resistance to heat flow of any of the layers listed in the coil heat transfer equation. The metals used in tube and fin areas of the coil are selected based on a compromise between heat transfer efficiency (Low R), external and internal corrosion resistance, structural strength, compatibility to manufacturing techniques and cost. The system designer for packaged air handlers rarely has a selection of coil materials. Even for air handlers where the coil is selectable, the material selection for coil tubes and fins is quite limited. Coils sold separately offer more material flexibility.

R_F is the fouling created by impurities that build up on the inside of the coil tubes. Like airside fouling it detracts from coil efficiency. Factors affecting internal fouling on a DX coil are the amount of oil in the evaporator and refrigerant velocity through the coil tubes. Excessive oil in the evaporator tends to cling to the wall of the tube and create a fouling effect. Low tube velocity also tends to allow oil to build up on the tube wall as well as particles of impurity, which circulate with the refrigerant. Additionally, refrigerant gas bubbles, which are formed when the refrigerant boils within the coil tubes, tend to stick to the tube wall at low velocities. Since conductive heat flow is greater through a liquid than a gas, a coil tube with gas coating the inside of the tube performs poorly compared to one with liquid coating the inside of the tube. The gas bubbles clinging to the inside tube wall at low refrigerant velocity act as a fouling factor. Increased tube velocity tends to create a scrubbing action and moves the gas bubbles along with the fluid stream. This also reduces the thickness of the oil layer and releases the particles of impurity from the tube wall. Proper coil circuit selection and coil control at partial capacity ensures adequate tube velocities to keep the waterside fouling factors to a minimum.

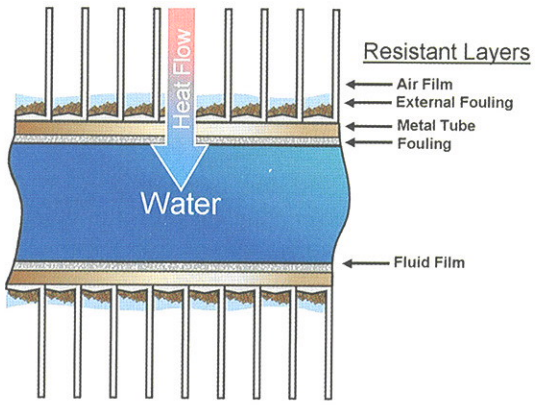


Figure 43

Chilled Water Coil Heat Transfer

R_{HTF} is the resistance to conductive heat transfer caused by the film of heat transfer fluid adhering to the inside of the coil tube. This waterside film resistance, like the airside film resistance, represents a significant portion of the total resistance to heat flow into the coil.

As with airside film resistance, increases in fluid velocity and turbulence within the tube tend to reduce the film thickness, thereby reducing its resistance to heat flow.

The next component in the coil heat transfer formula ($q_t = U * A * LMTD$) is coil surface area (ft^2), A. Anything that increases the total coil tube and finned heat exchange area increases its heat transfer capability. Things which accomplish this increase are increased face area, increased tube rows, addition of finning, closer fin spacing and closer tube spacing. Of these variables the ones which are most frequently controllable by the system designer are face area, coil rows and fins per inch. The equipment style and the manufacturer's versatility in product offering determine the selectability of these items.

Log Mean Temperature Difference and Counterflow

The final component in the coil heat exchange formula is the logarithmic mean temperature difference, LMTD between the air outside the coil and the heat transfer fluid within the coil tubes. A mean temperature difference is required because both air temperature and coil heat transfer fluid temperature are changing as they flow through the coil. This means the temperature difference at any point within the coil is different from that at another point.

Shown in Table 5 is a typical chart illustrating the air temperature changes as they progress through a chilled water coil. The entering air row of the coil absorbs the greatest amount of heat. Each additional row absorbs less than the one immediately upstream. The temperature change of both the water and air that is associated with each row of the coil can readily be seen. The airside figures summarize as shown in Table 5.

Table 5
Air Δt per Row

Row	Air Δt	% Total
1	9.0	41
2	6.0	27
3	4.3	20
4	2.7	12
TOTAL	22.0	100

Since the heat absorbed from the air enters the water within the coil, the waterside (water) load associated with each row will match the airside figures.

Log mean temperature difference is enhanced in chilled water coils by arranging the air and water for counterflow through the coil. This was the case for the water coil illustrated. Figure 44 shows a chilled water coil circuited for counterflow and parallel flow. While less efficient, the parallel flow arrangement, when used on chilled water coils, still provides a substantial percent of possible capacity.

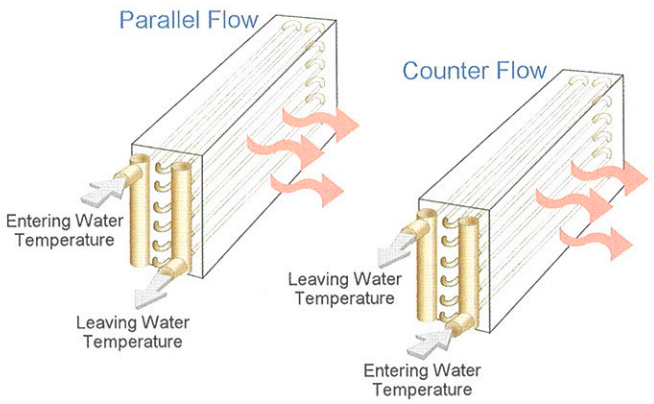


Figure 44

Parallel and Counter Flow

Direct expansion coils achieve optimum heat exchange efficiency (maximum LMTD) when refrigerant is piped for flow parallel to the airstream. This occurs when the refrigerant is fed into the entering air row of coil tubes. Most of the evaporator's capacity comes from the conversion of liquid refrigerant to vapor within the coil tubes (a latent heat absorption process for the refrigerant). The temperature at which this boiling process takes place (saturation temperature) is pressure dependent.

Since refrigerant flow through each coil circuit involves pressure drop, the saturation temperature drops within a coil tube the farther the refrigerant progresses in each circuit. This pressure and temperature drop helps to maintain a high temperature difference between air and the heat transfer fluid. The greater is the circuit waterside pressure drop, the greater will be this decrease in saturation temperature, and the better the heat transfer efficiency will appear for the parallel flow arrangement. Beyond a practical upper limit, however, extra pressure drop will detract from compressor and system capacity more rapidly than capacity will be added to the evaporator. Proper circuiting selection, when offered as a selectable product feature, will maintain coil refrigerant pressure drop within reasonable limits.

Heat removal capacity provided by a direct expansion coil results primarily from liquid refrigerant boiling within the coil tubes. A sensible heat addition to the refrigerant takes place in most evaporators as well. Once boiled into a saturated vapor, additional (sensible) heat is added to the refrigerant near the end of its journey through the refrigerant circuit. Superheat is planned into the coil's performance as a protective measure for all compressors. The TXV control device maintains it at a reasonable level.

Parallel flow of refrigerant and air through the coil optimizes the latent heat exchange process involved in boiling the refrigerant within the DX coil tubes. Unfortunately, the sensible heat exchange process involved in superheating the refrigerant vapor is inefficient when arranged in this manner. One way of resolving this problem is to feed the second row of tubes with refrigerant, continue to the leaving air row and then return to the entering air row to achieve the desired superheat. Where coils are row split this arrangement is more difficult to execute. Superheat quantity and quality suffer badly on the leaving air split of a conventional row split coil.

Some DX coil designs pipe refrigerant so that it is in a counterflow configuration to the airflow. While this is not optimal for heat transfer efficiency, it has advantages in simplicity of coil piping, quality of superheat and adaptability to a variety of coil splits. Where coil circuiting is a selectable parameter, selecting circuiting which gives low design coil pressure loss can minimize the efficiency loss of this type design.

The arrangement of flow of the heat transfer fluid within the coil relative to airflow through the coil is important. This impacts the log mean temperature difference in the coil heat exchange formula. Other factors influencing the LMTD are the general temperature levels of air and heat transfer fluid entering the cooling coil. Higher air entering and lower water or refrigerant entering temperatures increase log mean temperature difference, thereby increasing coil capacity. On the other hand, any drop in chilled water temperature or refrigerant temperature has a capacity reducing effect on the water chiller or compression equipment, respectively. These opposing capacity effects must be balanced by arriving at an operating point that gives optimal system performance. Most engineers accept 42 to 44° F entering water temperature to a chilled water coil and about 40 to 45° F saturation temperature leaving a DX cooling coil as design targets which represent a reasonable compromise between evaporator and system capacity, efficiency, and cost.

Waterside Heat Transfer

Waterside (or sometimes called tube-side) heat transfer, like airside heat transfer, is equal to the coil heat transfer. It can be quantified by the appropriate mass flow heat transfer formula for the fluid gaining heat within the coil tubes. For example, heat absorption by the chilled water within a chilled water coil at comfort air conditioning levels can be quantified:

$$q = 500 * gpm * RISE$$

Where:

- q = Total heat gain to the water within the coil (Btuh)
- 500 = A constant which converts volumetric water flow rate (gpm) to mass flow rate (pounds per hour)
- 500† = 60 min/hr * 8.33 lb/gal.
- gpm = Volumetric water flow rate through coil (gallons per minute)
- RISE = Water temperature increase from the time the water enters the coil until it leaves (F)

† Specific heat of the fluid is not shown in the equation but is essential in any mass flow heat exchange calculation. For water, specific heat is 1 Btu/lb °F.

An example using the waterside heat transfer equation for a chilled water coil follows:

Airside and Waterside Balance

The airside heat transfer and the waterside heat transfer of a coil should balance.

We can set the airside total heat equation equal to the waterside total heat equation and within reasonable tolerances we should get the same total heat transfer. Let's use a cooling coil as an example.

Airside:

cfm: 4,000

Enthaply (h) difference entering to leaving air: 7.0 Btu/lb

Waterside:

gpm: 18

Rise: 14° F

Airside heat transfer	=	waterside heat transfer
4.5 * cfm * Δh	=	500 * gpm * RISE
4.5 * 4,000 * 7	=	500 * 18 * 14
126,000 Btuh	=	126,000 Btuh

Application Topics

Chilled Water Coils for Heating Service

When hot water is used as the heating medium, the approach of using an existing chilled water cooling coil for heating has the advantage of reducing first cost. However, when using a coil in this way, the resulting heating capacity will greatly exceed the job requirements. If the capacity of the cooling coil is controlled by the throttling of chilled water, the control valve will be substantially oversized for the heating application. Because of this oversizing, problems can result in trying to maintain even leaving air temperatures and conditions in a space. Under these circumstances, it is preferable not to throttle the heating water but rather control the room temperature by other means such as face and bypass control or water temperature reset with changes in outdoor temperature. Because of freezing problems, this application normally should only be used where the mixed air temperatures entering the coil are always above 35° F. In locations where freezing conditions occur and this heating method is used, some necessary precautions must be taken to protect the coil from freezing.

Electric Heater Application Information

Each duct heater must be provided with an approved integral, automatic-reset temperature-limiting control or controllers to deenergize the circuit or circuits.

Resistance-type heating elements in electric space heating equipment must be protected (not greater than 60 amps). Equipment rated more than 4 amps and using resistance-type elements must have the heating elements subdivided, and each subdivided load should not exceed 48 amps).

Provide uniform and adequate airflow over the face of the heater in accordance with the manufacturer's instructions.

Heat pumps and air conditioners having duct heaters closer than 4 feet to the heat pump or air conditioner must have both the duct heater and heat pump or air conditioner approved for that type of installation and must be marked on the unit.

Ensure that the fan circuit is energized when any heater circuit is energized.

There must be a disconnect within sight of the heater for the heater, motor controller(s), and supplementary overcurrent protection device(s) of all fixed electric space heating equipment.

A return elbow or grille must be placed under any heater to prevent the heater or parts from falling into the conditioned space.

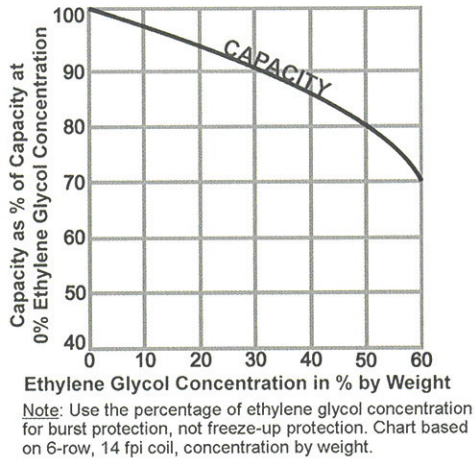
Antifreeze Effects

Antifreeze solutions, such as ethylene glycol and propylene glycol are commonly used. While they provide freeze protection, they have the following adverse impacts on the system:

- reduce coil heat transfer capacity
- increase coil pressure drop which increases water pumping energy
- reduce chiller capacity, resulting in higher equipment first cost

Generally, the use of antifreezes cause higher flow rates and much lower fluid Δt in the hydronic coils as compared to a fresh water selection. Shown in Figure 45 are effects of antifreeze solutions on a cooling coil.

In addition to the performance penalties, ethylene glycol's toxicity also raises environmental and safety concerns related to handling, exposure, system leaks, etc. These issues should be evaluated when choosing antifreeze. Propylene glycol fluids are recommended for use where incidental contact with potable water is possible, or where use of a propylene glycol-based fluid is required by state or local regulations.



Note: Use the percentage of ethylene glycol concentration for burst protection, not freeze-up protection. Chart based on 6-row, 14 fpi coil, concentration by weight.

Figure 45

Antifreeze Effects on Cooling Coil

Coil Corrosion Protection

Corrosion protection, while typically applicable to the packaged unit condenser coil, applies to the evaporator coil also. Applications such as paper mills, photo labs, beauty salons, or other facilities that have airborne contaminants that are recirculated to the evaporator coil require special evaporator coil treatment. On coastal applications, because outdoor air is brought in and mixed with the return air, the evaporator coil should have the same treatment as the outdoor coil. See Figure 46.

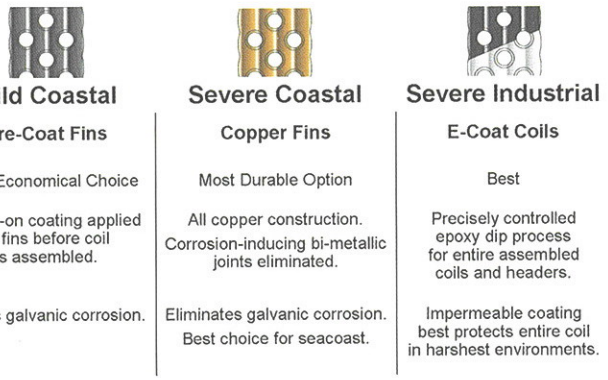


Figure 46

Corrosion Resistant Coil Options

Standard Coil Construction

The standard coil construction for packaged units is copper tubes with aluminum fins. This generally provides the highest performance for non-corrosive environments (e.g., non-polluted environments). Standard coil construction meets the requirements for the majority of applications and locations.

When dissimilar metals used in coils (such as the copper and aluminum) are mechanically connected in the presence of an electrolyte, a reaction occurs. This reaction is known as galvanic corrosion. There must be a bi-metallic couple (bond or contact) between two dissimilar metals in

the presence of an electrolyte solution. Galvanic corrosion will not occur if those ingredients (bi-metallic couple, two dissimilar metals, and an electrolyte solution) are not present. An electrolyte in the presence of the copper-tube and aluminum-fin bond is sufficient to initiate a corrosion reaction.

Electrolytes are electrically conductive solutions. Common electrolytes include sources of chloride contaminants such as seawater, road salts, pool cleaners, and household cleaning agents. These electrolytes are typically sodium or calcium chloride compounds. Other electrolyte sources include sulfur and nitrogen compounds generated by combustion of coal and fuel oils. Since it is common to experience salt-water contamination many miles from the coast, protection from ocean-borne electrolytes in inland areas may be necessary.

Elimination of the bi-metallic couple can eliminate galvanic corrosion. This can be accomplished with an all-copper coil or through isolation of the two metals with a protective coating. Three methods are commonly available to achieve coil protection.

Pre-Coated Aluminum-Fin Coils

Pre-coated aluminum-fin coils have a durable epoxy coating factory-applied to the fin. This option offers protection in mildly corrosive coastal environments, but is not recommended in severe industrial or severe coastal environments. Aluminum fin stock is coated with a baked-on epoxy coating prior to the fin stamping process. This process is known as "pre-coating." The dissimilar metals of the coil are insulated from one another by a thin layer of inert epoxy pre-coating material. As a result, the electrical connection between the copper and aluminum is broken, thus preventing galvanic action.

Copper-Fin Coils

Copper-fin coils eliminate the bi-metallic bond found on standard coil construction. A copper fin is mechanically bonded to the standard copper tube. All-copper tube sheets are also provided to enhance the natural resistance of all copper construction. A protective Mylar strip installed between the coil assembly and sheet metal coil support pan further protects the coil from galvanic corrosion. Durability in a coastal unpolluted marine environment can be substantially improved over the standard or pre-coated coil construction, since the bi-metallic construction is not present.

Coastal Environments

Copper-fin coils provide increased corrosion resistance in coastal environments where pollution sources are not present.

Copper is generally resistant to unpolluted coastal environments, since a natural protective film is formed to coat the copper surfaces. Furthermore, a mono-metal bond exists between the tube and the fin. Uncoated copper coils are not suitable for polluted coastal applications or industrial applications, since some pollutants attack copper.

Electro-Coated Coils

Electro-coated (E-coat) coils provide superior protection against many corrosive atmospheres with the exception of formic acid and nitric acid environments. Electro-coated coils have a durable and flexible epoxy coating uniformly applied over all coil surfaces after the coil is fabricated. A consistent coating is achieved through a precisely controlled electro-coating process that bonds a thin impermeable epoxy coating on the specially prepared coil surfaces. This can be applied to aluminum-fin coils or copper-fin coils. See Figure 47.

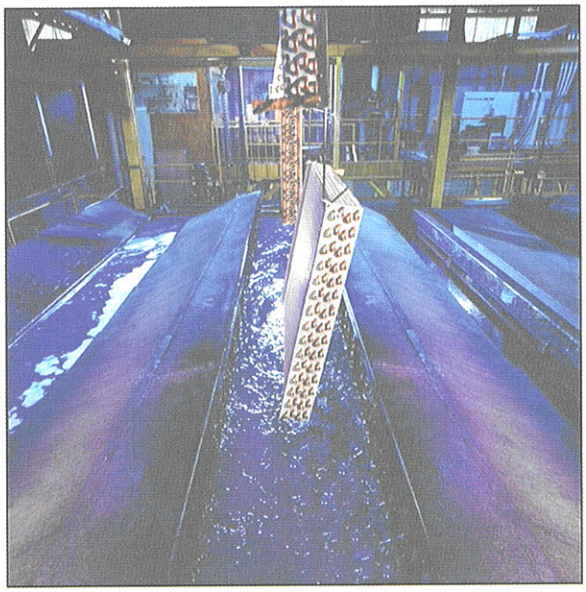


Figure 47

Electro-Coating Process

An electro-coating system applies a DC charge to the coil immersed in a bath of oppositely charged epoxy molecules. The molecules are drawn to the metal, forming an even, continuous film over the entire surface. An oven bake cures the coating uniformly to ensure consistent adhesion on all coil surfaces. Finally, a UV protective topcoat is applied to shield the finish from ultraviolet degradation.

Electro-coating is superior to conventional phenolic coatings, which are applied manually by dipping and baking. Electro-coating is a more durable, evenly coated, non-brittle, non-flaking coil protection product than phenolic coatings.

Coil Maintenance and IAQ

Coils that are kept clean both internally and externally can deliver maximum heat transfer efficiency. In addition, clean coils ensure the best possible indoor air quality. The building airflow passes constantly over the coils so coil maintenance is important to maintaining good overall IAQ.

Internal

Generally coils in closed pressurized chilled water and hot water systems stay clean by maintaining the chemical balance of the system. The potential for biological growth and corrosion is minimized. If the fluid in circulation is an antifreeze mixture, the fluid should be analyzed periodically to confirm the concentration percentage has been maintained and the corrosion inhibitor is still working. See Figure 48.

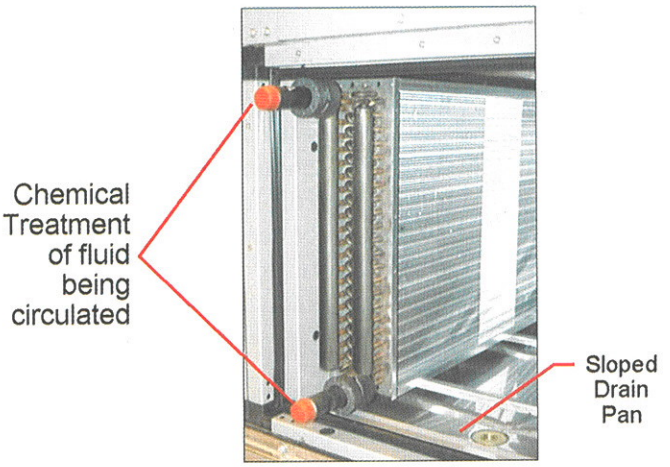


Figure 48

Coil Maintenance

The waterside fouling for chilled water coils is caused by corrosion, particle deposit and air bubbles, which cling to the tube wall. Corrosion can be controlled by proper chemical treatment of the water circulated within the chilled water loop. Since most chilled water systems used for comfort applications are not open to the atmosphere (i.e., closed loop systems), initial treatment of the water at the time of system filling is adequate. Open systems, on the other hand, require periodic treatment of the water. Chemicals are usually added which effect acidity/alkalinity and corrosion. In addition, antifreeze chemicals are added where the coil and/or piping will be exposed to freezing conditions. For a detailed discussion on water treatment, see TDP-641, Condensers and Cooling Towers.

For DX coils, the moisture content of the refrigerant should be checked and the condition of the compressor oil analyzed on a regular basis. Oil can break down if the compressor malfunctions. The acids can harm copper tubing.

External

To keep coils and drain pans clean, they can be washed with low pressure water and a coil cleaner or mild detergent if required. Care must be taken not to damage the fins on coils with too high a water force. In certain applications, like restaurants, where dirt and grease accumulation may be heavy, removal of the coil for cleaning its exterior surfaces may be necessary. Also, condensate drain lines must be kept open and flushed with water as required.

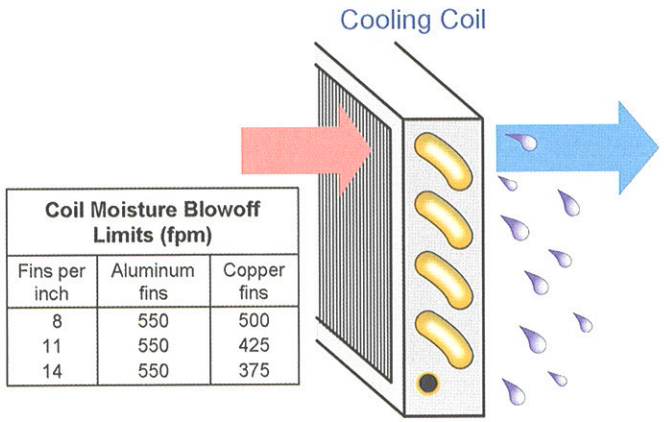
Providing proper filtration upstream of the coils is the single most important task that affects external coil maintenance and building IAQ.

Moisture Carryover

Cooling coils remove moisture by condensing out water from the airstream. This condensed water runs down the coil face, leaving water on the surface of the coil and falls into the drain pan. The design of the coil itself with respect to fin spacing, fin design, fin material, tube spacing, coil height, coil cleanliness, and moisture level of the entering air conditions, determines the maximum face velocity after which carryover (blow off) will occur. These may vary slightly from manufacturer to manufacturer.

Moisture carryover from cooling coils refers to the entrainment into the airstream of this water as a result of too high a coil face velocity. Carryover is undesirable because the water will enter areas of the air handler where it can cause damage or leak into the building. The introduction of moisture (condensate) back into the airstream can also lead to loss of relative humidity control. See Figure 49.

If the coil fins are not clean, the water may carryover even though the face velocity is within the maximum stated by the manufacturer. This can result from oils deposited on the coil fins during manufacturing or during operation on the jobsite. Coil cleaners are available to remove the film.



Coil Moisture Blowoff Limits (fpm)		
Fins per inch	Aluminum fins	Copper fins
8	550	500
11	550	425
14	550	375

Figure 49

Moisture Carryover

Carryover is not detrimental as long as it is caught in the coil drain pan. If it is not preventable, an eliminator section can be mounted downstream. Eliminator sections catch entrained water. However, this is not usually necessary because conventional means of preventing carryover like properly sized drainpans and adhering to maximum coil face velocities works best.

Velocity increase also brings with it increased airflow resistance which the fan must offset with increased horsepower. These negative byproducts must be balanced against heat transfer efficiency in arriving at a reasonable coil face velocity and fin/tube design.

Drain Pans and Condensate Trapping

Cooling coil sections must always contain a drain pan to collect the condensate water extracted from the air passing over the coil. ASHRAE Standard 62 requires a drain pan that will not allow standing water. To meet this requirement, drain pans will generally be sloped to a recessed bottom drain outlet. Drain pans will generally be constructed of galvanized or stainless steel or fire-resisant UV-rated plastic, and when part a of new air-handling unit design, may be coated with antimicrobial material to inhibit the growth of mold and bacteria.

Drain pans must be insulated to prevent the condensation of moisture on the outside of the unit casing. The insulation within the drain pan should be closed-cell foam and water resistant. Fiberglass is not acceptable for this purpose as it will absorb water, lose its insulating properties, and provide a source for mold and bacterial growth.

Condensate drain outlets must be properly trapped to isolate the air-handling system from the building drain system. Without traps on draw-thru units, air may be drawn into the air handler, introducing potentially objectionable smells and gases into the building. It can also impede water flow into the drain and cause water to backup into the drain pan. On blow-thru units, conditioned air can be lost, reducing the efficiency of the system.

Condensate Drains

Local codes and regulations must be checked for proper condensate piping practice regarding connection to open drains or sewers.

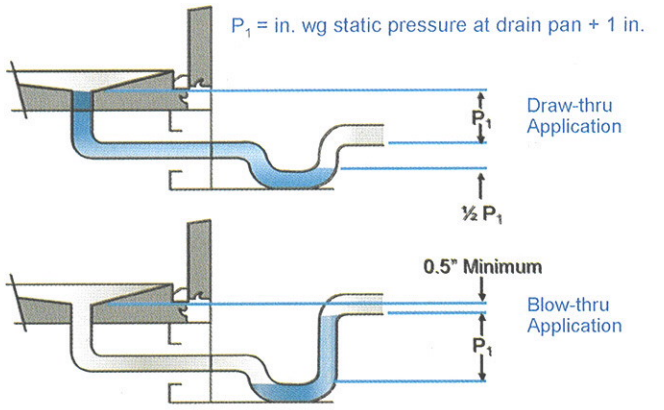


Figure 50

Condensate Trap Construction

The trap depth is dependent on static pressure, either positive or negative, at the drain location. When calculating trap depth on draw-thru or blow-thru applications, remember that it is not the total static pressure, but the upstream or downstream static resistance that should be used. For instance, when calculating the trap depth for a cooling coil condensate pan on the draw-thru side, trap against the coil pressure drop in that coil section and any other pressure drops upstream of it. See Figure 50.

For an example of a calculation for trap depth see TDP-611, Central Station Air-Handling Units.

For all units, provide condensate freeze protection as required. On units with internal spring isolators, be sure the unit is mounted to allow sufficient clearance for the required drain trap depth.

Coil Frosting

Coil frosting is the formation of frost or ice on the tubes and fins as a result of a coil surface temperature lower than 32° F. Condensed moisture can re-freeze on the fins and tubes resulting in partial blockage of airflow.

Frosting is most likely to occur if the saturated suction temperature of a DX coil has become too low. Improper fan operation, extremely dirty filters, or some other increase in system resistance may lead to an airflow reduction that can cause DX coil frosting.

Heat pump units use a defrost cycle to remove frost and ice from the coil.

Heat Pump Coils

Heat pump systems employ special indoor and outdoor coils that are designed to function as either an evaporator or condenser. Typically, the coils used are larger than a comparably sized cooling-only unit. In addition, the metering devices are different in order to accomplish both heating and cooling. When a heat pump unit is in cooling, it functions in the same manner as a cooling-only unit; the outdoor coil is the condenser and the indoor coil is the evaporator. However, when the unit is in heating, a 4-way valve is used to reverse the cycle, the outdoor coil is now the evaporator and the indoor coil is the condenser. In this way, heat is removed from the outdoor air and transferred to the indoor air. See Figure 51.

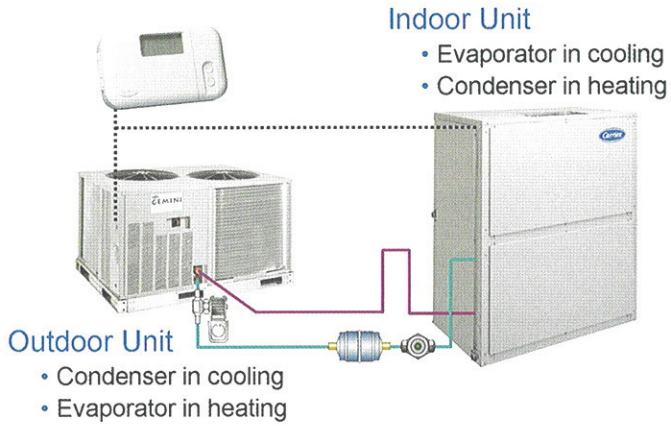


Figure 51

Heat Pump Coils

Heat pump system components are designed and tested as matched pairs and must only be applied according to the manufacturer's recommendations.

Coil Energy Recovery Loop

The coils used in a runaround loop are usually standard water coils. The water coils normally have a minimum of 4 rows up to 8 rows deep. Often the coil selected will be a 6 or 8 row coil to maximize heat transfer area. Sometimes the exhaust airstream contains corrosives, so a protective coating should be considered.

A coil energy recovery loop (or runaround loop) is actually a heat recovery system and distinct from individual pieces of equipment like energy wheels or fixed-plate heat exchangers. See Figure 52. Coil energy recovery loops are very flexible and well suited to industrial applications or comfort applications with remote supply and exhaust ductwork. They use standard finned-tube coils to transfer heat to and from an intermediate working fluid such as water or glycol-water solution. A pump circulates the fluid between the two coils

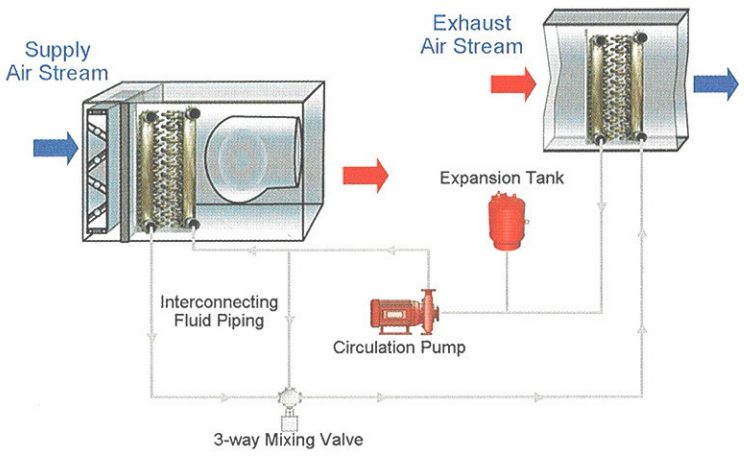


Figure 52

Coil Energy Recovery Loops

in a closed loop. The loop must be equipped with a flow control valve to modulate the heat transfer rate. Also, an expansion tank is necessary for thermal expansion and pressurization of the working fluid. The heat transfer rate must be controlled to prevent overheating or overcooling the supply airstream. If the outdoor airstream is very cold, an uncontrolled system may frost or freeze condensate on the coil in the exhaust stream.

Energy Recovery Loop

Maintenance requirements with an energy recovery loop are complicated because there are more component parts than with passive energy transfer equipment. The system also requires electrical power, motor controls, and temperature controls.

By regulating the fluid flow through each coil and blending warm fluid with the cold fluid entering the exhaust coil, the total heat transfer rate is limited to the maximum possible without freezing. Overheating is prevented in the same manner. Runaround loops are well suited for applications that must keep the two airstreams separate. They are useful for sensible heat recovery only and can be designed to transfer heat in either direction. They are also uniquely capable of simultaneous heat transfer between multiple locations using the same circulating system.

Spray Coils

A process that was popular in industrial applications in the past was a spray coil. Water is sprayed with nozzles onto a coil as shown in Figure 53. The coil is connected to a chiller. The spray water evaporates on the coil and provides a cooling effect.

Overall, the spray coil is not widely used anymore because the industrial user base such as the textile industry has shrunk and also because of the high humidity it produces. With the accompanying odor and building damage concern caused by mildew and mold growth, spray coil applications are less common. The spray coil is an inexpensive option to full mechanical cooling. Copper/copper coils are normally used for corrosive reasons.

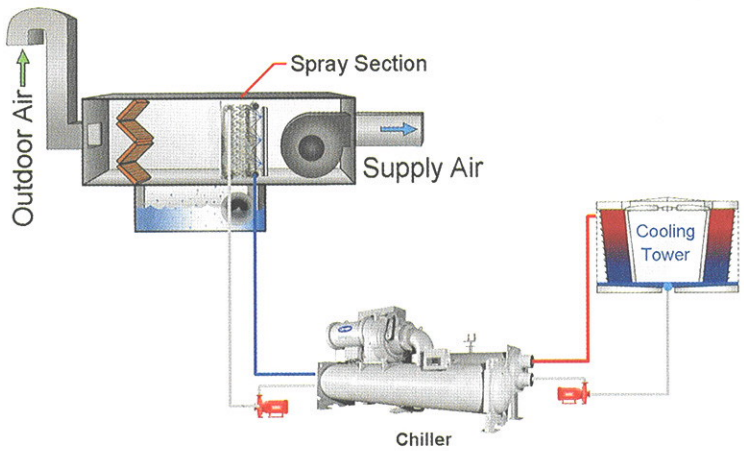


Figure 53

Spray Coils

Stacked Coils

Where air handler capacities – and consequently coil capacities – are large, individual coils may have to be stacked (combined) to meet the requirements. This approach basically involves stacking the separate coils one on top of another or in a staggered arrangement as shown in Figure 54, to produce the needed coil face area within the air-handling unit cabinet. The individual coils retain their coil casing as well as inlet and outlet connections. The coils are also frequently split internally in either face or row split fashion.

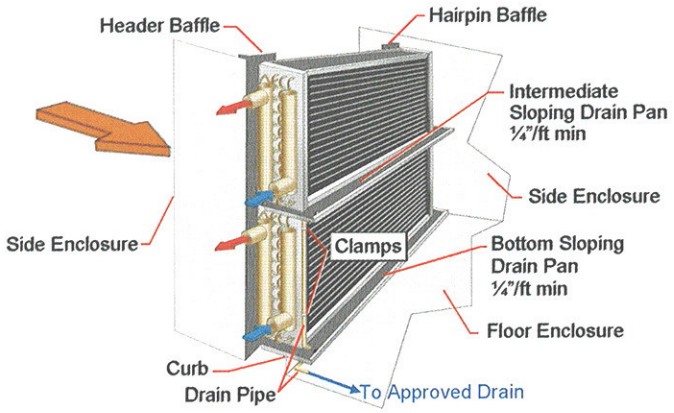


Figure 54

Stacked Coils

Care must be exercised in piping stacked or staggered coil banks. This is because it is desirable to maximize active coil face at partial load. For an example let's assume we have a staggered or stacked DX coil system (same concept applies to chilled water). If the coils are DX, and multiple condensing units are used, the best way to pipe them is to pipe each condensing unit to both coil sections. By doing so, the maximum amount of coil face remains active resulting in maximum moisture removal even if only one condensing unit is in operation.

Heating coils may also be stacked; however, in air-handling units this is less common than with cooling coils. The heating coil face velocity limitations are much higher so not as much surface is required for the same airflow as a cooling coil. Also, most heating applications can be handled by a one or two row coil, so if even more capacity is required, additional rows can be added versus larger face areas.

The staggered arrangement achieves the same result as stacking the coils but within a potential smaller air-handling unit cabinet. Care is taken by the manufacturer to achieve uniform airflow over both coil sections in either a stacked or a staggered arrangement.

Water Coil Control

Flow control valves should be carefully selected to match the operating characteristics of the system flow rate, coil pressure drop, close-off pressure, and type of action required. Most control valves have electrically operated actuators to stroke the valve open or closed. Other control valves have a full modulating proportional control. Spring-operated actuators are commonly found on heating coil valves. The spring return actuator will move the valve fully open in the event a power failure occurs. This is referred to as normally open.

Proportional control valves are most commonly used in HVAC applications. They modulate in response to a pneumatic or electronic control signal. They are used mostly to control fluid flow through heating and cooling coils. The valves are termed proportional because their output flow is not exactly linear in relation to the input signal. Despite their nonlinear response, these valves are an inexpensive way to control discharge air temperature off heating coils.

2-Way Valve Control

Two-way valves are commonly used in conjunction with face-and-bypass dampers to open when the outside air temperature drops below a predetermined set point – usually 55° F. The two-way, two-position control valve will open, allowing flow through the coil and the face-and-bypass dampers will modulate to maintain the discharge air temperature set point.

Two-way valves that are modulating type are used in variable flow cooling coil applications. This application requires the use of a VFD on the system pumps since the system flow varies due to the individual coils throttling the flow at part loads. See Figure 55.

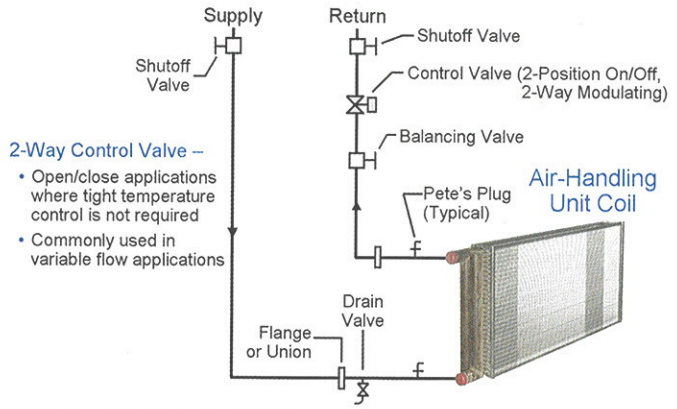


Figure 55

2-Way Valve Control

3-Way Mixing Valve Control

Three-way valves are typically used to provide a constant flow rate through the piped system while varying the flow through the hot water or chilled water coils. A system is shown in Figure 56. Three-way valves can be used in either a mixing or diverting type application. Three-way mixing valves offer good accuracy resulting in tight temperature control. Three-way valves are not used in variable flow systems since the system water flow remains constant due to the individual bypass line around each coil.

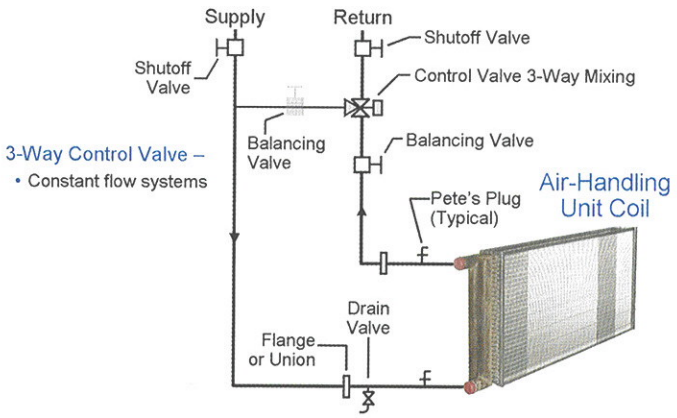


Figure 56

3-Way Mixing Valve

Face and Bypass Damper Control

Another control method that may be used with water (or steam) coils is face and bypass. With face and bypass control two sets of interlocked dampers are installed, generally in one section. The face dampers are located directly in front of the cooling coil and the bypass damper is located above the coil, controlling airflow through a bypass duct around the coil. See Figure 57.

This arrangement can be used with heating or cooling coils. They control heating capacity by diverting air around the coil, rather than throttling the hot water or steam to the coil. This minimizes the possibility of coil freeze-up in cold climates.

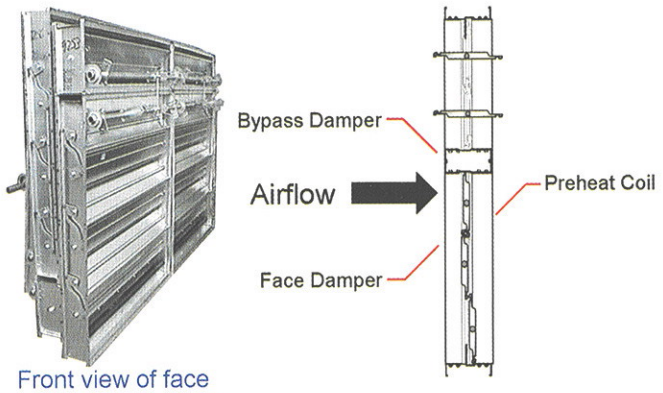


Figure 57

Face and Bypass Dampers

In cooling applications, face and bypass dampers are generally used only with chilled water coils, not direct expansion (DX) refrigerant coils. In chilled water applications they are effective in high latent load applications. In conjunction with a chilled water valve, they can reduce the leaving air temperatures off the cooling coil providing more dehumidification. This dehumidified air is then mixed with bypassed air to bring it up to an acceptable dry bulb temperature for distribution to the space.

On applications employing face and bypass sections, the fan selection and air distribution system must be designed for an air quantity about 10% above design dehumidified air volume. This additional air quantity compensates for leakage through a fully closed bypass damper and for air quantity variations when the dampers are in intermediate positions.

Traditionally, face and bypass has been discouraged in DX systems because if the controls allow the face dampers to close or significantly reduce airflow across the coil, the coil could freeze up and cause liquid refrigerant to flood back and cause compressor damage.

Often the coil used in a face and bypass arrangement will have no control valve. Temperature control is accomplished by bypassing air around the coil by fully closing the face damper. However, the radiant effect of the coil coupled with a backwash of the air after leaving the bypass can result in a residual heating effect. The amount of heat is a function of the design of the face damper and its ability to provide tight closure.

Steam Valve Control

See Figure 58 for a steam heating control diagram. When full design pressure is available, it should force the condensate out of the coil through the trap. When steam is throttled, it is possible for the pressure in the coil to be reduced to a point that will result in condensate hang-up in the coil. In some instances, the pressure may even drop into the vacuum range. At reduced steam flow, the condensate can be cooled so rapidly that coil freeze-up is virtually unavoidable. Standard or steam distributing construction is available for high and low-pressure applications. Standard steam type is the basic 5/8-inch or 1-inch tube steam coil, known as the single-tube design. The steam supply and condensate return headers and connections are normally at the opposite ends of the coil. Uniform steam distribution to each of the coil core tubes is accomplished by proper header assembly design. The steam supply connection should be located in the center of the header, with a perforated plate-type baffle located directly behind this connection. Properly sized orifices are located in each of the core tube entrances into the header. Steam distributing 5/8-inch and 1-inch tubes are the "freeze resistant" coils, known as the dual-tube design.

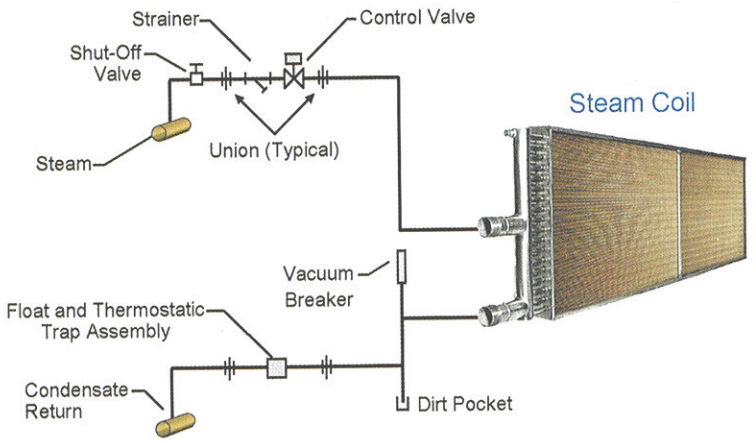


Figure 58

Steam Valve Control
(Photo courtesy of Heatcraft USA)

Steam Coil Drainage

Any steam coil can freeze when exposed to freezing temperatures without the proper drainage of the condensate.

This coil design utilizes a smaller inner tube, with precisely spaced, directional, orifice-type perforations. This is to help direct condensate flow to the return header.

U-bend (standard) steam coils can be used in applications where freeze protection is not a concern. This type of steam coil can be used when entering air temperatures are a minimum of 38° F. Standard coils should be controlled with only 2-way, on/off control valves so that condensate is not allowed to move so slowly (if throttled) that it could be susceptible to freezing.

Steam coils require a device to control the discharge of condensate from the coil but in the same time prevent any live steam from getting out. This device is called a trap and there are several types available. Traps are valves that sense the difference between steam and condensate and react automatically. There are two main types, those that are thermodynamically controlled, and those that are mechanically controlled. A widely used thermodynamically controlled trap uses a disc that closes to steam vapor, but opens to condensate which moves at a lower velocity than the vapor. Other thermodynamic types use pressure, liquid expansion, or a bi-metal element to function.

The mechanical trap may utilize a ball float or float and lever to prevent steam from passing through the coil unless it has condensed.

Electric Heater Control

In order to achieve incremental control of the heater output, heaters with multiple electrical circuits (stages) may be used. Although normal applications seldom require more then three stages, larger heaters may have six or more stages. The following devices may be used to control the electric heaters.

Step or program controllers generally operate heaters with more than three steps. They are normally operated by a modulating room or duct thermostat which energizes a varying number of steps depending on the deviation from the control set point. Depending on the type of step controller, various inputs are used, such as 0-135 ohm potentiometer, 0-16 vdc, 0-15 psi pneumatic, etc.

A silicon control rectifer (SCR) is a solid-state device that controls by modulating the flow of alternating current to heating components wired to the SCR. Depending on the capacity of the heater, several SCRs may be required. For close control, the SCRs offer the best system. However, in larger capacity heaters, the initial investment in SCRs can be expensive. In such a case, a step control-SCR vernier system should be considered.

A step control-SCR vernier system combines the advantages of the relatively low cost of the step controller with the modulation obtainable with SCR control. The system may use electric, electronic, or pneumatic step controllers and may be operated by various control signals. The first step on and the last step off would be the SCR step.

Most of the components used for electric heaters can also be built into a remote control panel. Remote control panels are used when code requirements do not allow certain features to be built into the terminal box of a heater, or the terminal box is in an inaccessible location when it is desired to have all control components in a readily accessible situation.

Coil Freeze Protection Considerations

The exposure of hot water, chilled water and steam preheat or reheat coils to subfreezing temperatures, either by accident or intent, creates the possibility of freeze-up within the coil tubes and may cause costly damage. The prevention of freeze-up requires consideration of the problem in the apparatus design and layout, in the selection of the equipment, and in the choice of the control methods.

Greater amounts of outdoor air are required by the current ASHRAE 62 standard than in previous years to maintain good IAQ. Elevated outdoor amounts are required to properly dilute indoor contamination levels and provide occupant comfort. However, with increased outdoor amounts comes the potential to freeze the coil in winter. Therefore, knowledge of the methods to prevent coil freeze-up is important.

Freezestat

A device that is often used is a freeze thermostat (or freezestat). This is used for coils that are not intended to heat entering air below 32° F. A freezestat is a coil sensor that is positioned across the face that shuts down the fan and closes the outdoor damper if the coil temperature reaches 35° F.

Air Blender

One method for freeze-up prevention is to ensure uniform entering air temperatures across the face of the coil. When the mixing of outdoor and return air takes place upstream of a heating coil, mixing should be promoted by introducing the denser and colder outdoor air at the top of the plenum. Provide as much airway length as possible. Use of an air blender is also recommended. Air handler manufacturers can provide a high quality air blender downstream of the mixing box as shown in Figure 59. The air blender helps mix the outdoor and return air by creating a churning action in the airstream. The airside pressure drop is not large, typically less than 0.25 in. wg. Blenders require little or no maintenance, however, they can increase unit length.

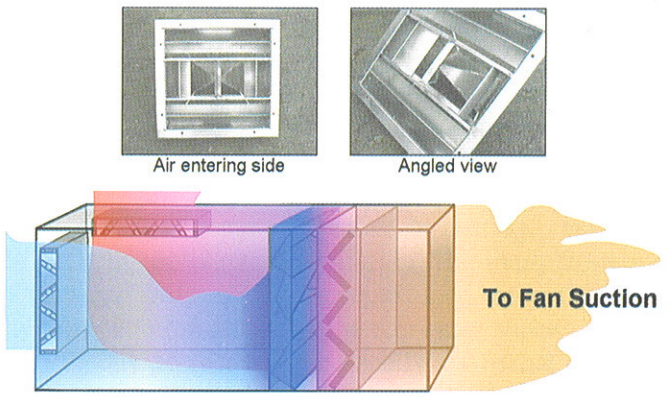


Figure 59

Air Blenders

Antifreeze Solution

A method of freeze prevention is to use an antifreeze mixture in the system. Glycol/water antifreeze solutions are capable of providing two levels of protection for the water system: burst protection and freeze protection. Burst protection allows for the fluid in the chilled-water loop to become slushy, but never to fully freeze solid to the point of rupturing or collapsing tubing. Freeze protection requires a greater glycol concentration so that the chilled fluid will not freeze. Adding glycol lowers the temperature at which the fluid freezes.

Antifreeze is expensive but is always protecting the system. Manufacturers often prefer this method of freeze protection also because it is very effective. However, the heat transfer of the coil is negatively affected depending on the percentage of antifreeze and type used.

Preheat with Energy Recovery

Another option is to provide preheating of the incoming ventilation air through a recovery device. This saves energy and introduces mixed air to the system at a higher temperature. There are many forms of air-to-air energy recovery available to transfer heat from the exhaust air stream to the incoming ventilation airstream. Coil runaround loops directly involve the use of separate coils in the exhaust and supply airstreams. For a complete discussion on systems that recover energy, see TDP-645, Energy Recovery.

Pumped Coils

Some designers like to use a pumped coil for freeze protection. Moving water will not freeze as quickly as standing water. The pump is activated to run continuously when the temperature of the outside air is at or below freezing. Figure 60 shows a pumped coil system schematic.

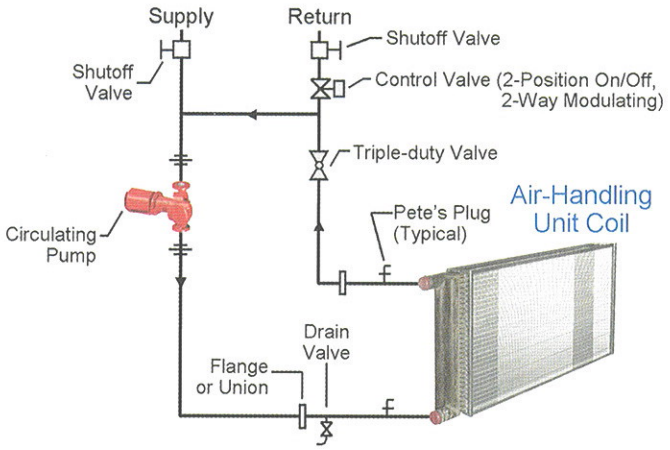


Figure 60

Pumped Coil

Steam Coil Considerations

Single-tube steam and hot water coils may be used for the tempering of subfreezing air, but the heating medium should not be modulated at entering air temperatures below 35° F. The steam should be supplied from the entering airside of the coil. Use of an inner distributing type coil is recommended.

A basic requirement for adequate freeze-up protection of steam coils is the positive and complete drainage of condensate from the tubes. Any type of steam coil may be damaged if condensate is allowed to accumulate and freeze due to poor design of the system or coil. For this reason an ideal position for a steam preheat coil is with the tubes vertical and with the condensate header at the bottom. For either horizontal or vertical airflow, steam preheat coils installed with tubes horizontal should be pitched downward toward the condensate header to facilitate drainage. Many steam distributing tube coils feature tubes internally pitched for either horizontal or vertical airflow. The use of a properly sized vacuum breaker can prevent hang up of condensate in the tubes.

For either steam or hot water, it is wise not to use a throttling valve to control the coil if the entering air temperature to the coil can be less than 32° F. The reduced flow and resulting low velocity in the tubes can increase the chances of freeze-up.

Cooling Coil Design Parameters

Load Estimation and Coil Selection

There are three important reasons for accurately estimating building cooling and heating loads and selecting coils and/or equipment to match those loads as closely as possible. They are:

- Equipment selection can be optimized.
- Part-load stability can be assured.
- Operating cost can be minimized.

From an equipment standpoint, unfortunately, it is common practice in heating, ventilating and air conditioning to overestimate loads for both cooling and heating. In fact, many designers routinely add a safety factor to the figures resulting from industry-accepted load estimating techniques. When equipment is selected for the job it is often difficult to get an exact capacity match with design load. As a result, the equipment capacity decided upon is usually above that indicated by the load estimate. The overestimate that can occur during load estimation is compounded during equipment selection. This practice adds extra cost and oversizes the job twice. Extra safety factors can result in part load capacity control problems and potential loss of humidity control resulting in IAQ problems.

For air handlers, moderately overestimating additional capacity can result in increased coil capacity while the air handler and compression equipment size remain unchanged. If the overestimate is more severe, a larger air handler size may be required. This could cause an approximate 10% air handler equipment cost increase, thereby possibly affecting the compression/condensing equipment size also.

Total installed cost increases if there is an unnecessary size increase in the air handler, compression or condensing equipment. The cost for electrical service, air ducting, air distribution terminals, refrigeration piping, rigging and installation labor are responsible for the higher cost.

Capacity Steps

The more closely the capacity step can match the coil load (i.e., the smaller the capacity step), the more smoothly and efficiently the system can operate at partial load. This situation is aided by avoiding equipment oversizing, and by selecting compression systems with more steps of capacity control.

The recent increases in the amount of ventilation air recommended in ASHRAE Standard 62-2004 have increased the importance of selecting and controlling cooling coils and air-conditioning equipment for optimum performance at part load. As with all air-conditioning equipment, the air-cooling coil has a limited, stable operating capacity range. Oversizing of equipment wastes some of this stable operating range.

Coil Psychrometrics

Illustrated on a psychrometric chart on Figure 61 is the airside performance of a 6-row, direct expansion cooling coil. The airside performance curve would look similar for a chilled water coil. The saturated evaporator temperature is the refrigerant temperature matching the pressure at the leaving end of the evaporator. The entering coil condition is t_{ma} for a draw-thru air-handling unit application. The numbered points (2 through 6) represent the air conditions leaving each row of the coil. The rows near the entering air face of the coil provide the greatest sensible and total capacity. The change for each row of finned tubes in dry bulb temperature and enthalpy, respectively, demonstrates this fact. Cooling coils first drive the air being treated toward the saturation curve and then cause the air leaving additional rows to move farther down along this curve. The last few rows the air passes over are providing primarily latent capacity. For this reason, latent capacity produced by coils with many rows is a higher percentage of total capacity than that produced by coils with fewer rows.

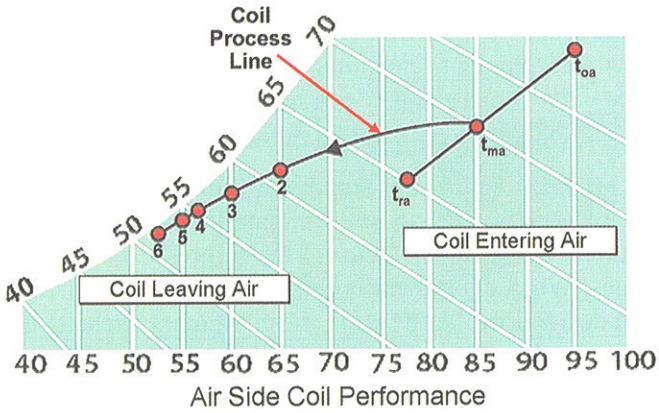


Figure 61

Psychrometric Chart – Coil Process Line

For a detailed explanation of coil psychrometrics, see TDP-201, Psychrometrics, Level 1: Introduction and TDP-204, Psychrometrics, Level 4: Theory.

Cooling Coil Requirements

The lines plotted on the psychrometric chart in Figure 62 represent the various processes that the air undergoes as it travels through the system. The two processes of particular interest here are the room process and coil process.

The room process line is important because it represents the effect the space sensible and latent loads have on the space temperature and humidity. The cooling coil must provide capacity great enough to offset all sensible and latent loads that occur in the building. If the coil sensible capacity is less than the sensible load generated, the space dry bulb temperature will rise above the design level. If latent capacity is inadequate, space relative humidity will climb above the design level.

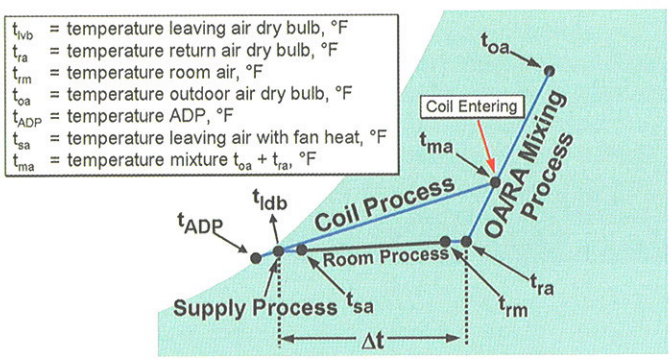


Figure 62

Psychrometric Chart – Room Process Line

The ratio of sensible and latent loads that occur in the space establishes the slope of the room process line. Since total load is the sum of the sensible and latent loads, the ratio may also be expressed in terms of sensible to total load. This is called the room sensible heat factor (RSHF). Stated in terms of an equation, RSHF = room sensible load/ room total load. Therefore, the slope of the room process line follows the RSHF line. The higher the RSHF, the lower the slope. High latent loads (low RSHF) will have steep RSHF and room process lines.

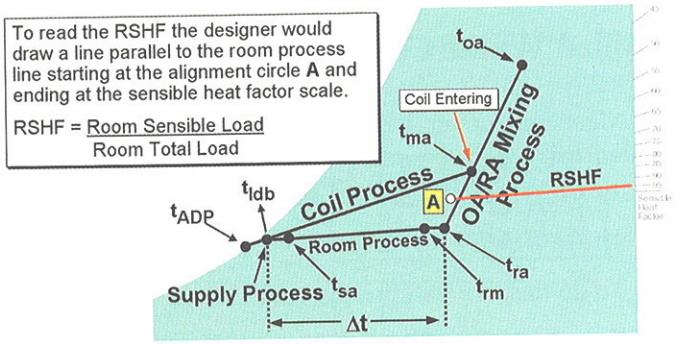


Figure 63

Cooling Coil Requirements

Sensible Heat Transfer

The RSHF concept is important because it is used to establish the temperature at which the coil must operate to properly handle the load in the correct proportions.

If the room process line is drawn from the room design point through the saturation curve on the psychrometric chart, the point at which the room process line intersects the saturation curve is called the apparatus dew point (ADP). This defines the average surface temperature at which the cooling coil must operate to handle the loads. The higher the latent load, the steeper the room process line and the lower the resulting ADP.

Coil Selection Examples

The coil apparatus dew point (average surface temperature) has a significant impact on the selection of the cooling coil. The ADP establishes the leaving water temperature that the chiller must supply, or the saturated suction temperature at which a condensing unit must operate. It also helps in selecting the coil airflow rate and air handler size. After the air handler size is selected, the cooling coil and heating coil rows, fins and circuiting can be determined based on the requirements of the job.

Coil Selection

The wide range of coils offered in central station air handlers allow the designer to closely match the sensible and latent capacity and other input parameters to the appropriate loads.

When one parameter is satisfied, other parameters will often miss the requirement. Common practice in these situations is to select a system and dehumidified air quantity that will match total sensible heat load with equivalent sensible capacity. Total capacity will be allowed to exceed or run slightly below. In this situation, any excess total capacity that occurs at peak load (design) conditions will result in room relative humidity below the original designer's target. This is considered desirable with respect to comfort applications.

Most commercial load estimating programs today will also do the psychrometric calculations needed for coil and system component selection. Shown is a load estimate summary with the coil selection data highlighted.

Chilled Water Coil Selection

Most coil or air handler manufacturers offer software to select their coils. Let's select a chilled water coil to satisfy the load requirements shown. For purposes of this exercise, we will use the Carrier AHUBuilder® software.

The first step is to select an air handler/coil size.

To do this, we will use the formula:

Face Area = Airflow (cfm)/design face velocity

Coil Face Area = 8758/500 = 17.5 ft^2 → Select a size 17 air handler.

Enter the selection parameters from the load estimate for entering air and water. The program will also allow the user to select the balancing criteria for both the airside (total capacity, sensible capacity), and the waterside (gpm or fluid temperature rise). See Figure 64.

The coil performance data for all coils will be displayed for analysis. Our goal is to select the most economical coil that best meets the job requirements.

To make evaluation easier, AHUBuilder has a "4-item browse mode" which sorts the data by user selected parameters. This allows the designer to quickly and easily compare the most pertinent data.

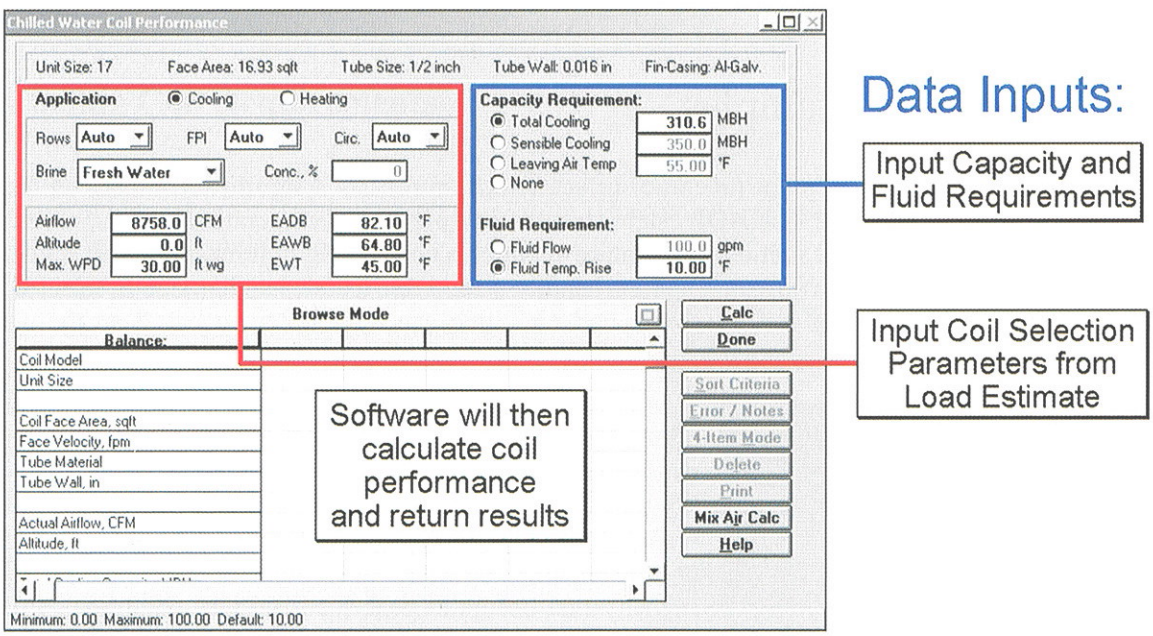


Figure 64

AHU Builder Selection Software – Chilled Water Coil

For this application, two coils closely meet the load requirements as shown in Figure 65. They are an 8-row, 11-fin per inch, double-circuited coil and a 6-row, 14-fin per inch, full-circuit coil. We will choose the 6-row coil as it has a lower first cost and the air friction is less than that of the 8-row coil. This will save fan energy and result in a more energy efficient system. The chosen coil has a higher pressure drop but better latent capacity.

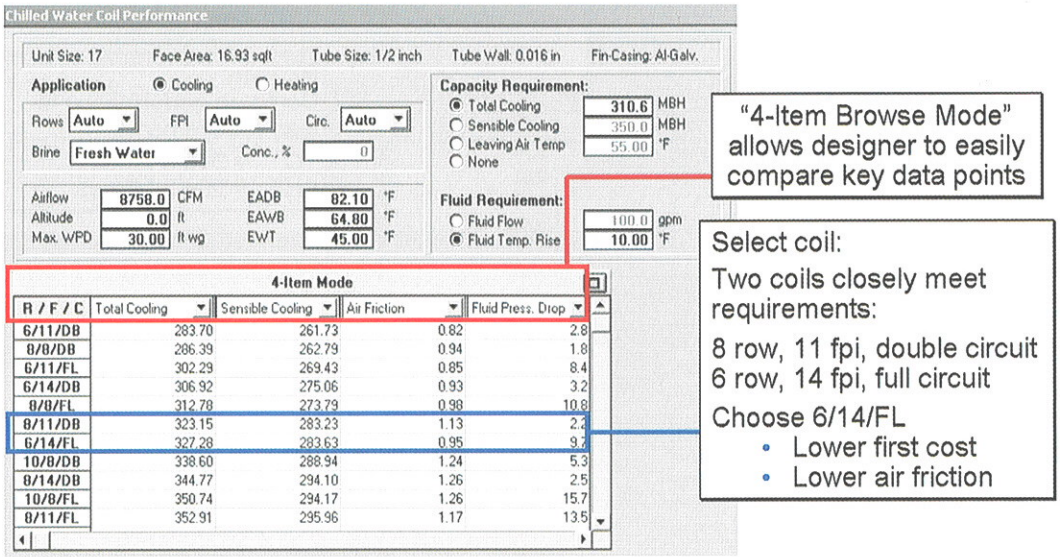


Figure 65

Chilled Water Coil Selection

Direct Expansion Coil Selection

Now let's select a direct expansion (DX) coil for this same job. The inputs will be similar to the chilled water coil, with a few differences. For DX coils, face or row split must be selected. The program will also balance the coil with a condensing unit. We will choose a nominal 30-ton condensing unit. See Figure 66.

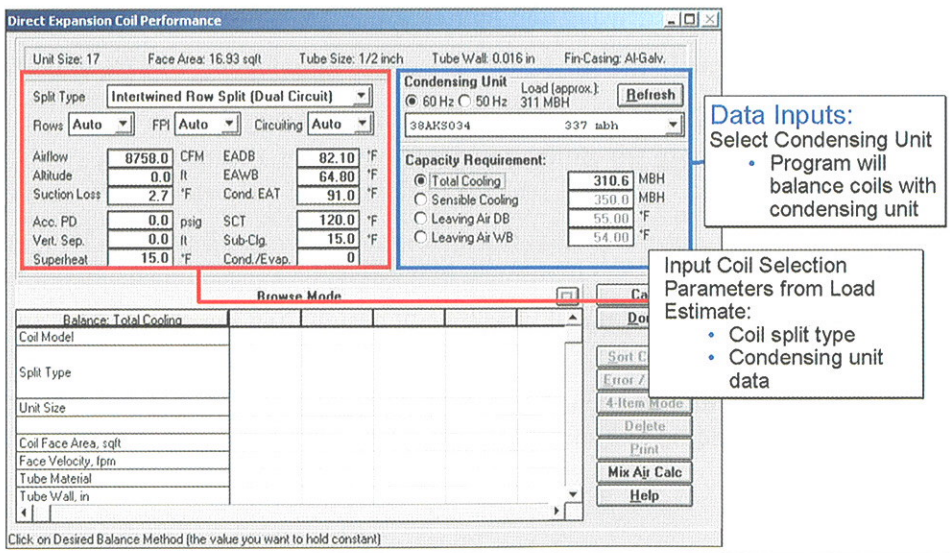
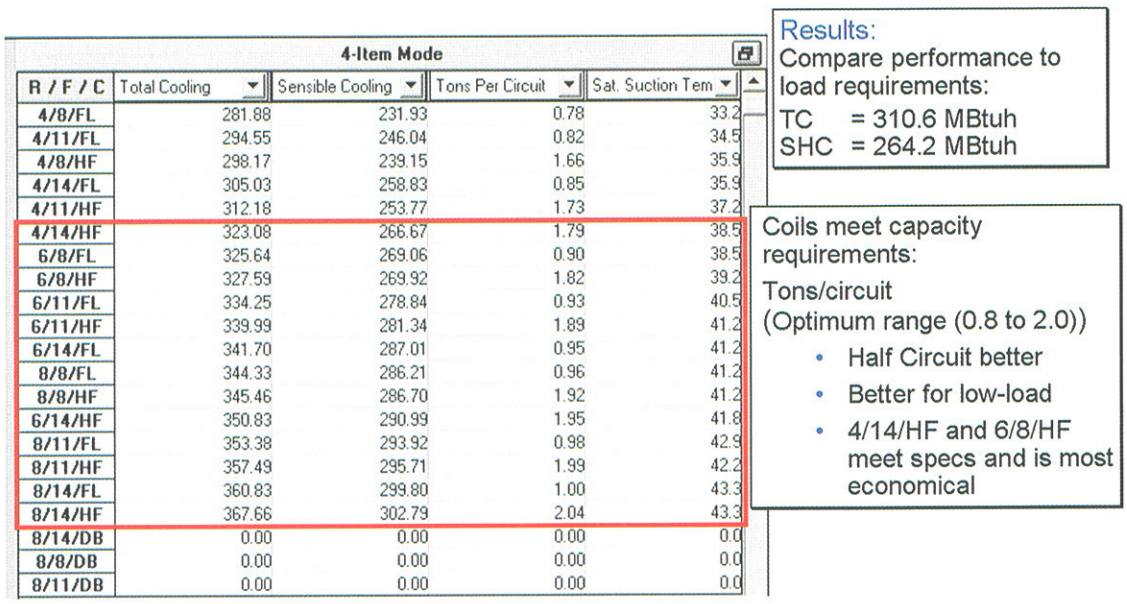


Figure 66

DX Coil Selection

Shown in Figure 67 are the results sorted for capacity, tons/circuit and saturated suction temperature. Looking at the data, many coils meet the capacity requirements. However, the half-circuited coils have higher circuit loading still within the optimum range of 0.8 to 2.0 tons per circuit. They make a better choice as they may eliminate the need for evaporator staging. Looking at the data, the two best matched coils are the 4-row, 14-fin per inch half-circuit coil and the 6-row, 8-fin per inch, half circuit. The 4-row coil is the most economical as we saw earlier.



4-Item Mode

R / F / C	Total Cooling	Sensible Cooling	Tons Per Circuit	Sat. Suction Tem
4/8/FL	281.88	231.93	0.78	33.2
4/11/FL	294.55	246.04	0.82	34.5
4/8/HF	298.17	239.15	1.66	35.9
4/14/FL	305.03	258.83	0.85	35.9
4/11/HF	312.18	253.77	1.73	37.2
4/14/HF	323.08	266.67	1.79	38.5
6/8/FL	325.64	269.06	0.90	38.5
6/8/HF	327.59	269.92	1.82	39.2
6/11/FL	334.25	278.84	0.93	40.5
6/11/HF	339.99	281.34	1.89	41.2
6/14/FL	341.70	287.01	0.95	41.2
8/8/FL	344.33	286.21	0.96	41.2
8/8/HF	345.46	286.70	1.92	41.2
6/14/HF	350.83	290.99	1.95	41.8
8/11/FL	353.38	293.92	0.98	42.9
8/11/HF	357.49	295.71	1.99	42.2
8/14/FL	360.83	299.80	1.00	43.3
8/14/HF	367.66	302.79	2.04	43.3
8/14/DB	0.00	0.00	0.00	0.0
8/8/DB	0.00	0.00	0.00	0.0
8/11/DB	0.00	0.00	0.00	0.0

Figure 67

DX Coil Selection Results

Now, check the selected coil at part-load operation with selected condensing unit to determine if any coil split deactivation is required with the selected condensing unit. Our example will be a 4-row, 14-fin, half-circuit coil that contains 2 TXVs. To check this, it is necessary to get the unloading capability from the condensing unit literature.

The selected condensing unit, the 38AKS034 is capable of unloading to 67% of design.

Tons/Ckt at part load = Full Load Tons/Ckt * Minimum Unloading

= 1.79 * 0.67

= 1.19

This results in acceptable tons per circuit with two active TXVs.

Result must be above the minimum allowed by coil design. (0.6 ton/ckt for the Carrier coil in this example.)

If resulting tons per circuit had been below the minimum, one split of the coil would have to be deactivated by closing a liquid line solenoid valve to one TXV when compressor is unloaded.

Heating Coil Selection

Hot Water Coil

Selecting a hot water coil involves entering minimum required data after which the selection software will arrive at an optimized coil selection. See Figure 68.

First, the entering air temperature and the entering water temperature are required.

> ***Oversized Heating Coils***
>
> *Often, after the air-handling unit size is fixed, the resulting heating coil face area tends to be a bit oversized. That is because the cooling coil selection face velocity of approximately 500 fpm is low relative to acceptable heating coil face velocities that can be greater than 900 fpm.*

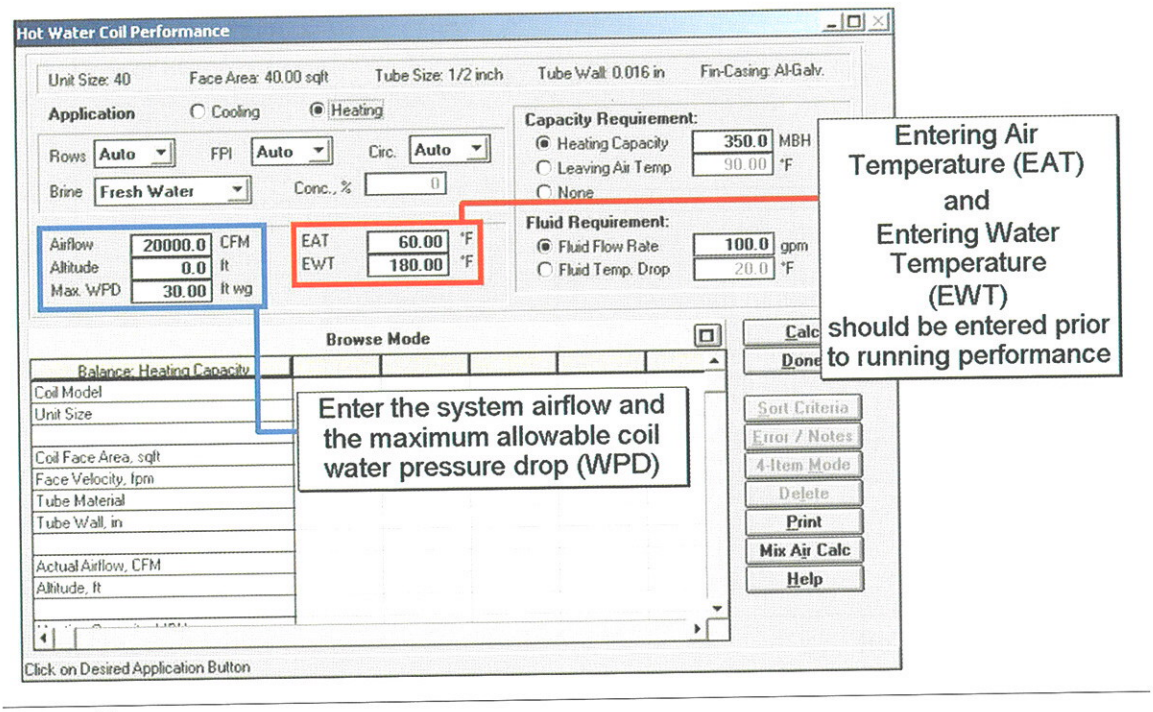


Figure 68

Hot Water Coil Selection

Next, the required capacity is entered along with either the gpm or the desired fluid temperature difference. We will use 100 gpm in this example.

Several coil row and fin combinations are displayed. As we see, our flow rate of 100 gpm results in far too much coil capacity.

If we reduce the flow to only 10 gpm through the coil, we can make the required capacity. The software allows the user to list coil performance details in a specific order of importance. In this case, the priority chosen was:

1. heating capacity
2. leaving air temperature
3. air friction
4. flow rate

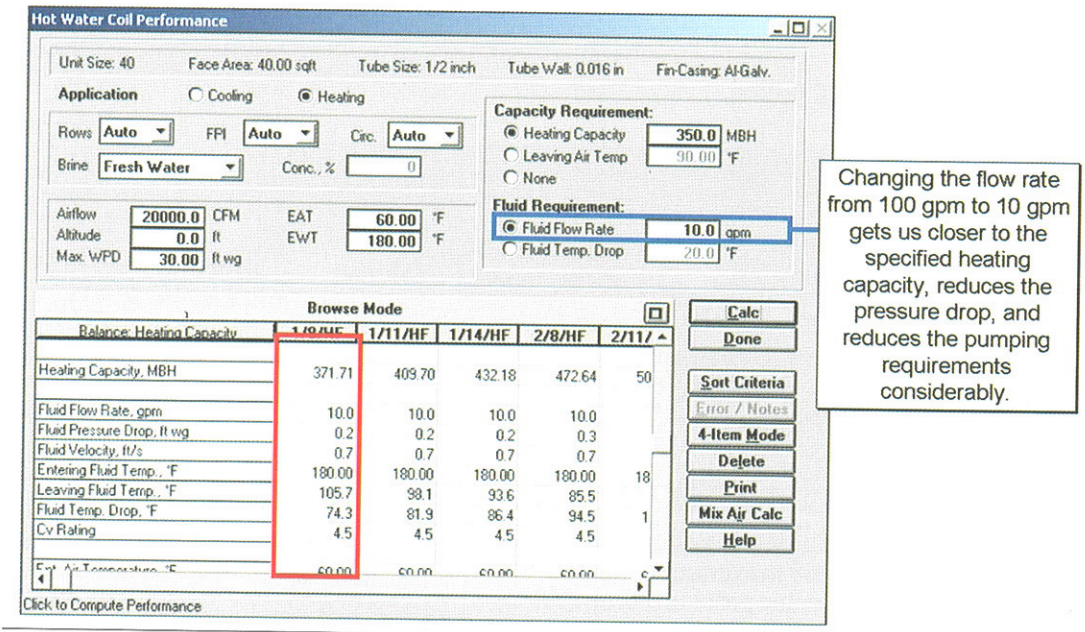


Figure 69

Hot Water Coil Selections

Electric Heating Coil

To make a selection for an electric heating coil, the user must first input the voltage for the heater and then select between an open wire or a sheathed element heater. See Figure 70.

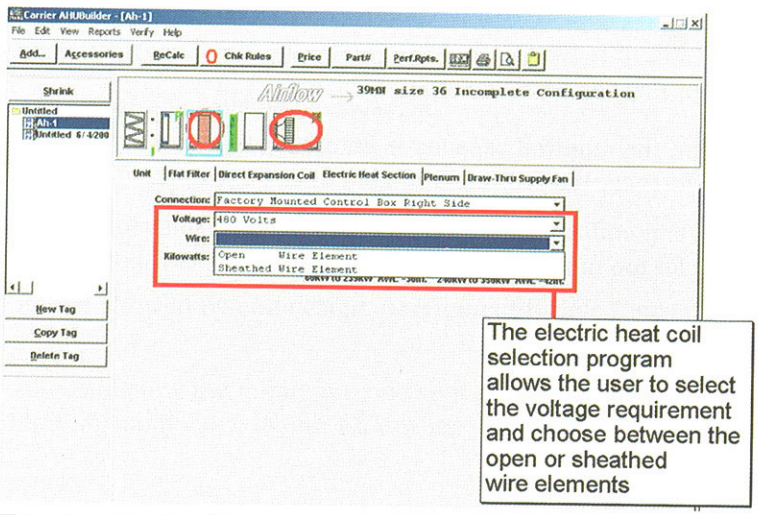


Figure 70

Electric Heat Coil Selection

Next, the desired kW is input. Once again, the air handler size has been previously fixed based on the cooling coil face velocity so the airflow has already been established. Standard heater sizes are displayed for that size of unit.

Steam Heating Coil

For a steam coil selection, the following parameters are input first:

- airflow
- altitude
- entering air temperature
- available steam pressure

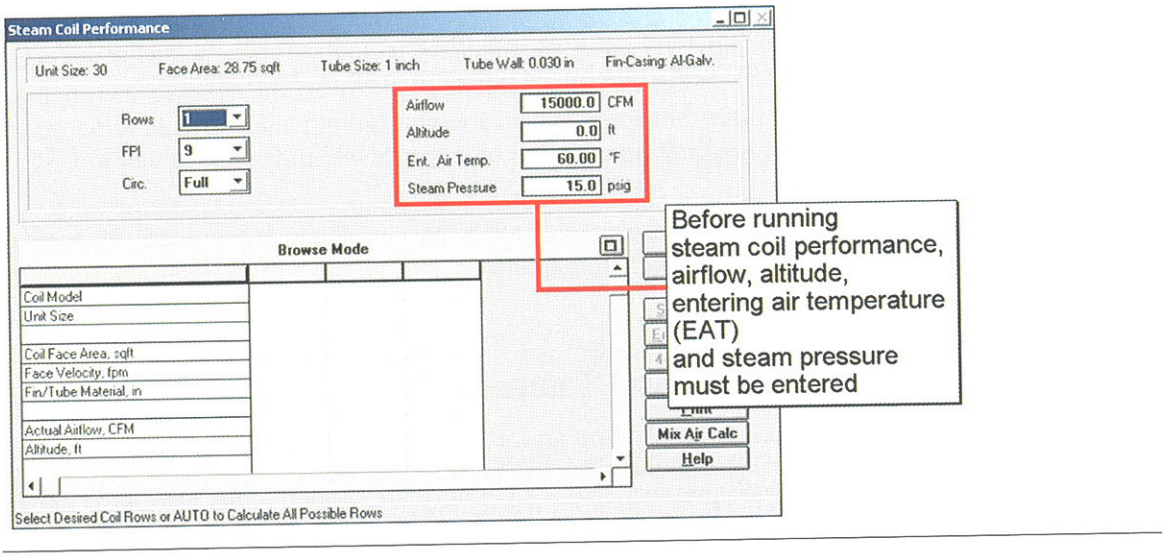


Figure 71

Steam Coil Selection

Coils are calculated and the heating capacity, leaving air temperature, air friction and, in the case of a steam coil, the condensate load in lb/hr is output.

Preheat Coils with Face-and-Bypass

Face-and-bypass dampers are typically used with preheat coils. The preheat coil is sized according to heating capacity requirements and coil face velocity limits.

Face-and-bypass dampers are commonly used in colder climates where there is a greater threat of coil freeze-ups. Full hot water or steam flow can be maintained through the coil while air is re-directed around the coil through the bypass dampers for temperature control.

ARI Certification and Coil Testing

ARI (Air Conditioning and Refrigeration Institute) is an independent non-profit organization. One of the main goals of ARI is to ensure that heating and cooling equipment delivers the performance ratings represented in catalog literature and computerized selection software. ARI has developed Standard 410, which establishes rating criteria and procedures for measuring and certifying product performance. Cooling and heating coil parameters like capacity, air friction, and water head loss are tested (within tolerances) to what the manufacturer says they will be. In this way, products are rated on a uniform basis, so that buyers and users can properly make selections for specific applications.

To qualify for ARI certification, coils are subjected to laboratory performance testing by the manufacturer. Once ratings under simulated (or actual) operating conditions have been obtained, they are submitted to ARI for verification. Chilled water, hot water, and direct expansion coils are included. Once certified, the ARI seal may then be affixed to the coil casing. ARI annually selects a significant portion of each manufacturer's production models to be tested by an independent laboratory under contract to ARI to ensure ongoing compliance with ARI guidelines.

Coil Testing, Proof and Leak Test

Coils are quality tested for defects during and after the manufacturing process. A typical proof test procedure for a manufactured coil would be to submerge the coil in water and pressurize it to 300 psig and check for leaks.

Working and Design Pressure and Temperature

The proof leak tests must exceed the typical coil design pressure or the maximum allowable working (operating) pressure. Since pressure and temperature are related, coils also have maximum allowable temperature limitations. Working pressure and temperature are the actual pressure and temperature the coil will be subjected to in everyday use. The Design pressure and temperature are the maximum allowable working pressure and temperature for the construction.

Standard ARI 410 requires testing by a manufacturer on a representative coil within a family of identically designed coils. That data is used to generate capacity data for other coils within the same family type.



Figure 72

Coil Testing

Summary

Coils serve the important function in an HVAC system of transferring heating or cooling from the water or refrigerant to the conditioned airstream. Each type and application of an HVAC system can utilize different coils. Coils are used in residential equipment, indoor packaged products, rooftop units, ductwork, air terminals (VAV boxes), and central station air-handling units. Air-handling units have the most variation in coil applications, so this TDP is focused on that category.

To properly apply heating and cooling coils, a knowledge of their construction and important features is necessary. Each coil component (including rows, fins, headers, inlet, outlet, circuiting, and face area) affects the final performance of the coil. With an understanding of coil components, a proper coil selection is possible.

The main types of coil that are used in the HVAC industry are direct expansion (DX) and chilled water for cooling. Hot water, steam, or electric coils can be used for heating duty. DX coils are matched with individual condensing units. The condensing units can be air or water-cooled. DX systems utilize refrigerant lines between the coil and the condensing unit. Chilled water coils utilize water piping, which can distribute the water over greater distances than DX coils. For that reason, chilled water is used in the larger buildings utilizing multiple coils.

The basic heat transfer properties of coils are airside transfer, waterside transfer, and an overall formula. The airside heat transfer should equal the waterside heat transfer. There are many factors that affect heat transfer, such as air film resistance, external fouling, tube and fin resistance, and fluid film resistance.

There are common application topics that affect coil selection and design. Important topics are moisture carryover, effects of antifreeze, corrosion protection, and condensate trapping.

The coil must be selected after the cooling and heating loads for the area being served have been established. The ratio of sensible and latent loads in the space establish the slope of the room process line and coil process line on the psychrometric chart. The leaving coil conditions reflect the correct ratio of sensible to total heat based on the load requirements. This is called sensible heat factor.

Selection of heating and cooling coils is done by computer software. The user must comprehend the coil properties and performance characteristics presented in this TDP module since most selections yield several alternatives, one of which may be the best overall.

A review of common coil piping methods and agency certification and testing methods for heating and cooling coils establishes a solid foundation on coils in the HVAC industry.

Work Session

1. Name five typical HVAC application types where coils are used.

_____

_____

_____

_____

_____

2. What is the difference between a draw-thru and blow-thru air handler configuration?

_____

_____

_____

3. What is the most popular fin and tube material combination applied to comfort commercial air cooling coils?

_____

4. Name the typical tube diameters used in HVAC coils.

_____

5. Using the coil picture shown, identify the following things:

a) Vent _____

b) Face area _____

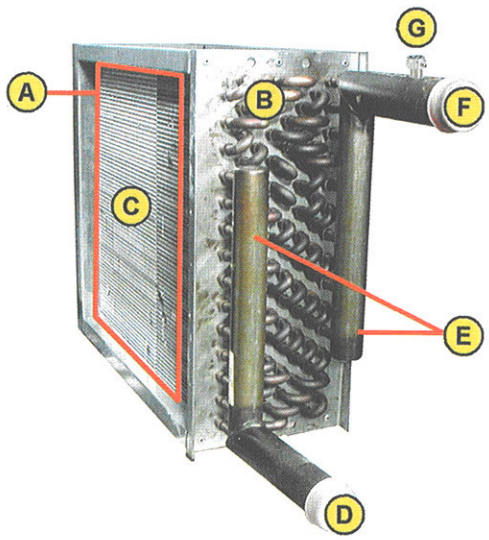
c) Outlet _____

d) Coil tubes _____

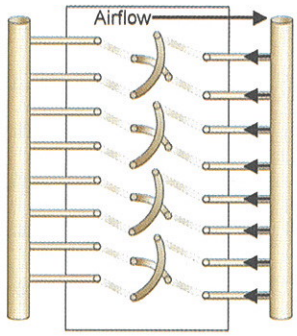
e) Coil fins _____

f) Supply and return headers _____

g) Inlet _____



6. For the coil shown here,



   a) What circuiting does it have? _____
   b) How many rows does it have? _____
   c) Is it a water or DX coil? _____
   d) What is its tube face? _____

7. Should the water inlet be located on the side of the coil that air enters or the side of the coil that air leaves and why?

_____

_____

_____

8. Describe what is meant by non-ferrous headers.

_____

_____

9. A designer has selected an air handler with a 4-row coil which is available in either half or full circuiting. The tube face of the coil is 8.

   a) Which circuiting has the longest circuits? _____
   b) How many passes does the refrigerant make in each circuit of the full-circuited coil? In the half-circuited coil? _____
   c) Which circuiting will have the higher tube-side pressure drop? _____
   d) Why? _____

_____

_____

10. Why are coil splits offered on DX coils but not chilled water coils?

_____

_____

_____

11. From most expensive to least, list the cost ranking factors for coils.

    Highest
    
    Lowest

    _____
    _____
    _____
    _____

12. What are the three things that limit the minimum load of a DX system at part load?

    _____
    _____
    _____

13. The general equation for heat transfer across the finned tube surface of a heat exchanger like a chilled water or DX air cooling coil is $q_t = U * A * LMTD$. What component is directly influenced by dirt build-up on the outside of the cooling coil?

    _____
    _____

14. In the equation shown above, why is a simple Δt not adequate?

    _____
    _____

15. What is a typical design saturated refrigerant temperature for the evaporator of a DX system?

    _____

16. What is a typical design water temperature entering a chilled water coil?

    _____

17. Accurate load estimation and equipment sizing is beneficial in three major areas. What are they?

    _____
    _____
    _____

18. The actual cooling coil airside performance curve is not straight. In a 4-row coil, the row nearest the entering air surface of the coil provides _____% of the cooling capacity.

19. Which is an accurate statement for bypass factor? _____

    a) A decimal expression of the coil's airside heat transfer inefficiency

    b) A decimal expression of the coil's airside heat transfer efficiency

    c) A decimal expression of the air that gets through coil untreated

    d) A decimal expression of the air which leaks through the mixing box dampers when closed

20. Name six methods used for heating coil freeze protection.

_____

_____

_____

_____

_____

_____

Notes:

Appendix

Work Session Answers

1. Five typical coil applications are:
   - Residential systems
   - Commercial packaged units
   - Duct-mounted coils
   - Air terminals, such as VAV boxes
   - Central station air-handling units

2. Draw-thru configuration positions the coil upstream of the fan and motor. Blow-thru configuration is downstream of the fan and motor. Blow-thru needs a diffuser plate to help distribute the air coming off the fan discharge evenly over the coil.

3. Copper tubes/aluminum plate fins

4. 5/16, 3/8, 1/2, 5/8, 3/4, and 1 inch

5. a) Vent G
   b) Face Area A
   c) Outlet F
   d) Coil tubes B
   e) Coil fins C
   f) Supply and return headers E
   g) Inlet D

6. a) full
   b) 4
   c) water
   d) 8

7. The water inlet should be located on the coil on the side that air leaves the coil in order to accomplish counterflow to maximize LMTD (heat transfer)

8. Non-ferrous headers contain no iron or steel and are typically made of copper.

9. a) half
   b) 4, 8
   c) half circuited coil
   d) Because every other tube in the face is fed with water so the individual circuits have a longer path of flow within the coil, hence more pressure drop.

10. Water coils use either flow modulation or water temperature adjustment as a capacity control technique. For that reason, splitting the coil is not a necessary method to achieve capacity control.

11. a) face area
    b) rows
    c) fins per inch
    d) circuiting

12. The three things that limit the minimum load of a DX system at part load are:
    - The TXV low load limit
    - The nozzle's low load limit capability
    - The circuits in the coil ability to entrain oil (expressed in terms of tons/ckt)

13. The overall conductance value or the "U" value.

14. Because both air temperature and heat transfer fluid temperature are changing as they flow through the coil. This means the temperature difference at any point is different from that at another point.

15. Typically 40 to 45° F

16. 42 to 44° F

17. The three major areas where load estimation and equipment sizing is beneficial are:
    - Equipment load can be optimized
    - Part load stability can be assured
    - Operating cost can be minimized

18. 41%

19. c)

20. Six methods used for heating coil protection are:
    - use of an antifreeze mixture in the circuiting
    - drainage of the coil in winter
    - use a pumped coil (constant circulation)
    - use of a face and bypass damper with constant flow of fluid through the coil
    - use of an air blender to ensure outdoor air and return air mixing
    - use of a smaller coil dedicated to the outdoor air intake

Range of Standard Rating Conditions

	Cooling Coils			Heating Coils		
Item	Volatile Refrigerant	Cold Water	Cold Ethylene Glycol Solution	Steam	Hot Water	Hot Ethylene Glycol Solution
Standard air face velocity, std. ft/min	200 to 800	200 to 800	200 to 800	200 to 1500	200 to 1500	200 to 1500
Entering air dry-bulb temperature, °F	65 to 100	65 to 100	65 to 100	-20 to 100	0.0 to 100	-20 to 100
Entering air wet-bulb temperature, °F	65 to 80	60 to 85	60 to 85	–	–	–
Tube-Side fluid velocity, std. ft/s	–	[1]1.0 to 8.0	[2]1.0 to 6.0	–	[1]0.5 to 8.0	[2]0.5 to 6.0
Entering fluid temperature, °F	–	35 to 65	0.0 to 90	–	120 to 250	0.0 to 200

[1] Application of coils at low velocity can lead to excessive fouling.
[2] Application of coils at low velocity can lead to possible air entrapment

Notes:

Prerequisites:

To obtain the highest benefit from this module, it is suggested that participants have prerequisite knowledge from the TDPs listed below, or equivalent.

TDP No.	Book Cat. No.	Instructor CD Cat. No.	Title
TDP-102	796-026	797-026	ABCs of Comfort
TDP-103	796-027	797-027	Concepts of Air Conditioning
TDP-201	796-030	797-030	Psychrometrics, Level 1: Introduction
TDP-301	796-034	797-034	Load Estimating, Level 1: Overview
TDP-400	796-037	797-037	Principles of Mechanical Refrigeration, Level 1: Fundamentals
TDP-611	796-049	797-049	Central Station Air Handlers

Learning Objectives:

After reading this module, participants will be able to:

- Identify where coils are used in HVAC applications.
- Define the component parts of coils.
- Describe the various types of heating and cooling coils and their applications.
- Identify the heat transfer properties of coils.
- Describe methods for piping and control of water and direct expansion coils.
- Identify key coil selection criteria and factors affecting performance.
- Select chilled water, DX, and heating coils.

Supplemental Material:

None

Instructor Information

Each TDP topic is supported with a number of different items to meet the specific needs of the user. Instructor materials consist of a CD-ROM disk that includes a PowerPoint™ presentation with convenient links to all required support materials required for the topic. This always includes: slides, presenter notes, text file including work sessions and work session solutions, quiz and quiz answers. Depending upon the topic, the instructor CD may also include sound, video, spreadsheets, forms, or other material required to present a complete class. Self-study or student material consists of a text including work sessions and work session answers, and may also include forms, worksheets, calculators, etc.



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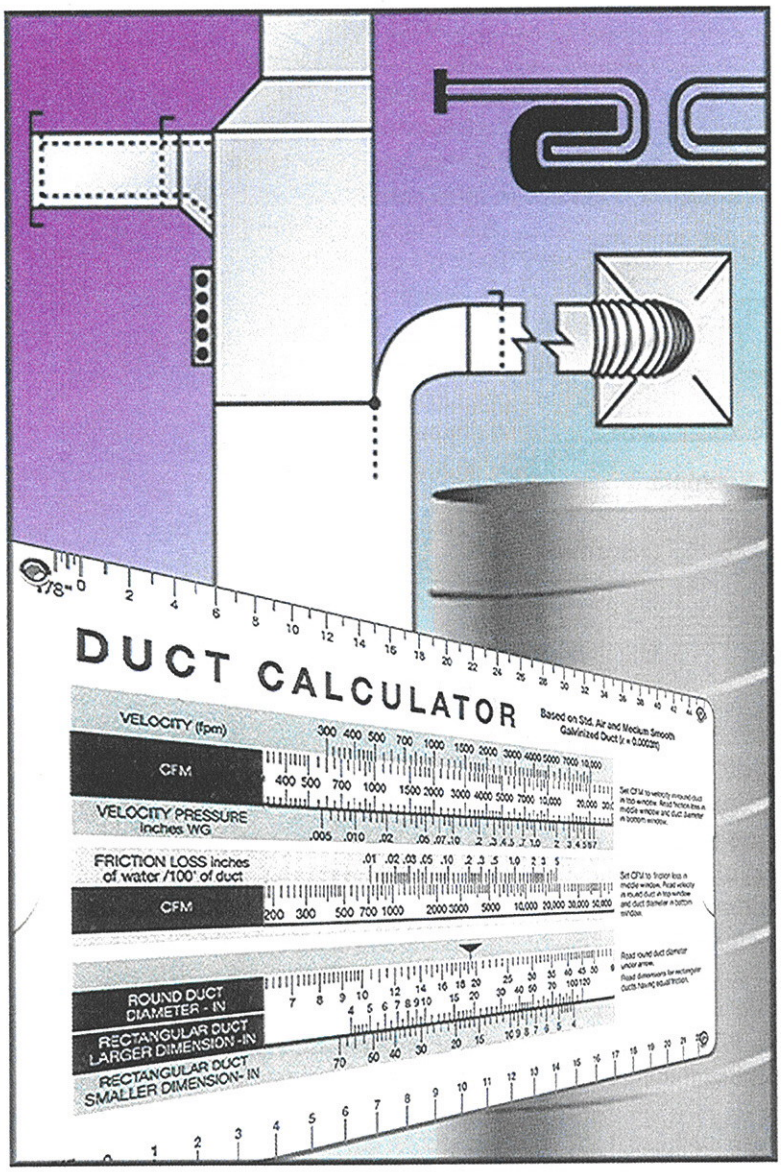


COMMERCIAL DISTRIBUTION SYSTEMS

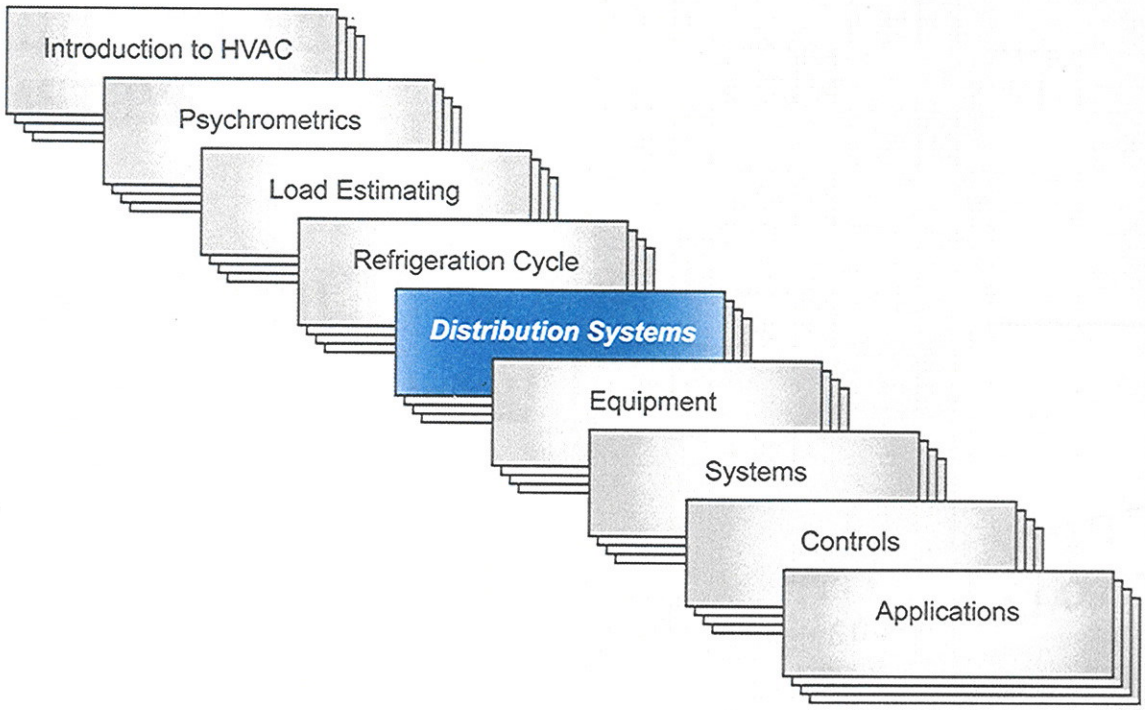
Duct Design

Level 1: Fundamentals

Technical Development Program

Technical Development Programs (TDP) are modules of technical training on HVAC theory, system design, equipment selection and application topics. They are targeted at engineers and designers who wish to develop their knowledge in this field to effectively design, specify, sell or apply HVAC equipment in commercial applications.

Although TDP topics have been developed as stand-alone modules, there are logical groupings of topics. The modules within each group begin at an introductory level and progress to advanced levels. The breadth of this offering allows for customization into a complete HVAC curriculum – from a complete HVAC design course at an introductory-level or to an advanced-level design course. Advanced-level modules assume prerequisite knowledge and do not review basic concepts.



This module will look at the way commercial duct design creates an airflow conduit for interconnecting an air handler, rooftop unit, or fan coil with VAV and CV terminals and/or room air distribution devices as a means of delivering conditioned air to the occupants of a building. A step-by-step design process will be presented covering such aspects of duct design as zoning, load determination, layout, sizing, and determining static pressure losses for system fan selection. After completing the module, participants will be able to manually size ductwork using either a friction chart or a duct calculator. The second level TDP of duct design will cover the modified equal friction method of duct design, along with additional sizing and layout recommendations.

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The information in this manual is offered as a general guide for the use of industry and consulting engineers in designing systems. Judgment is required for application of this information to specific installations and design applications. Carrier is not responsible for any uses made of this information and assumes no responsibility for the performance or desirability of any resulting system design.

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Introduction

This Technical Development Program (TDP) covers the fundamental principles of ***duct system*** design for commercial building applications. The most popular ***duct sizing*** method – ***equal friction*** – is covered in detail. ***Modified equal friction***, incorporating many of the benefits of ***static regain***, is presented in the related TDP-505, Duct Design, Level 2. Although many other duct sizing methods exist (e.g. velocity reduction, T-method, extended plenum, constant velocity, static regain), none are widely used by designers and are beyond the scope of this training module. The reader should refer to other publications for information on these sizing methods.

It is recognized that the use of manual ***duct calculators*** is normal, and that computer-aided duct design is becoming more popular; however, it is important to learn the manual friction chart method of duct sizing that is the foundation of these other methods. This will provide the knowledge necessary to recognize possible design errors and understand the effects of various design decisions. Once properly covered, use of Carrier's Duct Calculator for equal friction sizing will be presented.

Proper duct design requires performing load estimates to determine the zone and space cfm that the duct system will distribute. Once the cfm has been determined, the duct system components can be laid out. This includes locating the supply and return diffusers and registers to provide adequate air distribution to the spaces. Load estimating and room air distribution principles are covered in detail in other related TDPs.

This TDP will cover each duct design step in sufficient detail to permit the participant to lay out and size ductwork into a coordinated system that is energy efficient and cost effective to fabricate, install and commission. The Level 2 Duct Design TDP will present many areas of design enhancement, such as SMACNA Duct Construction Standards, duct design code requirements, fitting selection using loss coefficients, avoiding acoustic issues, unique VAV system duct features, and using life cycle cost analysis as a design criteria.

Level 1 Duct Design develops various aspects of sizing in detail because an oversized duct system will be difficult to balance and will increase the installed cost of the system. An undersized duct system will create higher than necessary air pressure drops, generate ***noise***, and will not deliver the required airflow quantities.

Work sessions are included as part of this program to assist the participant in evaluating his or her understanding of these fundamental principles and sizing parameters.

Duct Design Criteria

Several factors must be considered when designing a duct system. Generally, in order of importance, they are as follows:

- Space availability
- Installation cost
- Air ***friction loss***
- Noise level
- Duct heat transfer and airflow leakage
- Codes and standards requirements

Space Availability

The sizing criteria will often be defined by the space available to run the ductwork (Figure 1). Ceiling ***plenums***, duct ***chases***, and obstructions such as walls and beams often dictate that a certain size duct be used, regardless of whether or not it is the best size from a first cost or air friction loss perspective. There are most likely other building system components competing for the available space. Coordination is required to avoid sprinkler piping, power and communication conduit, light fixtures, and audio speakers. Header ducts and runouts are easier to locate, especially out in the perimeter areas of the floor. Larger trunk and branch ducts require greater coordination with equally large piping and conduit service utilities that tend to get located in the core areas of the building.

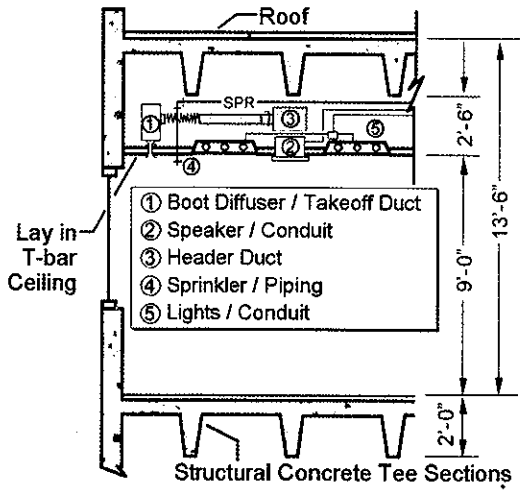


Figure 1

Fitting In the Ductwork

Duct Terms

Before we go any further, let's look at a simple duct system (Figure 2) and define some of the terms we will be using in this TDP (also, see Glossary). The trunk (or main) duct is the supply or return duct that connects to the air source (e.g. air handling unit, rooftop unit or fan coil) and distributes the air around the building. Branch ducts extend outward from the trunk duct, forming a tree pattern across the floor. Runout ducts connect VAV (variable air volume) and CV (constant volume) terminals to a branch duct or directly to the trunk duct. A takeoff (as a fitting) either connects a runout duct to a branch or trunk duct in order to distribute air to a terminal, or connects (as a duct) the header duct to the room air distribution devices (diffusers, registers or grilles). The header duct distributes zone air from the terminals.

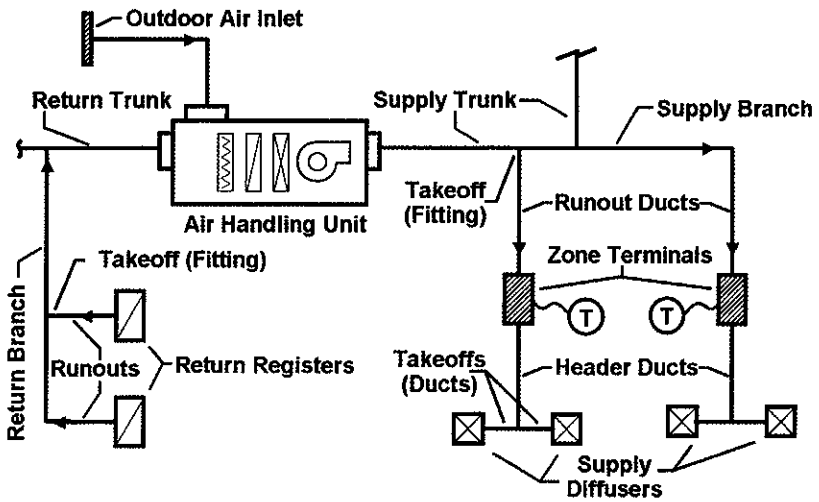


Figure 2

Duct Terms

Installation Cost

First cost is often quite important. First cost is not only impacted by the size of ducts and types of materials used to construct the ductwork, but also by the number and complexity of the ***duct fittings***, and the height/ complexity of the site conditions impacting duct installation labor.

The ductwork portion of the example system costs shown in Figure 3 represent 15 percent of the total, indicating that most of the dollars are spent on creating the heating and cooling capacity. Keep in mind that duct system costs are predominantly labor, representing upwards of 85 percent of the total installed number. Designers need to think of labor-saving designs, and be prepared to consider many suggestions from the installing contractor on design modifications that will economize the fabrication and installation for their shop and field practices.

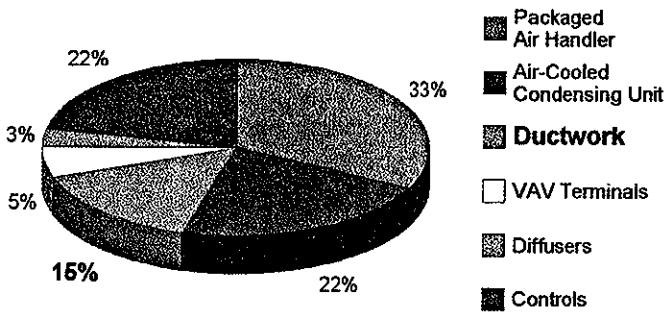


Figure 3

Example of HVAC Cost Breakdown

Ductwork

Air Friction Loss

Air friction loss is affected by the duct size and shape as well as the material and fittings used. For instance, round galvanized sheet metal has the lowest friction loss per linear foot, while flexible ductwork has the highest friction loss per linear foot. Also, the quality of fittings has a direct effect on the overall pressure drop of a duct system. Look to use smooth, efficient fittings with low turbulence to reduce the duct system air pressure drop, and use as few fittings as possible to lower the installation cost. A direct route using round duct with fewer fittings and size changes can have an overall friction loss that compares favorably with a similarly sized rectangular system with a longer route and size changes at each branch duct; but it will always be the more economical design.

Round ducts

Noise Level

An undersized duct system, that is, one with higher velocities, creates noise that is often objectionable to the occupants. Poorly selected or installed fittings also create ***turbulence***, which creates additional noise and air pressure drops. ***Dampers*** used for balancing need to be located out of the turbulence and not too close to the diffusers and registers in the space. There are many ways to limit noise creation (Figure 4) that need to be followed when designing ductwork.

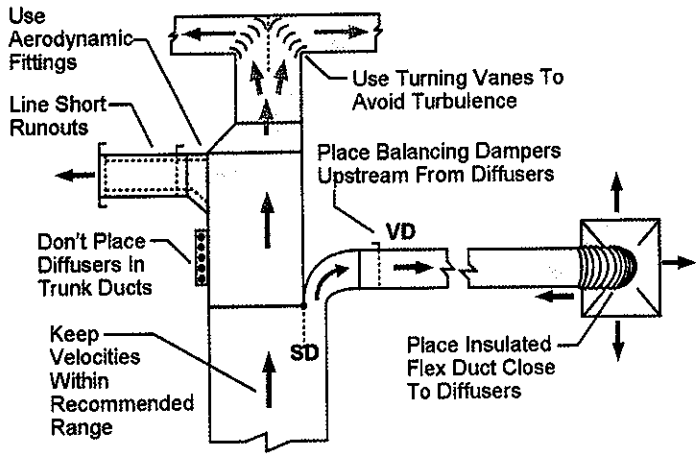
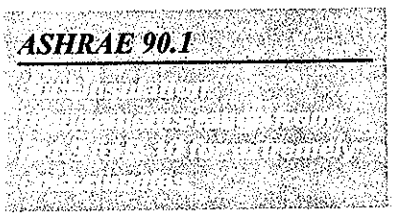


Figure 4

Limit Noise Creation

Heat Transfer and Leakage

Ductwork that runs through very warm or very cold areas can incur a heat gain or loss that effectively reduces the capacity of the cooling and heating equipment, and will likely result in occupant discomfort and higher operating costs. Leaky ducts have the same energy-wasting effect, and may create odors and stained ceiling tiles if duct ***thermal insulation*** becomes wet from the formation of ***condensation*** at the leak points.



ASHRAE 90.1 Energy Code dictates appropriate levels of insulation and joint seal levels for all ductwork in order to minimize these energy-wasting conditions. Figure 5 shows the extent of sealing required. Two extensive duct insulation tables in ASHRAE 90.1 cover all usages and climate areas.

Minimum Duct Seal Level

ASHRAE 90.1 Table 6.2.4.3A

Duct Location	Duct Type			
	Supply		Exhaust	Return
	≤ 2 in. w.c.	> 2 in. w.c.		
Outdoors	A	A	C	A
Unconditioned Spaces	B	A	C	B
Conditioned Spaces * *	C	B	B	C

Duct Seal Levels

ASHRAE 90.1 Table 6.2.4.3B

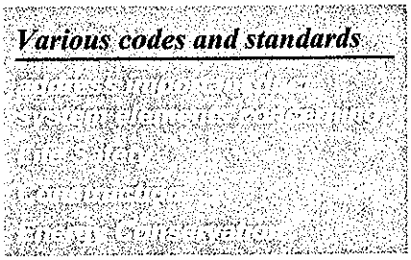
Seal Level	Sealing Requirements *
A	All transverse joints and longitudinal seams, and duct wall penetrations. Pressure-sensitive tape shall not be used as the primary sealant.
B	All transverse joints and longitudinal seams. Pressure-sensitive tape shall not be used as the primary sealant.
C	Transverse joints only

Figure 5

Sealing Ductwork Reprinted by permission from ASHRAE Standard 90.1 Copyright (2001) American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (www.ASHRAE.org)

Codes and Standards

HVAC duct systems are addressed in a number of building construction codes. Now that the International Codes Council's family of publications is being adopted across the United States, it is safe to say that familiarity with the International Building Code, International Mechanical Code, and International Energy Conservation Code will capture most of the code-related requirements for duct systems. Always check with your specific project code requirements for additional design related issues.



Theory and Fundamentals

Now that we have spent time going over duct design criteria, and before we spend time discussing a number of practical duct design issues, we need to cover a few basics of airflow fluid dynamics, present a couple airflow formulas, and talk about the relationship between static, velocity, and total pressures. When completed, we will cover some fundamentals on duct friction, fitting losses and duct sizing methods before moving on to the duct design process steps.

Law of Conservation of Energy – Bernoulli's Law

Objects may contain either potential energy or kinetic energy. Potential energy is derived from the object's relative position, that is, its location, when compared to a reference position. For instance, a truck parked at the top of a hill contains potential energy due to its mass and the force of gravity, which try to make the truck roll downhill. The brakes produce friction, which is greater than the force of gravity. If you release the brakes, the truck will roll downhill and the potential energy will be converted into kinetic energy. The word kinetic means motion, so the faster the truck rolls, the more kinetic energy it has, and the less potential energy it contains (Figure 6).

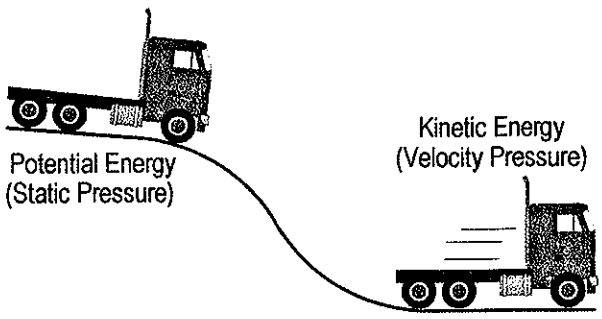


Figure 6

Potential Energy vs. Kinetic Energy

A typical fan wheel, driven by an electric motor, creates pressure and flow because the rotating blades on the impeller impart kinetic energy to the air by increasing its velocity. The air leaving the fan contains air pressure (energy) in two different forms:

Static pressure (P_S)

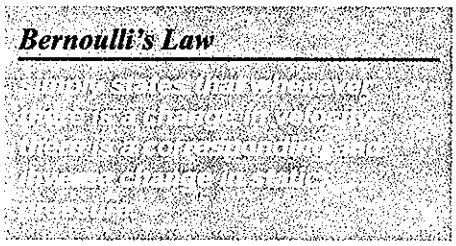
Velocity pressure (P_V)

As the high-velocity air exits the fan, the total pressure consists mostly of velocity pressure. This velocity pressure begins to convert into static pressure in the first few feet of supply air duct. Both velocity pressure and static pressure exist throughout the entire air system. Whenever the duct changes cross-sectional flow area, there is a conversion of velocity pressure into static pressure.

Static pressure is a force that is exerted against the sides of the duct wall equally in all directions. Static pressure is essentially the potential energy component in the air. An inflated balloon contains all static pressure (Figure 7). If you release an inflated balloon, the static pressure inside creates a flow of air out of the balloon, converting the static pressure into velocity pressure.

The sum of the static pressure and velocity pressure values is called the ***total pressure*** and is represented by the following equation:

$$P_T = P_S + P_V \qquad \text{Equation 1}$$



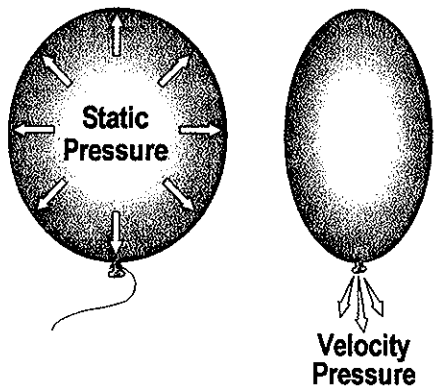
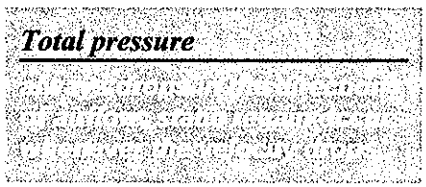


Figure 7

Static Pressure vs. Velocity Pressure

Daniel Bernoulli, a 16th century Swiss mathematician, physician, and physicist, developed a concept now known as Bernoulli's Law. He discovered that when velocity increases, static pressure decreases by the same amount, causing the total pressure to remain constant. Likewise, a decrease in velocity causes an increase in static pressure. This increase in static pressure is also referred to as ***static regain***. Total pressure, however, always decreases in the direction of airflow due to friction losses in the duct. This principle is illustrated in Figure 8.



For example, let's assume that the area of duct section ① is 1.0 sq ft and that the area of section ② is 0.6 sq ft.

According to the laws of physics and fluid flow, the velocity (V) of a substance, in this case air, is equal to the flow quantity (Q) divided by the cross-sectional area (A) of the flow conduit, in our case the duct. This can be stated as follows:

$$V = Q/A \qquad \text{Equation 2}$$

For airflow, Q is cubic feet per minute (cfm), for velocity, V is feet per minute (fpm), and area, A is square feet (sq ft).

Referring to Figure 8, if we assume that the airflow rate is 1000 cfm, we can calculate the velocity in both sections of duct as follows:

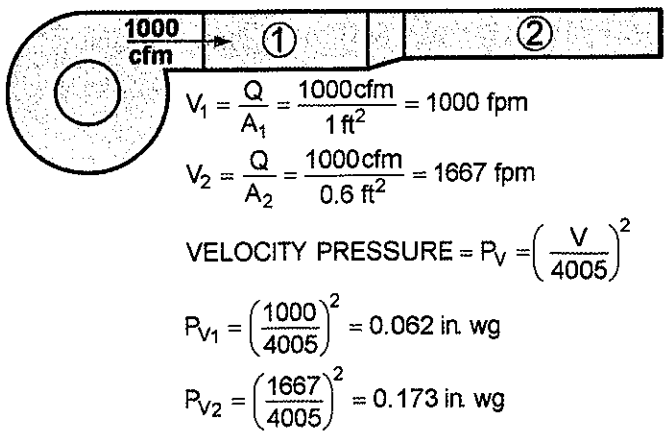


Figure 8

Velocity Pressure Conversion

$V_1 = 1{,}000$ cfm / 1.0 ft^2 = 1000 fpm

$V_2 = 1{,}000$ cfm / 0.6 ft^2 = 1667 fpm

Velocity pressure (P_v) for air at standard conditions (density (ρ) = 0.075 lb$_m$/ft^3) may be calculated by the following equation:

$$P_V = \rho V^2 / 2 g_c C_P = 0.075 V^2 / (64.4 * 5.193) = (V / 4005)^2$$

P_V is the velocity pressure in inches of water gauge (in. wg).

ρ is the material density

g_c is the gravitational constant, 32.2

C_P is a constant to reconcile units.

If we apply this equation to our example, we yield the following velocity pressure values:

$$P_{V_1} = (1000/4005)^2 = 0.062\ in.wg$$

$$P_{V_2} = (1667/4005)^2 = 0.173\ in.wg$$

From these calculations, we see that at the same cfm, the smaller duct (section ②) has a higher velocity and a higher velocity pressure. According to Bernoulli's Law, we should expect the static pressure to go the opposite way. Therefore, an increase in velocity pressure corresponds to a reduction in static pressure and vice versa. This means that the static pressure at Point ② is less than the static pressure at Point ① by the difference in velocity pressure (if we neglect the friction losses).

This principle is also analogous to water flowing through a water hose connected to a pressure nozzle. As the water flows through the hose and passes through the nozzle, the velocity is increased because the flow area has been greatly reduced at the nozzle. Even though it seems as if the water pressure has increased at the outlet of the nozzle, this actually represents only an increase of velocity. The actual static pressure of the water has decreased and the nozzle has converted all of the available static pressure of the water into velocity pressure.

Friction Loss in Ducts

When air flows through a duct, it encounters many obstacles along the way, such as elbows, transitions, and fittings. In addition, the surfaces of the duct walls are not completely smooth, creating friction as the air flows through the duct. Overall, duct friction loss is affected by these and many other factors, including:

- Air velocity
- Duct size and shape
- Duct material roughness factor
- Duct length

Changing any one of these variables will affect the friction loss. All of these factors contribute to what is commonly referred to as the friction loss. Other synonymous terms are often used when referring to friction loss in air systems such as "air pressure drop," or "static pressure loss." In most cases, these terms are measured in inches of water gauge (in. wg) per 100 feet equivalent length (EL). Equivalent length is covered in detail later under fitting losses.

Recommended Duct Velocities for Ductwork and HVAC Components

Recommended duct velocities generally range between 600 and 1300 fpm for commercial buildings, as shown in Figure 9. Table 2 in the Appendix presents detailed recommended maximum duct velocities. These are recommended maximum velocities for lower pressure class duct systems.

RECOMMENDED & MAXIMUM DUCT VELOCITIES RANGES

Designation	Schools, Theaters & Public Buildings
Fan Outlets	1300 – 2200
Main Ducts	1000 – 1600
Branch Ducts	600 – 1300
Branch Risers	600 – 1200

Velocities are for net free area.

DESIGN VELOCITIES FOR HVAC COMPONENTS

Louvers	- Intake	400 fpm
	- Exhaust	500 fpm
Filters	- Electrostatic	150-350 fpm
	- HEPA	250 fpm
	- Bag / Cartridge	500 fpm
	- Pleated	750 fpm
Heating Coils	- Steam / Water	500-1000 fpm
Cooling Coils	- DX / Water	400-500 fpm

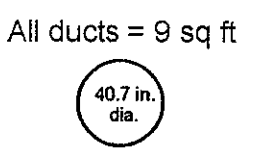
Figure 9

Duct and Design Velocities

Effects of Shape

At a given velocity, round ducts have the lowest static pressure drop per linear foot of any duct shape. Figure 10 indicates the relationship between duct shape and friction loss.

Notice the ratio of perimeter to area is lowest for a round duct. Less duct surface area translates into less air friction loss. As you increase the aspect ratio (longest dimension divided by the shortest dimension) the ratio of perimeter to area increases proportionally. The last example in the figure illustrates a duct with an aspect ratio of 9:1. This is a highly undesirable shape for an air duct, as the friction loss is nearly twice that of a square duct of the same cross-sectional area. In addition, ducts with large aspect ratios have more heat gain than smaller aspect ratio ducts.



Aspect Ratio	Perimeter (ft)	Ratio of Perimeter to Area	Equivalent Round Duct (in.)	Friction At 15,000 cfm (in. wg / 100' EL)
1:1	10.7	1.18:1	40.7	0.070
1:1	12	1.33:1	39.4	0.086
2.3:1	13	1.45:1	38.7	0.095
4:1	15	1.67:1	37.2	0.113
9:1	20	2.22:1	34.5	0.156

Figure 10

Effects of Shape, Ducts of Equal Area

Recommended Maximum Aspect Ratio

Large aspect ratio ducts are more difficult to reinforce structurally and may exhibit what is commonly referred to as "tin canning." Tin canning occurs when the fan turns on and off and the duct walls actually flex in and out due to the air pressure changes inside the duct, causing a loud, disturbing, thumping sound. Since it happens each time the fan turns on or off, you can imagine the occupant's dissatisfaction if intermittent fan operation was part of the control sequence.

Round ducts are often used for branch ducts off rectangular main ducts. Round duct unfortunately requires a larger height clearance when compared to rectangular duct. This is illustrated in Figure 10 with a 3-ft by 3-ft square duct having a cross sectional area of 9.0 sq ft. The equivalent round duct diameter for the same area is 39.4 in. This means that an additional 3.4-in. of height is required when using a round duct for this particular example. Saving space is one reason many designers use rectangular trunk ducts.

Rule of Thumb

Flat oval ducts do not take up less space than rectangular ducts, but they are inherently stronger when made with spiral lock seams and are quieter, like round ducts. Flat oval combines some of the best features of both, at a cost comparable to round ductwork.

Surface Roughness of Ducts

Duct material roughness (ε) refers to the inside surface of the duct; the rougher the surface, the higher the friction loss. Most duct sizing tables use the roughness factor for smooth, galvanized sheet metal as the reference value (ε = 0.0003 ft). This value of ε is based on bare sheet metal with joints every 4-ft. For other duct construction materials such as duct board, flexible duct, or duct liner, a multiplier of the measured duct length is used to correct for the higher roughness values. Duct material roughness multipliers are included as Table 4 in the Appendix.

Recommended Friction Rates - *f*

When sizing ducts for a particular pressure class, the designer will usually choose a design friction rate, abbreviated *f*, that is the desired friction loss in inches water gauge per 100 feet of equivalent length (in. wg/100 ft EL) of duct, commonly written as just in. wg. Pressure class is discussed in more detail later under Design Step 4.

The design friction rate is determined based on the desired velocity of the air in the first section of ductwork. As can be seen in Figure 11, system pressure class and duct duty also help determine the design static pressure loss to use in sizing ductwork.

Ductwork	Friction Rate Range (in. wg / 100 ft EL)
Pressure Classes ½, 1, 2	0.10 to 0.15
Pressure Class 3	0.20 to 0.25
Pressure Classes 4, 6, 10	0.40 to 0.45
Transfer Air Ducts	0.03 to 0.05
Outdoor Air Ducts	0.05 to 0.10
Return Air Ducts	80% of above supply duct values

Notes:
1. Higher friction rates should only be used when space constraints dictate.
2. Using higher friction rates permits smaller ducts but raises horsepower (energy) and velocity (noise).
3. Maximum aspect ratio is 4:1 unless space constraints dictate greater aspect ratios.
4. When diffusers, registers, and grilles are mounted to supply, return, and exhaust ducts, velocities should not exceed 1500 fpm or noise will result.

Figure 11

Recommended Friction Rates (f)

Note:

Fitting Dynamic Losses

Wherever turbulent flow occurs in a duct section, brought about by either a change in area or direction (Figure 12), losses in the fitting are greater than just the static pressure loss due to friction. These are called the dynamic losses and are additive to the losses determined for the straight lengths of duct.

There are two commonly used methods for determining the losses due to duct fittings (elbows, transitions, etc.): the equivalent length method and the total pressure method. The focus of this publication will be on the equivalent length method. The total pressure method is discussed briefly, but greater detail can be found in the Duct Design, Level 2 TDP. While there are common fitting equivalent length tables in the Appendix for most round and rectangular elements and common elbows (Tables 5, 6 and 7), you should refer to Carrier's System Design Manual, Part 2, for additional information whenever you come across an unusual situation not covered in the Tables.

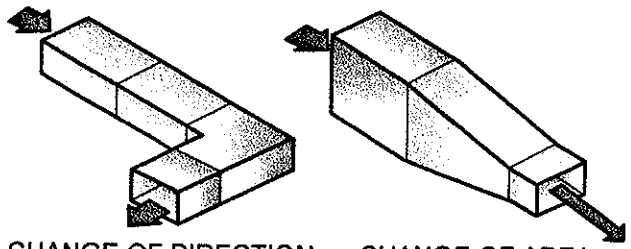


Figure 12

Dynamic Fitting Losses

Equivalent Length Method

Equivalent length (EL) is a concept used when referring to components in the duct system other than straight sections of duct, such as elbows, transitions, and other fittings. As can be seen in the example (Figure 13), the fitting total loss is converted into the same (equivalent) loss as a section of straight duct. For instance, using Table 7 in the Appendix, the 32-in. by 20-in. rectangular elbow would have an EL of 18 ft of straight duct. Let's now assume that the design friction rate for the system is 0.12 in. wg. By converting the friction losses of the fittings into EL of straight duct, you can add them directly to the actual length of straight ducts (measured to the fitting centerline) to obtain the total equivalent duct length of the ductwork.

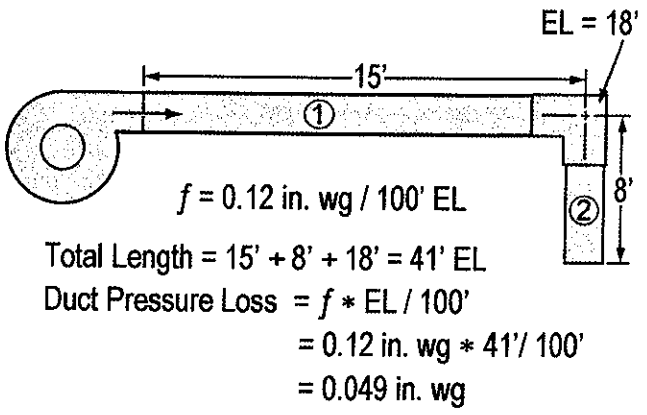
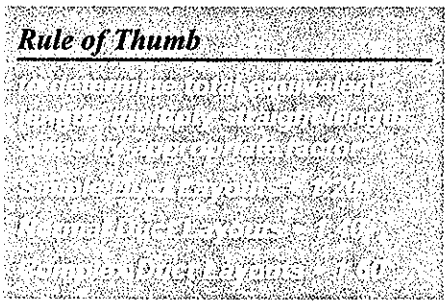


Figure 13

Using Equivalent Length

To calculate the total friction loss of the duct system, including fittings, multiply the friction rate times the total duct length, divided by 100 ft, ($f * EL/100$) as indicated in Figure 11.



The equivalent length method is widely used for residential and many light commercial applications. However, for larger, more extensive commercial duct systems, the method can become quite time consuming and not as precise as using the total pressure method.

Use of Fitting Loss Coefficients

Empirical testing of actual fittings have given the designer a dimensionless dynamic loss coefficient "C" that represents the number of velocity heads lost or gained at a fitting. This coefficient is multiplied times the velocity pressure of the air flowing through the fitting to get the total pressure loss of the fitting. In Duct Design Level 2, the designer will learn how to select more efficient fittings and create self-balancing designs that equalize duct circuit pressure losses.

System Effect

A duct system with an improper fan outlet or inlet connection will result in a reduced airflow quantity, less than the fan manufacturer's published ratings. Fans are generally rated, designed and tested with open inlets and with a section of straight duct connected to the outlet of the fan. In reality, most installations do not have open inlets or adequate lengths of straight duct attached to the fan outlet. This condition is commonly referred to as ***system effect***. System effect is a de-rating factor used to predict the reduction in actual fan performance caused by physical limitations placed on the fan system. When there is less than 100 percent effective length of straight duct directly off the fan outlet, like an abrupt transition or elbow, system effect should be considered.

The effective length of the discharge duct depends on the particular characteristics of the fan design. In general, it is a length of duct within which the velocity of the airflow reaches a uniform velocity profile. This is generally defined as a length of 2½ duct diameters for ducts with velocities of 2500 fpm or less.

A considerable amount of subjective judgment must be applied when working with system effect factors, as there may be a wide variation in different manufacturers fan designs. It is important to check for system effect in the total pressure analysis of the fan. The reader should refer to ASHRAE and SMACNA publications and Carrier's TDP-612, Fans: Features and Analysis, for additional information on system effect.

Duct Sizing Methods

The most common methods of sizing duct sections after the fan outlet in use today are:

- Equal friction
- Modified equal friction
- Static regain

Equal Friction

With the equal friction method, as the name implies, ducts are sized for an equal (constant) friction loss per unit length. In its purest form, this uniform friction loss per linear foot of duct is held constant for the entire duct system. The equal friction method is the most widely used method for sizing lower pressure systems. This method automatically reduces the velocity of the air in the direction of flow. Therefore some "regain" of static pressure is created; however it occurs in unknown amounts and is not usually accounted for.

Once initial sizes are calculated, the total pressure of all sections should be calculated and noted. Sections should then be resized to equalize the pressure at all junctions. This is demonstrated later in the example problem.

The equal friction method is generally used when sizing supply and return systems in CV (Constant Volume), and exhaust systems.

The primary disadvantages of the equal friction method are:

- There is difficulty in balancing branch flow rates, even with balancing dampers.
- It cannot ensure a uniform, constant inlet pressure to variable air volume devices and terminals.

Modified Equal Friction

To overcome these disadvantages, modifications to the equal friction procedure have been made that take advantage of the static regain effect. This sizing method is sometimes referred to as the ***modified equal friction*** design procedure. This procedure is used whenever the duct system is non-symmetrical or for systems with both long and short duct runs. By downsizing the shorter duct runs with lower friction loss there is less of a requirement to "choke" the airflow by the use of balancing dampers; results in smaller ducts, which saves money. The modified equal friction design procedure is described in detail in the Duct Design, Level 2 TDP.

Static Regain

The static regain method of duct sizing is a more complex and detailed method and may be used to size supply duct systems of any pressure/velocity class. The advantages of lower fan brake horsepower (bhp) and self-balancing sizing to the VAV terminals often outweigh the extra work involved in performing the calculations. In this method, the velocities are systematically reduced, resulting in a conversion of the velocity pressure into static pressure, which overcomes a portion of the air friction loss in the next downstream section of duct.

This phenomenon is not totally efficient, however. A regain efficiency of 75 percent is assumed for most average duct systems. This method is also advantageous because it creates a self-balancing system, in that all gains and losses are proportional to the velocities. This makes it especially useful for higher-velocity systems, where the turbulence and noise generated by air balancing dampers prohibits their use.

The main disadvantage of the static regain method is the larger duct sizes that result. This translates into higher first cost; however, the reduced static pressure on the fan often results in a lower fan operating cost over the life of the system.

Static regain manual sizing of ducts is not covered in the TDP series. It can be effectively applied using computer duct design software and should be studied using the program instructional manual.

Other Methods

While other duct sizing methods exist, such as the T-method, velocity reduction, total pressure, and constant velocity methods, they are not widely used and are not included in this text.

Work Session 1 Fundamentals

This is a good time to complete Work Session 1 and test your knowledge of the fundamentals, including design criteria and theory.

Duct Design Process Steps

Duct design should be thought of as a simple straightforward process that occurs over many segments of HVAC system design. The duct design steps should be performed in the order shown, though much iteration may be required for some of the steps before the HVAC system is finally designed. There is a separate duct system for each independent air handling system (air source) within the building.

Design Step 1: Determine Number of Zones

We begin by determining the number of unique temperature control zones that will be required. Perimeter and core areas should be separated into individual zones, depending on variations in internal loads and building exposure (N, S, E, or W). Further subdividing of the building into additional control zones may be necessary (Figure 14). Executive offices, conference rooms and computer rooms are all examples of additional zones that may be required.

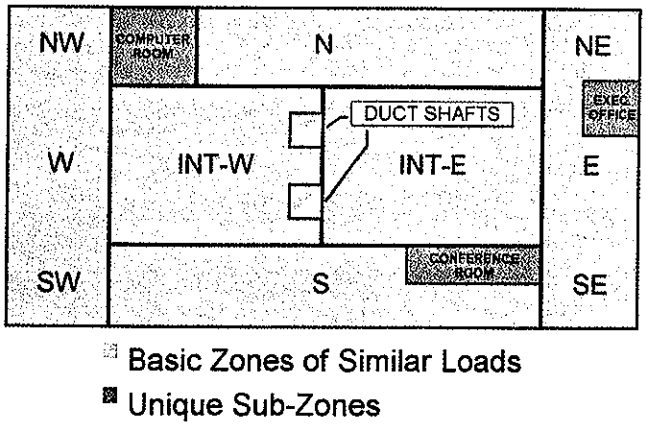


Figure 14

Design Step 1: Determine Number of Zones

Zoning helps us understand how the air is divided up and delivered by the trunk and branch ducts to either VAV or CV terminals or directly to diffusers. This is helpful later on when we get into initial duct layout. Further discussion on zoning can be found in TDP-702 Comfort Control Principles, and ENG-01 Comfort Design Made Simple.

Design Step 2: Perform Cooling and Heating Load Estimates

Accurate cooling and heating load estimates are important to all aspects of HVAC system design. An inaccurate load estimate can result in oversized or undersized equipment and ductwork, leading to an inadequate, poorly performing system and, likely, an unhappy customer.

Computer software programs can help the designer with the space, zone and block load airflows needed to begin the actual duct design steps. The system block load is simply defined as the maximum cooling load (coil load) for the air source that occurs within a design year. For most residential and commercial buildings located in the Northern Hemisphere, this maximum load occurs late in the afternoon during the hottest summer month, typically 4:00 p.m. in July or August. Load Estimating Level 3, Block and Zone Loads (TDP-302), covers the topic in greater detail.

Design Step 3: Determine Space, Zone and Block Airflows

The results of a load estimate calculation include the airflow quantities required for conditioning each of the spaces and zones, and for sizing the main coils and system fan. The room-by-room airflow quantities are used to size the supply diffusers and return grilles, as well as the take-offs and header ducts that serve them. If duct design was begun with assumptions for supply airflow Δt or cfm/sq ft, these values will need to be verified after final load estimates are completed. If the airflows change there is a very real chance that the ducts will need resizing.

The sum of the zone airflow quantities is the total amount of air the fan must deliver, also referred to as the ***total supply air quantity***. This value, obtained from the maximum block load, is also used to size the first section of supply duct off the fan. The sum of these airflow quantities is used to calculate the branch duct sizes and the trunk ducts that serve them.

Load estimating and room air distribution principles are covered in greater detail in other TDP publications. The scope of this program will be limited to the task of duct sizing.

Design Step 4: Select Duct Material, Shape and Insulation

After calculating the cooling and heating loads, the designer decides which materials and shape of duct to use, and how to insulate it to control energy loss/gain and limit noise levels. But even before beginning these first duct design steps, a quick word on the new classifications for duct systems now that low pressure and high pressure, or low velocity and high velocity classifications are no longer used.

System Classification

According to SMACNA (Sheet Metal and Air Conditioning Contractors' National Association, Inc.) standards, duct systems should be classified with a numerical pressure class as shown in Figure 15. These pressure-velocity relationships have replaced the older terminology. These older terms were rather vague and have been replaced with static pressure classification values.

Note in Figure 15 that pressure classes from ½-in. to 3-in. are designated as either positive (+) or negative (-) pressure. Pressure classes 4-in. and above are for positive pressure systems only.

Static pressure classifications are much more useful than terms such as low velocity or low pressure, because they may be used to establish the required duct construction materials and reinforcing. The designer and sheet metal contractor may use these values to establish the required metal gauge, reinforcing, and maximum duct dimensions allowable to prevent failure.

Static Pressure Class (in. wg)	Pressure Range (in. wg)	Maximum Velocity (fpm)
±0.5	0 to 0.5	2000
±1	>0.5 to 1	2500
±2	>1 to 2	2500
±3	>2 to 3	4000
+4	>3 to 4	4000
+6	>4 to 6	*
+10	>6 to 10	*

* Determined by designer

Figure 15

Pressure-Velocity Duct Classifications

Therefore, if someone refers to a SMACNA Pressure Class 4 system, according to the chart, the pressure is greater than 3.0-in. wg and up to 4.0-in. wg, with a maximum velocity of 4000 feet per minute (fpm).

While we are discussing pressure-velocity classifications, it is a SMACNA-recommended practice to note the duct drawings with the design static pressure classes. This can be seen in the Symbols List in the Appendix in the first few symbols on the left-hand side of the List. If this is done, the installing contractor now knows how to construct the duct, along with the size, whether it is lined or not, and in which direction the air flows.

What Can Be a Duct and What is it Made of?

A duct, quite simply, is a passageway or conduit made of noncombustible material for moving air from one place to another. Traditionally ducts have been made of metal, first iron and carbon steel (still used for kitchen hood exhaust), or copper (weather tight). Now, almost all are made from galvanized or coated steel, stainless steel or aluminum (Figure 16). Rigid fiberglass ductboards with a reinforced exterior vapor barrier and flexible vinyl round ducts have also been used extensively as a duct construction material.

Duty / Material	Galvanized Steel	Carbon Steel	Stainless Steel	Aluminum	Fiberglass Board	FRP	PV Steel	Gypsum Board
HVAC	X				X			
Flues		X						
Moisture-laden			X	X				
Kitchen		X	X					
Fume Hood			X			X	X	
Air Shafts	X							X
Underground							X	

FRP = Fiberglass Reinforced Plastic
PV Steel = PVC-coated steel

Figure 16

Common Duct Material Applications

Ducts can also be airflow passageways, such as plenums above ceilings and under ***access flooring***. They can be vertical air shafts made of drywall or masonry (not easily made air tight) as well. Whenever a supply or return plenum is employed in a duct system, care needs to be taken to make sure no other combustible or hazardous materials are present within the plenum.

Most commercial applications use galvanized sheet metal for the main ductwork and either flexible duct or sheet metal for the runouts to the air distribution devices. Flat oval spiral ductwork is used on higher-pressure class systems and is sized similar to round. Once a round duct size is determined, a conversion table (Table 8 in the Appendix) is used, as with rectangular ducting, to find an equivalent flat oval size.

There is generally a trade-off between duct material and price. Fiberglass duct board is relatively inexpensive and easy to install; however, the pressure loss is approximately 30 percent higher than that of smooth galvanized sheet metal. Flexible (flex) duct is easy to install but exhibits a much greater (three times higher) pressure loss per linear foot as compared to smooth sheet metal, even when the flex duct is hung straight. If flexible duct is allowed to unnecessarily sag and bend, the pressure loss will be much greater. Poorly installed flexible duct is one of the most common field problems and should be avoided.

How are Ducts put Together, Sealed and Insulated?

All ducts start out as flat sheets that are cut, bent, and formed into shapes that have seams and joints, usually at fittings or at the connecting of successive straight sections (Figure 17). Round and flat oval ducts can be wound on machines, creating a continuous spiral lock seam that can be quite air tight without further sealing. SMACNA has developed many systems of sheet metal gauges, reinforcing and joint types that allow a contractor to build a duct to the specified size and pressure class that works best for their fabrication machinery and field installation techniques. The designer needs to remain open to variations from the base design that still meet the requirements, hopefully, more cost effectively, too.

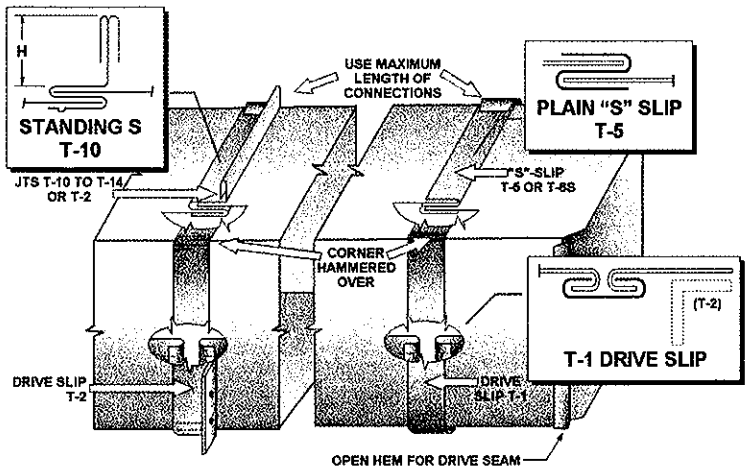


Figure 17

Duct Assembly

Insulating the ductwork is now required by the Energy Code, but care needs to be taken here too so that reinforcing and joint types do not conflict with the insulation. Insulation is often applied as an external flexible wrap on the exterior of the ducts after assembly and installation. Proper choice of low profile reinforcement and joint styles allows for easier insulating of the ductwork without worry of puncture. Internal insulation may appear more costly, but on larger size ductwork, where the reinforcing issue is more prevalent, shop-applied, internal coated, rigid fiberglass insulation board may prove to be cost effective. With erosion-resistant coating and proper field coating of the joint seams, there should be no concern about airborne particulates, and the ductwork is now quieter with the benefits of the fiberglass sound dampening reducing high frequency ***breakout noise.*** Review of Table 4 shows that *f* has now nearly double, so additional fan bhp will be required. Lined ductwork is more difficult to clean, and may harbor microorganisms if allowed to get dirty and damp.

Note:

One final comment about duct insulation: remember to allow extra room for the insulation. Ducts with a 1 in. internal liner will be built 2 in. larger on each dimension (height and width) to account for the liner. Duct dimensioning on contract documents is always the inside airway clear sizes. In other words, a 10-in. by 12-in. duct with internal liner must be fabricated 12-in. by 14-in. to accommodate the liner. Likewise, when laying out externally-insulated ductwork, be sure to allow for the extra height and width required.

Design Step 5: Lay Out Ductwork from AHU to Air Distribution Devices

The primary purpose of an air duct system is to deliver the proper amount of conditioned air from the air source to the conditioned space, and then to return the air from the space back to the air source, as shown in Figure 18. Not all duct systems have both supply air and return/exhaust air ductwork. For instance, an exhaust system has only exhaust ductwork, while a hospital operating room duct system may use all (100 percent) outdoor ***ventilation air***, in which case the return air ductwork would function to relieve the air to the outside. In addition, there are circumstances where it is advantageous to utilize the ***plenum*** space above the ceiling for return airflow.

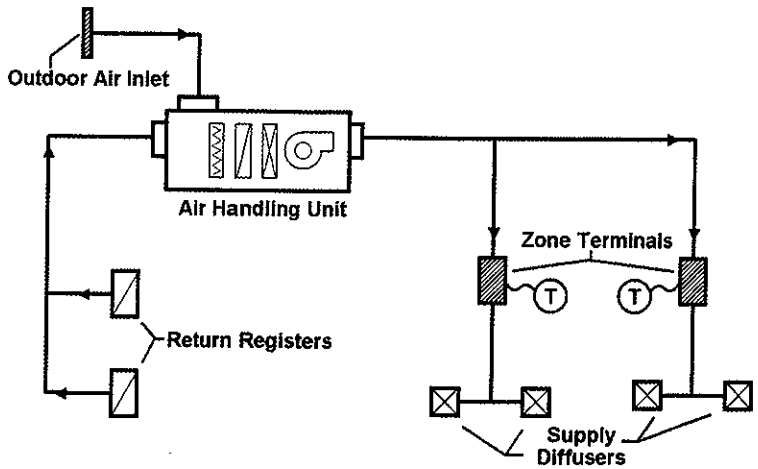


Figure 18

Design Step 5 — Lay Out Ductwork from AHU to Air Distribution Devices

A typical air-side for an HVAC system consists of a fan, supply air ductwork with transitions and fittings, supply diffusers, supply and return registers, return grilles, and return air ductwork, cooling and heating coils, and filters. The air in the room is essentially "still air" (15 – 50 fpm) and is typically at atmospheric pressure (zero gauge pressure).

To begin the layout process, produce a sketch of where to route the duct from the fan to each of the zones, maintaining straight lines without any unnecessary turns and bends. A semi-transparent paper, such as "onion skin" or "velum," can be used to sketch the duct system over the architectural floor plan. Be sure to refer to the reflected ceiling plans and structural drawings to avoid interference with plumbing, sprinklers, lighting, and structural members.

Try to allow at least a few feet of full-size straight ductwork directly downstream of the fan before you make any turns, take-offs or size changes. If it is necessary to have an elbow close to the fan, always turn the air in the same rotational direction as

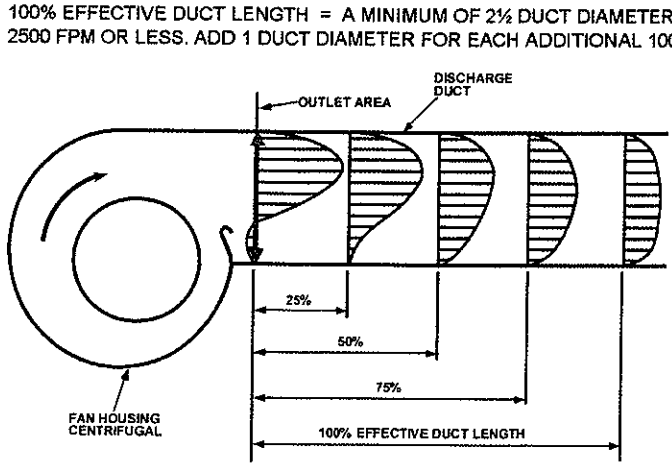


Figure 19

Fan Outlet System Effect

the fan wheel. The air in the duct is very turbulent coming off the fan discharge. The air needs a few feet of straight duct to establish a uniform velocity profile so that all of the energy from the fan can be converted into pressure. Any energy not converted into pressure becomes turbulence and vibration, which will likely lead to a noisy system.

Wherever possible, ducts should not be located in extreme temperature areas such as hot attics to minimize heat gain and loss. If you must route duct through extreme temperature environments, be sure to use adequate insulation to minimize thermal losses.

Do not forget the return air duct system design. They are sized the same ways as the supply ducts, though usually at a slightly lower friction rate. Some buildings use a ceiling plenum return for the return air; so, a fully ducted return air system may not be required.

Fit Trunk Duct to Building

Try to pick simple layouts for the ***trunk ducts*** that fit the building shape and usage patterns (Figure 20). Trunk ducts and ***terminals*** that are close by should both be located above corridors and central common spaces where service access is easy and any noise would not be bothersome.

Branch ducts and terminal runouts should be less than 25-30 ft, so the trunk layout should fit the basic shape of the building. If the building were relatively narrow, a single "spine" duct running down a central corridor would work well and be cost effective. As the building widens out, two trunks connected across the middle create an effective "H" pattern that once again should end up above circulation and/or common spaces.

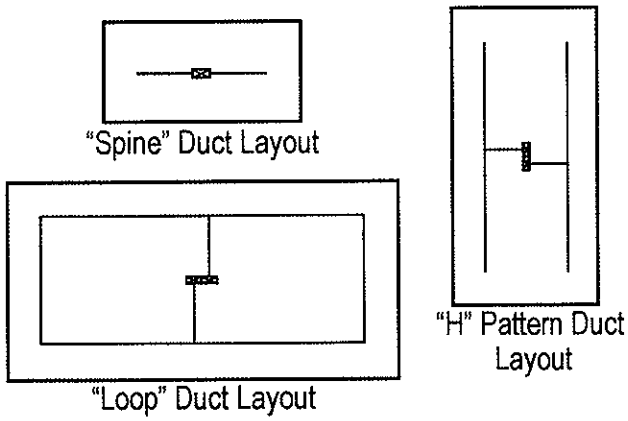


Figure 20

Trunk Duct Layouts

Another arrangement that works very well (especially on larger multi-story buildings) is to connect the ends of the H, creating a "loop." The use of this type of layout in ***variable air volume*** (VAV) systems evens out zone airflows throughout the day and can permit diversity downsizing of "common" portions of the trunk duct (discussed further in the Level 2 Duct Sizing TDP).

Once the trunk duct and branch ducts have been located, the zone terminals and ***air distribution devices*** can be connected, completing the initial duct layout. Supply outlet selection for each space in the zones, which completes the supply duct layout, is based on the zone airflow quantities calculated in the load estimate performed previously. Supply outlets would be selected based on their published and rated airflow quantity, pressure drop, throw, and noise criteria. Remember to use simple, symmetrical layouts to keep static pressures low and minimize noise generation.

Create a System Sizing Schematic

With layout completed from the air-handling unit to the air distribution devices, we can now take our simple single-line schematic diagram of the HVAC system and annotate it with duct sizing information. First, note the duct lengths between changes in airflow, or devices, and the design cfm at all air inlets and outlets. Figure 21 shows cfm and dimensions of the duct sections, along with "flags" or "labels" for the individual sections of equipment in our system that are connected by the duct system.

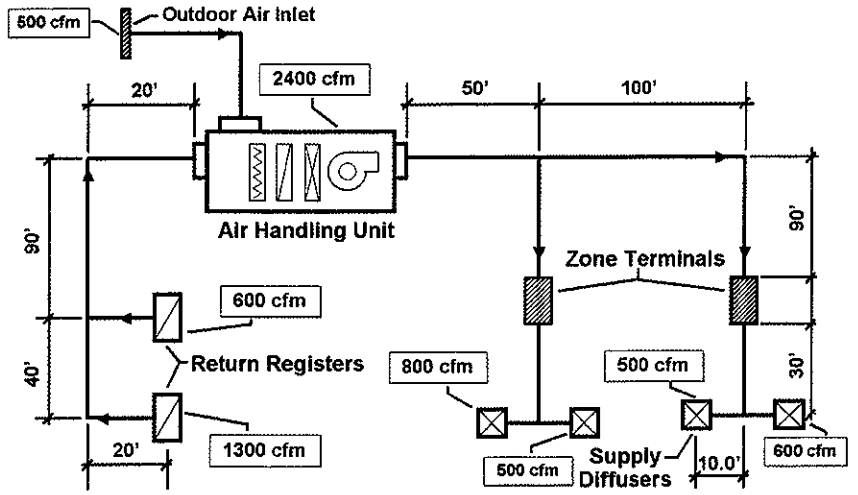


Figure 21

System Sizing Schematic

Design Step 6: Summarize Duct cfm and Label Duct Schematic

Beginning at the air outlets and inlets, add the airflow quantities from each supply diffuser and return register to the branch duct sections, then finally back to the main trunk ducts, finishing at the air source.

Because of VAV system load diversity, this total airflow quantity, made up of the zone load cfms, would not equal the block load cfm. These are the quantities flagged in red on the system sizing schematic that we just drew.

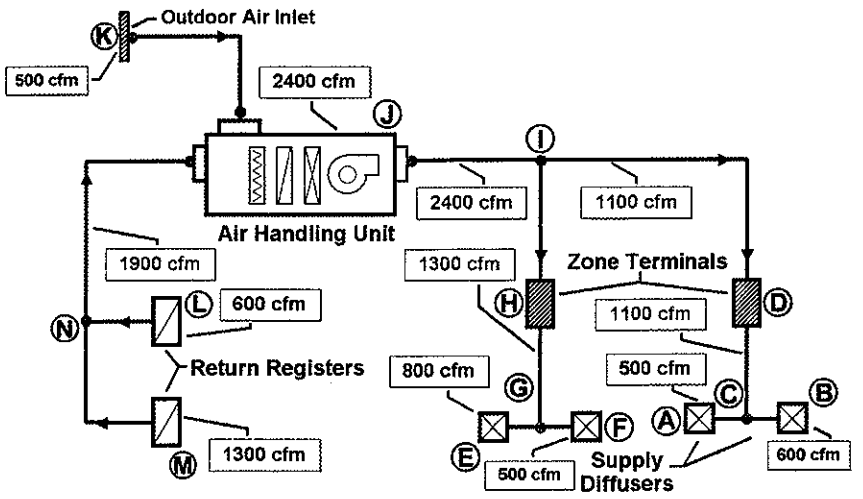


Figure 22

Design Step 6 — Summarize Duct cfm and Label Duct Schematic

Next add nodes and letter indicators at each duct system inlet and outlet, at each point where the ductwork joins or separates, and at each piece of equipment or component connected to the duct system.

Use a Duct Sizing Worksheet

On all but the simplest duct system, it is recommended that designers use a simple worksheet to keep track of the input information off the system schematic, record equipment selection data and list the results from the duct sizing efforts. We can now transfer the equipment labels and airflows and duct section lengths from Figure 21, plus the cfm duct section and nodes/labels from Figure 22 to a duct sizing worksheet (Figure 23). A blank duct sizing worksheet can be found in the Appendix. In the next step, such a worksheet will provide an easy means of determining the duct run, or circuit, of greatest pressure loss that is used in the fan selection. The full procedure can be seen in detail when going through the Equal Friction Sizing Example (Example 3, pg 33).

DUCT SIZING WORKSHEET

PROJECT NAME: DUCT SIZING WORKSHEET DATE: 10/1/04

SYSTEM: EXAMPLE SUPPLY - LONGEST RUN PAGE: 1 OF 1

Duct Run From -To	Duct Section (element)	Lining (in.)	Insul. (in.)	Other Item	Airflow	Velocity in Round duct (fpm)	Velocity Pressure P_v	Fitting Value n	Length (ft) L	Equiv. Length (ft) EL	Material Correction Factor	Friction Loss /per 100' duct	Friction Loss (in. wg)	Known Loss (in. wg)	Round Duct Size (in.)	Equivalent Rectangular Size (W x H)	Total Item Loss (in. wg)	Cumulative Loss (in. wg)
J-A	J-I	-	1	-	2400	-	-	-	50	-	-	-	-	-	-	x	-	-
	I-D	-	1	90° Elbow	1100	-	-	-	190	-	-	-	-	-	-	x	-	-
	D	-	-	TERMINAL	1100	-	-	-	-	-	-	-	-	-	-	x	-	-
	D-C	1	-	-	1100	-	-	-	30	-	3.2	-	-	-	-	x	-	-
	C	-	-	TEE	500	-	-	-	-	-	-	-	-	-	-	x	-	-
	C-A	-	1	DAMPER	500	-	-	-	10	-	-	-	-	-	-	x	-	-
	A	-	-	OUTLET	500	-	-	-	-	-	-	-	-	-	-	x	-	-

Figure 23

Using a Duct Sizing Worksheet

Design Step 7: Size Ductwork from Fan, Out to Extremities

The next task is to determine supply and return ductwork sizes that will be required to deliver the correct amount of air to the spaces. Begin with the first section of duct off the fan. This section is sized based on an assumed initial velocity. Subsequent duct sections should be sized by the appropriate sizing method, from the fan out to the most distant points. The equal friction method of duct sizing is illustrated in this section.

Select your fittings from the tables in the Appendix, looking for the lowest equivalent length values that work with your project duct design criteria, especially space availability and installation cost.

Once the duct sizes and fittings are known, the designer can then determine the total static pressure that the fan must overcome to deliver the required airflow.

You should always use a Duct Sizing Worksheet, like the one found in the Appendix, to organize your duct layout data and summarize your sizing and static pressure calculations.

Duct Sizing Using the Friction Chart

Although many manufacturers and industry societies offer duct calculator wheels and slide rules to simplify the design process, this training module will first teach the traditional Round Duct Friction Loss Chart (Chart 1 in Appendix, called just the Duct Friction Chart). Once you understand where the values come from, you will find that using a duct calculator or computer software program will assist you with the task of duct sizing. Figure 24 outlines that steps involved in using the Duct Friction Chart.

Once a round duct size has been determined, it can be converted into *rectangular* dimensions of similar airflow performance. That is, a rectangular duct with the same pressure loss at the same airflow rate (see Table 1 in the Appendix). *Example 2* will illustrate the relationship between equivalent round and rectangular duct sizes.

To illustrate how to use the Duct Friction Chart for round duct, consider the following example.

1. Select desired velocity in first duct section.
2. Enter friction loss chart, read round duct diameter at intersection of cfm and velocity lines.
3. Read resulting friction loss value at bottom of friction chart; verify that it is acceptable.
4. If sizing round duct, you have completed sizing the first duct section. Proceed to the next duct section using desired friction rate.
5. If sizing rectangular duct, you must convert round sizes to equivalent rectangular sizes using Table 1 in the Appendix.

Figure 24

Sizing with the Duct Friction Chart

Example 1 – Using the Duct Friction Chart

Using a design friction rate of 0.10 in. wg, what size round duct is required if the velocity is to be maintained at no greater than 1500 fpm and the flow rate is 1800 cfm?

Refer to Figure 25. The first step is to locate the design friction rate along the Friction Loss scale, in our case 0.10 in. wg. Next, locate the airflow value (1800 cfm) along the Air Quantity scale.

Now locate the point of intersection between these two values. The duct diameter lines run from the upper left to the lower right sides of the chart. The intersection point falls between the 16-in. and 18-in. lines (Point ①).

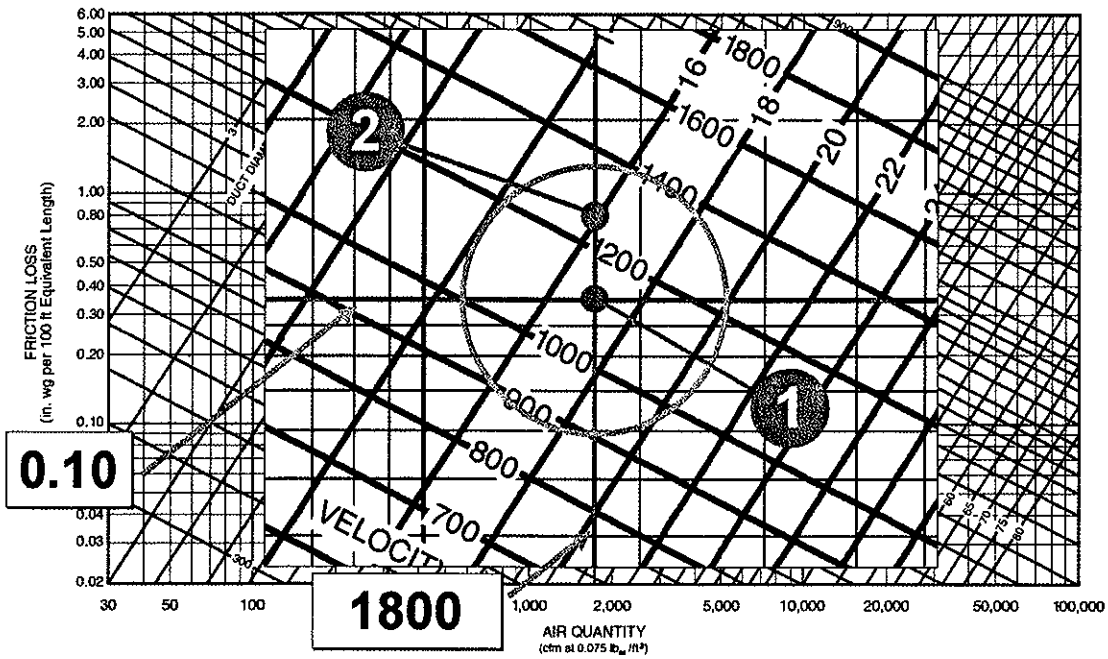


Figure 25

Round Duct Friction Chart Example (full size chart in Appendix)

Next, locate the velocity lines and make sure that the velocity is acceptable. The velocity lines run from the lower left to the upper right corners of the friction loss chart. The velocity is between 1000 and 1200 fpm, which is below our requirements. Which size do you use, the 16-in. or the 18-in. round duct? Generally, when sizing round ducts, you should round up to the next larger size, in our case an 18-in. round duct.

Finally, from the friction chart, read the actual friction loss for an 18-in. duct. It is approximately 0.08 in. wg. (This can be seen by plotting values on the full-size blank chart in the Appendix.)

What happens if we decide to use the 16-in. duct instead of the 18-in. duct? Since the airflow quantity is the only constant value, all of the other variables will change when we change the duct size. To see the effects of using the smaller duct, locate the intersection point where the 16-in. duct meets the 1800 cfm line (Point ②). This is above of our previous condition. This corresponds to a velocity of approximately 1300 fpm, still below our maximum requirement of 1500 fpm.

However, our friction rate has increased to 0.14 in. wg. As a designer you will have to decide if this higher friction rate is acceptable. For short duct lengths it is probably not a significant factor. When designing duct systems, however, it is best to stay as close as possible to the initial design friction rate, in our case 0.10 in. wg (Figure 11 recommendation).

Circular Equivalent Diameters of Rectangular Ducts

Duct sizing is generally done first as round ducts, then when required they are converted to rectangular sizes of equivalent friction rate. You cannot simply calculate the area of the circle (round duct), and then use a rectangular duct with the same cross-sectional area. Doing so would create a duct with a higher pressure drop than the round duct of the same area. This goes back to the ratio of perimeter to area shown in Figure 10. The velocity in a rectangular duct with equivalent friction rate will be less than the velocity in the round duct. This is necessary so that the pressure losses for the two ducts are equal.

To accomplish this conversion from round duct to an equivalent rectangular duct, refer to Table 1 in the Appendix.

Example 2 – Converting From Round to Equivalent Rectangular Ducts

Convert a 22-in. round duct to an equivalent rectangular duct with a maximum aspect ratio of 4:1.

Figure 26 shows a portion of Table 1. The rectangular duct dimensions are listed along the top and left side scales. The complete table is provided in the Appendix.

Table 1 — Circular Equivalent Diameter, * Equivalent Area of Rectangular Ducts for Equal Friction

SIDE	6		8		10		12		14		16		18		20		22	
	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)
10	0.38	8.4	0.52	9.8	0.65	10.9												
12	0.46	9.1	0.62	10.7	0.78	12.0	0.94	13.1										
14	0.52	9.8	0.72	11.5	0.91	12.9	1.09	14.2	1.28	15.3								
16	0.59	10.4	0.81	12.2	1.03	13.7	1.24	15.1	1.46	16.4	1.67	17.5						
18	0.66	11.0	0.90	12.9	1.15	14.5	1.39	16.0	1.64	17.3	1.88	18.5	2.11	19.7				
20	0.72	11.5	0.99	13.5	1.27	15.2	1.54	16.8	1.81	18.2	2.08	19.5	2.34	20.7	2.61	21.9		
22	0.78	12.0	1.08	14.1	1.38	15.9	1.68	17.6	1.98	19.1	2.28	20.4	2.57	21.7	2.87	22.9	3.15	24.0
24	0.84	12.4	1.16	14.6	1.49	16.5	1.82	18.3	2.15	19.9	2.48	21.3	2.80	22.7	3.12	23.9	3.44	25.1
26	0.90	12.8	1.25	15.1	1.60	17.1	1.96	19.0	2.32	20.6	2.67	22.1	3.02	23.5	3.37	24.9	3.72	26.1
28	0.96	13.2	1.33	15.6	1.71	17.7	2.10	19.6	2.48	21.3	2.86	22.9	3.25	24.4	3.62	25.8	4.00	27.1
30	1.01	13.6	1.41	16.1	1.82	18.3	2.23	20.2	2.64	22.0	3.05	23.7	3.46	25.2	3.87	26.6	4.28	28.0
32	1.07	14.0	1.49	16.5	1.92	18.8	2.36	20.8	2.80	22.7	3.24	24.4	3.68	26.0	4.11	27.5	4.55	28.9

Figure 26

Circular Equivalent Diameters

Now, the trick is to find a combination of dimensions that meet the criteria stated above; that is, a maximum aspect ratio of 4:1 and a 22-in. round duct. Note there are two numbers under each column, *Area (sq ft)* and *Diam. in.* Scanning down the first column of Table 1 in the Appendix (6-in. side) you are looking for a diameter of 22-in. It is not under the first column, as the largest diameter is a 16.8-in. duct.

Now look under the 8-in. column. There is a 22-in. diameter duct listed for the 64-in. side duct. However, the 8:1 aspect ratio (64 x 8) violates the 4:1 aspect ratio limit.

Continue over to the 10-in. column. Read down till you see the 21.9-in. diameter. This equates to a 46-in. duct, which makes our aspect ratio 4.6:1, still too large.

Read over to the 12-in. column, then read down to the 21.9-in. diameter corresponding to a 36-in. duct. That works, as the aspect ratio is 3:1 (36 x 12).

There are other combinations of values that also work, such as: 30 x 14, 26 x 16, 24 x 18, and 20 x 20.

Designers generally attempt to use the larger dimension for the width and the smaller dimension for the height. This is due to the fact that available space to run duct above most ceilings is usually in short supply. Also, try to maintain one of the duct dimensions (either height or width) whenever possible when transitioning from one duct size to the next. This saves money during fabrication by simplifying the transition.

Duct Sizing Using Duct Calculators

Duct calculators, whether wheels or slide rules, are manual nomographs that perform the same function as entering velocity and airflow/friction rate into the Friction Loss Chart and determining a round duct size. Most duct calculators contain additional scales and charts to allow you to find equivalent rectangular sizes, determine the velocity pressure, and find the friction loss for a variety of materials.

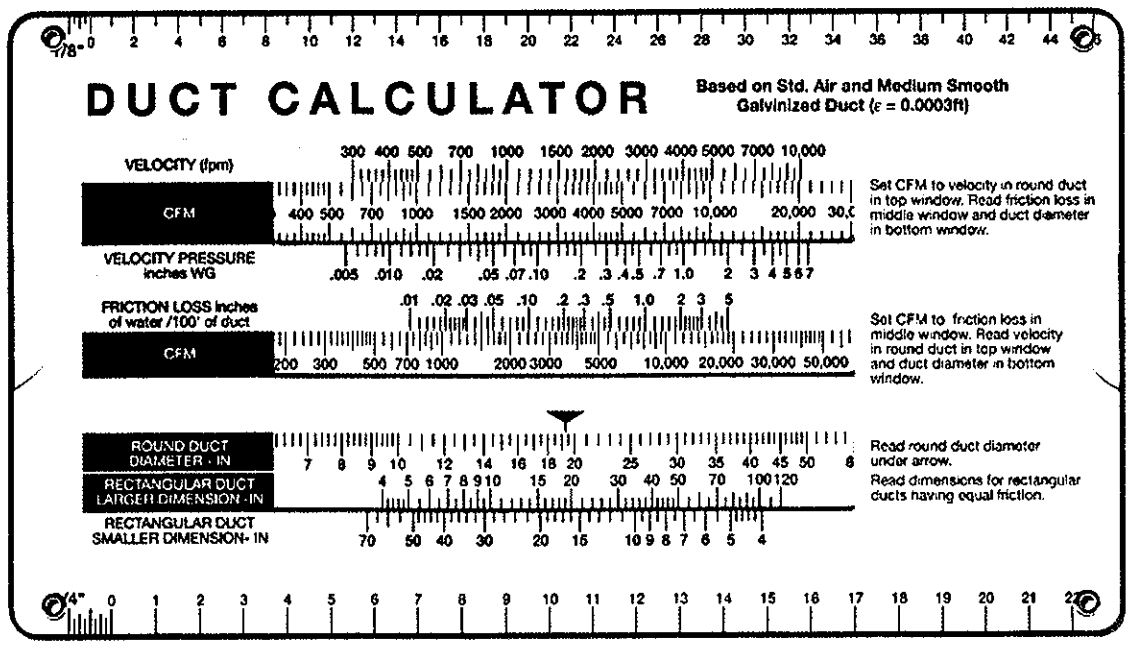


Figure 27

Sizing with the Duct Calculator (front)

Carrier offers the handy pocket-sized duct calculator (Figure 27), with Instructional Manual, that does all the tasks mentioned above. A short discussion on the use of the duct calculator follows. Once you have mastered the Duct Friction Chart, rework some of the earlier examples and Example 3 after the Summary to see how a duct calculator can be used for sizing.

The duct calculator allows you to quickly and easily determine duct sizes for round and rectangular duct systems. It works for a variety of duct construction materials and unique inside surface roughness factors (ε). All of the information contained in the round duct friction chart, as well as the conversion chart for converting round to rectangular ducts, has been incorporated into this slide chart. Values indicated on the front are based on standard air (70° F, 29.92 in. Hg) and medium-smooth galvanized sheet metal duct with a specified roughness factor ε = 0.0003 ft.

Approximate equivalent friction losses are indicated on the backside of the duct calculator (Figure 28) for duct board or duct liner, flexible metal duct, and flexible vinyl-coated ductwork with a helical wire core. There is also a table that lists recommended and maximum duct velocity ranges for fan outlets, main and branch ducts, as well as branch risers and a table of design velocities for various HVAC components.

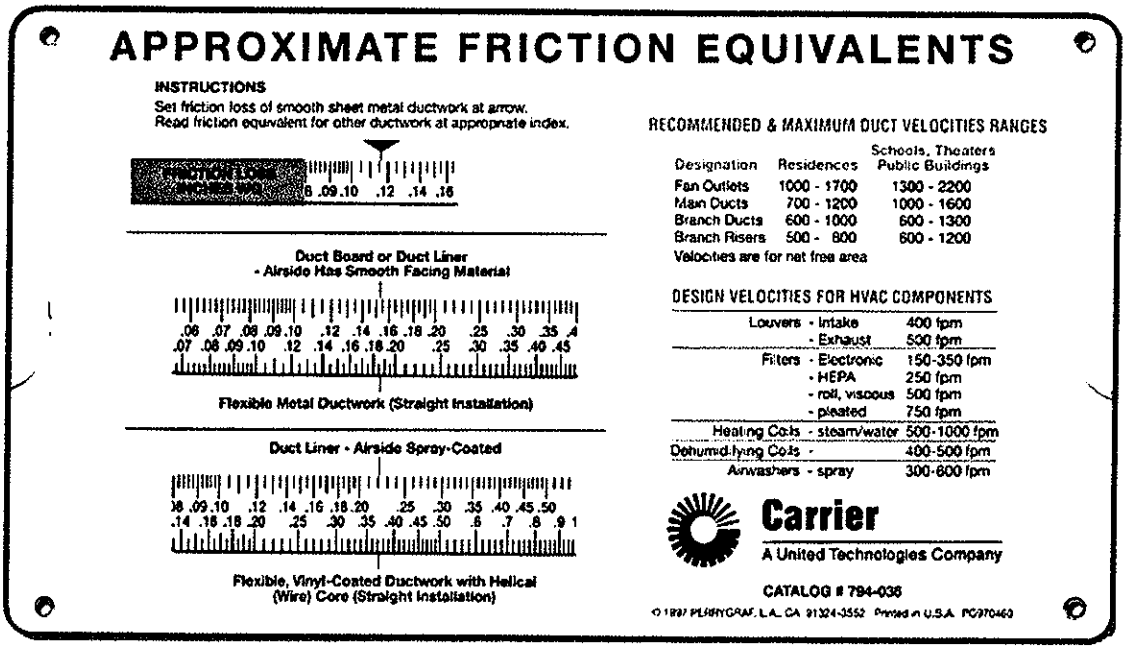


Figure 28

Duct Calculator (reverse)

Using your duct calculator, let's take a closer look at the various scales on the front side, as indicated in Figure 27.

There are three sliding scale windows on the front side of the duct calculator. The top scale is the VELOCITY (fpm) scale. Directly underneath the velocity is the CFM (airflow quantity) scale, followed by the VELOCITY PRESSURE (in. wg). In most cases, you will not have any use for the velocity pressure and we will not use it in this training module.

First, let's look at an actual example of how we can use the duct calculator for calculating duct pressure loss (Figure 29). Let's assume that we have a 12-in. round duct with a flow rate of 700 cfm. Line up the black pointer on the bottom window with the 12-in. round duct (Point 1).

In the top window, notice where the 700 cfm line and the velocity lines meet (Point 2). At that point you can read a velocity of approximately 900 fpm. Next, in the center window, read the friction loss where its scale intersects the 700 cfm line directly below it (Point 3). This is read as 0.10 in. wg/100' EL.

Now determine the sizes for rectangular ducts of the same friction rate by reading them on the bottom two scales. Notice that the larger dimension duct is indicated on the upper scale and the smaller dimension is indicated on the bottom scale. There are many possible combinations of rectangular dimensions that will work.

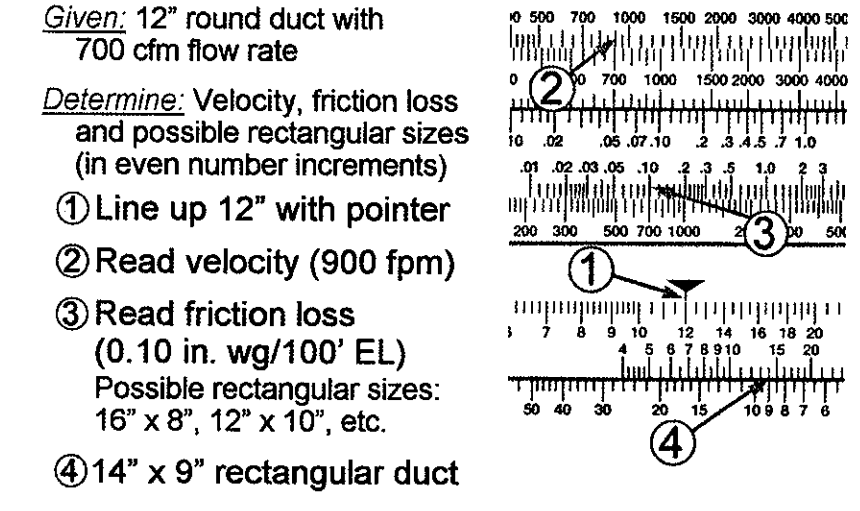


Figure 29

Duct Friction Loss Calculation Example

Keep in mind that aspect ratios should be kept below 4:1 if at all possible. Most sheet metal fabrication shops prefer to build rectangular ductwork in even number dimensions (2-in., 4-in., 6-in., etc.) so we should not select an 11-in., 9-in., 7-in., or 5-in. duct.

You can always upsize or downsize one or both of the dimensions to make them even numbers. Slide the scale and line up any two even numbers that are close to your original values. Be sure to make the upper scale the larger dimension. Keep in mind, however, that when you slide the scale to a new position, your velocity and friction loss values will also change.

For instance, according to the duct calculator (Point 4), a 14" x 9" rectangular duct will also work for our example. Because we cannot use 9-in. duct, we need to either round up or round down to the next even size, 10-in. or 8-in., respectively. Let's round up to a 14" x 10" duct. Slide the scale up so that the 14-in. on the black scale matches the 10-in. scale on the green scale. Notice that since we are now using a slightly larger than necessary duct, the friction rate, read in the center window, has dropped from 0.10 down to a little less than 0.08 in. wg/100' EL.

When using the duct calculator, you need only two values to arrive at the remaining values. For instance, if you know the maximum velocity and the cfm, you can read all of the other values (friction loss, round duct size, and rectangular duct size). The only exception is that rectangular duct velocity cannot be read directly from the duct calculator. The velocity scale is for round duct only. Therefore, the duct calculator may be used to size duct on new designs or to verify existing ductwork for pressure losses and sizes.

Now let's flip the duct calculator over and review the information on the backside.

The scales on the back are for calculating the approximate friction loss equivalents for materials other than medium-smooth sheet metal ductwork. These scales are shown in Figure 30.

Let's perform a friction loss conversion from smooth sheet metal ductwork to the other materials. Line up the FRICTION LOSS arrow (top scale) with 0.08 in. wg. Read the friction losses for the other materials in the appropriate windows. For duct board or duct liner, the friction loss is 0.105 in. wg. When installed properly, flexible metal ductwork has a friction loss of approximately 0.13 in. wg. Duct liner with an airside spray coating has a friction loss of about 0.15 in. wg, and flexible vinyl-coated duct (flex) has a friction loss of 0.26 in. wg, which is more than three times higher than that of smooth sheet metal.

Given: **Friction loss for sheet metal duct = 0.08 in. wg**

Determine: **Friction loss for other duct materials**

- **Duct board = 0.105 in. wg**
- **Metal flex (installed straight) = 0.13 in. wg**
- **Duct liner with airside spray coating = 0.15 in. wg**
- **Flexible, vinyl-coated duct (flex) = 0.26 in. wg**

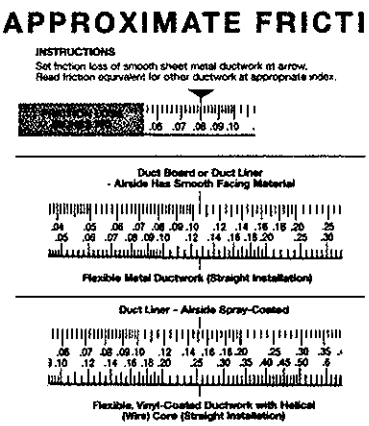


Figure 30

Conversion of Friction Loss Factor

There is another very useful feature of this friction equivalent chart. Let's suppose we are using a duct with spray-coated liner. In addition, let's assume that we want to design the system for a maximum pressure loss of 0.10 in. wg/100' EL. How do we proceed? Simply line up the 0.10 in. wg value on the Duct Liner - Airside Spray-Coated Scale (Point 1 on Figure 31). Read the top scale at the black pointer as 0.052 in. wg (Point 2). Let's round this value down to 0.05 in. wg for simplicity. Remember, the duct calculator sizes indicated on the front side are for smooth sheet metal; however, we now know that the equivalent friction loss needs to be 0.05 in. wg. Now flip the duct calculator over and line up the friction loss scale at 0.05 in. wg with the desired cfm. You may now proceed as before to complete your duct sizing.

Given: **Spray-coated liner and a design friction rate of 0.10 in. wg**

Determine: **The equivalent friction factor for smooth sheet metal duct**

① Line up 0.10 in. wg on Duct Liner – Airside Spray-Coated Scale

② Read equivalent friction factor of 0.052 in. wg.

Use this equivalent friction factor on front side and proceed with design.

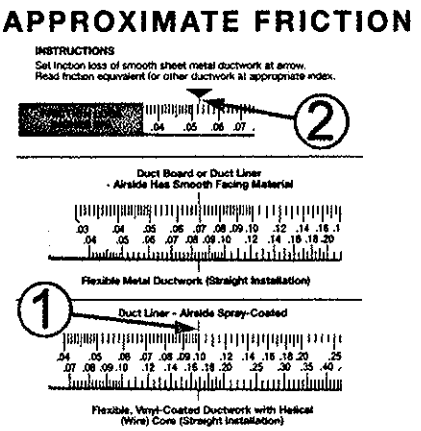
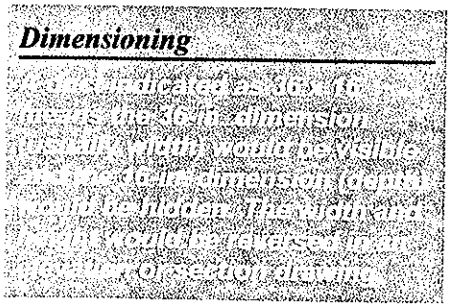


Figure 31

Friction Loss Factor Conversion Example

Changing One Dimension at a Time

Ductwork is specified and constructed to whole inch increments, not fractions. Rectangular ductwork can be constructed to "odd" number dimensions (7-in., 9-in., 11-in., etc.), but generally most fabricators prefer to manufacture fittings in even number dimensions to minimize inventory requirements and scrap.

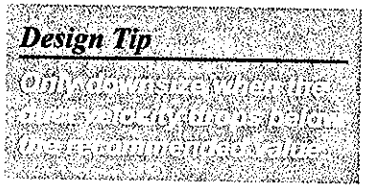


Round duct, especially flexible-type, is readily available in both even and odd dimensions up to 10-in., then in 2-in. increments.

When looking at a set of plans, there are generally several views available: a plan view, an elevation view, and a section view. The plan view, which is the most common duct drawing, the standard convention for representing duct dimensions is to always list the first number of the duct dimension as the visible dimension.

Extended Plenum

A concept that embraces limiting the number of size reduction fittings, or transitions, in trunk ducts and terminal header ducts is the extended plenum design. Instead of making a size change at every diverging or converging fitting, the supply or return duct main is run for a substantial number of fittings at the same size. The selected size fits the design guidelines at the mid-point cfm value and is slightly over and under-sized at the ends. The higher static pressure loss of the under-sized section is compensated for by the static regain that occurs along the remaining length as velocity reduction occurs. Overall poundage of material is often higher, but labor for fabrication and installation is much less.



A variation, semi-extended plenum, provides for size changes every two or three fittings, still providing many of the benefits without as wide a variation in airflow parameters. Do not exceed 25 ft between size changes.

Return and Supply Air Plenums

Do not forget about the return air duct system design. Some buildings use a ceiling plenum return for the return air, therefore, a ducted return air system may not be required (Figure 32).

Generally, ceiling plenum returns should be used only on intermediate floors of a multiple-story building and not in top floor or attic areas due to the heat gain from the roof. If a plenum return is used on an upper floor the return air loading of the air source coil will be higher and must be accommodated. Even with ducted returns, if they are not insulated, plenum heat gains will affect cooling coil selections.

- Some multi-floor buildings use a ceiling plenum return.
- They are generally used only on intermediate floors due to heat gain from roof.
- A ducted return air is generally used on top floors.

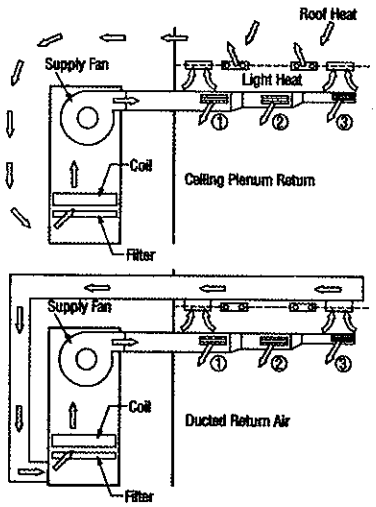


Figure 32

Ceiling Plenum and Ducted Return

Supply air plenum systems have had a resurgence with the advent of underfloor air distribution systems for offices. As with any plenum, air leakage must be contained, and thermal decay of the supply air temperature must be evaluated during design and equipment selection.

Design Step 8: Calculate Air System Pressure Losses

Once the duct sections have been sized and the static pressure friction losses calculated, you can summarize the air pressure losses for the greatest pressure loss circuit or run for both the supply and return ductwork. This is only done for the run with the greatest circuit loss because once selected the fan will always have enough pressure for the other duct runs. This total is often referred to as the ***external static pressure*** because these losses are "external" to the air source. Manufacturers of packaged equipment generally publish fan ratings and fan curves based on external static pressure; that is, internal losses due to coils, dampers, heat exchangers, and often filters, are accounted for in the cataloged fan ratings. This is not the case with applied equipment, so all internal unit air friction losses must be added to the external ductwork values to arrive at the total static pressure of the fan.

Double-check the calculated duct sizes to make sure they will fit into the allowable space. If not, adjust sizes or aspect ratios to fit. Do not forget to allow space for either internal duct liner or external duct insulation.

Design Step 9: Select Fan and Adjust System Airflows

In Step 8 we summarized the static pressure losses for the duct system. To this amount we add any unaccounted pressure losses due to filters, dampers and any other components in the duct system and equipment to arrive at a value referred to as the total static pressure. This total amount is the static pressure that the fan must overcome to deliver the required amount of air. This total amount is also used to select the required fan motor size.

While it is more correct to select the fan using the total pressure difference between the inlet of the fan and the outlet (referred to as the ***fan total pressure)***, in HVAC systems, fans are commonly selected by using the difference between the fan inlet and fan outlet static pressures, not subtracting the velocity pressure of the inlet (often too small to matter).

Once the fan is selected and installed, an air balance usually done by the testing and balancing contractor as part of commissioning the system. This involves measuring the fan speed (rpm) and total static pressure drop across the fan, as well as measuring and summarizing the airflows at all of the diffusers. These measured values are then compared against the required values from the cooling and heating load estimate. Balancing dampers are adjusted to attempt to deliver the correct amount of air to each zone. Often the actual airflow being delivered is less than or greater than required and the fan pulley (sheave) may require adjustment or resizing.

Summary

Like so much of HVAC system design, creating cost effective duct designs is as much an art as it is a science. Bernoulli's Law explains the relationship between velocity and static pressures, a very important concept to understand when sizing ducts, especially when static regain is desired.

Selecting fittings and thinking about the effects of turbulence in the ducts is critical to keep the final designs quiet, self-balanced and at a low system static pressure loss.

Manual duct sizing and selection of fittings using charts and tables present the designer with the optimum situation for reinforcing duct design principals and gaining experience in balancing the many design criteria that often create conflicting requirements. Once learned through this manual chart method, use of a duct calculator or design software can be properly applied to speed up the process and facilitate using the static regain effect to improve system performance.

Work Session 2 – Duct Sizing

This is a good time to complete Work Session 2 and test your knowledge on duct sizing.

Example 3 – Equal Friction Sizing Example Using the Duct Friction Table

This detailed work session is included for your practice at applying the skills covered in this duct design module. It is recommended that you take time to work the example soon after the material is presented to confirm the principals and give you time to discuss any issues with the instructor before going on to the next topic. In actual practice you would not lay out a duct system like this with diffusers in the trunk duct because it would be too noisy.

This method assumes that the user has already completed the required steps as outlined in the previous "Duct Design Process Steps."

The following design procedure is recommended when using the equal friction method:

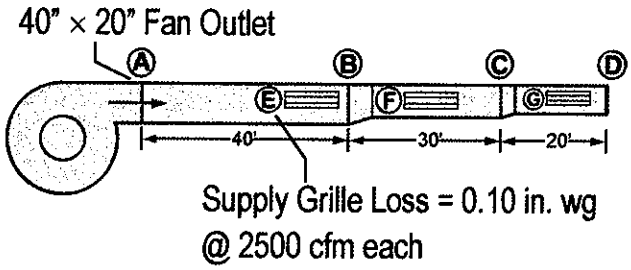


Figure 33

System Total Static Pressure and Fan Selection

The first section of duct downstream of the fan is not sized by any particular method (equal friction or static regain). Rather, it is sized based on an assumed initial velocity. Designers sometimes size the first section of duct to be the same size as the fan outlet connection, provided the outlet velocity is within an acceptable range. If the air source discharge velocity is quite high, quickly increase the duct to bring it within recommendations. Remember to try to provide several feet of straight duct directly off the fan to prevent system effect from occurring. Recommended and maximum velocities are listed in Table 2 Recommended Maximum Velocities.

Remember that the friction loss values that appear on the duct friction chart are for round, galvanized, smooth sheet metal duct. If you are using a duct material other than galvanized sheet metal, you will need to correct the friction factor indicated. This is done by applying a material correction factor, which is listed in Table 4 Duct Roughness Multipliers. If you are not using round duct you must first size the system as round duct then convert the round sizes to equivalent rectangular dimensions by using Table 1 Friction Loss Chart for Round Duct.

The use of flex duct should be minimized and used in no greater than 6 to 8-ft lengths since the equivalent pressure drop of flex duct is over three times that of galvanized sheet metal.

In the example duct system shown in Figure 33, we have a 40-in. by 20-in. fan connection. Since most fan outlets are rectangular in shape, you will generally have to supply a transition from the fan outlet dimensions to the dimensions of the first duct section. Let's assume that we will be using rectangular, smooth sheet metal duct for this system. From Table 2, for a general office, we will use a maximum velocity of 1500 fpm, which is the controlling factor for noise generation. The total airflow in section A-B is the sum of all diffusers, $7500 = 2500\ cfm * 3$.

Since we now know the velocity and airflow we can solve for the area:

$$A = Q / V = 7500\ cfm / 1500\ fpm = 5.0\ ft^2$$

Now enter Table 1 (Circular Equivalent Diameter) and locate a combination of duct dimensions with an area equal to or greater than 5.0 sq ft. From Table 1 we find that there are many possible combinations of duct dimensions with an area of 5.0 sq ft. Since the fan outlet is 40 x 20, let's check the velocity using the same size duct as the fan outlet. From Table 1 for a 40 x 20 duct, the equivalent area is 5.07 sq ft, which is adequate. Therefore, a transition from the fan to duct section is not required and there is no fan outlet gain or loss to consider. In Table 1, under each column of duct dimensions, there is a value labeled "Diam, in." This is the diameter for a round duct with an equivalent friction loss. For a 40" x 20" rectangular duct, the round equivalent is 30.5 in.

Next, on the Duct Friction Chart, locate the 7500 cfm line on the vertical axis. Follow this line to the right until it intersects with the 30.5 in. round duct line. Now read vertically to determine the friction loss value, in our case 0.085 in. wg/100 ft EL. This means that for a flow rate of 7500 cfm, for a 30.5 in. round duct, the friction loss for a 100 ft length section will be 0.085 in. wg. But section A-B is only 40 ft long, the corresponding friction loss will therefore be:

$$\Delta P_S = 0.085 * (40/100) = 0.034 \text{ in. wg}$$

This corresponds to a velocity of approximately 1500 fpm, which may be read from the friction chart. We will use this friction rate ($f = 0.085$ in. wg) to size the remaining duct sections.

Using the Duct Sizing Worksheet included in the Appendix, enter the information we know thus far.

The next element in the system is outlet E in section A-B with an airflow rate of 2500 cfm. The outlet loss is given as 0.10 in. wg. You should consult actual diffuser, grille and register catalogs when designing duct systems to determine the appropriate friction losses. Add this known loss value to the worksheet. Consider the duct length from the fan to outlet E, section A-E, to be the same 40 ft we just used for section A-B.

Next we come to duct run B-C. Since the airflow quantity is reduced significantly (7500 - 2500 = 5000 cfm) we need to transition to a smaller duct. To determine the required rectangular dimensions of section B-C we must first find the required round size then convert it to a rectangular equivalent. Refer to the Friction Loss Chart and use an equal friction rate of 0.085 in. wg. Determine the intersection of the airflow quantity (5000 cfm) with the friction rate. The intersection point falls very close to the 26 in. diameter line. Now refer to Table 1 and locate a combination of rectangular dimensions that are close to a 26 in. diameter. There are several possibilities. As mentioned previously, it is generally a good idea to try to maintain at least one of the dimensions when transitioning duct sizes. This makes the fitting less difficult to fabricate and saves money too. Let's maintain a constant duct height of 20 in. and vary the duct width. From Table 1 we will select a 28" x 20" size for section B-C. This size is equivalent to a 25.7 in. round duct with a velocity of approximately 1400 fpm and a friction rate of 0.095 in. wg. Here the duct section is only 30 ft long, so the static pressure loss will be less. Compute the value and enter it into the Worksheet.

Continuing, we can now calculate the friction loss due to the transition that starts section B-C. Refer to Table 5 (Friction of Rectangular Duct System Elements), specifically the contraction fitting, using a 30° angle for least amount of static pressure loss. According to the table notes, the P_S loss is equal to the change in velocity pressure (downstream – upstream) times the fitting value n. The velocity pressures can be looked up on the Friction Loss Chart or from Table 3. Using $n = 1.02$, and finding $P_{V1} = 0.062$ (remember, the upstream 30.5-in. duct only has 5000 cfm in it too) and $P_{V2} = 0.122$, the Table 5 formula would give a fitting loss of:

$$\Delta P_S = 1.02 * (0.122 - 0.062) = 0.061 \ \ in.wg$$

Since there is static regain from the decrease in velocity due to the outlet, use a value less than calculated, say 50 percent in this case (loss becomes 0.031). In the Duct Design, Level 2 TDP you will see that empirically derived fitting loss coefficients will give more reliable data in many cases.

For all practical purposes, this is an insignificant amount. You should note that the total pressure loss due to a fitting is highly influenced by the velocity. Had the velocity in the previous example been say twice as high, the pressure loss would have increased by a factor of four times due to the "squared" relationship between pressure and velocity. Add these values to the worksheet.

The last element in duct run B-C is outlet F, with a flow rate of 2500 cfm. Again, this loss is given as 0.10 in. wg. Add this value into the Worksheet.

The final duct run is C-D. Similar to the previous section, we have another transition since the airflow is reduced to 2500 cfm. We will maintain our previous duct height of 20" and locate the width required. As before, we must determine the required round duct size and then convert it to an equivalent rectangular size. From the Friction Loss Chart find the round duct size corresponding to the friction rate of 0.09 in. wg. A 20-in. round duct meets our requirements for section C-D. Next, in Table 1 locate a round duct with a 20-in. diameter that corresponds to a duct height of 20 in. A 20 in. by 16 in. duct has an equivalent diameter of 19.5-in., a round duct velocity of 1200 fpm, and a corresponding friction rate of 0.10 in. wg. Here too the duct length has shortened to 20 ft, so losses will reduce as well. Complete the calculations and add these values to the worksheet, remembering that again we have an outlet G.

Now we need to calculate the loss due to the transition at the start of section C-D. This transition is the same type as the one before. Since we now know the sizes of both sections, we can once again determine the velocity pressures and plug them into the table formula, again using the loss reduction adjustment of 50 percent to account for static regain.

$$\Delta P_S = 1.02 * (0.096 - 0.032) * 50\% = 0.033 \ in.wg$$

We are now ready to calculate the static pressure friction loss of each duct run. From the Duct Sizing Worksheet (Figure 34) we see that the cumulative static pressure loss for duct run A-E is 0.134 in. wg, for duct run A-F it is 0.194 in. wg, and the longest duct run A-G it has increased to 0.247 in. wg. Since our system was simple, the Worksheet was tabulated fully for each of the three duct runs. Alternatively, only the individual duct sections and outlets could have been entered. Then, the potential highest pressure loss runs would need to be checked, which in our case was easily the longest run. On large systems this is somewhat of a tedious task, making experience a good teacher for searching out and checking only those runs that exhibit the potential for large pressure drop. If you choose to use a duct design software program to size the ductwork, the highest pressure drop run will be a program output after the sizing is done.

Comparing the friction losses of the three runs, or circuits, they are within 0.12 in. wg of each other. Therefore, air balancing will be easy to accomplish by the use of diffuser opposed blade control dampers.

In this example there is no return ductwork, so the fan can be selected at 0.25 in. wg total static pressure, verifying that the fan was rated with no inlet ductwork.

Remember to allow room for either internal duct lining or external duct wrap (insulation) when designing duct systems.

This completes our calculations for the equal friction method sizing example.

Figure 34

Duct Sizing Worksheet (Example 3)

DUCT SIZING WORKSHEET

PROJECT NAME: EQUAL FRICTION EXAMPLE

DATE: 10/1/04

SYSTEM: SUPPLY

PAGE: 1 OF 1

Duct Run From-To	Duct Section (element)	Lining (in.)	Insul. (in.)	Other Item	Airflow	Velocity in Round duct (fpm)	Velocity Pressure P_V	Fitting Value n	Length (ft) L	Equiv. Length (ft) EL	Material Correction Factor	Friction Loss / per 100' duct	Friction Loss (in. wg)	Known Loss (in. wg)	Round Duct Size (in.)	Equivalent Rectangular Size (W x H)	Total Item Loss (in. wg)	Cumulative Loss (in. wg)
A-E	A-B	-	1	-	7500	1500	0.140	-	40	-	-	0.085	0.034	-	30.5	40 x 20	0.034	0.034
	E	-	-	OUTLET	2500	-	-	-	-	-	-	-	-	0.10	-	-	0.10	0.134
A-F	A-B	-	1	-	7500	1500	0.014	-	40	-	-	0.085	0.034	-	30.5	40 x 20	0.034	0.034
	B-C	-	1	-	5000	1400	0.122	-	30	-	-	0.095	0.029		25.7	28 x 20	0.029	0.063
	B	-	1	TRANS	5000 / 5000	1040 / 1400	0.062 / 0.122	1.02	-	-	-	-	0.061	-50%	-	40 x 20 / 28 x 20	0.031	0.094
	F	-	-	OUTLET	2500	-	-	-	-	-	-	-	-	0.10	-	-	0.10	0.194
A-G	A-B	-	1	-	7500	1500	0.140	-	40	-	-	0.085	0.034	-	30.5	40 x 20	0.034	0.034
	B-C	-	1	-	5000	1400	0.122	-	30	-	-	0.095	0.029		25.7	28 x 20	0.029	0.063
	B	-	1	TRANS	5000 / 5000	1040 / 1400	0.062 / 0.122	1.02	-	-	-	-	0.061	-50%	-	40 x 20 / 28 x 20	0.031	0.094
	C-D	-	1	-	2500	1200	0.096	-	20	-	-	0.10	0.02	-	20.7	16 x 20	0.02	0.114
	C	-	1	TRANS	2500 / 2500	715 / 1200	0.032 / 0.096	1.02	-	-	-	-	0.065	-50%	-	28 x 20 / 16 x 20	0.033	0.147
	G	-	-	OUTLET	2500	-	-	-	-	-	-	-	-	0.10	-	-	0.10	0.247

NOTES: All duct sizes indicated are inside clear dimensions.

LARGEST STATIC PRESSURE LOSS (in. wg) 0.247

FOR RUN = A - G

Work Session 1 – Fundamentals

1. Define the following terms:

   Total Pressure: _____

   Velocity Pressure: _____

   Static Pressure: _____

2. Which of the following affects duct friction loss? (Choose all that apply): _____

   a.) duct size
   b.) duct length
   c.) thickness of duct wrap
   d.) air velocity
   e.) duct construction material
   f.) fitting type

3. True or False? A fan begins to convert static pressure into velocity pressure in the first few feet of supply duct. _____

4. A bowling ball sitting still at the top of a hill contains what type of energy? _____

   If this same ball is rolling downhill it contains mostly what type of energy? _____

5. Define Bernoulli's Law in simple terms: _____

   _____

   _____

6. *Given:* An airflow of 2250 cfm through a round duct and a square duct with the same cross-sectional area, 1.5 sq ft.

   Calculate: The velocity of the air flowing through each duct. _____

   _____

7. Which shape duct (round or square) has the highest pressure loss per unit length? _____

   Why? _____

8. *Given:* Refer to the ducts in Question 7 above, with a cross-sectional area of 1.5 sq ft.

   Calculate: The maximum allowable airflow through the duct if the maximum velocity of the air in the duct is 1,000 feet per minute.

   _____

9. Increasing the aspect ratio of a duct causes which of the following to occur? (Choose all that apply):

a.) the ratio of perimeter to area increases
b.) the weight of the duct increases
c.) the duct heat gain increases
d.) the duct cross-sectional area increases
e.) the fabrication cost is increased
f.) the friction loss increases

10. Given a design friction rate of 0.08 in. wg/100 ft EL for a smooth sheet metal supply duct, what is the equivalent friction rate for flexible, vinyl-coated ductwork with helical wire core?

11. The phenomenon of system effect is caused by which of the following conditions?

(Choose all that apply):

a.) an undersized duct system
b.) an oversized fan
c.) an elbow located directly at the fan outlet
d.) turning vanes in an elbow
e.) a very short supply duct
f.) a very long supply duct

Work Session 2 – Duct Sizing

1. The _____ is the total amount of air that the fan must deliver.

2. Define the term block load cfm _____

   _____

   _____

3. True or False? Flex duct should be used as much as possible because it is easier and less expensive to install. _____

   Explain your answer _____

   _____

   _____

4. List the advantages and disadvantages of using lined ductwork: _____

   _____

   _____

   _____

5. Why should you provide several feet of straight duct immediately after the fan? _____

   _____

   _____

   _____

6. True or False? It is a good design practice to downsize trunk duct sections after each branch take-off to save money. _____.

   Why? _____

   _____

   _____

   _____

7. Describe the recommended procedure for sizing the first section of duct downstream of the fan.

_____

_____

8. True or False? Ceiling plenums should be used for return air on all floors of a building whenever possible. _____.

   Why? _____

_____

_____

_____

_____

_____

9. Choose the correct answer. The sum of all pressure losses external to the HVAC unit is referred to as the: _____

   a.) duct friction loss
   b.) total friction loss
   c.) external static pressure
   d.) velocity pressure
   e.) total static pressure
   f.) total fitting loss

10. Which of the following best describes a duct system with an operating static pressure of 3.25 in. wg?

    a.) high velocity
    b.) SMACNA Pressure Class 3
    c.) high pressure
    d.) low velocity
    e.) SMACNA Pressure Class 4

11. Choose the correct answer. The sum of all pressure losses from the duct system plus any internal unit losses such as coils, filters, etc. is referred to as the:

    a.) duct friction loss
    b.) total friction loss
    c.) external static pressure
    d.) velocity pressure
    e.) total static pressure
    f.) total fitting loss

12. True or False? The equal friction method of duct design means that all duct sections have an equal friction loss. _____.

13. True or False? The equal friction method should not be used to design return air duct systems. _____.

14. Which of the following are advantages of the equal friction method as compared to the static regain method? (Choose all that apply):

a.) duct system is self-balancing
b.) lower friction loss
c.) easier to design
d.) results in smaller ducts
e.) reduced fan energy requirements
f.) reduced installation costs

15. What are the advantages of the modified equal friction method of duct design? _____

16. List the advantages and disadvantages of the static regain method of duct design: _____

17. Describe the phenomenon of static regain in ducts: _____

Appendix and Supplemental Material

The tables and supplemental material presented in the back of the module are presented both for teaching the duct design principles covered in this module and for the designer's continued use afterwards. Further information on duct design is available in many of the references. Please continue your growth in duct design and enjoy the knowledge and experience you gain in successfully applying this aspect of HVAC system design.

Chart 1 – Friction Loss Chart for Round Duct

Table 1 – Circular Equivalent Diameter, Equivalent Area, and Duct Class of Rectangular Ducts for Equal Friction

Table 2 – Recommended Maximum Duct Velocities for Low Velocity Systems (fpm)

Table 3 – Velocity Pressures

Table 4 – Duct Roughness Multipliers

Table 5 – Friction of Rectangular Duct System Elements (from SDM-2, Table 10)

Table 6 – Friction of Round Elbows (from SDM-2, Table 11)

Table 7 – Friction of Rectangular Elbows (from SDM-2, Tabl3 12)

Table 8 – Flat Oval Equivalent Duct Sizes

Symbols for HVAC Systems (from SMACNA)

Duct Sizing Worksheet

Glossary

References

Work Session 1 Answers

Work Session 2 Answers

Chart 1 – Friction Loss Chart for Round Duct

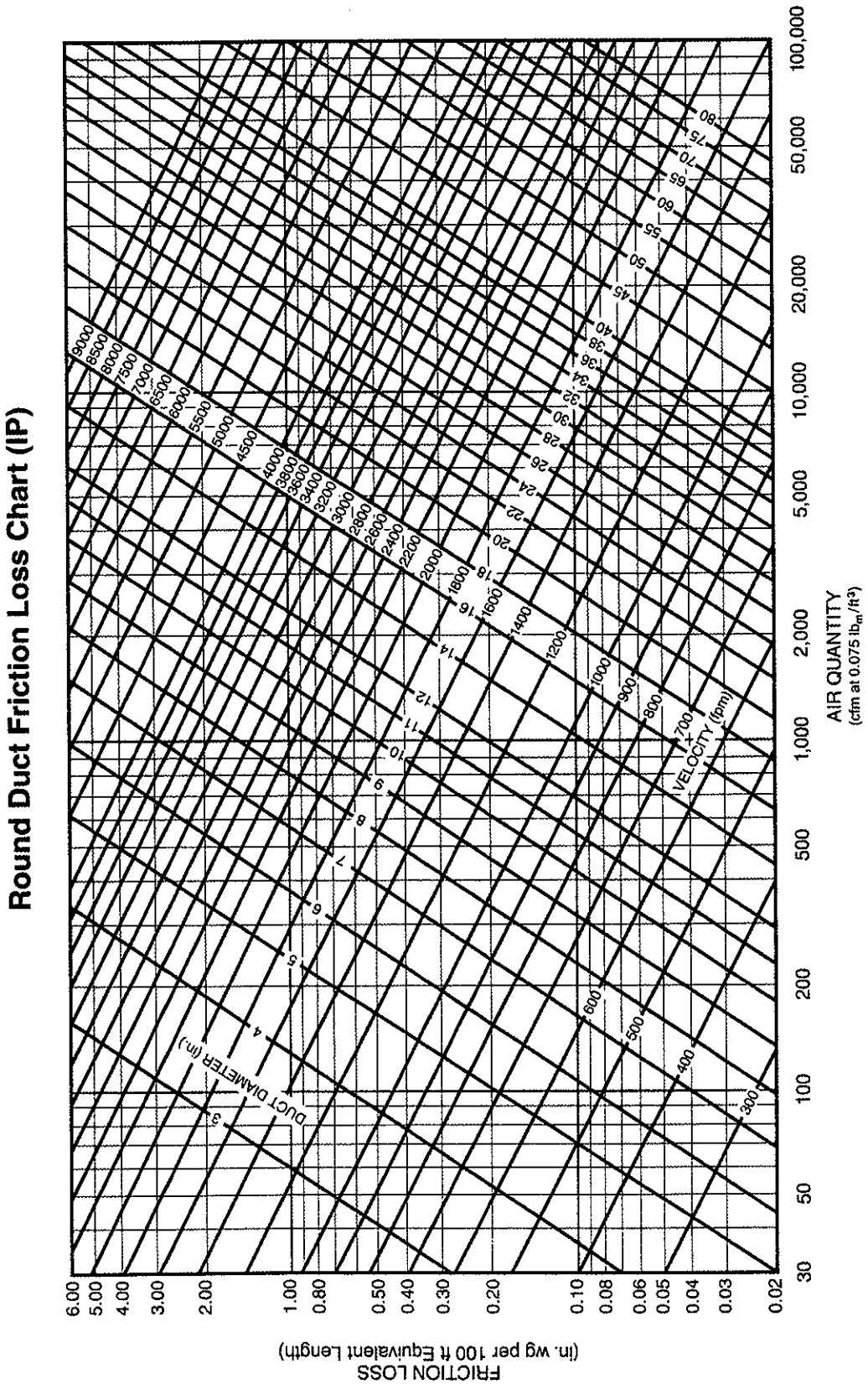


Table 1 – Circular Equivalent Diameter*, Equivalent Area of Rectangular Ducts for Equal Friction

SIDE	6		8		10		12		14		16		18		20		22	
	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)
10	0.38	8.4	0.52	9.8	0.65	10.9												
12	0.46	9.1	0.62	10.7	0.78	12.0	0.94	13.1										
14	0.52	9.8	0.72	11.5	0.91	12.9	1.09	14.2	1.28	15.3								
16	0.59	10.4	0.81	12.2	1.03	13.7	1.24	15.1	1.46	16.4	1.67	17.5						
18	0.66	11.0	0.90	12.9	1.15	14.5	1.39	16.0	1.64	17.3	1.88	18.5	2.11	19.7				
20	0.72	11.5	0.99	13.5	1.27	15.2	1.54	16.8	1.81	18.2	2.08	19.5	2.34	20.7	2.61	21.9		
22	0.78	12.0	1.08	14.1	1.38	15.9	1.68	17.6	1.98	19.1	2.28	20.4	2.57	21.7	2.87	22.9	3.15	24.0
24	0.84	12.4	1.16	14.6	1.49	16.5	1.82	18.3	2.15	19.9	2.48	21.3	2.80	22.7	3.12	23.9	3.44	25.1
26	0.90	12.8	1.25	15.1	1.60	17.1	1.96	19.0	2.32	20.6	2.67	22.1	3.02	23.5	3.37	24.9	3.72	26.1
28	0.96	13.2	1.33	15.6	1.71	17.7	2.10	19.6	2.48	21.3	2.86	22.9	3.25	24.4	3.62	25.8	4.00	27.1
30	1.01	13.6	1.41	16.1	1.82	18.3	2.23	20.2	2.64	22.0	3.05	23.7	3.46	25.2	3.87	26.6	4.28	28.0
32	1.07	14.0	1.49	16.5	1.92	18.8	2.36	20.8	2.80	22.7	3.24	24.4	3.68	26.0	4.11	27.5	4.55	28.9
34	1.12	14.4	1.57	17.0	2.03	19.3	2.49	21.4	2.96	23.3	3.42	25.1	3.89	26.7	4.36	28.3	4.82	29.7
36	1.18	14.7	1.65	17.4	2.13	19.8	2.62	21.9	3.11	23.9	3.61	25.7	4.10	27.4	4.59	29.0	5.09	30.5
38	1.23	15.0	1.73	17.8	2.23	20.2	2.75	22.4	3.27	24.5	3.79	26.4	4.31	28.1	4.83	29.8	5.36	31.3
40	1.28	15.3	1.80	18.2	2.33	20.7	2.87	22.9	3.42	25.0	3.97	27.0	4.51	28.8	5.06	30.5	5.61	32.1
42	1.34	15.6	1.88	18.5	2.43	21.1	3.00	23.4	3.57	25.6	4.14	27.6	4.72	29.4	5.29	31.2	5.87	32.8
44	1.39	15.9	1.95	18.9	2.53	21.5	3.12	23.9	3.71	26.1	4.32	28.1	4.92	30.0	5.52	31.8	6.13	33.5
46	1.44	16.2	2.02	19.3	2.62	21.9	3.24	24.4	3.86	26.6	4.49	28.7	5.12	30.6	5.75	32.5	6.38	34.2
48	1.49	16.5	2.09	19.6	2.72	22.3	3.36	24.8	4.01	27.1	4.66	29.2	5.31	31.2	5.97	33.1	6.63	34.9
50	1.54	16.8	2.17	19.9	2.81	22.7	3.48	25.2	4.15	27.6	4.83	29.8	5.51	31.8	6.20	33.7	6.88	35.5
52			2.24	20.2	2.91	23.1	3.59	25.7	4.29	28.0	4.99	30.3	5.70	32.3	6.42	34.3	7.13	36.2
54			2.31	20.6	3.00	23.5	3.71	26.1	4.43	28.5	5.16	30.8	5.90	32.9	6.63	34.9	7.37	36.8
56			2.37	20.9	3.09	23.8	3.82	26.5	4.57	28.9	5.33	31.2	6.09	33.4	6.85	35.4	7.62	37.4
58			2.44	21.2	3.18	24.2	3.94	26.9	4.71	29.4	5.49	31.7	6.27	33.9	7.07	36.0	7.86	38.0
60			2.51	21.5	3.27	24.5	4.05	27.3	4.85	29.8	5.65	32.2	6.46	34.4	7.28	36.5	8.10	38.5
62			2.58	21.7	3.36	24.8	4.16	27.6	4.98	30.2	5.81	32.6	6.65	34.9	7.49	37.1	8.34	39.1
64			2.65	22.0	3.45	25.1	4.27	28.0	5.12	30.6	5.97	33.1	6.83	35.4	7.70	37.6	8.57	39.6
66			2.71	22.3	3.54	25.5	4.39	28.4	5.25	31.0	6.13	33.5	7.01	35.9	7.91	38.1	8.81	40.2
68			2.78	22.6	3.62	25.8	4.49	28.7	5.38	31.4	6.28	33.9	7.20	36.3	8.12	38.6	9.04	40.7
70					3.71	26.1	4.60	29.1	5.51	31.8	6.44	34.4	7.38	36.8	8.32	39.1	9.27	41.2
72					3.80	26.4	4.71	29.4	5.65	32.2	6.59	34.8	7.56	37.2	8.52	39.5	9.50	41.7
74					3.88	26.7	4.82	29.7	5.78	32.5	6.75	35.2	7.73	37.7	8.73	40.0	9.73	42.2
76					3.97	27.0	4.92	30.0	5.90	32.9	6.90	35.6	7.91	38.1	8.93	40.5	9.96	42.7
78					4.05	27.3	5.03	30.4	6.03	33.3	7.05	36.0	8.09	38.5	9.13	40.9	10.18	43.2
80					4.13	27.5	5.14	30.7	6.16	33.6	7.20	36.3	8.26	38.9	9.33	41.4	10.40	43.7
82					4.22	27.8	5.24	31.0	6.29	33.9	7.35	36.7	8.43	39.3	9.53	41.8	10.63	44.1
84							5.34	31.3	6.41	34.3	7.50	37.1	8.61	39.7	9.72	42.2	10.85	44.6
86							5.45	31.6	6.54	34.6	7.65	37.4	8.78	40.1	9.92	42.6	11.07	45.0
88							5.55	31.9	6.66	34.9	7.80	37.8	8.95	40.5	10.11	43.1	11.29	45.5
90							5.65	32.2	6.79	35.3	7.94	38.2	9.12	40.9	10.30	43.5	11.50	45.9
92							5.75	32.5	6.91	35.6	8.09	38.5	9.28	41.3	10.50	43.9	11.72	46.4
94							5.85	32.8	7.03	35.9	8.23	38.8	9.45	41.6	10.69	44.3	11.94	46.8
96							5.95	33.0	7.15	36.2	8.38	39.2	9.62	42.0	10.88	44.7	12.15	47.2
98							6.05	33.3	7.27	36.5	8.52	39.5	9.78	42.4	11.07	45.0	12.36	47.6
100									7.39	36.8	8.66	39.8	9.95	42.7	11.25	45.4	12.57	48.0
102									7.51	37.1	8.80	40.2	10.11	43.1	11.44	45.8	12.78	48.4
104									7.63	37.4	8.94	40.5	10.28	43.4	11.63	46.2	12.99	48.8
106									7.75	37.7	9.08	40.8	10.44	43.7	11.81	46.5	13.20	49.2
108									7.87	38.0	9.22	41.1	10.60	44.1	12.00	46.9	13.41	49.6
110									7.99	38.3	9.36	41.4	10.76	44.4	12.18	47.3	13.62	50.0
112											9.50	41.7	10.92	44.7	12.36	47.6	13.82	50.3
114											9.64	42.0	11.08	45.1	12.55	48.0	14.03	50.7
116											9.77	42.3	11.24	45.4	12.73	48.3	14.23	51.1
118											9.91	42.6	11.40	45.7	12.91	48.6	14.44	51.4
120											10.05	42.9	11.55	46.0	13.09	49.0	14.64	51.8
122											10.18	43.2	11.71	46.3	13.26	49.3	14.84	52.2
124											10.32	43.5	11.87	46.6	13.44	49.6	15.04	52.5
126											10.45	43.8	12.02	46.9	13.62	50.0	15.24	52.9
128											10.58	44.0	12.18	47.3	13.80	50.3	15.44	53.2
130													12.33	47.5	13.97	50.6	15.64	53.5
132													12.48	47.8	14.15	50.9	15.83	53.9
134													12.64	48.1	14.32	51.2	16.03	54.2
136													12.79	48.4	14.50	51.6	16.23	54.5
138													12.94	48.7	14.67	51.9	16.42	54.9
140													13.09	49.0	14.84	52.2	16.62	55.2
142													13.24	49.3	15.01	52.5	16.81	55.5
144													13.39	49.6	15.19	52.8	17.00	55.8

*Circular equivalent diameter (d_e). Calculated from $d_e = 1.3 * ((ab)^{.625}/(a+b)^{.25})$

Table 1 – Circular Equivalent Diameter*, Equivalent Area of Rectangular Ducts for Equal Friction (Cont.)

SIDE	24		26		28		30		32		34		36		38		40	
	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)
10																		
12																		
14																		
16																		
18																		
20																		
22																		
24	3.75	26.2																
26	4.07	27.3	4.41	28.4														
28	4.37	28.3	4.74	29.5	5.11	30.6												
30	4.68	29.3	5.08	30.5	5.47	31.7	5.87	32.8										
32	4.98	30.2	5.41	31.5	5.83	32.7	6.26	33.9	6.67	35.0								
34	5.28	31.1	5.74	32.4	6.19	33.7	6.64	34.9	7.09	36.1	7.53	37.2						
36	5.57	32.0	6.06	33.3	6.54	34.6	7.02	35.9	7.50	37.1	7.98	38.2	8.45	39.4				
38	5.87	32.8	6.38	34.2	6.89	35.6	7.40	36.8	7.91	38.1	8.41	39.3	8.91	40.4	9.41	41.5		
40	6.16	33.6	6.70	35.1	7.24	36.4	7.78	37.8	8.32	39.0	8.85	40.3	9.38	41.5	9.91	42.6	10.43	43.7
42	6.44	34.4	7.02	35.9	7.59	37.3	8.15	38.7	8.72	40.0	9.28	41.3	9.84	42.5	10.40	43.7	10.95	44.8
44	6.73	35.1	7.33	36.7	7.93	38.1	8.53	39.5	9.12	40.9	9.71	42.2	10.30	43.5	10.88	44.7	11.46	45.8
46	7.01	35.9	7.64	37.4	8.27	38.9	8.89	40.4	9.52	41.8	10.14	43.1	10.75	44.4	11.37	45.7	11.98	46.9
48	7.29	36.6	7.95	38.2	8.60	39.7	9.26	41.2	9.91	42.6	10.56	44.0	11.20	45.3	11.85	46.6	12.49	47.9
50	7.57	37.2	8.25	38.9	8.94	40.5	9.62	42.0	10.30	43.5	10.98	44.9	11.65	46.2	12.33	47.5	13.00	48.8
52	7.84	37.9	8.56	39.6	9.27	41.2	9.98	42.8	10.69	44.3	11.40	45.7	12.10	47.1	12.80	48.4	13.50	49.7
54	8.12	38.6	8.86	40.3	9.60	41.9	10.34	43.5	11.07	45.1	11.81	46.5	12.54	48.0	13.27	49.3	14.00	50.7
56	8.39	39.2	9.16	41.0	9.92	42.7	10.69	44.3	11.46	45.8	12.22	47.3	12.98	48.8	13.74	50.2	14.50	51.6
58	8.65	39.8	9.45	41.6	10.25	43.3	11.04	45.0	11.84	46.6	12.63	48.1	13.42	49.6	14.21	51.0	14.99	52.4
60	8.92	40.4	9.74	42.3	10.57	44.0	11.39	45.7	12.21	47.3	13.03	48.9	13.85	50.4	14.67	51.9	15.48	53.3
62	9.19	41.0	10.04	42.9	10.89	44.7	11.74	46.4	12.59	48.0	13.44	49.6	14.28	51.2	15.13	52.7	15.97	54.1
64	9.45	41.6	10.33	43.5	11.20	45.3	12.08	47.1	12.96	48.7	13.84	50.4	14.71	51.9	15.59	53.5	16.46	54.9
66	9.71	42.2	10.61	44.1	11.52	46.0	12.43	47.7	13.33	49.4	14.24	51.1	15.14	52.7	16.04	54.2	16.94	55.7
68	9.97	42.8	10.90	44.7	11.83	46.6	12.77	48.4	13.70	50.1	14.63	51.8	15.56	53.4	16.49	55.0	17.42	56.5
70	10.23	43.3	11.18	45.3	12.14	47.2	13.10	49.0	14.06	50.8	15.03	52.5	15.98	54.1	16.94	55.7	17.90	57.3
72	10.48	43.8	11.47	45.8	12.45	47.8	13.44	49.6	14.43	51.4	15.42	53.2	16.40	54.8	17.39	56.5	18.38	58.0
74	10.74	44.4	11.75	46.4	12.76	48.4	13.77	50.3	14.79	52.1	15.80	53.8	16.82	55.5	17.84	57.2	18.85	58.8
76	10.99	44.9	12.02	47.0	13.06	48.9	14.11	50.9	15.15	52.7	16.19	54.5	17.23	56.2	18.28	57.9	19.32	59.5
78	11.24	45.4	12.30	47.5	13.37	49.5	14.44	51.4	15.51	53.3	16.58	55.1	17.65	56.9	18.72	58.6	19.79	60.2
80	11.49	45.9	12.58	48.0	13.67	50.1	14.76	52.0	15.86	53.9	16.96	55.8	18.06	57.5	19.15	59.3	20.25	60.9
82	11.74	46.4	12.85	48.5	13.97	50.6	15.09	52.6	16.21	54.5	17.34	56.4	18.46	58.2	19.59	59.9	20.71	61.6
84	11.98	46.9	13.12	49.0	14.27	51.1	15.41	53.2	16.56	55.1	17.72	57.0	18.87	58.8	20.02	60.6	21.18	62.3
86	12.23	47.3	13.39	49.6	14.56	51.7	15.74	53.7	16.91	55.7	18.09	57.6	19.27	59.4	20.45	61.2	21.63	63.0
88	12.47	47.8	13.66	50.0	14.86	52.2	16.06	54.3	17.26	56.3	18.47	58.2	19.67	60.1	20.88	61.9	22.09	63.6
90	12.71	48.3	13.93	50.5	15.15	52.7	16.38	54.8	17.61	56.8	18.84	58.8	20.07	60.7	21.31	62.5	22.54	64.3
92	12.95	48.7	14.19	51.0	15.44	53.2	16.69	55.3	17.95	57.4	19.21	59.3	20.47	61.3	21.73	63.1	23.00	64.9
94	13.19	49.2	14.46	51.5	15.73	53.7	17.01	55.8	18.29	57.9	19.58	59.9	20.87	61.9	22.15	63.7	23.44	65.6
96	13.43	49.6	14.72	52.0	16.02	54.2	17.32	56.4	18.63	58.4	19.94	60.5	21.26	62.4	22.58	64.3	23.89	66.2
98	13.67	50.1	14.98	52.4	16.31	54.7	17.64	56.9	18.97	59.0	20.31	61.0	21.65	63.0	22.99	64.9	24.34	66.8
100	13.90	50.5	15.25	52.9	16.59	55.2	17.95	57.4	19.31	59.5	20.67	61.6	22.04	63.6	23.41	65.5	24.78	67.4
102	14.14	50.9	15.51	53.3	16.88	55.6	18.26	57.9	19.64	60.0	21.04	62.1	22.43	64.1	23.82	66.1	25.22	68.0
104	14.37	51.3	15.76	53.8	17.16	56.1	18.57	58.3	19.98	60.5	21.40	62.6	22.81	64.7	24.24	66.7	25.66	68.6
106	14.61	51.7	16.02	54.2	17.44	56.6	18.87	58.8	20.31	61.0	21.75	63.2	23.20	65.2	24.65	67.2	26.10	69.2
108	14.84	52.2	16.28	54.6	17.72	57.0	19.18	59.3	20.64	61.5	22.11	63.7	23.58	65.8	25.06	67.8	26.54	69.8
110	15.07	52.6	16.53	55.1	18.00	57.5	19.48	59.8	20.97	62.0	22.47	64.2	23.96	66.3	25.47	68.3	26.97	70.3
112	15.30	53.0	16.78	55.5	18.28	57.9	19.79	60.2	21.30	62.5	22.82	64.7	24.34	66.8	25.87	68.9	27.40	70.9
114	15.53	53.4	17.04	55.9	18.56	58.3	20.09	60.7	21.63	63.0	23.17	65.2	24.72	67.3	26.28	69.4	27.83	71.4
116	15.75	53.7	17.29	56.3	18.83	58.8	20.39	61.1	21.95	63.4	23.52	65.7	25.10	67.8	26.68	69.9	28.26	72.0
118	15.98	54.1	17.54	56.7	19.11	59.2	20.69	61.6	22.28	63.9	23.87	66.2	25.47	68.3	27.08	70.5	28.69	72.5
120	16.21	54.5	17.79	57.1	19.38	59.6	20.99	62.0	22.60	64.4	24.22	66.6	25.85	68.8	27.48	71.0	29.11	73.1
122	16.43	54.9	18.04	57.5	19.65	60.0	21.28	62.5	22.92	64.8	24.57	67.1	26.22	69.3	27.88	71.5	29.54	73.6
124	16.65	55.3	18.28	57.9	19.93	60.4	21.58	62.9	23.24	65.3	24.91	67.6	26.59	69.8	28.27	72.0	29.96	74.1
126	16.88	55.6	18.53	58.3	20.20	60.9	21.87	63.3	23.56	65.7	25.26	68.0	26.96	70.3	28.67	72.5	30.38	74.6
128	17.10	56.0	18.78	58.7	20.47	61.3	22.17	63.8	23.88	66.2	25.60	68.5	27.33	70.8	29.06	73.0	30.80	75.1
130	17.32	56.4	19.02	59.1	20.73	61.7	22.46	64.2	24.19	66.6	25.94	69.0	27.69	71.3	29.45	73.5	31.22	75.7
132	17.54	56.7	19.26	59.4	21.00	62.1	22.75	64.6	24.51	67.0	26.28	69.4	28.06	71.7	29.84	74.0	31.63	76.2
134	17.76	57.1	19.51	59.8	21.27	62.4	23.04	65.0	24.82	67.5	26.62	69.9	28.42	72.2	30.23	74.5	32.05	76.7
136	17.98	57.4	19.75	60.2	21.53	62.8	23.33	65.4	25.14	67.9	26.96	70.3	28.78	72.6	30.62	74.9	32.46	77.1
138	18.20	57.8	19.99	60.5	21.79	63.2	23.62	65.8	25.45	68.3	27.29	70.7	29.15	73.1	31.01	75.4	32.87	77.6
140	18.41	58.1	20.23	60.9	22.06	63.6	23.90	66.2	25.76	68.7	27.63	71.2	29.51	73.6	31.39	75.9	33.28	78.1
142	18.63	58.4	20.47	61.3	22.32	64.0	24.19	66.6	26.07	69.1	27.96	71.6	29.86	74.0	31.77	76.3	33.69	78.6
144	18.84	58.8	20.70	61.6	22.58	64.3	24.47	67.0	26.38	69.5	28.29	72.0	30.22	74.4	32.16	76.8	34.10	79.1

*Circular equivalent diameter (d_e). Calculated from $d_e = 1.3 * ((ab)^{.625}/(a+b)^{.25})$

Table 1 – Circular Equivalent Diameter, * Equivalent Area of Rectangular Ducts for Equal Friction (Cont.)

SIDE	42		44		46		48		50		52		54		56		58	
	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)
42	11.50	45.9																
44	12.04	47.0	12.62	48.1														
46	12.59	48.0	13.19	49.2	13.79	50.3												
48	13.13	49.1	13.76	50.2	14.39	51.4	15.02	52.5										
50	13.66	50.0	14.32	51.2	14.98	52.4	15.64	53.6	16.29	54.7								
52	14.19	51.0	14.89	52.2	15.58	53.4	16.26	54.6	16.94	55.7	17.62	56.8						
54	14.72	52.0	15.45	53.2	16.16	54.4	16.88	55.6	17.59	56.8	18.30	57.9	19.01	59.0				
56	15.25	52.9	16.00	54.2	16.75	55.4	17.49	56.6	18.24	57.8	18.97	59.0	19.71	60.1	20.44	61.2		
58	15.77	53.8	16.55	55.1	17.33	56.4	18.10	57.6	18.88	58.8	19.64	60.0	20.41	61.2	21.17	62.3	21.93	63.4
60	16.30	54.7	17.10	56.0	17.91	57.3	18.71	58.6	19.51	59.8	20.31	61.0	21.10	62.2	21.89	63.4	22.68	64.5
64	17.33	56.4	18.19	57.8	19.06	59.1	19.92	60.4	20.78	61.7	21.63	63.0	22.48	64.2	23.33	65.4	24.18	66.6
68	18.35	58.0	19.27	59.4	20.20	60.8	21.11	62.2	22.03	63.6	22.94	64.9	23.85	66.1	24.76	67.4	25.67	68.6
72	19.36	59.6	20.34	61.1	21.32	62.5	22.30	63.9	23.27	65.3	24.24	66.7	25.21	68.0	26.18	69.3	27.14	70.5
76	20.36	61.1	21.40	62.6	22.43	64.1	23.47	65.6	24.50	67.0	25.53	68.4	26.56	69.8	27.58	71.1	28.60	72.4
80	21.35	62.6	22.44	64.1	23.54	65.7	24.63	67.2	25.72	68.7	26.80	70.1	27.89	71.5	28.97	72.9	30.05	74.2
84	22.33	64.0	23.48	65.6	24.63	67.2	25.78	68.7	26.92	70.3	28.07	71.7	29.21	73.2	30.35	74.6	31.49	76.0
88	23.30	65.4	24.50	67.0	25.71	68.7	26.92	70.2	28.12	71.8	29.32	73.3	30.52	74.8	31.72	76.3	32.91	77.7
92	24.26	66.7	25.52	68.4	26.78	70.1	28.04	71.7	29.30	73.3	30.56	74.9	31.82	76.4	33.07	77.9	34.32	79.3
96	25.21	68.0	26.53	69.7	27.85	71.5	29.16	73.1	30.48	74.8	31.79	76.3	33.11	77.9	34.42	79.4	35.73	80.9
100	26.15	69.2	27.53	71.0	28.90	72.8	30.27	74.5	31.64	76.2	33.01	77.8	34.38	79.4	35.75	81.0	37.12	82.5
104	27.09	70.5	28.52	72.3	29.94	74.1	31.37	75.8	32.80	77.5	34.23	79.2	35.65	80.8	37.07	82.4	38.50	84.0
108	28.02	71.7	29.50	73.5	30.98	75.4	32.46	77.1	33.95	78.9	35.43	80.6	36.91	82.3	38.39	83.9	39.87	85.5
112	28.94	72.8	30.47	74.7	32.01	76.6	33.55	78.4	35.08	80.2	36.62	81.9	38.16	83.6	39.69	85.3	41.23	86.9
116	29.85	74.0	31.44	75.9	33.03	77.8	34.62	79.7	36.21	81.5	37.80	83.3	39.40	85.0	40.99	86.7	42.58	88.4
120	30.75	75.1	32.40	77.1	34.04	79.0	35.69	80.9	37.33	82.7	38.98	84.5	40.63	86.3	42.27	88.0	43.92	89.7
124	31.65	76.2	33.35	78.2	35.04	80.2	36.74	82.1	38.44	84.0	40.15	85.8	41.85	87.6	43.55	89.4	45.25	91.1
128	32.54	77.2	34.29	79.3	36.04	81.3	37.79	83.2	39.55	85.2	41.30	87.0	43.06	88.9	44.82	90.6	46.58	92.4
132	33.43	78.3	35.23	80.4	37.03	82.4	38.84	84.4	40.65	86.3	42.45	88.2	44.27	90.1	46.08	91.9	47.89	93.7
136	34.31	79.3	36.16	81.4	38.01	83.5	39.87	85.5	41.73	87.5	43.60	89.4	45.46	91.3	47.33	93.2	49.20	95.0
140	35.18	80.3	37.08	82.5	38.99	84.6	40.90	86.6	42.82	88.6	44.73	90.6	46.65	92.5	48.57	94.4	50.49	96.2
144	36.05	81.3	38.00	83.5	39.96	85.6	41.92	87.7	43.89	89.7	45.86	91.7	47.83	93.6	49.81	95.6	51.78	97.4

SIDE	60		64		68		72		76		80		84		88		92	
	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)	Area (ft²)	Diam. (in.)
52																		
54																		
56																		
58																		
60	23.46	65.6																
64	25.02	67.7	26.70	70.0														
68	26.57	69.8	28.36	72.1	30.14	74.3												
72	28.10	71.8	30.01	74.2	31.90	76.5	33.79	78.7										
76	29.62	73.7	31.64	76.2	33.66	78.6	35.66	80.9	37.65	83.1								
80	31.12	75.5	33.27	78.1	35.40	80.6	37.52	82.9	39.62	85.2	41.71	87.5						
84	32.62	77.3	34.88	80.0	37.13	82.5	39.36	85.0	41.58	87.3	43.79	89.6	45.99	91.8				
88	34.10	79.1	36.48	81.8	38.84	84.4	41.19	86.9	43.53	89.3	45.86	91.7	48.17	94.0	50.47	96.2		
92	35.57	80.8	38.06	83.5	40.54	86.2	43.01	88.8	45.47	91.3	47.91	93.7	50.34	96.1	52.76	98.4	55.17	100.6
96	37.03	82.4	39.64	85.3	42.23	88.0	44.82	90.7	47.39	93.2	49.95	95.7	52.50	98.1	55.04	100.5	57.56	102.7
100	38.48	84.0	41.20	86.9	43.91	89.7	46.61	92.4	49.30	95.1	51.98	97.6	54.65	100.1	57.30	102.5	59.94	104.8
104	39.92	85.6	42.75	88.5	45.58	91.4	48.40	94.2	51.20	96.9	54.00	99.5	56.78	102.0	59.55	104.5	62.30	106.9
108	41.35	87.1	44.29	90.1	47.24	93.1	50.17	95.9	53.09	98.7	56.00	101.3	58.90	103.9	61.78	106.4	64.66	108.9
112	42.76	88.5	45.83	91.7	48.88	94.7	51.93	97.6	54.96	100.4	57.99	103.1	61.00	105.8	64.01	108.3	67.00	110.8
116	44.17	90.0	47.35	93.2	50.51	96.2	53.68	99.2	56.83	102.1	59.97	104.9	63.10	107.6	66.22	110.2	69.32	112.7
120	45.57	91.4	48.85	94.6	52.14	97.8	55.41	100.8	58.68	103.7	61.94	106.6	65.18	109.3	68.42	112.0	71.64	114.6
124	46.95	92.8	50.35	96.1	53.75	99.3	57.14	102.4	60.52	105.3	63.89	108.2	67.25	111.0	70.60	113.8	73.94	116.4
128	48.33	94.1	51.84	97.5	55.35	100.7	58.85	103.9	62.35	106.9	65.84	109.9	69.31	112.7	72.78	115.5	76.23	118.2
132	49.70	95.5	53.32	98.9	56.94	102.2	60.56	105.4	64.17	108.5	67.77	111.5	71.36	114.4	74.94	117.2	78.51	120.0
136	51.06	96.8	54.80	100.2	58.53	103.6	62.25	106.8	65.98	110.0	69.69	113.0	73.40	116.0	77.09	118.9	80.78	121.7
140	52.41	98.0	56.26	101.6	60.10	105.0	63.94	108.3	67.77	111.5	71.60	114.6	75.42	117.6	79.23	120.5	83.04	123.4
144	53.76	99.3	57.71	102.9	61.66	106.3	65.62	109.7	69.56	112.9	73.50	116.1	77.44	119.2	81.36	122.1	85.28	125.0

*Circular equivalent diameter (d_e). Calculated from $d_e = 1.3 * ((ab)^{.625}/(a+b)^{.25})$

Table 2 – Recommended Maximum Duct Velocities for Low Velocity Systems (fpm)

APPLICATION	CONTROLLING FACTOR NOISE GENERATION Main Ducts	CONTROLLING FACTOR–DUCT FRICTION			
		Main Ducts		Branch Ducts	
		Supply	Return	Supply	Return
Residences	600	1000	800	600	600
Apartments Hotel Bedrooms Hospital Bedrooms	1000	1500	1300	1200	1000
Private Offices Directors Rooms Libraries	1200	2000	1500	1600	1200
Theaters Auditoriums	800	1300	1100	1000	800
General Offices High Class Restaurants High Class Stores Banks	1500	2000	1500	1600	1200
Average Stores Cafeterias	1800	2000	1500	1600	1200
Industrial	2500	3000	1800	2200	1500

Table 3 – Velocity Pressures

VELOCITY PRESSURE (in. wg)	VELOCITY (fpm)	VELOCITY PRESSURE (in. wg)	VELOCITY (fpm)	VELOCITY PRESSURE (in. wg)	VELOCITY (fpm)	VELOCITY PRESSURE (in. wg)	VELOCITY (fpm)
0.01	400	0.29	2150	0.58	3050	1.28	4530
0.02	565	0.30	2190	0.60	3100	1.32	4600
0.03	695	0.31	2230	0.62	3150	1.36	4670
0.04	800	0.32	2260	0.64	3200	1.40	4730
0.05	895	0.33	2300	0.66	3250	1.44	4800
0.06	980	0.34	2330	0.68	3300	1.48	4870
0.07	1060	0.35	2370	0.70	3350	1.52	4930
0.08	1130	0.36	2400	0.72	3390	1.56	5000
0.09	1200	0.37	2400	0.74	3440	1.60	5060
0.10	1270	0.38	2470	0.76	3490	1.64	5120
0.11	1330	0.39	2500	0.78	3530	1.68	5190
0.12	1390	0.40	2530	0.80	3580	1.72	5250
0.13	1400	0.41	2560	0.82	3620	1.76	5310
0.14	1500	0.42	2590	0.84	3670	1.80	5370
0.15	1550	0.43	2620	0.86	3710	1.84	5430
0.16	1600	0.44	2650	0.88	3750	1.88	5490
0.17	1650	0.45	2680	0.90	3790	1.92	5550
0.18	1700	0.46	2710	0.92	3840	1.96	5600
0.19	1740	0.47	2740	0.94	3880	2.00	5660
0.20	1790	0.48	2770	0.96	3920	2.04	5710
0.21	1830	0.49	2800	0.98	3960	2.08	5770
0.22	1880	0.50	2830	1.00	4000	2.12	5830
0.23	1920	0.51	2860	1.04	4080	2.16	5880
0.24	1960	0.52	2880	1.08	4160	2.20	5940
0.25	2000	0.53	2910	1.12	4230	2.24	5990
0.26	2040	0.54	2940	1.16	4310	2.28	6040
0.27	2080	0.55	2970	1.20	4380		
0.28	2120	0.56	2990	1.24	4460		

NOTES: 1. Data for standard air (29.92 in. Hg and 70° F)

2. Data derived from the following equation: $H_V = \left(\frac{V}{4005}\right)^2$ where: V = *velocity in fpm*

H_v = *pressue difference termed "velocity head" (in. wg)*

Table 4 – Duct Material Roughness Multipliers

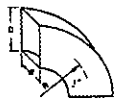
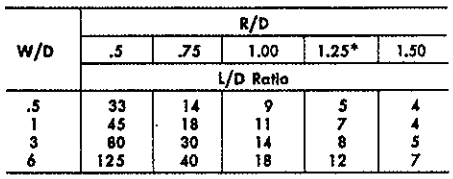
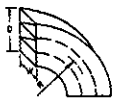
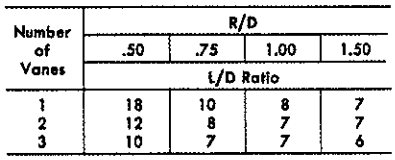
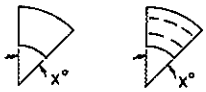
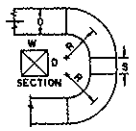
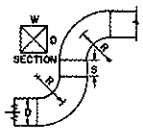
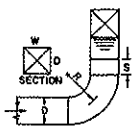
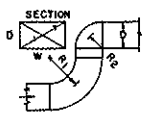
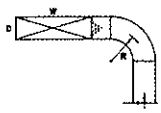
For internal ductwork surfaces other than smooth sheet metal, multiple equivalent lengths by:

Ductwork Description	Multiplier	
	Supply	Return
Rigid Fiberglass – Preformed Round Ducts – Smooth Inside	1.0	1.0
Rigid Fiberglass Duct Board	1.32	1.30
Duct Liner – Airside has Smooth Facing Material	1.32	1.30
*Flexible Metal Duct (Straight Installation)	1.6	1.6
Duct Liner – Airside Spray-Coated	1.9	1.8
*Flexible, Vinyl-Coated Duct with Helical Wire Core (Straight Installation)	3.2	3.4

INSTRUCTIONS: Multiply the measured length of each section by the appropriate multiplier to calculate the total footage equivalent to the same friction loss in smooth sheet metal ductwork.

*Flexible duct multipliers assume that the duct is installed fully extended.

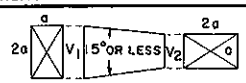
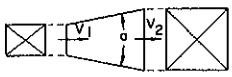
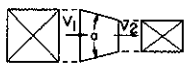
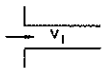
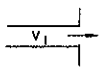
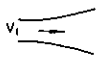
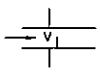
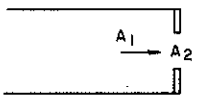
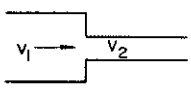
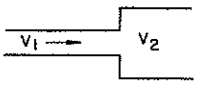
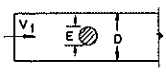
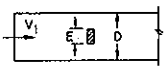
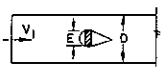
Table 5 – Friction of Rectangular Duct System Elements

ELEMENT	CONDITIONS	L/D RATIO †
Rectangular Radius Elbow 		
Rectangular Vaned Radius Elbow 		
X° Elbow 	Vaned or Unvaned Radius Elbow	X/90 times value for similar 90° elbow
Rectangular Square Elbow 	No Vanes	60
	Single Thickness Turning Vanes	15
	Double Thickness Turning Vanes	10
Double Elbow  **W/D = 1, R/D = 1.25***	S = O	15
	S = D	10
Double Elbow  **W/D = 1, R/D = 1.25***	S = O	20
	S = D	22
Double Elbow  **W/D = 1, R/D = 1.25* For Both**	S = O	15
	S = D	16
Double Elbow  **W/D = 2, R_1/D = 1.25*, R_2/D = .5**	Direction of Arrow	45
	Reverse Direction	40
Double Elbow  **W/D = 4, R/D = 1.25* for both elbows**	Direction of Arrow	17
	Reverse Direction	18

W/D	R/D				
	.5	.75	1.00	1.25*	1.50
	L/D Ratio				
.5	33	14	9	5	4
1	45	18	11	7	4
3	80	30	14	8	5
6	125	40	18	12	7

Number of Vanes	R/D			
	.50	.75	1.00	1.50
	L/D Ratio			
1	18	10	8	7
2	12	8	7	7
3	10	7	7	6

Table 5 – Friction of Rectangular Duct System Elements (Continued)

ELEMENT	CONDITIONS	VALUE OF n‡
Transformer 	$V_2 = V_1$ S.P. Loss = nhv_1	.15
Expansion 	"n" Angle "a" V_2/V_1 | 5° | 10° | 15° | 20° | 30° | 40° .20 | .83 | .74 | .68 | .62 | .52 | .45 .40 | .89 | .83 | .78 | .74 | .68 | .64 .60 | .93 | .87 | .84 | .82 | .79 | .77 S.P. Regain = $n(hv_1 - hv_2)$	
Contraction 	a | 30° | 45° | 60° n | 1.02†† | 1.04 | 1.07 S.P. Loss = $n(hv_2 - hv_1)$ ††Slope 1″ in 4″	
Abrupt Entrance 	S.P. Loss = nhv_1	.35
Bellmouth Entrance 		.03
Abrupt Exit 	S.P. Loss or Regain Considered Zero	
Bellmouth Exit 		
Re-Entrant Entrance 	S.P. Loss = nhv_1	.85
Sharp Edge Round Orifice 	A_2/A_1 | 0 | .25 | .50 | .75 | 1.00 n | 2.5 | 2.3 | 1.9 | 1.1 | 0 S.P. Loss = nhv_2	
Abrupt Contraction 	V_1/V_2 | 0 | .25 | .50 | .75 n | 1.34 | 1.24 | .96 | .52 S.P. Loss = nhv_2	
Abrupt Expansion 	V_2/V_1 | .20 | .40 | .60 | .80 n | .32 | .48 | .48 | .32 S.P. Regain = nhv_1	
Pipe Running Thru Duct 	E/D | .10 | .25 | .50 n | .20 | .55 | 2.00 S.P. Loss = nhv_1	
Bar Running Thru Duct 	E/D | .10 | .25 | .50 n | .7 | 1.4 | 4.00 S.P. Loss = nhv_1	
Easement Over Obstruction 	E/D | .10 | .25 | .50 n | .07 | .23 | .90 S.P. Loss = nhv_1	

* 1.25 is standard for an unvaned full radius elbow.

† *L* and *D* are in feet. *D* is the duct dimension illustrated in the drawing. *L* is the additional equivalent length of duct added to the measured duct. The equivalent length *L* equals *D* in feet times the ratio listed.

‡ The value n is the number of velocity heads or differences in velocity heads lost or gained at a fitting, and may be converted to additional equivalent length of duct by the following equation:

$$L = n * \frac{h_v * 100}{h_f}$$

where: L = additional equivalent length, ft

h_v = velocity pressure for V_1, V_2 or the difference in velocity pressure, in. wg (see SDM-2, Chart 7 or Table 8)

h_f = friction loss/100 ft, duct cross section at h_v in. wg (see SDM-2, Chart 7)

n = value for particular fitting

Table 6 – Friction of Round Elbows

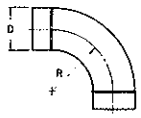
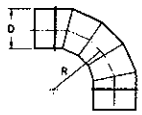
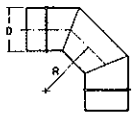
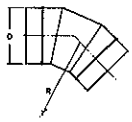
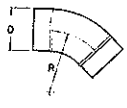
ELBOW DIAMETER (in.)	90° SMOOTH  R/D = 1.5	90° 5-PIECE  R/D = 1.5	90° 3-PIECE  R/D = 1.5	45° 3-PIECE  R/D = 1.5	45° SMOOTH  R/D = 1.5
	ADDITIONAL EQUIVALENT LENGTH OF STRAIGHT DUCT (FT)				
3	2.3	3	6	1.5	1.1
4	3	4	8	2	1.5
5	3.8	5	10	2.5	1.9
6	4.5	6	12	3	2.3
7	5.3	7	14	3.5	2.6
8	6	8	16	4	3
9	—	9	18	4.5	—
10	—	10	20	5	—
11	—	11	22	5.5	—
12	—	12	24	6	—
14	—	14	28	7	—
16	—	16	32	8	—
18	—	18	36	9	—
20	—	20	40	10	—
22	—	22	44	11	—
24	—	24	48	12	—

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Table 7 – Friction of Rectangular Elbows

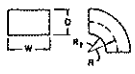
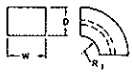
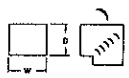
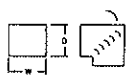
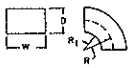
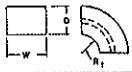
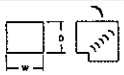
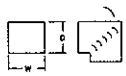
DUCT DIMENSIONS (in.)		RADIUS ELBOW NO VANES	RADIUS ELBOW—WITH VANES‡				SQUARE ELBOWS‡	
								
W	D	Radius Ratio† R/D = 1.25	R_1 = 6" (Recommended)		R_1 = 3" (Acceptable)		Double Thickness Turning Vanes	Single Thickness Turning Vanes
		ADDITIONAL EQUIVALENT LENGTH OF STRAIGHT DUCT (FT)						
				Vanes		Vanes		
96	48	31	45	2	43	3	40	60
	36	25	36	2	31	3	30	45
	30	22	31	2	38	2	25	37
	24	19	33	1	29	2	20	30
	20	16	28	1	25	2	17	25
72	48	28	44	2	41	3	35	60
	36	23	33	2	29	3	29	45
	30	21	28	2	33	2	25	37
	24	17	29	1	25	2	21	30
	20	15	23	1	19	2	18	25
	16	13	18	1	16	2	15	20
	12	12			15	1	11	15
60	48	27	41	2	39	3	33	60
	36	22	31	2	27	3	27	45
	30	19	25	2	31	2	23	37
	24	16	27	1	26	2	20	30
	20	14	22	1	21	2	17	25
	16	12	16	1	15	2	13	20
	12	10			14	1	10	15
48	96*	45	35	3				
	48	26	35	2	34	3	29	60
	36	20	26	2	22	3	23	45
	30	18	23	2	28	2	21	37
	24	15	24	1	21	2	18	30
	20	14	19	1	17	2	15	25
	16	11	15	1	14	2	12	20
	12	9			13	1	10	15
	10	8			11	1	8	12
	8	8			9	1	7	10
42	42	23	28	2	26	3	24	53
	36	20	24	2	21	3	22	45
	30	17	21	2	26	2	20	37
	24	15	21	1	19	2	16	30
	20	13	18	1	16	2	14	25
	16	11	14	1	13	2	12	20
	12	9			13	1	9	15
	10	8			10	1	8	12
	8	7			8	1	6	10
36	72*	34	27	3				
	36	19	22	2	19	3	20	45
	30	16	19	2	22	2	18	37
	24	14	20	1	22	2	15	30
	20	12	17	1	15	2	13	25
	16	10	13	1	12	2	11	20
	12	9			12	1	9	15
	10	8			9	1	8	12
	8	7			8	1	6	10
32	32	17	19	2	16	3	17	40
	30	16	18	2	21	2	17	37
	24	14	19	1	17	2	15	30
	20	12	16	1	14	2	12	25
	16	10	12	1	12	2	11	20
	12	8			12	1	8	15
	10	7			9	1	7	12
	8	6			8	1	6	10

Table 7 – Friction of Rectangular Elbows (Continued)

DUCT DIMENSIONS (in.)		RADIUS ELBOW NO VANES	RADIUS ELBOW—WITH VANES‡				SQUARE ELBOWS‡	
								
W	D	Radius Ratio† R/D = 1.25	R_1 = 6″ (Recommended)		R_1 = 3″ (Acceptable)		Double Thickness Turning Vanes	Single Thickness Turning Vanes
		ADDITIONAL EQUIVALENT LENGTH OF STRAIGHT DUCT (FT)						
				Vanes		Vanes		
28	28	15	14	2	17	2	14	34
	24	13	17	1	15	2	13	30
	20	12	15	1	13	2	12	25
	16	10	11	1	11	2	10	20
	12	8			11	1	8	15
	10	7			9	1	7	12
	8	6			8	1	6	10
24	96*	38	19	3			23	80
	72*	32	17	3			21	72
	48*	22	20	2	20	3	18	62
	24	13	16	1	14	2	12	30
	20	11	13	1	12	2	10	25
	16	10	11	1	10	2	9	20
	12	8			10	1	8	15
	10	7			8	1	7	12
	8	6			7	1	6	10
	6	5					4	8
20	80*	32	16	3			19	66
	60*	26	19	2			17	58
	40*	22	15	2	14	3	14	49
	20	11	12	1	10	2	10	25
	16	9	9	1	9	2	8	20
	12	7			9	1	7	15
	10	6			8	1	6	12
	8	5			7	1	5	10
	6	4					4	8
16	64*	26	9	3			14	48
	48*	21	12	2	12	3	12	43
	32*	15	11	2	9	3	11	38
	16	9	8	1	8	2	7	20
	12	7			8	1	6	15
	10	6			6	1	5	12
	8	5			6	1	5	10
	6	4					4	8
12	48*	19	8	2	8	3	10	33
	36*	16	7	2	7	3	9	30
	24*	11	8	1	8	2	8	26
	12	7			7	1	5	15
	10	6			5	1	5	12
	8	5			5	1	4	10
	6	4					3	8
10	40*	19	6	2	6	3	8	27
	30*	13	6	2	8	2	7	24
	20*	9	7	1	6	2	6	21
	10	5			5	1	4	12
	8	4			5	1	4	10
	6	4					3	8
8	32*	13	5	2	4	3	6	21
	24*	11	6	1	5	2	6	19
	16*	8	4	1	5	2	5	16
	8	4			4	1	3	10
	6	3					3	8
6	24*	10	4	1	4	2	4	15
	18*	8	3	1	4	2	4	13
	12*	6			4	1	3	11
	6	3					3	8

*Denotes Hard Bends as shown

Hard Bend 

Easy Bend 

†For other radius ratios, see *Table 10.*

‡For other sizes, see *Table 10.*

Vanes must be located as illustrated in *Chart 6, page 24,* to have these minimum losses.

References to charts and tables above relate to Carrier SDM-2

Table 8 – Flat Oval Equivalent Duct Sizes

Table 14-4 SPIRAL FLAT-OVAL DUCT (Nominal Sizes—U.S. Units)
(Diameter of the round duct which will have the capacity and friction equivalent to the actual duct size.)

	3	4	5	6	7	8	9	10	11	12	14	16	18	20	22	24	26	28	30	32	34	36	38	40
7		5.7																						
8	5.1		6.6	6.9																				
9	5.6	6.2		7.7																				
10		6.7	7.3		8.7	9.0																		
11	6.0		7.9	8.4		9.8																		
12	6.4	7.2		8.9	9.4		10.8	11.0																
13		7.6	8.4		10.1	10.6		11.9																
14	6.7		8.8	9.6		11.2	11.5																	
15	7.0	8.0		10.1	10.7					13.8														
16			9.3					13.4	13.6															
17	7.3	8.4					12.9																	
18					11.7	12.4				15.3														
19			10.0	11.0					15.0															
20							14.0	14.7			17.5													
21					12.6	13.5				16.7			19.9											
22				11.8					16.3			19.5												
23							15.1	15.7			18.9													
24						14.4							21.6											
25				12.5						18.0		20.9			23.9									
26								16.7						23.6										
27						15.2					20.2		23.1			25.9								
28				13.2						19.1					25.6									
29								17.7				22.3		25.2			27.9							
30						15.9					21.3					28.1								
31				13.8						20.1			24.5		27.2			29.9						
32								18.5				23.5					29.7							
33						16.6					22.4			26.6		29.3			32.0					
34				14.3						20.9			25.7					31.7						
35								19.3				24.7			28.7		31.3			34.0				
36						17.3					23.4			27.9					33.7					
37				14.9						21.9			27.0			30.8		33.4			36.0			
38								20.1				25.7			30.0					35.8				
39						17.9					24.4			29.2			32.8		35.4			38.0		
40										22.7			28.1			32.2					37.8			
41				15.4				20.8				26.8			31.3			34.9		37.4			40.0	
42											25.3			30.3			34.3					39.8		
43						18.6				23.5			29.1			33.5			37.0		39.5			42.0
44				15.9								27.7			32.5			36.4				41.8		
45								21.5			26.1			31.4			35.6			39.0		41.5		
46						19.1							30.2			34.7			38.5					43.8
47										24.3		28.6			33.7			37.8			41.1		43.5	
48								22.1						32.5			36.9			40.5				
49						19.6					26.9		31.1			35.9			39.9			43.1		45.6
50										25.0					34.8			39.1			42.6			
51								22.8				29.4		33.4			38.1			42.0			45.2	
52						20.2					27.7					37.0			41.2			44.7		
53										25.7			32.0		35.8			40.3			44.1			47.2
54								23.3				30.2					39.3			43.3			46.7	
55											28.4			34.4		38.1			42.5			46.2		
56										26.3			32.9					41.5			45.5			48.8
57								23.8				31.0			36.7		40.4			44.6			48.2	
58											29.1			35.3					43.7			47.6		
59										26.9			33.7			39.2		42.6			46.8			50.4
60								24.4			29.6	31.8		36.1	37.8		41.3		44.5	45.9		48.5	49.7	
62										27.5			34.5			40.1		43.5			48.1			51.8
64								25.1			30.5	32.7		37.1	38.9		42.5		46.0			50.2		
66								25.4		28.3		33.3	35.4		39.6			44.8		48.0	49.7		52.4	
68										28.6	31.2			38.0		42.0			47.1			51.9		54.5
70								26.0			31.5	34.1	36.2		40.6		44.4	46.0		49.4	50.9		54.1	
72								26.3		29.2									48.6		51.6	53.1		56.2

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Symbols for HVAC Systems

SYMBOL MEANING	SYMBOL
POINT OF CHANGE IN DUCT CONSTRUCTION (BY STATIC PRESSURE CLASS)	
DUCT (1ST FIGURE, SIDE SHOWN 2ND FIGURE, SIDE NOT SHOWN)	20 x 12
ACOUSTICAL LINING DUCT DIMENSIONS FOR NET FREE AREA	
DIRECTION OF FLOW	
DUCT SECTION (SUPPLY)	S 30 x 12
DUCT SECTION (EXHAUST OR RETURN)	E OR R 20 x 12
INCLINED RISE (R) OR DROP (D) ARROW IN DIRECTION OF AIR FLOW	R
TRANSITIONS: GIVE SIZES. NOTE F.O.T. FLAT ON TOP OR F.O.B. FLAT ON BOTTOM IF APPLICABLE	
STANDARD BRANCH FOR SUPPLY & RETURN (NO SPLITTER)	S R
WYE JUNCTION	
VOLUME DAMPER MANUAL OPERATION	VD
AUTOMATIC DAMPERS MOTOR OPERATED	SEC MOD
ACCESS DOOR (AD) ACCESS PANEL (AP)	OR AD
FIRE DAMPER: SHOW VERTICAL POS. SHOW HORIZ. POS.	FD AD
SMOKE DAMPER (S)	AD S
FIRE & SMOKE DAMPER - (A) SMOKE DAMPER - (S) RADIATION DAMPER - (R)	
TURNING VANES	
FLEXIBLE DUCT FLEXIBLE CONNECTION	
GOOSENECK HOOD (COWL)	
BACK DRAFT DAMPER	BDD

SYMBOL MEANING	SYMBOL
SUPPLY GRILLE (SG)	20 x 12 SG / 700 CFM
RETURN (RG) OR EXHAUST (EG) GRILLE (NOTE AT FLR OR CLG)	20 x 12 RG / 700 CFM
SUPPLY REGISTER (SR) (A GRILLE + INTEGRAL VOL. CONTROL)	20 x 12 SR / 700 CFM
EXHAUST OR RETURN AIR INLET CEILING (INDICATE TYPE)	20 x 20 GR / 700 CFM
SUPPLY OUTLET. CEILING, SQUARE (TYPE AS SPECIFIED) INDICATE FLOW DIRECTION	20 / 700 CFM
SUPPLY OUTLET. CEILING, SQUARE (TYPE AS SPECIFIED) INDICATE FLOW DIRECTION	12 x 12 / 700 CFM
TERMINAL UNIT. (GIVE TYPE AND OR SCHEDULE)	T.U.
COMBINATION DIFFUSER AND LIGHT FIXTURE	
DOOR GRILLE	DG 12 x 6
SOUND TRAP	ST
FAN & MOTOR WITH BELT GUARD & FLEXIBLE CONNECTIONS	
VENTILATING UNIT (TYPE AS SPECIFIED)	
UNIT HEATER (DOWNBLAST)	
UNIT HEATER (HORIZONTAL)	
UNIT HEATER (CENTRIFUGAL FAN) PLAN	
THERMOSTAT	T
POWER OR GRAVITY ROOF VENTILATOR - EXHAUST (ERV)	
POWER OR GRAVITY ROOF VENTILATOR - INTAKE (SRV)	
POWER OR GRAVITY ROOF VENTILATOR - LOUVERED	
LOUVERS & SCREEN	36 H x 24 L

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DUCT SIZING WORKSHEET

PROJECT NAME: _____

DATE: _____

SYSTEM: _____

PAGE: _____ OF _____

Duct Run From -To	Duct Section (element)	Lining (in.)	Insul. (in.)	Other Item	Airflow	Velocity in Round duct (fpm)	Velocity Pressure P_V	Fitting Value n	Length (ft) L	Equiv. Length (ft) EL	Material Correction Factor	Friction Loss f per 100' duct	Friction Loss (in. wg)	Known Loss (in. wg)	Round Duct Size (in.)	Equivalent Rectangular Size (W x H)	Total Item Loss (in. wg)	Cumulative Loss (in. wg)
																x		
																x		
																x		
																x		
																x		
																x		
																x		
																x		
																x		
																x		
																x		
																x		
																x		
																x		
																x		
																x		
																x		
																x		
																x		

NOTES: All duct sizes indicated are inside clear dimensions.

LARGEST STATIC PRESSURE LOSS (in. wg) [ ]

FOR RUN = _____

Glossary

access floor	a system of floor panels and supports that create a service utility plenum above the structural slab; allows using underfloor air distribution (UFAD)
air distribution device	opening through which air discharges from (outlet) or enters into (inlet) a duct; may be a grille, register, diffuser, or geometric shape, like a slot.
air handling unit (AHU) / air handler	equipment, usually connected to ductwork, to move and condition air; contains components such as fans, cooling coils, heating coils, filters and dampers to control return, outdoor and exhaust air
branch duct	a duct section of the same size or smaller, connected to a trunk duct; serves one or more zones.
breakout noise	transmission or radiation of noise from some part of a duct system to an occupied space in the building.
chase	vertical passage within a building for enclosing ducts, pipes or wires.
condensation	the process by which a vapor is changed into a liquid of the same temperature by removing heat from the vapor.
damper	a device used to regulate the flow of air.
duct	a conduit section used to move air, i.e. supply to the conditioned space, or return or exhaust from the space
duct calculator	slide rule or wheel used to manually size ducts and fittings based on parameters such as friction loss, velocity, static regain, and fitting coefficients.
duct circuit	path from a fan to an outlet or inlet; duct systems normally have many circuits which need to be balanced to a similar static pressure.
duct fitting	connector creating a change of direction and/or cross-sectional area within the ductwork.
duct sizing	calculation of dimensions of ducts and fittings for a given duct system.
duct system	interconnected sections of ductwork used to convey air or other gases from one location to another; may include fans, coils, terminals, and air distribution devices.
ductwork	series of interconnected ducts and fittings
equal friction duct sizing	method for calculating the size of ducts so that the frictional resistance per unit length is constant.
external static pressure	total static pressure difference of the duct system as measured between the equipment air outlet and inlet connections; see static pressure.
fan	a component that provides the energy to move air, typically through a duct system
fan static pressure	difference between fan total pressure and fan discharge velocity pressure.

friction loss	pressure loss created as a result of friction between a flowing fluid and its contact surface.
header duct	an extended plenum off a terminal unit discharge that distributes the zone air to the air distribution devices.
main duct / trunk duct	the supply or return duct connected to the air source; distributes or collects air from building spaces.
modified equal friction duct sizing	method in which ducts sized by the equal friction method are resized to increase or decrease the static pressure loss of a duct circuit to improve self-balancing characteristic of the duct system.
noise	a sound, especially one that is loud, harsh or confused or undesired.
plenum	enclosed space within a building where air flows, similar to duct-work. Also used to enclose mechanical and electrical services.
runout duct	a connector duct between a terminal unit and a branch or trunk duct.
static pressure	the potential energy component of air flowing in ductwork, represented by the Ideal Gas Law equation, PV=nRT, where the fan filling (static) pressure at the supply duct inlet is above atmospheric pressure due to the ductwork (V term in equation) outlet and friction losses.
static regain duct sizing	method for calculating the size of ducts so that the regain in static pressure due to decreased velocity between two points totally or partially compensates for the frictional resistance between the points.
system effects	inlet and outlet duct conditions at the fan that affect its performance and related testing, adjusting, and balancing work.
takeoff	as a duct, connects the header duct to the air distribution devices; as a fitting, connects a runout duct to a branch or trunk duct.
terminal unit	component located in or near the zone that controls the supply airflow and/or temperature to the zone spaces; may have a fan, heating coil, and/or cooling coil.
thermal insulation	material or assembly of materials used to provide resistance to heat flow.
total pressure	the sum of the static pressure and velocity pressure.
total supply air quantity	the sum of the zone airflow quantities; determined from the maximum block load.
trunk duct / main duct	the supply or return duct connected to the air source; distributes or collects air from building spaces.
turbulence (eddy flow)	fluid flow in which the velocity varies in magnitude and direction in an irregular manner throughout the mass.
variable air volume (VAV)	systems where the supply airflow to the space is varied to maintain the temperature set point; the supply air temperature is usually constant. This is contrasted with a constant volume (CV) system.
velocity pressure	the kinetic energy component of air flowing in ductwork; represented by the equation $P_v = (v/4005)^2$.
ventilation air	planned outdoor airflow that is provided to a space to replace oxygen and dilute odors and contaminants.

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Flexible Duct

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NFPA Standard 90A–Installation of Air Conditioning and Ventilating Systems.

Duct Liner

NFPA Standard 90A–Installation of Air Conditioning and Ventilating Systems.

Notes:

Work Session 1 – Answers

1. Total Pressure: The sum of velocity pressure and static pressure.
   Velocity Pressure: The force exerted in the duct system due to the relative speed of the air.
   Static Pressure: The force exerted in all directions when air pressure is either elevated above or lowered below the atmospheric pressure.
2. a.), b.), d.), e.), f.)
3. False – The fan converts velocity pressure into static pressure in the first few feet of duct.
4. Potential energy
   Kinetic energy
5. Whenever there is a change in velocity there is a corresponding and inverse change in static pressure.
6. From Equation 2: V = Q / A; V = 2,250 / 1.5 sq ft = 1,500 fpm [same for both round and square].
7. Square – Square duct has a higher perimeter to area ratio.
8. Solve Equation 2 for flow rate Q; Q = V * A; Q = 1,000 fpm * 1.5 sq ft = 1,500 cfm.
9. a.), b.), c.), e.), f.)
10. From Table 4 in the Appendix: Read multiplier of 3.2; f = 0.08 * 3.2 = 0.26 in. wg
11. c.), e.)

See Page 36 for Work Session 1 Equal Friction Duct Sizing Worksheet example.

Work Session 2 – Answers

1. total supply air quantity

2. The air source cfm that meets the maximum simultaneous cooling load for the zones. This is often less than the sum of the largest zone cfms because they usually do not occur at the same time or day of the year.

3. False – Flex duct has over three times the friction loss rate of smooth sheet metal and should be used in 8 ft or less lengths to minimize total friction loss.

4. **Advantages:** quieter (attenuates fan noise), eliminates need for external duct insulation;

   Disadvantages: increases duct friction loss, difficult to clean, may harbor microorganisms if allowed to get dirty and damp.

5. The air is very turbulent right off the fan and needs time to establish a uniform velocity profile. When fittings, transitions, etc. are located too close to the fan outlet it creates an additional friction loss value known as system effect.

6. False – Transition fittings cost more to fabricate and install than straight duct sections. You should attempt, whenever possible, to maintain at least one of the duct dimensions when transitioning from one duct size to another and should only downsize when the velocity drops below the recommended value.

7. The first duct section is sized based on an assumed velocity. Table 2 in the Appendix lists recommended design velocity values.

8. False – Return air plenums should not be used on top floors adjacent to roofs due to the heat gain from the roof being entrained in the return air. Also, under certain conditions the temperature of the structural beams or other materials located in the plenum may be colder than the dew point of the return air. Under these conditions condensation will form on these surfaces causing stained ceiling tiles and possibly wet surfaces that can harbor growth of microorganisms.

9. c.)

10. e.)

11. e.)

12. False – It means that all duct sections are sized using an equal friction rate. Variations in section friction losses are balanced by downsizing the shorter runs or by installing balancing dampers.

13. False – The equal friction rate is generally the preferred method for sizing return duct systems.

14. c.), d.), f.)

15. By downsizing the shorter duct runs with lower friction loss there is less of a requirement to "choke" the airflow by the use of balancing dampers; results in smaller ducts, which saves money.

16. Advantages: lower system pressure drop, creates a self-balancing duct system, lower fan operating cost. Disadvantages: larger duct sizes, more expensive to install, more difficult to design.

17. The duct velocities are systematically reduced as you proceed down the duct system from the fan, resulting in a conversion of the velocity pressure into static pressure. This increased static pressure overcomes a portion of the air friction loss in the next downstream duct section.

Prerequisites:

An understanding of air conditioning equipment and systems along with the airflows required to satisfy the heating and/or cooling loads. This knowledge can be gained from: TDP-103, ABCs of Comfort; TDP-301, Load Estimating, Level 1: Fundamentals; and TDP-302, Load Estimating, Level 2: Block and Zone Loads.

Learning Objectives:

After completing this module, participants will be able to:

- Describe fundamental theory and principles of airflow through duct systems.
- Apply a step-by-step process and evaluation criteria to duct design.
- Calculate air velocity and pressure loss in round and rectangular ducts and fittings.
- Diagram the ductwork layout for the actual building design and notate it with airflows, dimensions, and devices affecting sizing selections.
- Size supply and return air duct systems manually, with either a friction chart or duct calculator, using the equal friction method.
- Estimate the approximate total duct system static pressure losses in order to size the system fan.
- Integrate code and standard requirements for energy conservation into the design process.

Supplemental Material:

Form No.	Cat. No.	Title
T200-11	794-036	Duct Calculator

Instructor Information:

Each TDP topic is supported with a number of different items to meet the specific needs of the user. Instructor materials consist of a CD-ROM disk that includes a PowerPoint™ presentation with convenient links to all required support materials required for the topic. This always includes: slides, presenter notes, text file including work sessions and work session solutions, quiz and quiz answers. Depending upon the topic, the instructor CD may also include sound, video, spreadsheets, forms, or other material required to present a complete class. Self-study or student material consists of a text including work sessions and work session answers, and may also include forms, worksheets, calculators, etc.



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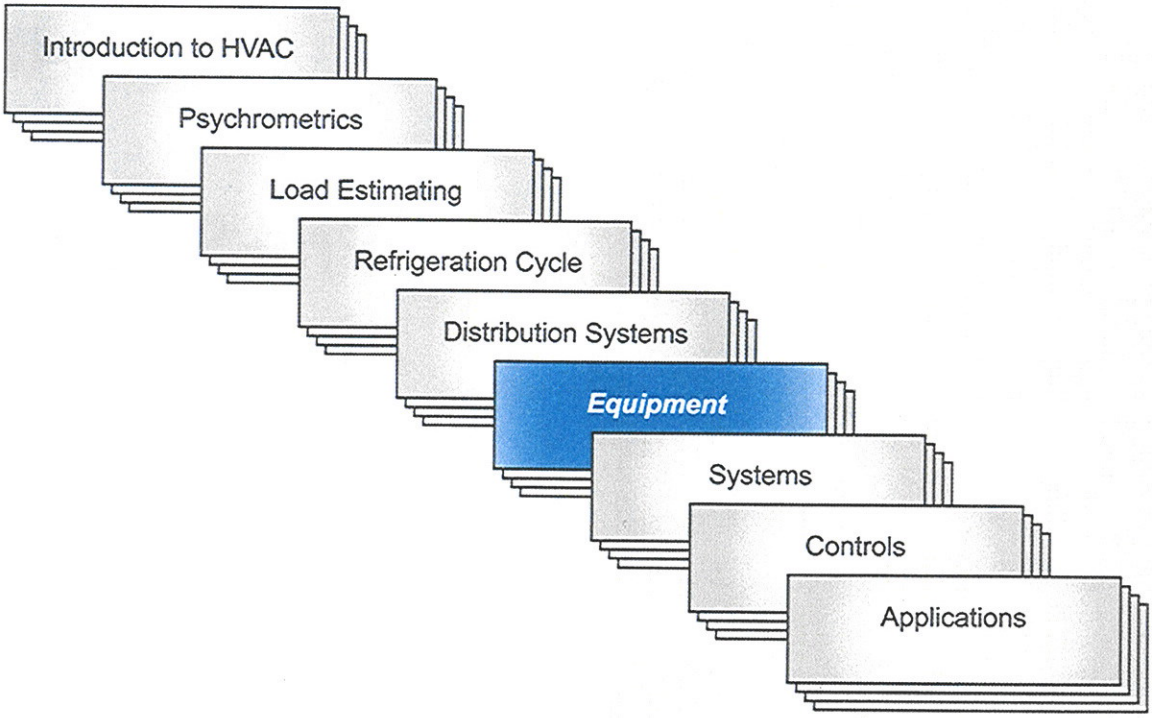
COMMERCIAL HVAC AIR-HANDLING EQUIPMENT

Central Station Air Handlers

Technical Development Program

Technical Development Programs (TDP) are modules of technical training on HVAC theory, system design, equipment selection and application topics. They are targeted at engineers and designers who wish to develop their knowledge in this field to effectively design, specify, sell or apply HVAC equipment in commercial applications.

Although TDP topics have been developed as stand-alone modules, there are logical groupings of topics. The modules within each group begin at an introductory level and progress to advanced levels. The breadth of this offering allows for customization into a complete HVAC curriculum – from a complete HVAC design course at an introductory-level or to an advanced-level design course. Advanced-level modules assume prerequisite knowledge and do not review basic concepts.



Air handlers do not just handle air. They also cool, heat, filter, and humidify. Central station air handlers are typically "built to order" with a wide variety of available options and accessories to choose from. Central station air handlers are available factory-designed for indoor use or for rooftop mounting. This TDP module will explain the types of equipment and the sectional components that comprise an air handler, both indoor and outdoor types, discuss modern construction methods for central station air-handling units, as well as the software programs used for selection.

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Introduction

Factory-assembled central station air-handling units are generally one of the first items of air-conditioning equipment selected after the cooling load estimate is completed. In the system design process, a chilled water or refrigerant temperature level is established under which the chiller or condensing units will operate. In turn, this temperature is used to determine the design requirements for the air-handling equipment, including coils and fans. Because of its effects on other system components, it is imperative for the designer to have a thorough understanding of central station air-handling equipment and how it should be selected and applied.

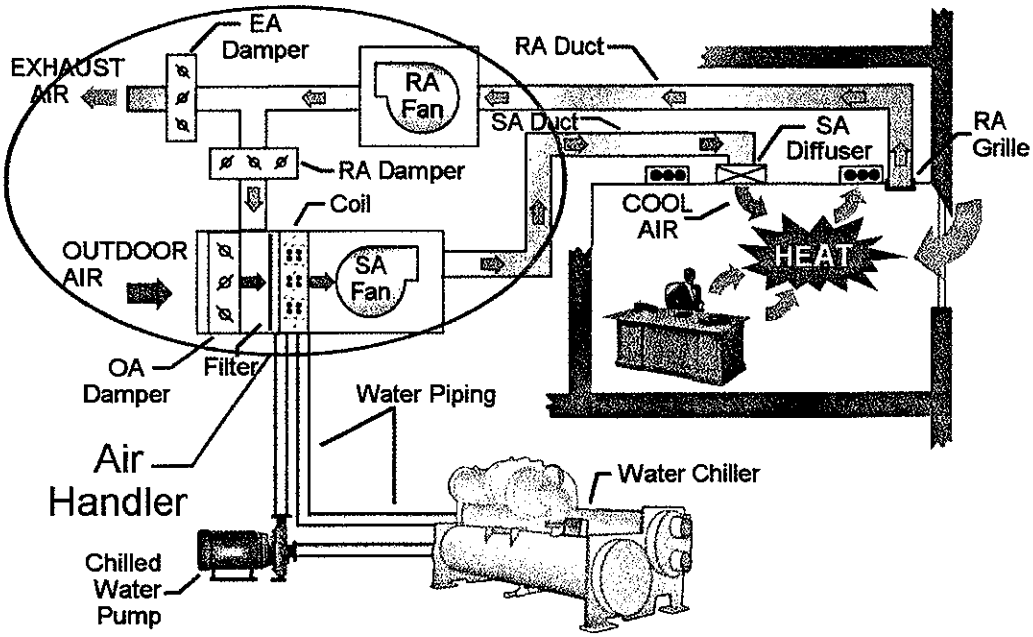


Figure 1

Where does the air handler fit in?

This TDP will outline the basic construction methods used in current central station air-handling units, the types of fans and their characteristics, common methods to modulate fans when used in VAV systems, and indoor air quality (IAQ) air-handling unit components like energy recovery and filtration sections.

Increasing concern about building health conditions and ventilation requirements has made IAQ a top consideration in today's HVAC equipment purchases. For air handlers, in addition to delivering conditioned air in the proper quantities and temperatures, effective filtration, minimal air leakage, energy efficiency and improved serviceability are also critical. As a result, the designer can no longer focus attention exclusively on the coil and fan selections. Casing design and performance should also be considered from the standpoints of thermal performance, air leakage, and serviceability. Therefore, this TDP module will explain both the components that make up an air handler, and how they are constructed.

Manufacturers are developing new products and integrating new materials and technologies to address these needs. More and more, features that were once available on "custom" units are being incorporated into "standard" air handlers.

Packaged, Central, and Custom Air Handlers

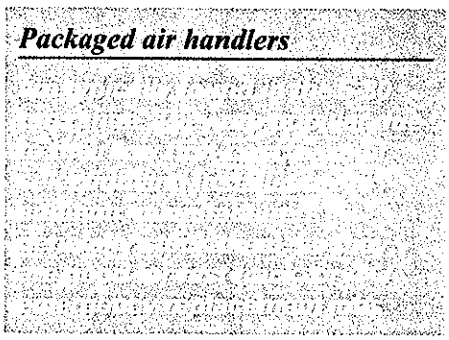
ARI Standard 430 defines central station air-handling units as "... a factory-made encased assembly consisting of a fan or fans and other necessary equipment to perform one or more of the functions of circulating, cleaning, heating, cooling, humidifying, dehumidifying and mixing of air; and shall not contain a source of cooling or heating other than gas or electric heat. This device is capable of use with ductwork having a total static resistance of at least 0.5 in. wg."

Although the term "fan coil" is frequently used interchangeably with air handler, ARI defines fan coils as being "non-ducted," or applied to systems operating with less than 0.25 inch static resistance.

In the commercial HVAC market, air handlers range from simple "packaged" air handlers up to approximately 15,000 cubic feet per minute (cfm) range, to large central station air handlers capable of delivering over 100,000 cfm. Let's take a moment to look at the differences between them.

Packaged Air Handlers

Packaged air handlers have a fixed fan and coil configuration. They are typically used for low pressure comfort cooling and heating applications that require less than two inches external static pressure. Packaged air handlers have a more limited set of options than central station air handlers, generally composed of heating coils, mixing boxes, and discharge plenums.



Characteristics

- **Fixed fan and coil configuration**
- **Limited to <15,000 cfm (30 tons)**
- **Low-static applications only**
- **Limited options available**
- **Indoor applications**
- **Single-wall casing**
- **Manufacturers stock them**

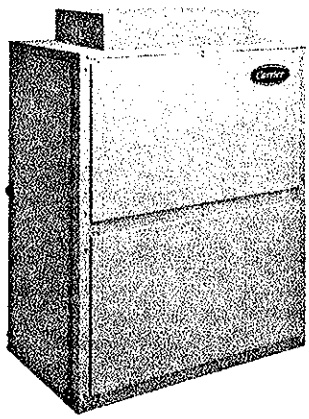


Figure 2

Packaged Air Handler

Most often, they are matched to split system condensing units and heat pumps, although optional chilled water coils may be available. Several manufacturers provide matched performance ratings with their condensing units and certify these ratings under ARI Standard 340/360 (Unitary Large Equipment). As a result, packaged air handlers are usually classified by nominal cooling tonnage, rather than airflow.

Central Station Air Handlers

In contrast, central station air handlers, sometimes referred to as "applied" or "built-up" air handlers, offer a wide range of component options to cover an almost limitless set of application needs, both commercial and industrial. Central station air handlers range in size from 1,500 cfm for small single-zone applications, to large, fully custom air handlers capable of delivering over 100,000 cfm to constant volume or variable air volume systems. Although the size range is large, the majority of comfort air-conditioning applications fall into the 1,500 to 50,000 cfm size range.

Characteristics

- Almost infinite component combinations
- Standard size range from 1,500 to 50,000 cfm
- Custom models available over 100,000 cfm
- Multiple fan and coil types and arrangements
- Draw-thru and blow-thru
- Wide range of accessory sections
- Indoor or outdoor models
- Built to order

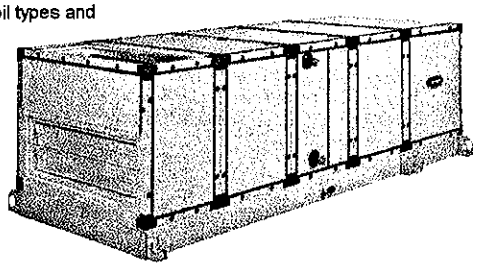


Figure 3

Central Station Air Handler

To simplify the design process and keep the time required to design, build, and deliver the air handler to a minimum, most manufacturers offer a "standard" catalog offering of pre-engineered sizes and components. These components are assembled by the designer in building-block fashion to suit the job requirements. In addition to traditional catalogs, some manufacturers offer computer selection software that facilitates the configuration and selection of a unit, along with detailed engineering drawings and data for submittal and ordering purposes.

Central station air handlers

are generally built to order due to their variety and complexity.

Custom Air Handlers

Custom air handlers are used where standard unit designs cannot easily be applied. Typical volumetric flow rates for custom air-handling equipment range from 50,000 cfm to 300,000 cfm or greater. However, custom equipment can also be found on a much smaller scale where space restrictions, strict sound requirements, low leakage rates or special materials and construction methods are required. These units are designed to meet the exact requirements of a given project and can include a staggering array of options and accessories.

Key Features:

- **Job-specific design**
  - Dimensions
  - Components
  - Configurations



Figure 4

Custom Air Handler

Custom air handlers

Custom air-handling equipment design is driven by the requirements of the application. When size restrictions are a major concern, the most efficient component selections are made to fit in the minimum available space. If sound is a primary concern, appropriate materials and components are selected and arranged within the cabinet to minimize the sound output power levels radiated through the cabinet, at the inlet and at the outlet of the unit.

Selection Basis for Central Station and Custom Air Handlers

The cataloged size of an air-handling unit often refers to the face area in square feet of the largest cooling coil that can be installed in that particular casing. Once the required airflow rate is established from the building load estimate and design requirements, the engineer will define a maximum permissible cooling coil face velocity. The maximum design air velocity is limited to prevent condensate on the coil surface from being blown off the coil surface and into downstream components. Coil velocity limits vary with coil fin material and coil design. Typical design air velocities for aluminum-finned coils are generally in the 500-550 feet per minute (fpm) range. For example, an application requiring 25,000 cfm operating at 500 fpm would require an air handler with a 50-square foot coil. This would then equate to a nominal size 50 air handler. Note that for a particular nominal size, actual coil face area can vary slightly between manufacturers, as they are generally rounded to a nominal unit size. Heating coils do not condense moisture, therefore do not exhibit moisture blowoff. Face velocities can be higher (900 fpm) and are limited by airside pressure drop.

Maximum cooling coil face velocity in feet per minute (fpm) is restricted by moisture blow-off.

Coil Moisture Blow-off Limits (fpm)			
Fins per inch	Aluminum	Copper	E-Coat
8	550	500	475
11	550	425	400
14	550	375	350

1. Limits apply to clean and properly maintained coils.
2. The limits shown are for Carrier 39 Series coils. Consult manufacturer for specific data.

Figure 5

Maximum Coil Velocity

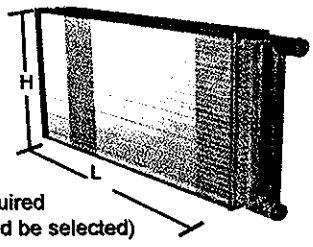
Without employing any energy recovery or gas-fired heating options, a custom air-handling unit is similar to a standard air handler in that it is typically designed around the total face area of the cooling coil required to meet minimum performance requirements. That is, the cooling coil dimensions are based on the maximum allowable face velocity, which varies with respect to coil duty and the tube and fin material options prescribed by the application.

The relationship between airflow volume (cfm), velocity (V) and area (A) is:

$$cfm = VA \quad \text{or} \quad A = cfm/V$$

Where:

$$A = H * L$$



Solution: A = cfm / V
A = 25,000 cfm / 500 fpm
A = 50 ft² cooling coil required
(nominal size 50 unit would be selected)

Figure 6

Air Handler Size Selection

An exception

However, regardless of the component, once the limiting factor is determined, its design and selection is optimized and the rest of the layout naturally revolves around this feature.

Basic Air Handler Unit Construction

Most central station air handler casings are designed using either "post and panel" or "structural panel" construction techniques. In addition, they may be designed for indoor or outdoor installation with single or double-wall panel construction. Let's take a closer look at these options.

Post and Panel Design

Post and panel designs employ a frame assembly using structural rails or posts that form a framework or "skeleton" to support the pre-formed casing panels and internal components. Once the frame is assembled, individual panels are attached to the rails to form the air handler enclosure. The panels are attached either with screws or quarter-turn quick release fasteners for faster installation and removal. The panels are generally gasketed or sealed to minimize air leakage. A benefit of post and panel construction is serviceability, as all panels can be removed individually for cleaning or direct access to internal components. In many cases, especially when quarter-turn fasteners are used, wall panels are mounted on hinges and function as access doors. This design technique eliminates the need for separate inset access doors, minimizing the potential sources for air leaks. Air leaks contribute to system inefficiency.

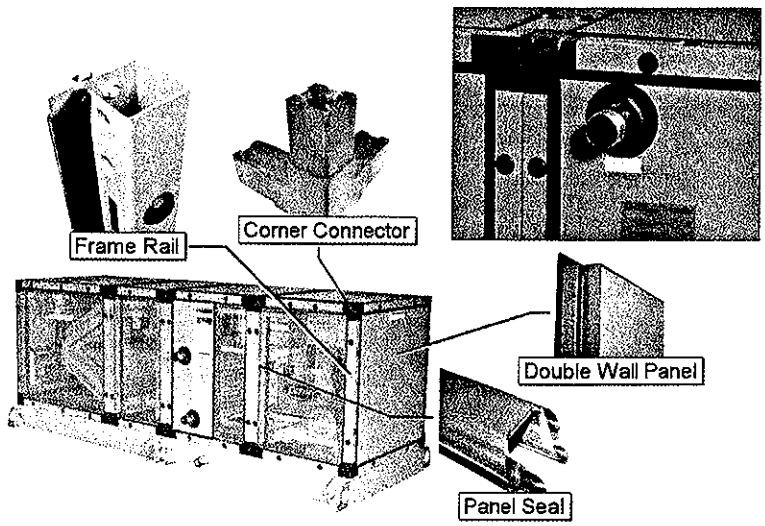


Figure 7

Post and Panel Construction

Post and panel construction lends itself to modular designed air handlers. One approach uses timesaving quick-connect latches that permit easy assembly and disassembly, eliminating the need for conventional bolt and screw assembly.

Structural Panel Design

In structural panel casing designs, the side panels are designed to carry the load of the top panels and internal components without the need for a separate frame structure. Load-bearing wall panels are fastened to the floor panels and the top or roof panels are supported by the walls. For service access, individual panels may be removed one or two at a time, but not all at once. As a result, structural panel designs rely more heavily on inset doors for service access.

One benefit of the structural panel design is that the manufacturer has greater flexibility to increase wall thickness for strength to handle extremely high static pressure applications, higher insulation requirements, or improved acoustic performance.

Structural panels

- Unit structure is achieved by load-bearing side wall panels
- Side panels support top panels and internal components
- Individual panels may generally be removed, but not all at once
- Inset doors are primary means of service access
- Design allows manufacturers to vary wall thickness more easily to accommodate high-pressure or high thermal resistance applications
- Panels and sections typically sealed with foam gaskets and caulking

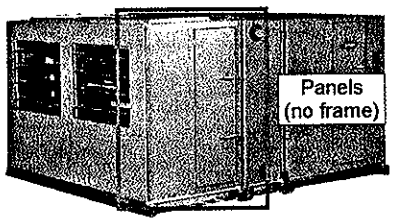


Figure 8

Structural Panel Casings

Casing Design and Materials

As a general industry standard, manufacturers use galvanized steel to build air-handling units. Galvanizing is a zinc coating process. Galvanizing is highly effective at preventing corrosion of the steel substrate with the following limitations:

- Bare steel can become exposed at the sheet metal edge where it is cut or where holes are punched. Rust may form at this edge, undermining the surface galvanizing.
- Scratches can expose bare steel allowing rust to form. Scratches may occur during the manufacturing process, installation, or maintenance.

Galvanizing also has the characteristic that the zinc coating may itself develop surface corrosion in the form of zinc oxide that is sometimes referred to as white rust. For these reasons, most manufacturers offer an option for painted casings.

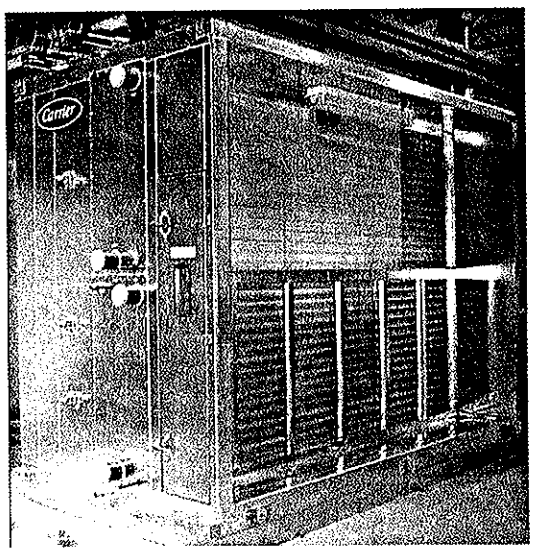


Figure 9

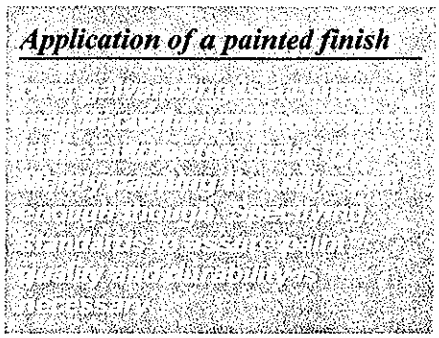
Galvanized Casing

Paint Quality and Durability Standards

American Society for Testing and Materials (ASTM) B 117 Standard Practice for Operating Salt Spray (Fog) Apparatus - describes the apparatus, procedure, and conditions required to create and maintain a salt spray (fog) test environment. Units are typically specified to withstand 500 hours salt spray.

ASTM G 85 Standard Practice for Modified Salt Spray (Fog) Testing - describes apparatus and procedures for cyclic testing of painted specimens, which is a more aggressive test of durability. A test duration of 125 hours is often specified.

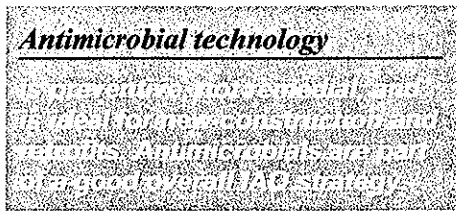
Application of a painted finish



With all the attention given to indoor air quality, alternate casing and lining materials such as paint, aluminum, and stainless steel, are becoming more widely used on comfort cooling systems.

Antimicrobial Coatings

Antimicrobial technology



Antimicrobial coatings are also becoming more commonplace to deter the growth of bacteria, fungus, and mold. They are applied to the inner liner of double-wall casings, fiberglass insulation, and condensate drain pans.

Antimicrobials are materials that prevent, inhibit, or suppress the growth of microbes, such as bacteria, mold, fungus, or yeast. Antimicrobials fall into two categories: organic (hydrocarbon based) and inorganic (metallic-ion based). They can be solids, liquids, or gases.

A multitude of materials exist; the key is to select the right technology for the application.

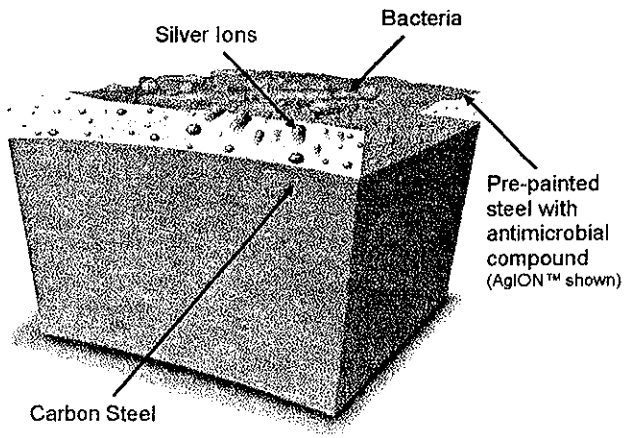


Figure 10

Cross Section of Coated Air Handler Panel

The following are important considerations when choosing an antimicrobial:

- Efficacy
  - Ensure that the chosen technology is right for the application
- Longevity
  - Understand the life expectancy / maintenance requirements
- Safety
  - Should pass long-term toxicity and biocompatibility tests
  - Meet UL Standard 723 (ASTM E-84) for smoke and flame spread

- Regulatory Status
  - The Environmental Protection Agency (EPA) and the Food and Drug Administration (FDA) regulate antimicrobials
  - Should be EPA registered for use in HVAC components
  - Should be FDA listed for some applications related to food service
- Cost
  - Affects feasibility of applications

An example of an inorganic-based antimicrobial product that has been EPA approved for HVAC applications, and FDA approved for food service is Carrier's AgION™. Let's take a look at how this works. The AgION™ antimicrobial compound is a fine powder that withstands over 1400° F.

The silver ions reside in an open zeolite matrix structure. The zeolite acts as an ion pump causing a controlled release of silver ions over the life of the product. Silver ions – silver atoms with an electrical charge – are attracted to oppositely charged hydrogen ions – commonly found in most bacteria and microbes – like a magnet. Once the two ions connect, the hydrogen ions are no longer available for other chemical bonds, abruptly halting the bacteria and microbes' respiration and growth.

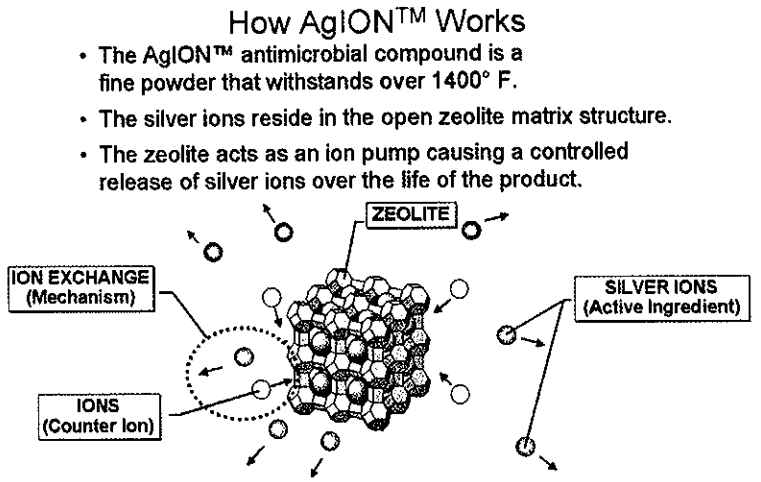
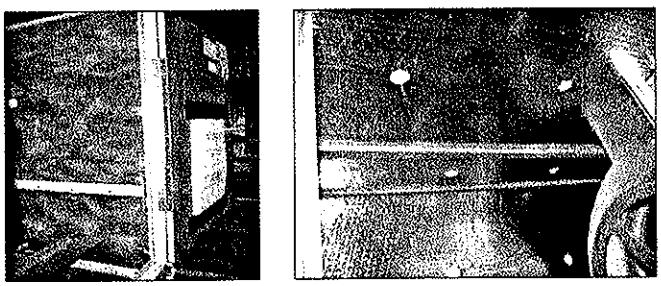


Figure 11

Silver Ion Antimicrobial

Insulation Types

Thermal performance of the air handler casing is important in today's energy-conscious market. Fiberglass is still the most widely used insulating material in air handlers. It provides good thermal performance at an economical cost and is easy to install. While it is available in many variations of thickness, density, rigidity and coatings, air handlers typically use two types of insulation; unfaced for double wall, and matte-faced for single wall units.



- 1-inch matte-faced fiberglass 1½ and 3 lb density available
- 1-inch foil-faced insulation reduces the potential for mold or bacteria and is easier to clean

Figure 12

Insulation – Single Wall Units

The R-value is the unit of measurement used to express the insulation's thermal resistance to heat transfer. The difference in thermal performance between density options is small. Only when you increase the thickness of fiberglass insulation do you see a difference in thermal performance. A 1-in. layer of insulation with a density of 1½ or 3 pounds yields the same approximate R-value. Increasing the thickness, however, would increase the R-value. The impact of insulation density is in sound transmission, where higher density provides greater attenuation.

While polyurethane foam insulations have been used extensively in appliances for years, they are just starting to be used in air handlers. With an R-value equal to about R-6 per inch, they can reduce the heat transfer through the casing by nearly 50 percent over fiberglass. Foams also offer several additional benefits over fiberglass:

- They do not absorb moisture. This is a very important point, because, wet fiberglass provides little insulating quality and can lead to the growth of mold and mildew and severe IAQ problems.
- They contain no fibers to get into airstream.
- They have a rigid board structure, so they eliminate any possibility of air leakage through the insulation.

Single-Wall vs. Double-Wall

The central station air handler market has begun to move away from single-wall construction to double-wall designs. The primary reason is the concern over the release of glass fibers from the insulation into the airstream and into the occupied building. The insulation used in single-wall air handlers is typically fiberglass with a rubberized or acrylic matte facing. The matte facing is designed to prevent erosion of the insulation that is exposed to air moving over it in the normal velocity ranges encountered in air-handling equipment. However, it is inevitable that some fiberglass fibers will be released into the airstream. Matte-faced insulation is also difficult to clean without damage to the facing. Facing damage will lead to erosion and increased fiber release. In recent years, a reinforced foil-faced insulation has been introduced to provide a higher level of cleanability in single-wall units.

Most codes also require that the air handlers, regardless of insulation type, meet the requirements of the National Fire Protection Agency (NFPA) Standard 90A for maximum limits on smoke generation and flame spread. This means that for single-wall units, the insulation itself must meet the NFPA standards, while on double-wall units the panel assembly must pass.

Double wall construction sandwiches the insulation, either fiberglass or foam between the outer casing and an interior metal liner. This prevents exposure of the insulation to the moving airstream, thereby minimizing the possibility that fiberglass particles can be carried into the space. In addition, the metal liner is easier to clean and may be painted or constructed of a variety of materials to suit the applications.

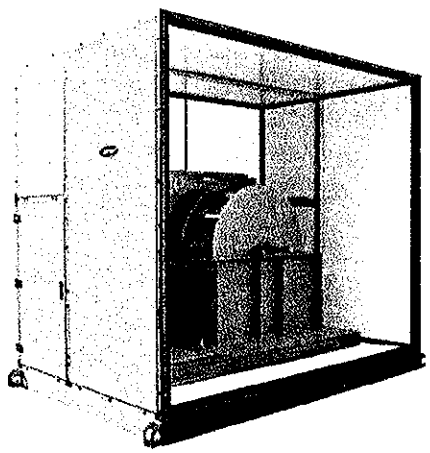


Figure 13

Double-Wall Construction

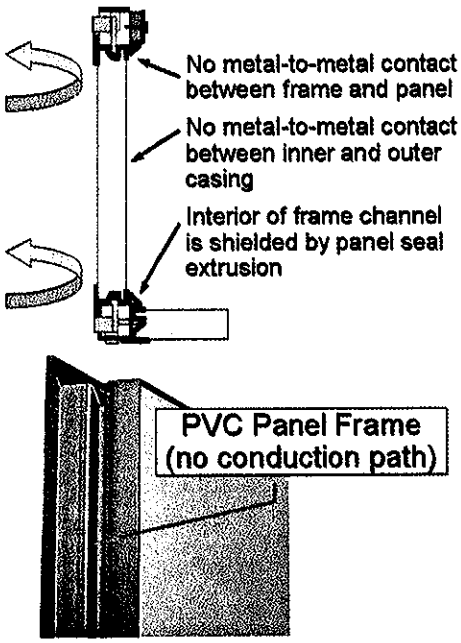


Figure 14

Thermal Break and Double-Wall Panel

Traditionally, engineers have specified panel metal gage for air handler casings. However, with recent advances in materials and technology, engineers recognize new laminate panel designs have allowed stronger, lighter, more thermally efficient casings to be developed. They also understand that strength is a function of entire structural design, not just surface skins and recognize that panel deflection is a more accurate measure of strength. As a result, they are replacing traditional sheet metal gage specifications with a maximum deflection specification for the panel assembly as shown.

Panels constructed with polyurethane foam form an extremely rigid, lightweight casing panel. In the panel shown here, the inner and outer skins have a flange formed along the edges to lock into an extruded PVC frame. When the panel is assembled, the cavity is filled with insulation that bonds to the inner surfaces of the steel pans and PVC frame to create a rigid, one-piece panel. The rigidity helps prevent the panel from deflecting under load, to minimize the chance for air leakage at the seal between the panel and the frame.

The PVC frame also provides a continuous thermal break to minimize heat transfer between the inner or outer surfaces of the panel, minimizing the possibility of the panels sweating.

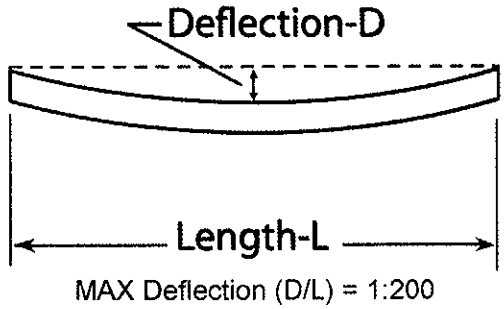


Figure 15

Panel Maximum Deflection

Seals

Seals are an integral part of a low-leak air handler. Advances in material and manufacturing technology have led to new products, which are being integrated into air handler product designs. Closed cell foam gaskets lose their resiliency with age, but are effective in joints that are not repeatedly opened, such as between sections. In a joint that may be opened from time to time, such as access or panel joint seals, foam gaskets may not reseal as well as when they were new. Joints that are frequently opened should have a resilient seal that is designed to maintain its shape and sealing ability after repeated compression cycles. As a result, automotive-type bulb seals are becoming more prevalent in standard air handlers today.

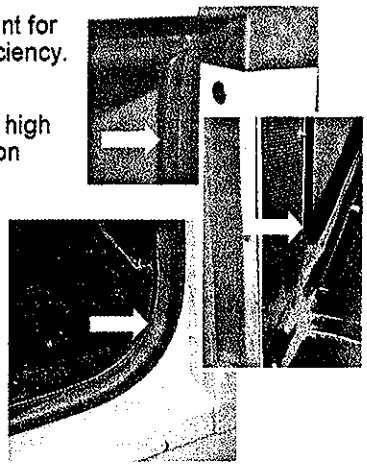


Figure 16

Casing Seals

Air Handler Types

Indoor Units

Indoor air handlers may be built in horizontal, vertical, or stacked configurations, and allow air inlet and discharge connections on top, bottom, and sides. This allows the designer greater flexibility to accommodate obstructions and other obstacles when making duct connections. To facilitate rigging and installation into existing equipment rooms or tight spaces, indoor units are frequently shipped to the jobsite in separate sections. Once in place, the sections would be connected together and installation completed. Care should be taken to allow adequate service clearances in accordance with manufacturers' recommendations.

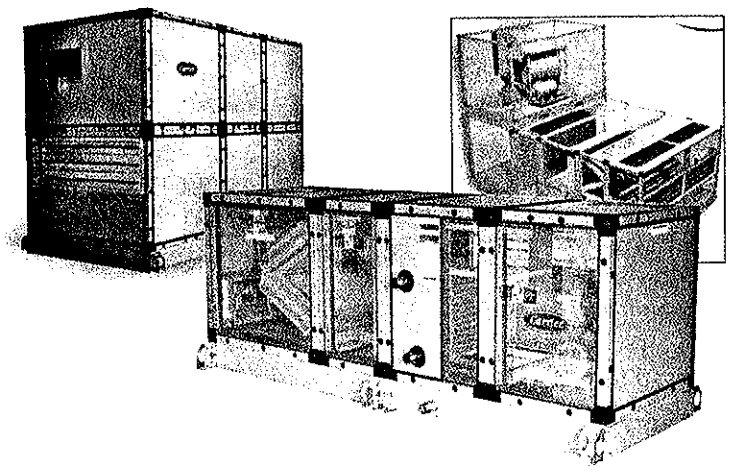
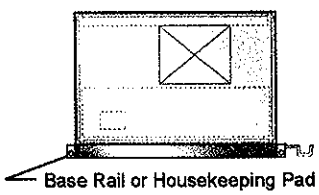


Figure 17

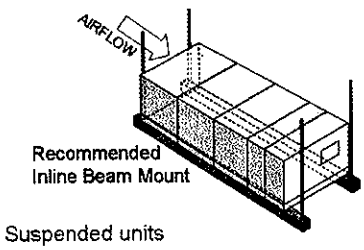
Indoor Air Handler Types – Horizontal, Vertical, Stacked

Units may be floor-mounted or suspended. Floor-mounted units may require space beneath them to allow room for construction of a condensate drain line trap. Since the condensate drain outlet is usually at the lowest point in the unit, the unit may have to be raised off the floor. This can be done by means of a concrete housekeeping pad or structural steel rails under the unit. However, most manufacturers can provide base rails ranging from 6 to 12 inches in height as an integral part of the unit to save the contractor time and money. If the base rails are high enough, the condensate drain trapping can be accomplished within the base rail height.



Floor-mounted units

- AHU must be elevated enough to allow for condensate drain trapping
- Factory-supplied base rails (full perimeter recommended)
- Field-constructed rails or concrete housekeeping pad
  - Added labor and material costs



Suspended units

- Follow manufacturer recommendations on unit support
- Locate units over NON-critical spaces to prevent:
  - Water leaks that could damage sensitive equipment or materials
  - Sound that could be objectionable
  - Restricted service access

Figure 18

Indoor Unit Mounting Considerations

Outdoor Units

Units intended for outdoor installation are designed to withstand weather conditions such as sun, wind, rain, and snow. The casings are generally constructed using G-90 mill-galvanized steel, which has a higher corrosion resistance (more zinc) than typically used on indoor units. Many manufacturers also offer painted casings for appearance as well as added durability. The exterior should have the appropriate salt-spray rating for the installation's requirements. Additionally, stainless steel or aluminum construction of the exterior panels should be used where corrosion poses serious risk to unit integrity.

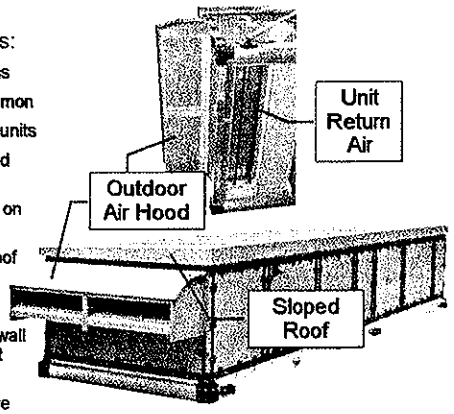


Figure 19

Outdoor Air Handler Design Features

Most outdoor units are designed to sit on a continuous curb like a packaged rooftop unit. They may also be installed on slabs, structural steel rails, or piers.

Because air handlers frequently operate at high static pressures, rain and outside air may be drawn in through any small cracks or openings. As a result, outdoor units generally employ the following design features:

- single-piece designs or weather-tight connection flanges
- sloped roofs that overhang the side panels to shed rainwater and provide positive drainage of melting snow. On units with single-side access, roof should slope away from the access-side of the unit. Roof panels should have standing-seams construction to allow water runoff without risk of penetration of the exterior skin.
- horizontal or bottom inlet air openings. Top inlets and discharge connections are discouraged due to potential leak risk
- high quality seals or caulk on all seams
- air inlet hoods with mist eliminators or weather-tight louvers
- exhaust hoods with insect screens
- double-wall construction with at least 2-inch walls to minimize thermal heat transfer and provide additional sealing surfaces

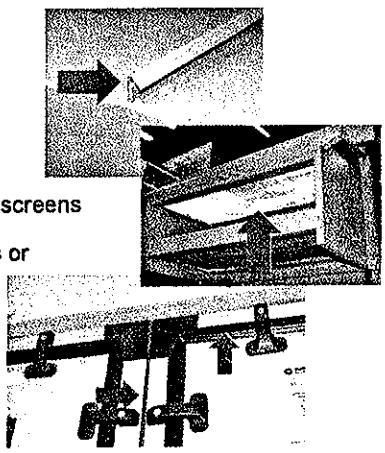


Figure 20

Additional Outdoor Air Handler Design Features

Draw-Thru and Blow-Thru

Central station air-handling units are generally classified as "draw-thru" units or "blow-thru" units, based on the relative position of the cooling coil with respect to the supply air fan. In draw-thru units, the cooling coil is located upstream on the inlet side of the fan. The fan draws, or pulls, the air through the cooling coil and other components and discharges it from the fan outlet directly into supply air ductwork. Draw-thru units are by far the most common, as they provide even air distribution over the cooling coil.

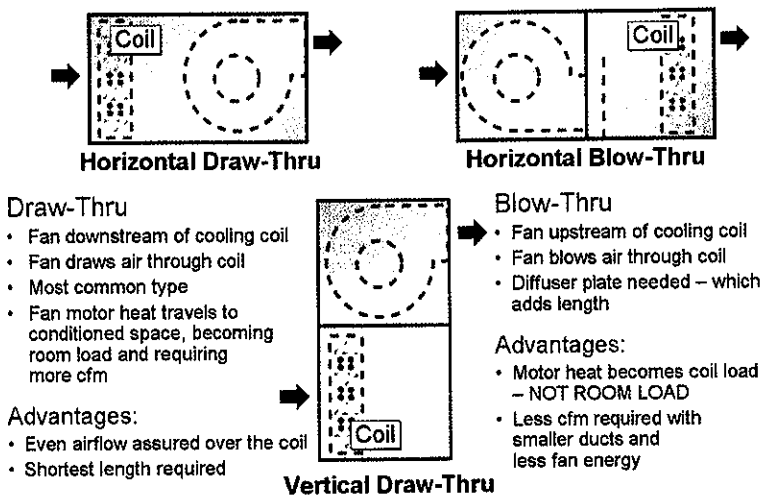
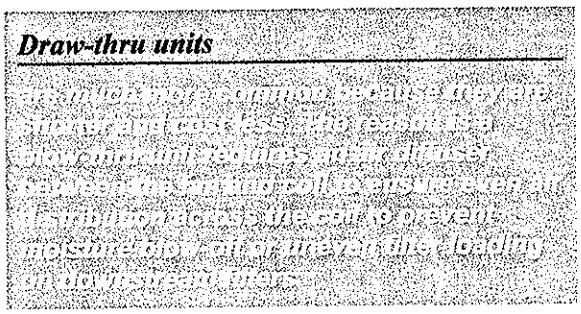


Figure 21

Draw-Thru and Blow-Thru Configurations

In a blow-thru unit, the cooling coil is on the discharge side of the fan, which blows or pushes the air through the coil. Blow-thru units have the advantage of allowing a reduction in the supply airflow vs. draw- thru units because the fan motor heat is immediately absorbed by the cooling coil as an equipment load, not transferred to the supply duct and conditioned space.



Coils

Most air handlers will typically include a section with cooling coils, heating coils or both. Cooling and heating coil sections are generally designed with tracks or coil mounting rails that accept a family of coils of various rows and circuiting.

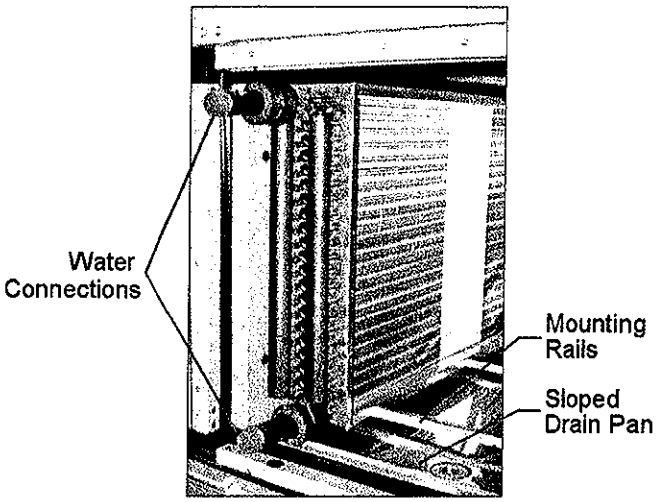


Figure 22

Cooling Coil Section

Types and Construction

Cooling coils are designed for use with chilled water or liquid refrigerant (direct expansion, or DX cooling coils). These are available in a variety of row depth, fin spacing, fin design, fin material, circuiting, and, in the case of DX coils, a variety of coil split arrangements.

Coil row offerings may vary from 2 to 12 rows with 4, 6, and 8 being predominantly used in the comfort air-conditioning industry.

Aluminum and copper are the materials used almost exclusively for both the tubes and fins of cooling coils, with copper tubes being used for the vast majority of commercial comfort applications. Aluminum fins are mechanically bonded to copper tubes by an expansion process as shown in Figure 24. Aluminum fins are used most extensively for air-conditioning duty. Copper fins, being more expensive, have limited usage but are definitely required as a minimal precaution against coil corrosion in applications where the coil is exposed to airborne materials such as hydrogen sulfide, sulfer

- Types
  - Chilled water (CW)
  - Direct expansion (DX)
  - Variety of rows, fin spacing, and splits
- Materials (Common)
  - Plate fins
    - Aluminum
    - Copper
  - Copper tubes
    - Multiple wall thicknesses
- Casings
  - Galvanized steel
  - Stainless steel

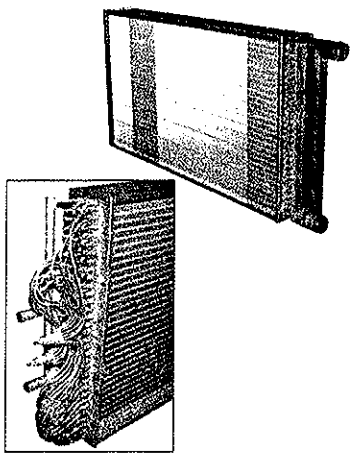


Figure 23

Cooling Coils

dioxide or (in high concentrations) carbon dioxide. Highly corrosive atmospheres, such as those encountered in many industrial process applications, may require special coil coatings to insure protection beyond that afforded by copper fins. Each of these situations should be individually analyzed with the aid of consultation from the equipment manufacturer. A more detailed description of coil construction features, circuiting, performance, and selection can be found in TDP-614, Coils: Direct Expansion, Chilled Water, and Heating.

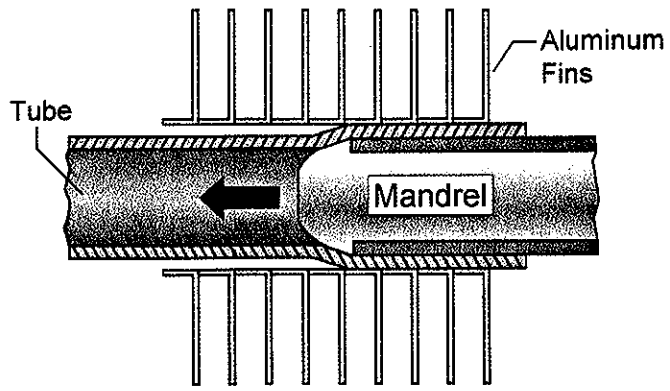


Figure 24

Mechanical Bonding Fins and Tube

Drain Pan

Cooling coil sections will always contain a drain pan to collect the condensate water extracted from the air passing over the coil. ASHRAE Standard 62 requires drain pans that do not allow standing water. To meet this requirement, drain pans will generally be sloped to a recessed bottom drain outlet. Drain pans will generally be constructed of galvanized or stainless steel, and may be coated with antimicrobial coatings to inhibit the growth of mold and bacteria.

Drain pans must be insulated to prevent the condensation of moisture on the outside of the unit casing. The insulation within the drain pan should be closed-cell foam, which is water resistant, and should be the full thickness of the casing insulation at the drain outlet to provide adequate thermal resistance. Fiberglass is not acceptable for this purpose as it will absorb water, lose its insulating properties, and provide a site for mold and bacterial growth.

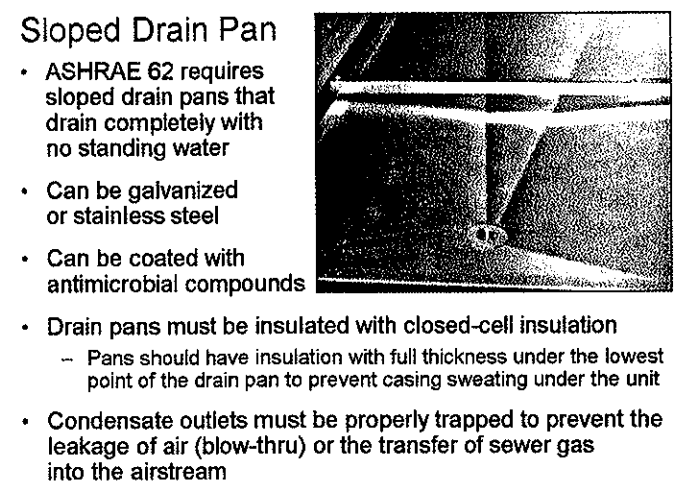


Figure 25

Cooling Coil Sections – Drain Pan

Condensate Drain Trapping

Condensate drain outlets must be properly trapped to isolate the air-handling system from the building drain system. Without traps on draw-thru units, air may be drawn into the air handler, introducing potentially objectionable smells and gases into the building. It can also impede water flow into the drain and cause water to backup into the drain pan. On blow-thru units, conditioned air can be lost, reducing the efficiency of the system.

The trap depth will be dependent on static pressure, either positive or negative, at the drain location. When calculating trap depth on draw-thru or blow-thru applications, remember that it is not the total static pressure, but the upstream or downstream static resistance that is trapped against. For instance, when calculating the trap depth for a cooling coil condensate pan on the draw-thru side, trap against the coil pressure drop in that coil section and any other pressure drops upstream of it.

When calculating trap depth on draw-thru or blow-thru applications, remember that it is not the total static pressure, but the upstream or downstream static resistance that is trapped against. For instance, when calculating the trap depth for a cooling coil condensate pan on the draw-thru side, trap against the coil pressure drop in that coil section and any other pressure drops upstream of it. Example of draw-thru trap:

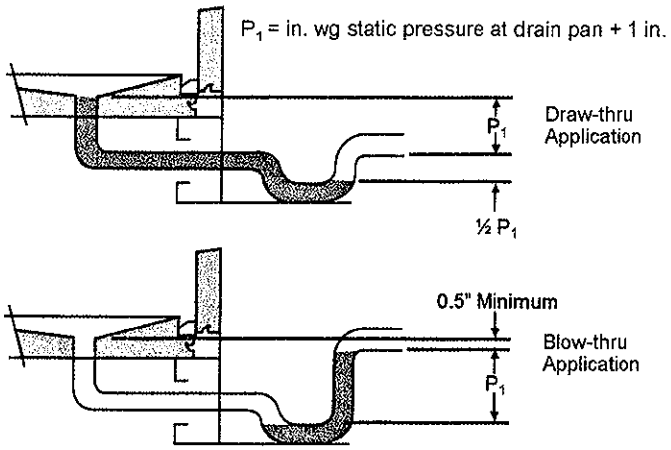


Figure 26

Condensate Trap Construction

- Return duct 0.5 in. static
- Mixing box 0.4 in. static
- Filters 0.2 in. static
- Heating coil 0.2 in. static
- Cooling coil 1.2 in. static

Total 2.5 in. static

To calculate trap depth,

If calculating the trap dept for the cooling coil, the total trap static would be 2.5 in. plus 1 in. safety factor (P_1 = negative static pressure +1 in.) as shown.

Traps on draw-thru units must store enough condensate to prevent losing the drain seal at start-up. The "minimum ½ P_1" dimension ensures that enough condensate is stored.

To determine the trap dimension for blow-thru units, find the coil's maximum positive pressure and add ½ in. safety factor. This figure is normally the fan total static pressure (P_1 = fan total static pressure).

For all units, provide condensate freeze-up protection as required. On units with internal spring isolators, be sure the unit is mounted to allow sufficient clearance for the required drain trap depth.

Fan Section Characteristics and Performance

Central station air handlers offer the designer the option to position fans in a variety of locations within the air handler to operate as supply, return, or exhaust fans. In addition, multiple fan wheel types increase the flexibility of an air-handling unit to meet differing application requirements. Each fan location has its advantages and disadvantages, making each a better application under specific circumstances.

Once an air handler unit size, configuration, and component selections are complete, the fan can be selected. Due to the variety of components offered, air handler fans are selected on the basis of total static pressure (TSP), rather than external static pressure (ESP), which is common on packaged HVAC products. Therefore, static pressure losses of all components of the airside system, both inside the air handler (dampers, filters, coils, etc), and external (ducts, dampers, diffusers) must be included in the fan total static pressure requirements. Most manufacturers provide selection software to aid in this process. Quite often, more than one fan will meet the performance requirements. At this point, the system designer must select the best fan, considering first cost, operating cost, and acoustics. For variable air volume (VAV) applications, part load stability must also be evaluated.

Supply Fan

Every air handler will have a supply air (SA) fan that must overcome all static pressure losses in the duct system, including internal unit components. Supply fans are offered in both draw-thru and blow-thru configurations. A draw-thru supply fan is intended to have the fan discharge connected directly to a supply duct fitting. The inlet will be connected to other sections of the air handler. A blow-thru supply fan differs from a draw-thru unit in that the casing is connected to additional downstream sections such as coil, filter and discharge plenum sections. Blow-thru fans with downstream coils or filters should have a diffuser plate, as shown by the dotted line in Figure 27, mounted downstream to break up the high velocity discharge air profile leaving the fan to ensure even air distribution over the downstream components. The diffuser may be built into the fan section, or a separate section type, depending on the manufacturer.

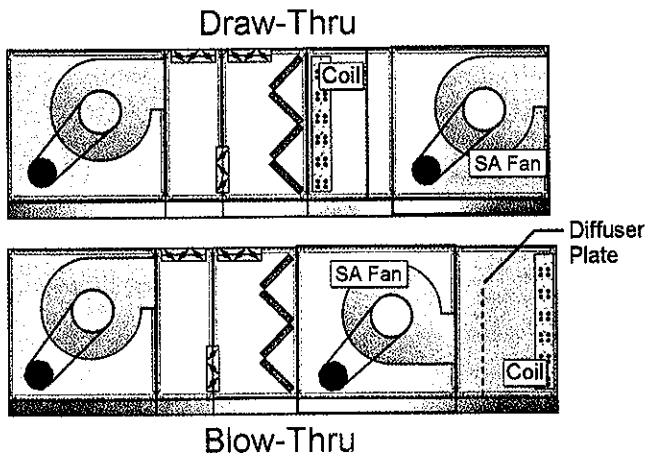
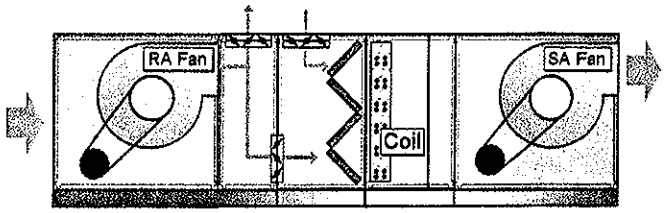


Figure 27

Supply Fan Locations

Return Fan

Return air (RA) fans are used in applications with long or complex return duct runs. Without them, the supply fan would over-pressurize the building in an attempt to push the air back through the return. This can cause problems opening doors, excessive exfiltration, excessive pressurization of membrane roofs, and other building problems. Return fans are configured to run anytime the supply fan is operating, and are generally sized to handle the same airflow as the supply fan, less any direct exhaust (such as toilets, fume hoods, etc) that is not returned to the AHU.



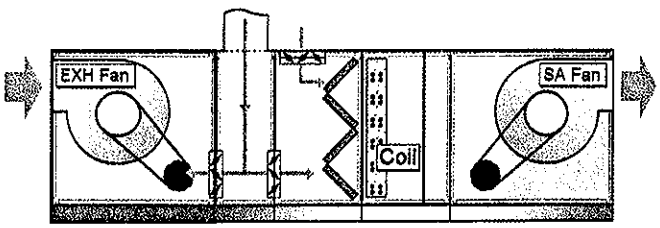
- Prevents excessive building pressurization
- Used in systems with high return duct pressure drops
- Runs only when supply fan is operating
- Generally controlled by building pressure sensor or tracks the supply fan on VAV systems
- Sized for less airflow than supply fan

Figure 28

Return Fan Features

Exhaust Fans

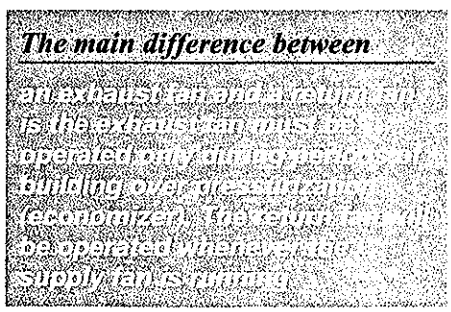
Exhaust fans are generally used where the return duct static losses do not require a return fan. The exhaust fan purpose is to alleviate temporary building over-pressurization conditions, which would exist during periods of time when the unit is in economizer or "free cooling" mode, bringing in a high percentage of outdoor air and barometric relief dampers cannot be used effectively. During these times, the mixing box return air damper would be closed or nearly closed. The exhaust fan would then be started to evacuate the exhaust air from the building and maintain reasonable building pressures.



- Prevent excessive building pressurization
- Operates only during "economizer mode" or when a high percentage of ventilation air is being drawn in
- Return air damper closed or nearly closed
- Sized to prevent building overpressurization

Figure 29

Exhaust Fan Features



Fan Discharge Arrangements

Discharge options provide the system designer with the flexibility to assemble air-handling unit combinations that fit the mechanical space and permit duct alignment that is efficient and simple to fabricate.

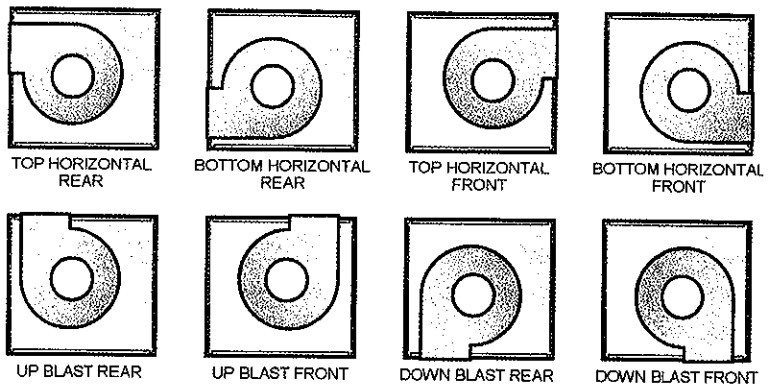


Figure 30

Fan Discharge Arrangements

Centrifugal Fan Types

As mentioned previously, most air handler manufacturers also offer multiple fan wheel types. While custom air handlers sometimes incorporate the use of axial, tubular, and mixed flow fans in high airflow applications, they will not be discussed in detail in this book. Refer to TDP-612, Fans: Features and Analysis, for discussion on these types of fans. Centrifugal fans are most commonly used in air handlers, which we will discuss here.

Twin fan arrangements (two fan wheels on a common shaft) are often used by custom manufacturers and some packaged air handlers to provide more uniform airflow across coils and are often selected to reduce the aspect ratio often associated with large flow rates.

Critical applications

Forward-Curved Impeller

Forward-curved (FC) centrifugal fans have a large number of blades that are curved in the direction of airflow. They produce large volumes of air at low static pressures, typically up to about 5 in. wg. They are fairly lightweight and provide a cost-effective solution to many applications. As a result, they are generally the designer's first choice when the application permits.

Characteristics:

- Most commonly used wheel in HVAC
- Light weight – lowest cost
- Operates at static pressures up to 5 in. wg max.
- 24 to 64 blades
- Low rpm (800-1200 rpm)

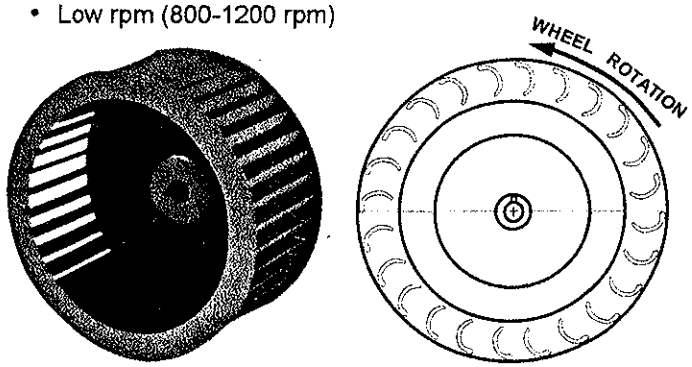


Figure 31

Forward-Curved Wheel Design

DWDI fan wheels

pull air in both sides at once. SWSI wheels (shown here) utilize a single side inlet.

Backward-Inclined Impeller

Backward-inclined (BI) centrifugal fans have blades with tips that point in a direction opposite their direction of rotation. They have fewer, heavier blades than forward-curved fans, and as a result they operate at higher speeds, producing lower volumes for a given fan size, but are capable of producing higher static pressures.

Characteristics:

- For medium to high air capacity and pressure applications (4 -10 in. wg)
- Comparison to airfoil:
  - Less efficient
  - Higher sound levels

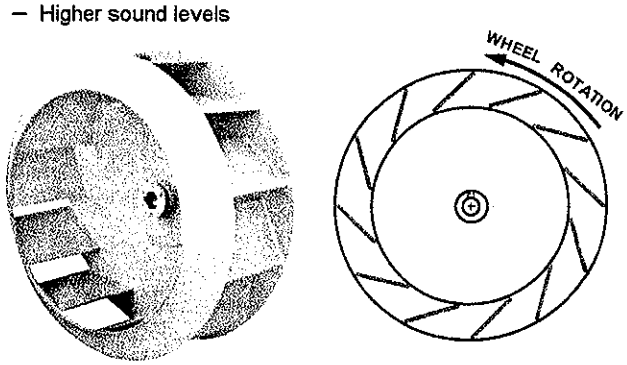


Figure 32

Backward-Inclined Wheel Design

Airfoil

Airfoil (AF) centrifugal fans are an aerodynamic variation of the backward-inclined fan. In contrast to the backward-inclined single-thickness blades, airfoil fan blades are double-thickness, with an airfoil cross-section. As such, they have similar performance characteristics, but with improved efficiency. They are generally preferred over their backward-inclined counterparts, even though they have slightly higher cost.

Characteristics:

- Aerodynamically designed – double thickness airfoil shape for maximum efficiency
- For medium to high air capacity and pressure applications (4 -10 in. wg)
- Lower sound levels than backward inclined fans
- Higher cost due to efficiency of design

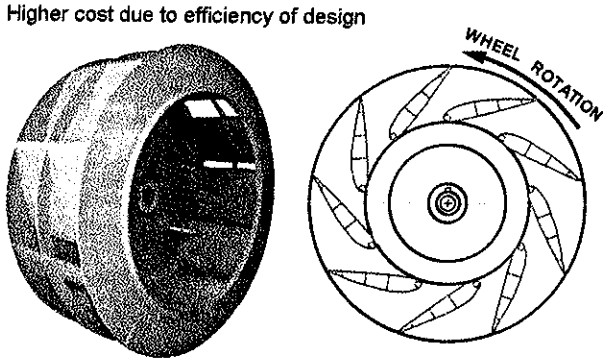
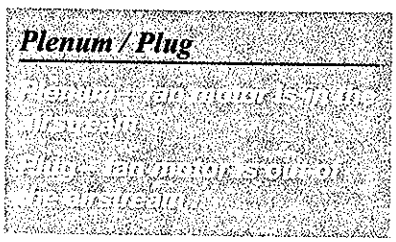


Figure 33

Airfoil Wheel Design

Plenum Style

Plenum fans are un-housed centrifugal fans that pressurize a plenum chamber rather than accelerate air into a duct. Generally, plenum fans use single-width, single-inlet (SWSI) airfoil wheels. The ability to field connect outlet ducts to the plenum in many configurations is the salient benefit of a plenum fan.



Characteristics:

- Single-Width; Single-Inlet (SWSI)
- Operates at static pressures up to 10 in. wg
- Best application with limited space or multiple ducts

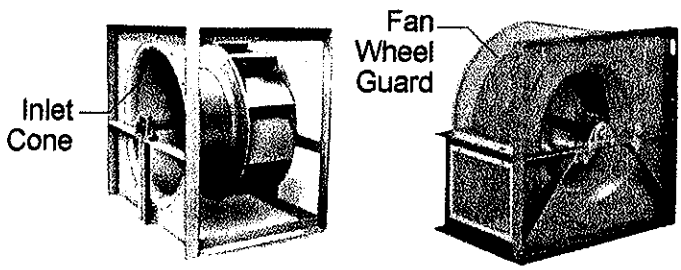


Figure 34

Plenum fans (SWSI)
Fan wheel guard photo courtesy of Barry Blower

VAV Fan Volume Control

Central station air handler fan sections are frequently used in variable air volume (VAV) systems. As its name implies, the airflow required by the system varies based on the position of the dampers in the VAV terminals. As a result, the fans need to employ some means of part load control. Let's take a look at the most common methods. Refer to TDP-613, Fans in VAV Systems for a complete discussion on fan modulation.

Riding the Fan Curve

The simplest method is called riding the fan curve. No additional control devices are necessary, and therefore it can be an easy, cost-effective solution. The fan simply reacts to a reduction in airflow according to its pressure-volume characteristics. Since the fan is running at a constant speed, the performance will track along its constant speed curve.

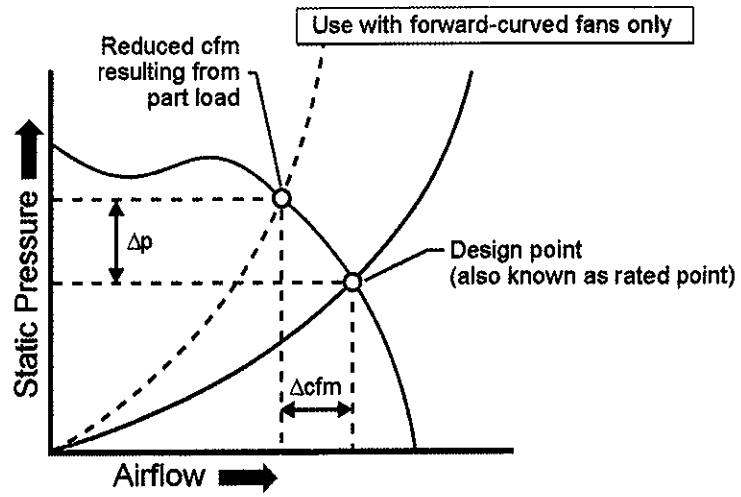


Figure 35

Riding the Fan Curve

This method should be used only on forward-curved centrifugal fans, which have a relatively flat system curve. In other words, large changes in airflow only result in small changes in static pressure. Care must be used to ensure that the additional static pressures encountered at low flow do not exceed the VAV terminal damper limits. Speed curves of backward-inclined and airfoil are too steep for this method of control to be used.

Inlet Guide Vanes

Figure 36 shows a close-up view of a typical inlet guide vane (IGV) assembly. The IGV is fastened to each inlet of the fan and is controlled from a fully open to fully closed position by a static pressure regulator using a sensor in the supply ductwork. A damper actuator moves the control arm to modulate the vanes. Inlet guide vanes reduce the fan capacity and required brake horsepower by reducing the area of the fan inlet. By adding pre-swirl to the air as it enters the fan inlet, they increase the fan efficiency. The drawbacks to IGVs are high first cost, setup and maintenance costs, and the increased pressure drop of the guide vane assembly. They can also be a source of additional unwanted noise. For these reasons, they are rapidly being replaced with variable frequency drives.

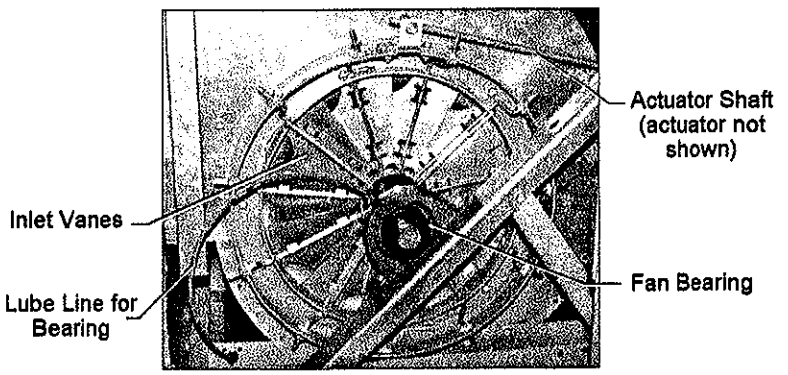
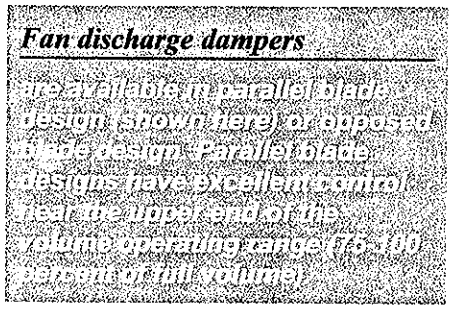


Figure 36

Inlet Guide Vanes

Discharge Dampers

Fan discharge dampers may be applied only to forward-curved centrifugal fans as shown in Figure 37. They must not be used on backward-inclined or airfoil impeller designs, as damage can occur to the fan housing, duct, and dampers due to the high pressures that are possible if the damper was to close. While they reduce fan capacity with an accompanying brake horsepower savings, they are rarely used today.



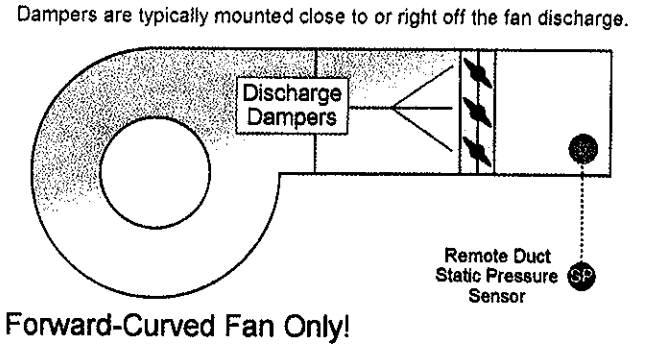


Figure 37

Discharge Dampers

Variable Frequency Drives

Over recent years, improved reliability and lower first cost have allowed variable frequency drives (VFD) to replace inlet guide vanes as the most common method of fan volume control. They provide the most flexibility with the least maintenance. They also yield the highest energy savings of all the volume control methods and do not contribute to fan noise.

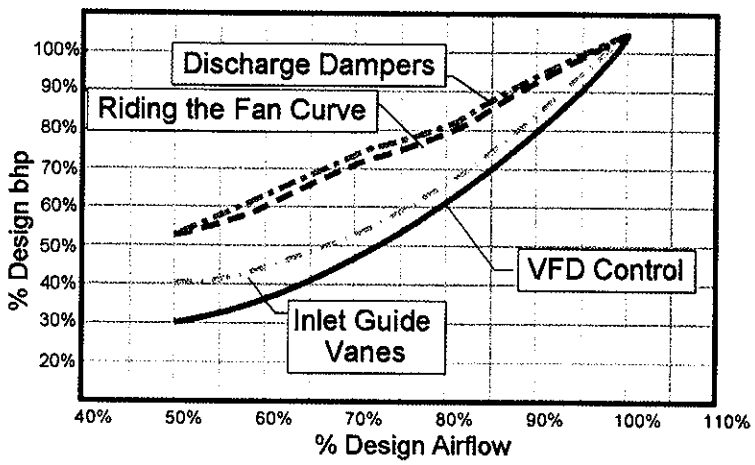


Figure 38

Comparison of Fan Modulation Methods

AMCA Fan Class

The Air Movement Control Association, Inc. (AMCA) has established classifications for centrifugal fans that define fan class as a function of minimum operating performance (AMCA Standard 99). There are three classifications: Class I, II, and III, with Class III being the highest performance. The class standards are based on a concept of mean brake horsepower per square foot of outlet area. A representative graph of performance for a given type of centrifugal fan is shown.

Performance is defined as a function of fan static pressure (in inches of water gauge) and fan outlet velocity (in feet per minute). Outlet velocity is used rather than airflow volume to enable the classification to apply to both large and small fans.

A system designer has two criteria, fan performance and fan construction, when selecting a fan. By defining fan performance, the AMCA classifications indirectly represent the structural limitations of a centrifugal fan design. For a given size of fan, a Class III version will be more sturdy and rugged than a Class II version simply because of the static pressure, velocity, and horsepower requirements to perform in that class. Either fan, however, may be operated at the same performance point (i.e., in the Class II range) and work equally well for the application. Even though a Class II fan will work, the system designer may specify Class III simply to get a more ruggedly built fan.

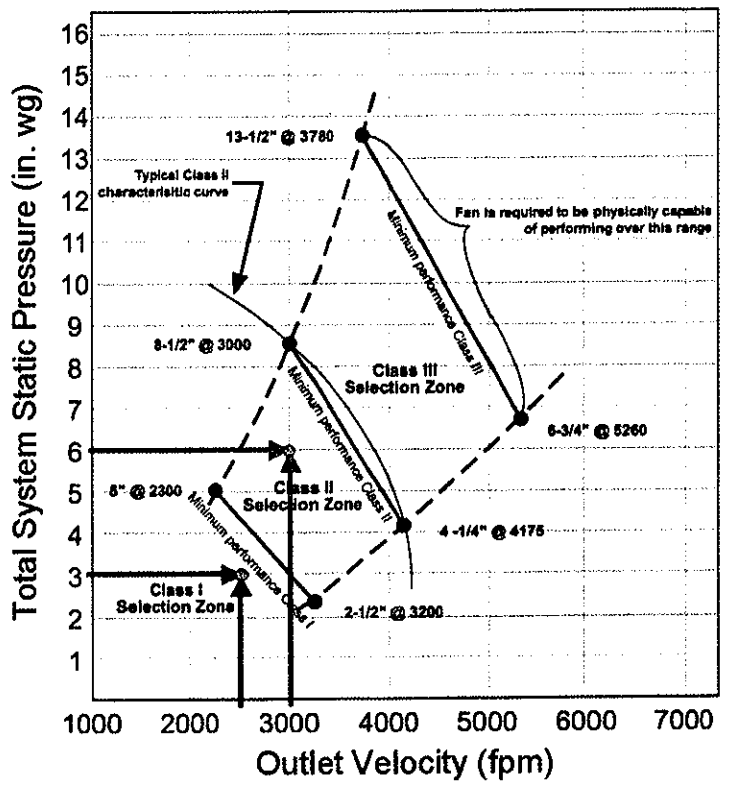


Figure 39

AMCA Fan Construction Class

If the fan discharge velocity is 3000 fpm and the total system static pressure is 6 in. wg, the operating conditions fall within the AMCA Class II range and a Class II fan should be considered for this application. If the fan discharge velocity is 2500 fpm and the total system static pressure is 3 in. wg, the operating conditions fall within the AMCA Class I range and a Class I fan could be used for this application.

Fan Components

Fan Mounting

Depending on the application, a system designer may choose to specify the air-handling unit with internal isolation of the fan and motor, or choose to externally isolate the entire unit. Coil sections, mixing boxes, and other modules, may be isolated as a complete assembly. In this latter circumstance, internal fan isolation is redundant. Fan sled, or assembly, isolators with a one or two-inch static deflection are most common.

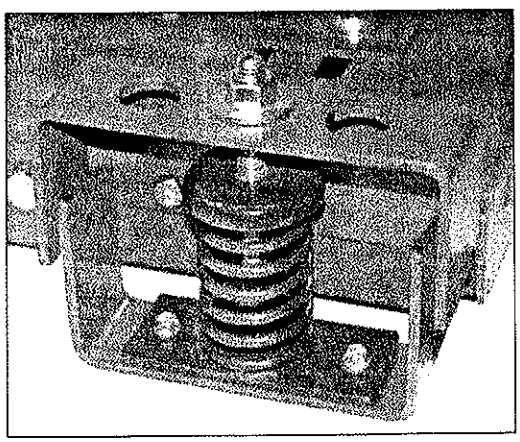


Figure 40

Fan Spring Isolation (Standard 2-inch)

Discharge Isolation

The fan discharge must be isolated to prevent transmission of vibration and noise to the attached ductwork. The traditional method of accomplishing this transition is with a short section of flexible, canvas duct.

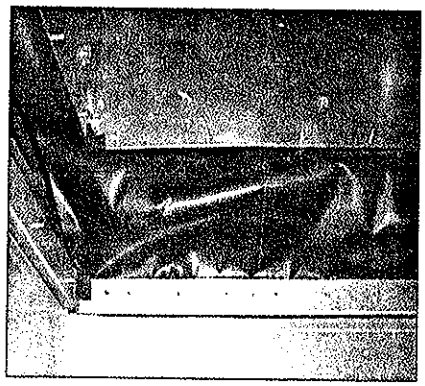


Figure 41

Traditional Canvas Connector

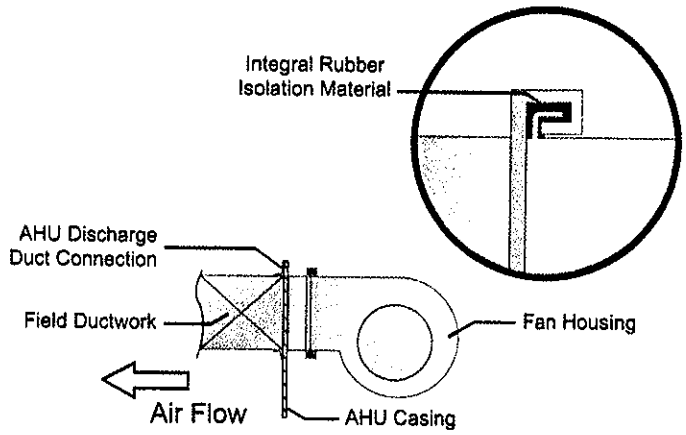


Figure 42

Rubber Connector

Some manufacturers use a vibration-absorbing rubber isolator that performs the same as a rubber-in-shear isolator in lieu of canvas.

Bearings

Fan manufacturers use several different types of bearings in their product line. That is because the bearings on a small gas furnace are subject to a different loading than those on a large central station air-handling unit. The important terms that one should understand apply to many bearing types. Fan manufacturers work with bearing suppliers to establish a level of quality and assure the bearing life expectance required by the HVAC industry.

Bearing Life

The life of a bearing is a function of the number of revolutions it experiences before developing evidence of fatigue in the moving elements. The terms that have been used in the industry are B_{10}, L_{10} and B_{50} or L_{50}. The terms B_{10} and L_{10} mean the same thing, as do B_{50} and L_{50}. The current terms to be used are L_{10} and L_{50}.

The American Bearing Manufacturer's Association (ABMA) defines L_{10} as the bearing life associated with a 90 percent reliability rate when operating under normal conditions. Normal operation means the bearing was kept clean, properly lubricated, operated at a reasonable temperature, and free of dust and debris with perfect alignment. In reality, this may not be the case, so the actual life of the bearing can be shortened based on the application conditions. However, following the manufacturer's installation and maintenance requirements will help extend the life to the manufacturer's specified values.

The designation L_{50} indicates the duration in hours that one half (50 percent) of the bearing can be expected to survive without showing evidence of failure. Conversely, it is the life at which one half of the bearings can be expected to fail. Thus a bearing with a longer L_{50} life rating for a given application can be expected to perform more reliably than another bearing with a shorter L_{50} life rating. L_{50} life equals five times the L_{10} life.

To get a L_{50} life equal to a L_{10} 100,000 life, you must specify the L_{50} life to be 500,000 hours.

Bearing life is useful when specifying a level of bearing construction. When required to provide a given life such as L_{10} all equipment manufacturers must supply the same capability bearing for the same given application. A 100,000 hour L_{10} bearing will have a life over twice as long as 40,000 hour L_{10} bearing and hence should last longer on a similar field application.

Hours and Years

How long is 200,000 hours? The following table converts hours to years based on different daily usage.

Hours	Years		
	8 hours per day	16 hours per day	Continuous Duty
40,000	13.7	6.8	4.6
100,000	34.2	17.1	11.4
200,000	68.4	34.2	22.8
400,000	137.0	68.4	45.8
500,000	171.0	85.6	57.0
1,000,000	342.0	171.0	114.0

Bearing Selection

Most manufacturers select their bearings as an integral part of the air-handling unit fan design. Some of the main selection criteria include shaft diameter and weight, motor horsepower range, weight and location on the shaft, maximum fan speed, fan wheel weight, and the direction of belt pull.

Ball bearings with stamped steel housings are well suited for applications with light loads, as in smaller equipment. The use of these bearings is limited to fan products with ¾ inch and smaller diameter shafts, and one horsepower and smaller motors, such as small fan units.

Air-handling units will tend to use ball, spherical, or tapered roller pillow block or flange-mount bearings. Once the application exceeds the speed limit for the contact seal and lubrication capabilities of the solid housing, a pillow block bearing is typically specified. The pillow block design incorporates a friction-free seal and a larger grease cavity. Higher speeds can then be attained and the rollers become the limiting factor instead of the seal.

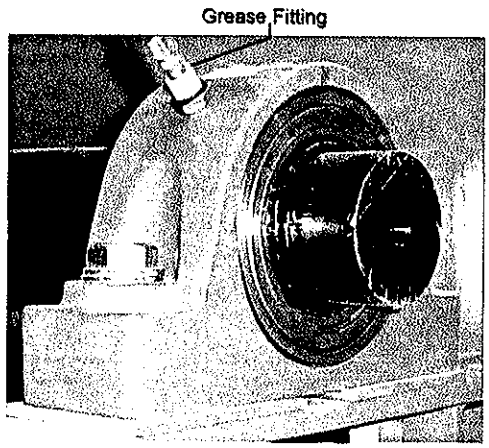


Figure 43

Typical Pillow Block Bearing

To enhance accessibility, it is often desirable to extend the bearing lubrication lines to the drive side of the fan. In some cases customers want the lubrication lines and fittings extended to the cabinet exterior so that bearing lubrication can be performed without stopping the unit. But, customers should also consider the downside of extended lube lines. Bearings should be inspected at the time of lubrication to look for improper operating conditions or signs of failure. If lube lines fail or vibrate loose, lubricating grease may never reach the bearing, creating an ideal condition for premature bearing failure. Also, bearings can be over-lubricated, in which case seals are dislodged, allowing the surplus lubricant to escape.

Drives

The one fan component that is most subject to wear is the drive. The primary function of the fan sheaves and V-belts is to transfer the speed and horsepower of the motor to the fan. In addition, the drive components must be selected to withstand the high-torque overload condition and belt slippage that occurs when the fan starts.

The fan drives are either fixed drive or adjustable drive. When a unit is furnished with an adjustable drive, the fan sheave diameter can be changed to fine tune the fan speed and performance.

The service factor indicates the drive's ability to perform reliably during overload conditions. Continuous duty fans are often specified with a higher service factor.

- **Belt drive is most common on central station units**
  - Adjustable pitch for low bhp
  - Fixed-pitch for bhp >15
- **Service factors typically 1.5 or higher**
- **Drive losses typically 3 to 5%**
  - Typically NOT included in fan selection software

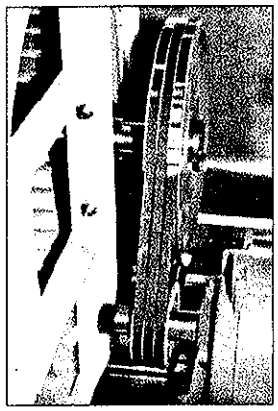


Figure 44

Types of Motor and Drive Arrangements

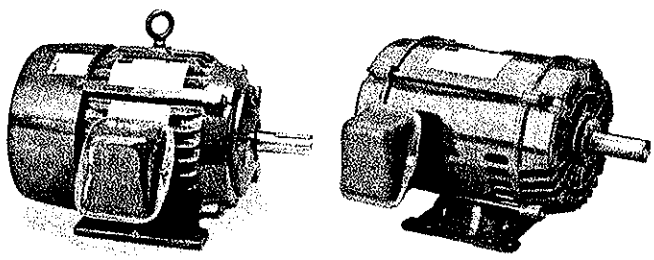
Overload Service Factors						
Percent Overload	0	25	50	75	100	150
Service Factor	1.0	1.1	1.2	1.3	1.4	1.5
Note: Design horsepower is equal to the indicated horsepower times the service factor. For continuous duty (16-24 hours per day) add 0.1 to the service factor						

The fan drives are either fixed drive or adjustable drive. When a unit is furnished with an adjustable drive, the fan sheave diameter can be changed to fine tune fan speed and performance. Although this is good for balancing the system, it is suggested that variable pitch drives be replaced with fixed pitch drives once the desired drive speed is obtained since variable pitch drives are more difficult to keep in balance.

Motors

Customers may have specific motor requirements dictated by environmental conditions or other system considerations. In addition, motors used with variable frequency drives must be specifically rated for use with VFDs.

A totally enclosed fan-cooled motor is a motor that is equipped for exterior cooling by means of a built-in fan. TEFC motors are used when water may splash or drip on the motor. An open drip proof motor has openings for ventilation air to cool the windings inside. The openings are positioned such that droplets from above would not interfere with the motor operation.



Totally Enclosed Fan-Cooled (TEFC) Motor

Open Drip Proof (ODP) Motor

Figure 45

Fan Motor Types

The Energy Policy Act of 1992 (EPAct) enacted by Congress set minimum efficiency levels for motors manufactured alone or as a component of another piece of equipment using general purpose motors rated for continuous duty and induction motors of NEMA designs A and B. Design A and B refers to the insulation class used in the motor and is based on the temperature rise above the cooling medium (air).

Unit Accessories

Most air-conditioning jobs today require more than just the basic air-handling unit components. Air handlers must also be responsible for mixing, heating, cleaning, and humidifying the air. Therefore, a number of accessory components are available which may be used to provide these supplementary functions. Much of the flexibility of application, which central station air handlers possess, comes from the variety of basic unit components (fan types, coil types, etc.) as well as accessory items which are offered with the unit.

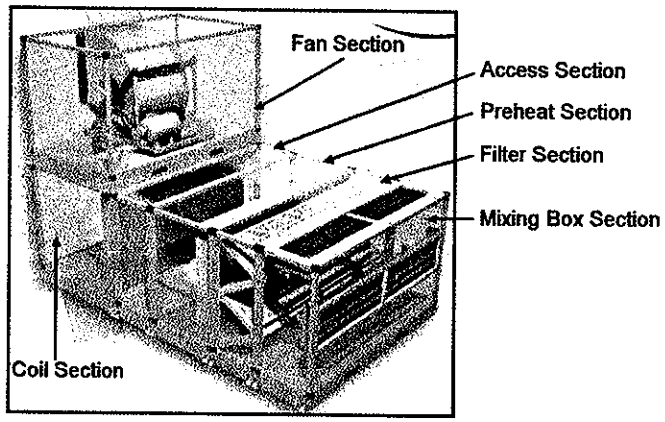


Figure 46

An Example of an Air Handler with Multiple Components

Filter Sections and Filters

Filtration is an important factor in maintaining indoor air quality and system efficiency. Multiple filtration options give the system designer necessary flexibility to specify the arrangement, media type, and media efficiency necessary to meet a customer's unique requirements. While the system application will dictate the level of filtration required, the internal air handler components (coils, fans, etc.) must be protected from airborne contaminants. For this reason, most air handlers will have filtration installed in a draw-thru position following the mixing box, and upstream of the first component or coil section. Critical applications such as hospitals, and clean rooms, and laboratories may also have additional filtration, commonly referred to as "final filtration," in the blow-thru position following the fan. This additional filtration traps fan belt dust and any contaminants that may have been drawn into the unit through casing leaks or access door openings.

The efficiency of air filtration is a matter of engineering design that is oriented to the application for which the space is intended. The particulate matter which filters are intended to remove varies from leaf-size particles all the way down to objects as tiny as bacteria and gaseous fumes. This broad range of particle size precludes the design of a single, all-purpose filter. As a result, multiple-filter designs have been developed, each offering a different efficiency, to accommodate different applications.

Filter Types and Ratings

Driven by market demand for improved indoor air quality, ASHRAE developed a new standard for testing and rating filtration efficiencies. Primarily aimed at higher efficiency filtration, ASHRAE Standard 52.2 provides a more absolute method of testing and rating filtration performance, particularly in the 0.3 to 1.0 micron particle size range. Instead of the traditional dust spot and average arrestance ratings used previously, new filter efficiency ratings are expressed in terms of Minimum Efficiency Reporting Values or MERV numbers, which range from 1 to 20, with 20 being the highest efficiency. While there is no exact correlation between past efficiency ratings and the new ratings, they can be approximated.

Filter Classifications Based on Particle Size			
Particle Size (Microns)	Typical Pollutant	General Efficiency	Typical Filter Type
0.1 to 0.3	Sea Salt, Carbon Dust, Combustion Smoke	Ultra-High	HEPA/ULPA
0.3 to 1	Tobacco Smoke, Copier Toner, Bacteria	High	Bag or Cartridge
1 to 3	Legionella, Auto Emissions, Welding Fumes	Medium	Bag or Cartridge
3 to 10	Pollen, Mold, Dust Mites, Sanding Dust	Low	Pleated or Throwaway

Reference ASHRAE Std 52.2-1999 for more detailed information.

Figure 47

Filter Selection Criteria

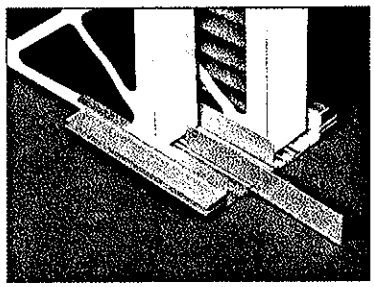
For more detailed explanation of filter ratings, see TDP-644, Filtration.

Each manufacturer of central station air handlers offers its own line of filtering systems. The efficiency range and types of filtering systems offered may vary from manufacturer to manufacturer but some of the most commonly used types will be described here.

Panel Filters

Panel filters are the least expensive and the most commonly used type of filter in the comfort air-conditioning industry today. As its name implies, these filters have a flat, rectangular shape. A number of standard rectangular sizes are available, typically in depths of one, two, or four inches. Panel filters achieve what is generally categorized as low efficiency filtration, up to approximately 30 percent, or MERV 6. This typically represents the minimum acceptable quality of filtration for comfort air-conditioning applications.

- Most commonly used for air conditioning
- Minimum filtration capability
- Up to MERV 6
- Throw-away:
  - Cardboard frames (shown)
  - Metal or plastic screens
  - Fiber or spun glass fill
  - Adhesive coating
  - Pleated media available
- Permanent
  - Metal frame and filter media
  - Cleanable



- High velocity in vertical arrangement (shown above)
  - 400 to 700 fpm
  - Commonly called "flat filters"
  - Shortest airway length

Figure 48

Panel Filter

Panel filters are designed in both throwaway and permanent types. The throwaway type usually has cardboard frames, with either metal or plastic perforated screens for the entering and leaving air sides. The filtering media can be made from any of several types of fiberfill; spun glass fiber and hemp fiber being two commonly used types. Frequently, the density of the fill increases from the entering to the leaving air surface of the filter. An adhesive coating is usually applied to the filter media while the filter is being made to improve its particle-holding capability.

The design of permanent style panel filter is very similar to that described for the throwaway. The major difference is that the framework and filter media are metal.

Panel filters are usually installed on the intake side of the fan upstream of the coil(s). They can function as either high velocity or low velocity filters depending upon the filter cabinet design in which they are installed.

Flat Filter Section

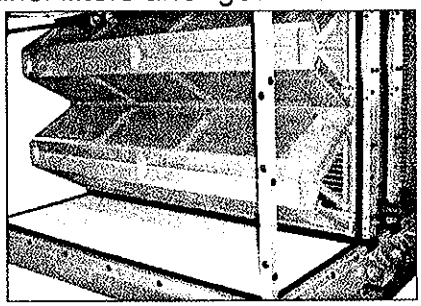
A high velocity filter, or flat filter section, orients the panel filters in a straight vertical manner. This arrangement yields the shortest possible airway length, which is important for installations where space is limited. It is also the most cost effective. The velocity through the filters approximates that for the cooling coil. Normal face velocities range from 400 to about 550 feet per minute.

Angle Filter Section

Figure 49 shows a typical low-velocity filter, or angle filter section. Angle filter sections use panel type filters arranged in a series of V-banks to achieve lower air velocities across the filters than at the cooling coil. Typical filter face velocities of 350 feet per minute or below enable the panel filter to operate more efficiently than in a high velocity flat application.

As with most types of filter sections, access to the low velocity filter section for filter replacement or servicing is accomplished through an access door at the end of the casing. Angle filter arrangements provide increased contaminant-holding capability, which lengthens the interval between filter changes in comparison to a high velocity filter arrangement.

Panel filters arranged in a "V" bank



- **Commonly referred to as "angle filter" sections**
- **Reduces velocity through filters to about 350 fpm**
  - **Better filtration efficiency**
  - **Lower pressure drop vs. high velocity (flat) filter section**

Figure 49

Angle Filter Section

Bag Filters

One alternative to panel filters, which offers medium efficiency filtration up to approximately 75 percent or MERV 12, is a bag filter. This filter removes fine dust particles that cause surface soiling.

The bags are constructed of a special blend of paper and fiber in the shape of a bag. This shape creates an extensive filter surface area, which results in low air velocities across the filter. Standard as well as extra-length bags are typically offered, the latter being used where larger-than-normal air quantities are involved.

- **Medium to high efficiency**
- **Up to MERV 12**
- **Removes "fine" dust particles**
- **Greater filter area**
  - **Lower velocities**
  - **More dirt absorption**
- **Throw away when dirty**
- **High velocity pre-filters recommended**

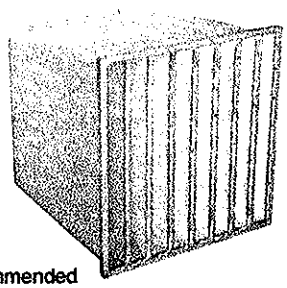


Figure 50

Bag Filter

Since these filters have a higher efficiency and more complex design than panel filters, they cost considerably more. Therefore, high velocity pre-filters are usually installed upstream of the bag filter to remove large particulate matter and reduce the replacement frequency of the bag filter. Most bag filter sections provide pre-filter tracks in the filter section for this purpose.

Cartridge Filters

Cartridge (or box) filters provide medium to high efficiency filtration in the 75 to 95 percent range (MERV 12-16).

Cartridge filters have pleated media installed in a rigid frame constructed of metal or plastic. The filters are typically 6 or 12 inches deep. There are two styles of filter cartridges: cartridges with header frames and those without, to accommodate either side-loading or front-loading filter racks.

Filters without headers have a simple rectangular box shape. They mount into a front-loading filter frame that is gasketed to minimize air bypass around the filter.

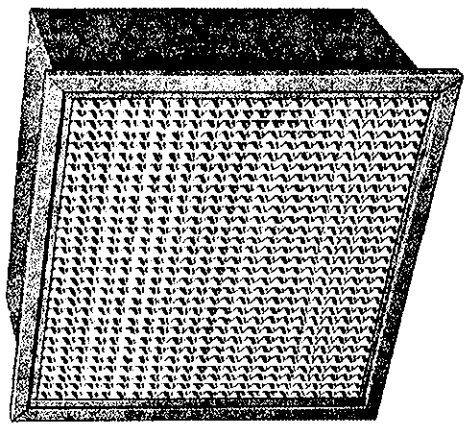


Figure 51

Cartridge filters will not sag at reduced cfms in VAV systems.

The filter is held in place by retaining clips or springs in a front-loading filter rack. Due to the cost of cartridge filters, pre-filters are generally used upstream to trap the larger particulates and extend the life of the cartridge. Cartridge filters with headers may be mounted in front-loading racks as described earlier, or into side-loading racks, similar to flat or angle filter sections. The filter has a one-inch header frame around the filter, which is used to hold it in place in the filter track.

The advantage of side-loading filters over front-loading is that they require a shorter airway length, as they do not need upstream space for service access to replace the filter. They are also faster to replace than front loading.

HEPA Filters

HEPA stands for High-Efficiency Particulate Air. These filters are also referred to as "absolute" filters since they achieve efficiencies of 99.97 percent or higher (MERV 16 or higher) and can remove particles as small as 0.3 microns or less from the airstream. Applications for this type of filter include hospital operating rooms, research laboratories, pharmaceutical plants and any other applications where atmospheric contamination must be prevented.

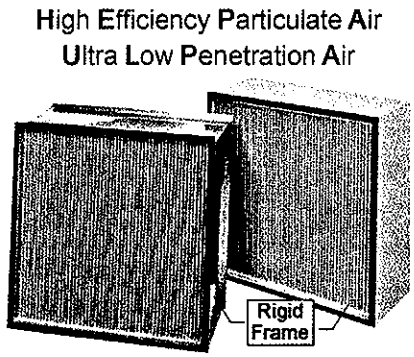


Figure 52

HEPA/ULPA Filters

HEPA filters are installed in special front-loading frames specially designed to prevent any air leakage through them. The filter frames are very rigid to eliminate deflection under pressure. They are also gasketed and caulked in place, typically with silicone-based adhesive caulking. HEPA filters are almost always installed as "final" filters.

Due to the extremely high cost of HEPA filters, upstream filtration of the types described previously should always be used to minimize the HEPA filter replacement.

"Ultra-Low Penetration Air" Filters

An ULPA filter is an extended paper media filter built in a rigid frame having a minimum particle-collection efficiency of 99.999 percent on 0.12 micrometers. The media area can range from 89 square feet to 242 square feet with an initial resistance range of 0.31 in. wg to 1.0 in. wg. Final pressure drop measurements are from 0.70 in. wg to 2.0 in. wg.

Electrostatic Filters

The electrostatic filter removes particulate matter from the air by placing an electrical charge on particles and collecting them on electrostatically-charged surfaces. Efficiencies for this type of filtering method are comparable to that of bag filters. However, electrostatic filters have a lower pressure drop than bag filters. Saving the replacement media cost associated with bag filters makes the electrostatic filtering method attractive for certain applications.

- Dirt particles are electrically charged
- Ionizing grids (opposite charge) in the filter attract charged dirt particles
- Efficiency similar to bag filters
- Permanent and cleanable
- Can be separate and field coupled to AHU
- Preceded by upstream low-efficiency pre-filters

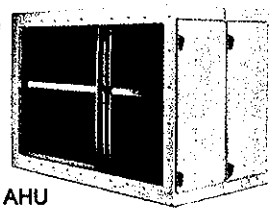


Figure 53

Electrostatic Filter

There are currently very few central station air-handling manufacturers who offer electrostatic filters as a part of their standard commercial product line. Large commercial applications usually utilize the appropriate size unit purchased as a separate component from one of the large filter manufacturing firms, making any necessary field modifications to couple the unit with the air handler. Usage of electrostatic filters is predominantly restricted to the residential and light commercial markets.

Impact of Filter Pressure Drop on Fan Selection

You may want to select an air handler with dirty filters so that, in theory, the unit will have enough horsepower to deliver the same amount of air when the filters are dirty. On a constant volume unit, that would only work if the unit contained an airflow measuring station and could adjust the flow accordingly using a VFD. Otherwise, the point of operation moves along the rpm line as the static pressure in the system changes.

Three things happen when you order the fan with sheaves selected for dirty filters:

1. The sheaves will be replaced with a smaller size by the air balancer because the airflow is too high when filters are clean. When the filters load up, airflow is reduced.

2. If an air balance is not performed, the cooling coil may exhibit moisture carryover due to the considerable increase in airflow.
3. The fan motor trips out on overload with the forward-curved centrifugal fan because of the increase in bhp.

Example: Selecting a forward-curved fan for dirty filter pressure drop

In this example, we are showing a forward-curved fan rated at 15,000 cfm, 1010 rpm, 17.8 hp, and selected with 100 percent dirty 60-65 percent cartridge filters and pre-filters. The dirty filters result in a total static pressure (TSP) of 4 in. wg. Clean filters, however, would have resulted in a TSP of only 2.55 in. wg.

If we follow the 1010 rpm line down to 2.55 in. wg reflecting the clean filter TSP, we would have a clean filter cfm of 21,000! Also note that the horsepower goes from 17.8 bhp to about 28 bhp because the forward-curve fan continues to draw hp as the cfm increases.

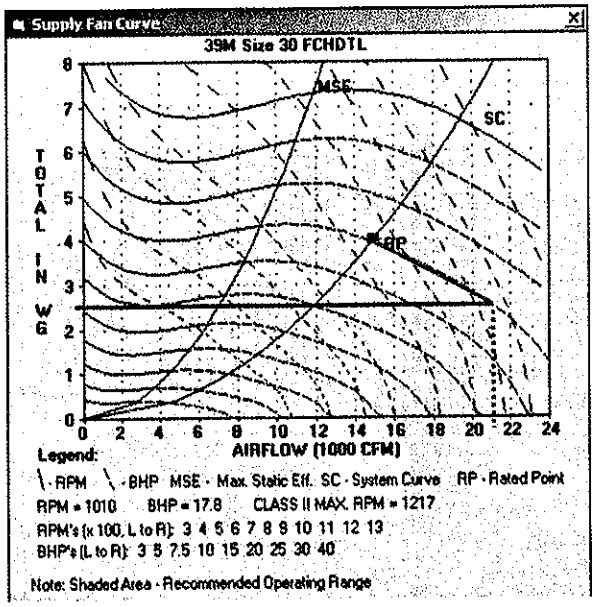


Figure 54

Forward-curved fan selection using dirty filter pressure drop

If dirty filters need to be taken into consideration, do one of the following:

1. Make the final fan selection with the clean filter rpm but use the motor horsepower requirement for dirty filters.
2. Make the final fan selection with the dirty filter rpm and use the motor horsepower requirement for dirty filters only if the engineer plans on using a VFD and airflow measurement station or if it is a VAV system.
3. Use an airfoil fan when the difference between dirty and clean filter pressure drop is greater than 1 in. wg. That way, the difference between clean and dirty airflow is minimized.

Example: Using an airfoil fan

The design airflow is 15,000 cfm, the total static pressure with dirty filters is 4.0 in. wg, the resulting fan rpm is 2210, the clean filter TSP is 2.25 in. wg.

To find the operating point with clean filters, follow the 2210 rpm line down to 2.55 in. wg. Clean cfm will be 16,700 cfm. Since airfoil fans are non-overloading (bhp lines run parallel with rpm lines) the bhp does not change. However, coil blow off is possible due to the increase in airflow.

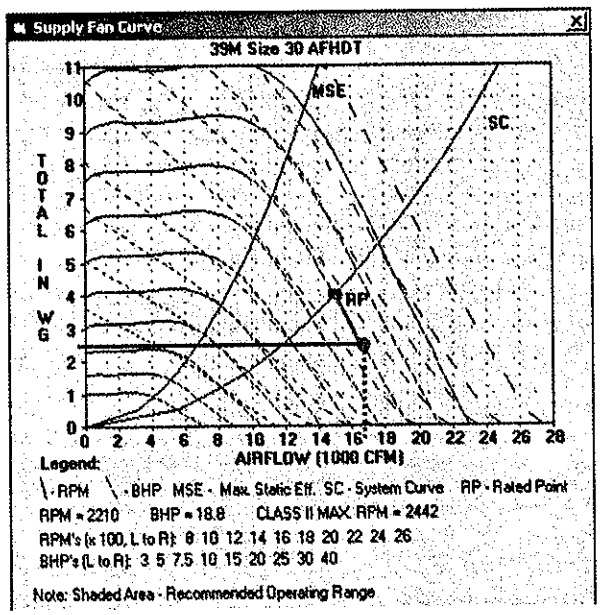


Figure 55

Airfoil Fan Example

Access and Plenum Sections

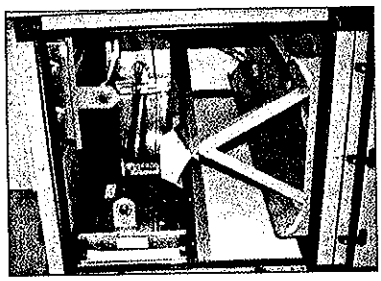
Access and plenum sections are used to provide space between other components for service access, and for additional components to be installed at the jobsite, such as humidifiers, sound attenuators, etc. They are generally available in a variety of airway lengths and can be equipped with drain pans if necessary. Plenum sections are simply access sections without doors. They are useful where cost is a primary concern and the section does not require frequent access.

Blow-thru plenums are basically a section with a discharge air opening that is used primarily to facilitate duct connection. On units with bottom openings, safety grates are frequently used to prevent people or tools from falling in.

Mixing Box Section

The mixing box is an air plenum with two air inlets, each of which contains a volume control damper. When installed in its normal position, at the entering-air end of the air-handling unit, this accessory will provide the proper mixture of return and outdoor air.

The interconnecting linkage between the dampers closes one while the other is opening. While this example shows the damper intakes on the bottom and rear of this component, either opening can also be located on the top, or sides of the mixing box for added flexibility.



- Dampers
  - Parallel and opposed blades
  - Various leakage rates available
    - Check manufacturer's data
- Recommend:
  - Parallel blade for return air
  - Opposed blade for outdoor air

- Plenum with two air inlets
  - Outdoor air
  - Return air
- Positions controlled to provide:
  - Ventilation air
  - Free cooling (economizer)
  - Indoor air quality (CO_2)
- Dampers are interconnected
  - When one opens, the other closes

Figure 56

Mixing Box Characteristics

Manufacturers may offer several damper options that provide different leakage rates, and pressure drops. Low-leakage dampers are important when an air-handling unit cycles on and off. When the unit is off, loose-fitting dampers allow unconditioned outdoor air to leak into the building, conditioned air to leak out, or both. Leakage through a damper that is supposed to be closed means that energy is being wasted and the air-handling system is less efficient than it could be.

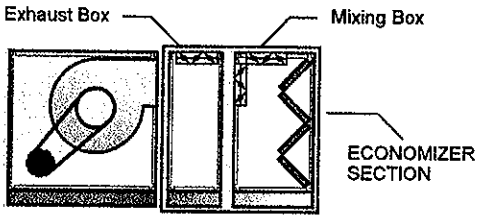
For most continuous-duty units, the outdoor air and return air dampers are rarely fully closed (except during economizer or unoccupied operation). During normal operation, the more important damper criteria are low-pressure loss and quality construction.

Manufacturers use various terms

such as low-leakage and ultra-low-leakage to describe the leakage rates. Therefore, the most meaningful comparison is to quote actual leakage volume or leakage per square foot of damper area.

Exhaust and Economizer Sections

An exhaust box is generally used immediately downstream of a return air fan section to allow air to be exhausted from the system. Some manufacturers offer separate exhaust box sections, which can be mated with a standard mixing box. Others offer a single section containing exhaust, return and ventilation air dampers in a single section. This is commonly referred to as an economizer section.



- Used downstream of return fan
- Provides outlet to relieve excess air from building
- Maintains space pressurization
- Dampers are controlled by space pressure sensor, or linked to mixing box dampers

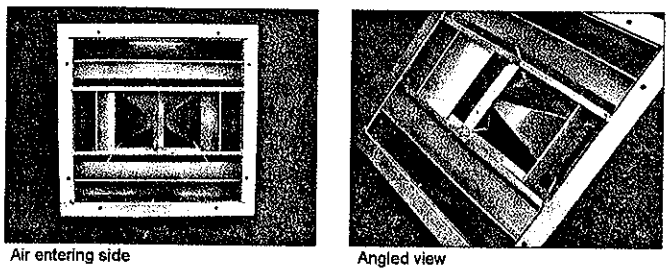
Figure 57

Exhaust and Economizer Box

Air Mixers

Under certain winter conditions, outdoor air entering an air-handling unit may not mix well with return air. Variations in air quantities and temperatures contribute to the problem. The colder outdoor air will stratify to the lower part of the unit, increasing the risk of coil freezing.

Static air mixers are stationary devices made up of multiple, directional-turning vanes that purposely cause turbulence in the combined airstream (outdoor air plus return air) to promote mixing. They are generally installed immediately down-stream of the mixing box section.



Installed downstream of Mixing Boxes

Directional turning vanes

- Air velocity 1000 fpm through mixer
- Causes turbulence and forces air to mix
- Minimizes temperature stratification
- Helps prevent coil freeze-ups or nuisance freezestat tripping

Figure 58

Static Air Mixer

The quantity of air going through the mixing section creates a pressure drop and causes the air to pass through the passages in the mixing section at some nominal design velocity. The mixing efficiency for the mixer is specified at various reference airflow points.

While they perform reasonably well in constant volume systems, in variable air volume (VAV) applications, they quickly lose their mixing effectiveness as airflow is reduced below the design rate. This is typical in VAV air handlers where the unit rarely operates at design airflow, especially immediately after a morning warm-up cycle has terminated and the unit starts the cooling mode. This is the point when most freezestat trips occur. Airflow through the unit is reduced to a very low level because the building's cooling requirement is minimal.

The total system cfm drops to the sum of the minimum airflow set points of the terminals. Therefore, it is extremely important that the designer evaluates the minimum system airflow and ensures that it is above the minimum required by the air mixer.

Face and Bypass Damper Sections

Another accessory that may be used with an air-handling unit is a face and bypass damper section. Two sets of interlocked dampers are installed, generally in one section. The face damper is located directly in front of the cooling coil and the bypass damper is located above the coil, controlling airflow through a bypass duct around the coil.

This arrangement can be used with heating or cooling coils. They control heating capacity by diverting air around the coil, rather than throttling the hot water or steam to the coil. This minimizes the possibility of coil freeze-up in cold climates.

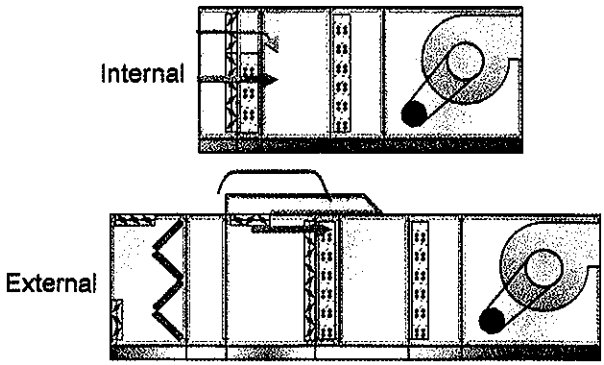


Figure 59

Face and Bypass

In cooling applications, face and bypass dampers are generally used only with chilled water coils, not direct expansion (DX) refrigerant coils. In chilled-water applications they are effective in high latent load applications. In conjunction with a chilled-water valve, they can reduce the leaving air temperatures off the cooling coil providing more dehumidification. This dehumidified air is then mixed with bypassed air to bring it up to an acceptable level for distribution to the space.

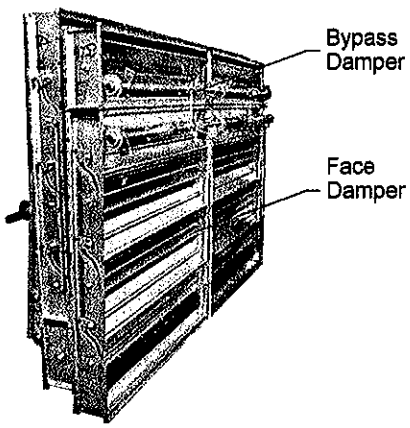


Figure 60

Face and Bypass Dampers

On applications employing face and bypass sections, the fan selection and air distribution system must be designed for an air quantity about 10 percent above design dehumidified air volume. This additional air quantity compensates for leakage through a fully-closed bypass damper and for air quantity variations when the dampers are in intermediate positions.

Traditionally, face and bypass has been discouraged in DX systems because if the controls allow the face dampers to close or significantly reduce airflow across the coil, the coil could freeze up and cause liquid refrigerant to flood back and cause compressor damage.

Humidifiers

Maintaining proper humidification is an integral component of indoor air quality. When humidity levels are low, higher room temperatures are generally necessary to maintain thermal comfort. Low relative humidity also increases respiratory complaints caused by drying membranes in the nose, throat, and respiratory system.

Very low humidity (and very high humidity) is detrimental to human comfort, health, and productivity. In winter, controlling relative humidity to approximately 30 percent and 35 percent at normal room temperatures has been shown to be the optimum range.

There are several types of humidifiers, differentiated by the moisture source. The spray humidifier section consists of a header and spray nozzles with a strainer which may be field installed.

The sprays operate on a water pressure of 15 psi or more. The humidifier should be installed directly ahead of the chilled water coil to permit drain water to flow to the condensate pan and out through the condensate piping. Spray humidifiers may result in dust accumulation in the occupied space if the supply water has any dissolved salts. Spray humidifiers may be used with copper or aluminum finned coils when naturally soft city water is available. If the water is contaminated, dissimilar fin and tube materials may result in a significant reduction of coil life due to corrosion. When industrial gases such as hydrogen sulfide, sulfur dioxide or carbon dioxide are present in the spray water or in the outdoor air, use of an all copper coil is recommended.

When face and bypass damper control is used, controls should be set so that there is always a minimum flow of air through the spray apparatus. A solenoid valve should be installed in the water line or other suitable precautions taken to shut off sprays when the fan is not running

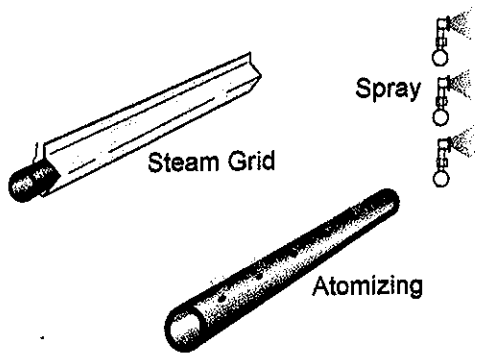


Figure 61

Types of Humidifiers

The atomizing spray humidifier assembly is made of a single horizontal pipe placed on the leaving side of the cooling coil and above the condensate pan. Brass, cone-shaped, 1/8" nozzles with a spray angle of 70 degrees are screwed into the pipe. Each nozzle has a fine mesh brass strainer. To minimize nozzle clogging, it is recommended that the supply pipe also have a strainer. The atomizing sprays operate on city water pressure of 15 psi or higher.

The steam grid humidifier is constructed of an open or slotted pan that encloses a steam pipe with drilled holes in the pipe to distribute the steam. The pan is pitched to facilitate condensate drainage. A condensate trap and liquid leg is required when operating under negative pressure to prevent "hold-up" of condensate in the pan. The steam supply to this humidifier should be odor free to prevent build-up of objectionable odors in the conditioned space. Automatic water make-up must also be provided to maintain the proper water level in the steam generator.

These types of humidifiers

Spray humidifiers and atomizing spray humidifiers are not recommended for use on blow-thru type units. On these units, the humidifiers are located on the positive pressure side of the fan and spray types may cause leakage at the joints in the casing. Steam grid types are, however, acceptable for blow-thru units.

Ultraviolet Light Germicidal Lamps

The quality of indoor air is difficult to assess. But, in all situations where IAQ is a problem, there are three factors present: a contaminant, susceptible occupants, and a mechanism to transport the contaminant. In many cases the transport mechanism is the HVAC system.

In HVAC systems, bioaerosols are of particular concern. Bioaerosols are airborne particles that may be living organisms, spores, or fragments of living organisms. They include pathogens (viruses, mold, and bacteria), allergens (mold and bacteria that cause allergic reaction), and toxins. In the air, bioaerosols behave like gases in that they float in the air and can remain suspended for hours. UV-c lamps kill stationary particles, not particles in the moving airstream.

- According to BOMA (Building Owners & Managers Assoc.), the average condensate drain pan is cleaned only 3.5 times per year
- Condensate drain pan and evaporator coil may be sources of mold/bacteria growth under certain conditions (68-86° F)
- Limits biological growth on evaporator coil and condensate drain pan
  - Maintains design air pressure drop
  - Helps reduce maintenance costs
- General rule of thumb sizing:
  - 1 lamp per 12 ft² of coil surface @ 12" distance from coil

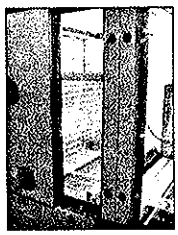




Figure 62

UV-c Germicidal Lamps

The best placement

An ultraviolet germicidal lamp is most effective if it is on the downstream side of the cooling coil over the drain pan. This permits simultaneous irradiation of both areas by preventing the bioaerosols at their source. They also have the opportunity to enter the airstream or recirculate during multiple passes.

Ultraviolet germicidal irradiation can effectively kill biological contaminants in an HVAC system. Within the entire spectrum of ultraviolet light, UV-c, the portion designated as the "c" bandwidth (200-290 nanometer wavelength) has been proven effective at killing contaminants by disrupting cellular growth.

By preventing biological accumulation, UV-c lamps prevent coil fouling and the resulting increase in pressure drop. This is an added energy-saving benefit.

Energy Recovery

Although many building owners, architects and engineers recognize the benefits of introducing more outdoor air into a conditioned space, many are concerned about the impact on equipment and operating costs. As a result, heat recovery technologies, which transfer heat between exhaust and ventilation airstreams, are becoming more and more popular. There are

ASHRAE Standard 62

Ventilation for Acceptable Indoor Air Quality:

- Defines the minimum outdoor air ventilation rates required to achieve acceptable indoor air quality
- Standard has been adopted by all building codes

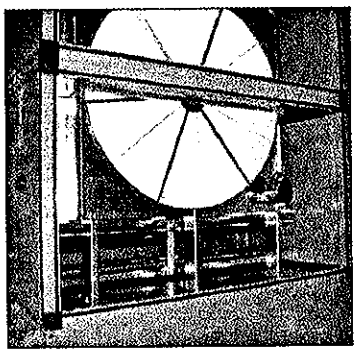


Figure 63

Why Energy Recovery?

two basic seasonal scenarios for air-to-air energy recovery in comfort applications. The first is to transfer heat and moisture from the exhaust stream to the supply stream during winter months. The second is the reverse function: to transfer heat and moisture from the supply stream to the exhaust stream in the summer.

Providing a comfortable and healthy indoor environment for building occupants is a primary concern for HVAC design engineers. ASHRAE (American Society of Heating, Refrigeration and Air-Conditioning Engineers) Standard 62, Ventilation for Acceptable Indoor Air Quality, contains ventilation design requirements for commercial and residential buildings. This standard, which is referenced in part or whole by most building codes in the United States, recommends outdoor air quantities per person to avoid adverse health effects. With half of all illnesses attributable to indoor airborne contaminants, the EPA (Environmental Protection Agency) has declared indoor-air quality a public health priority.

The latest revision of the ASHRAE standard for ventilation (62N) takes into account the non-people related indoor contaminants (off gassing from wallpaper, carpets, etc.) to come up with a blended rate of cfm per square foot and cfm per person. For example, if the non-people ventilation requirements for a certain space are 2 cfm per square foot for a 1000 square foot zone, the people-related ventilation requirements are 5 cfm per person, and the zone is designed for 100 people, the effective ventilation rate is 2 * 1000 = 2000 cfm (non-people) + 5 * 100 - 500 cfm (people) = 2500/10 = 25 cfm per person.

There are many possible applications of air-to-air energy recovery. Any building with comfort conditioning and relatively high levels of outdoor air, especially where latent cooling is a significant portion of the outdoor air requirement, is a likely application. Buildings with significant occupancy levels are usually good candidates. Typical applications may include schools and universities, gymnasiums, office buildings, hospitals, nursing homes, and retail shopping centers, to name a few.

There are also many heat recovery technologies available from which to choose. The most common types: coil energy recovery loop, energy recovery wheel, fixed-plate heat exchanger and heat pipe will be briefly described here.

It is important to note that ARI has established a rating and certification standard for air-to-air energy recovery ventilating equipment called ARI Standard 1060. For manufacturers who list their equipment under this standard, the designer and owner can have confidence and assurance that the manufacturers' performance data has been independently verified by ARI to be accurate and that it will perform as expected when installed. The scope of ARI 1060 includes energy wheels, fixed-plate heat recovery, and heat pipe equipment. For more information on energy recovery, refer to TDP-645, Energy Recovery Systems.

Coil Energy Recovery Loop

Often referred to as a runaround loop, a coil energy recovery loop is actually a heat recovery system and distinct from individual pieces of equipment like energy wheels or fixed-plate heat exchangers. Coil energy recovery loops are very flexible and well-suited to industrial applications or comfort applications with remote supply and exhaust ductwork. They use standard finned-tube coils to transfer heat to and from an intermediate working fluid such as water or antifreeze solution. A pump circulates the fluid between the two coils in closed loop. The loop must be equipped with a flow control valve to modulate the heat transfer rate. Also, an expansion tank is necessary

for thermal expansion and pressurization of the working fluid. The heat transfer rate must be controlled to prevent overheating or overcooling the supply airstream. If the outdoor airstream is very cold, an uncontrolled system may frost or freeze condensate on the coil in the exhaust stream.

Maintenance requirements

- Separate coils in exhaust and supply air
- Fluid pumped between coils
- Sensible heat transfer only
- Suited for applications where:
  - No cross-contamination of airstreams is allowed
  - Exhaust and supply airstreams are some distance apart
- Can use multiple coils

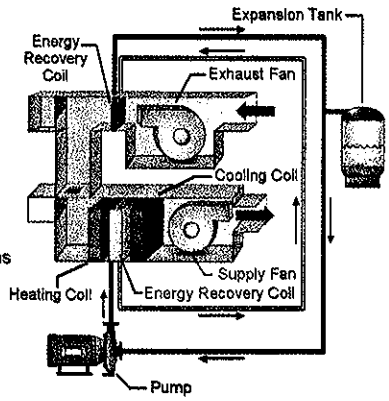


Figure 64

Coil Energy Recovery Loop Characteristics

By regulating the fluid flow through each coil and blending warm fluid with the cold fluid entering the exhaust coil, the total heat transfer rate is limited to the maximum possible without freezing. Overheating is prevented in the same manner. Runaround loops are well-suited for applications that must keep the two airstreams separate. They are useful for sensible heat recovery only. They can be designed to transfer heat in either direction. They are also uniquely capable of simultaneous heat transfer between multiple locations using the same circulating system.

Energy Recovery Wheels

An energy recovery wheel is also called a heat wheel and an enthalpy wheel. Although many people use the phrases interchangeably, the term heat wheel is used to distinguish a sensible-only application. Enthalpy wheel is used to distinguish a combined sensible and latent energy transfer application. In both cases, the fundamental piece of equipment is nearly identical. This discussion will use the term energy wheel as an all-inclusive class of rotary, air-to-air energy exchanger.

A rotary air-to-air energy exchanger has a revolving wheel filled with an air-permeable medium having a large internal surface area. The exchanger is designed to be positioned between two adjacent ducts with opposing flow directions. This establishes a counter-flow heat exchange pattern. The wheel rotates between 10 and 60 revolutions per minute depending on the application. When the wheel passes through the higher-temperature airstream, the media temperature increases as heat is transferred and stored in the individual filaments.

- Wheel rotates across exhaust and supply airstreams
- Can transfer both sensible and latent heat
- Effectiveness up to 80%
- Wheel is self cleaning

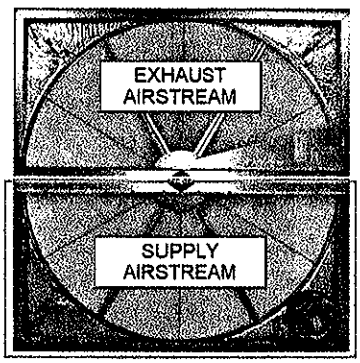


Figure 65

Energy Recovery Wheels

When the media wheel rotates into the low-temperature airstream, the filaments are cooled and release heat. This form of heat transfer is purely sensible and is driven by a temperature gradient between the high-and low-temperature airstreams. In most comfort applications, the temperature difference is relatively small as compared to some process energy recovery applications.

Latent heat transfer can occur in one of two ways. First, if the temperature and humidity gradients are sufficient, water may condense on the cold media in the high-humidity airstream. Moisture droplets held in the media are carried to the low-humidity airstream where they evaporate. The second and more reliable means of moisture transfer happens when the heat transfer media in the rotary wheel is coated with a desiccant film. In the high-humidity airstream, desiccant material adsorbs moisture molecules by the process of vapor diffusion. The molecules ride the rotating wheel and are released (desorbed) into the low-humidity airstream.

Although the overall process is considered energy transfer, the moisture absorption-desorption process described is actually a mass transfer event. It is driven by the water vapor partial-pressure gradient between the high-humidity and low-humidity airstreams. It is not dependent on the temperature gradient driving sensible heat transfer. This means that heat transfer can flow in one direction and moisture transfer can occur in the opposite direction because they are driven by different mechanisms.

Fixed-Plate Heat Exchanger

The heat transfer core of a fixed-plate heat exchanger is made from alternate layers of plates, formed and sealed at the edges to create two adjacent but separate airflow paths. Their most distinct advantage is that they have no moving parts. Energy transfer across the plates from one airstream to the next is completely passive and driven by the thermal gradient. Fixed-plate heat exchangers are useful for sensible heat transfer only. Latent heat transfer can occur only if the plates are made from water vapor-permeable material. Cross-flow is the most common arrangement. Counter-flow and parallel-flow exchangers have design and manufacturing complications that restrict their use. Units are available in many different materials, sizes and patterns, and are often modular, allowing combinations to fit large airflow applications. Plates are commonly spaced between 0.10 and 0.50 in. apart. Aluminum is the most commonly used material although plastics have been used without substantially affecting heat transfer efficiency. Airstreams are separated by folding, gluing or welding the plate edges and forming separate air paths. Spacing of the air paths is accomplished by molding integral dimples or ribbed patterns into the plates, or with external separators. Fixed-plate heat exchangers can be manufactured so there is little or no leakage between the airstreams. Most units are constructed with drains for removing condensate and wash water.

- Alternating layers of plates
- Form two separate but adjacent air passages
- Counter-flow or cross-flow designs
- Transfers heat between airstreams
- Generally sensible only (aluminum plates)
- Latent with vapor-permeable plates
- No moving parts

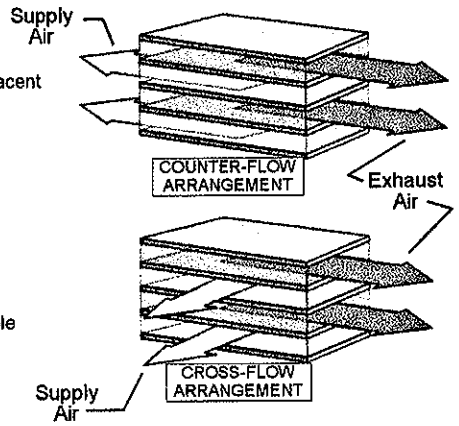


Figure 66

Fixed Plate Heat Exchanger Characteristics

Heat Pipe Heat Exchanger

A heat pipe heat exchanger has the appearance of an ordinary finned coil, but each successive tube is independent and not connected to any other tube. Each tube is built with an internal capillary wick material. The tube is evacuated, filled with a non-toxic refrigerant, such as R-134a, and individually sealed. With the tubes installed horizontally, one-half of the heat exchanger will act like an evaporator and the other half acts like a condenser. The high-temperature airstream passes through the evaporator half of the unit and the low-temperature airstream passes through the condenser half. The high-temperature airstream passes over one-half of all the tubes. As the working fluid (refrigerant) is warmed and vaporized in the evaporator half, the internal vapor pressure gradient drives the gas to the condenser end of the tube. In the condenser end, the fluid releases the latent energy of vaporization as it condenses, thereby warming the low-temperature airstream. Liquid refrigerant returns to the evaporator end through the internal wick.

- Finned-tube coil
- Tubes not connected
- Each tube charged with refrigerant
- Coil spans exhaust and ventilation airstreams
- Sensible heat transfer between air and coils
- Refrigerant in coils evaporates and condenses to transfer heat from one airstream to the other
- Works on vapor pressure gradient along tube
- Several arrangements available

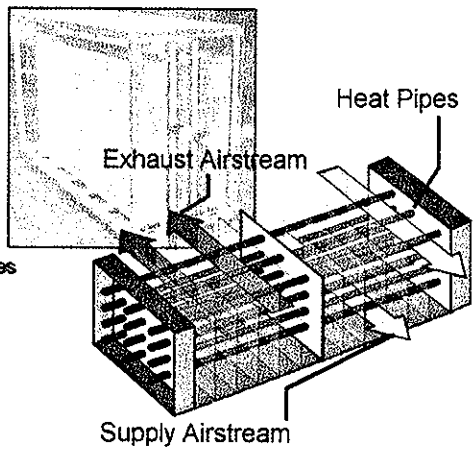


Figure 67

Heat Pipe Characteristics

Heat pipes are usually constructed of copper tubes with aluminum fins. Tubes are installed horizontally, but tilting the tubes one direction or the other can control the amount of heat transfer. For instance, operating the heat pipe unit with the evaporator end lower than the condenser end improves the liquid refrigerant flow back to the evaporator and increases heat transfer capacity. Reversing the arrangement retards liquid refrigerant flow and decreases capacity. Heat pipes are useful for sensible heat transfer only. Some latent benefit is achieved if a hot, humid outdoor airstream is cooled sufficiently to condense moisture on the evaporator end of the unit. Heat pipes can be installed so that cross-contamination between the two airstreams is near zero.

Additional Air Handler Configurations

Most of what we have discussed so far has been related to single duct, single discharge air systems. Let's explore when air handlers are used in other configurations.

Dual-Duct

Dual-duct is a blow-thru unit with a dual-coil section downstream of the fan. The coil section contains a cooling coil and a heating coil. The coils are installed in parallel with separate air outlets, one delivering cold air, the other hot or neutral air. The cooling and heating coils are commonly referred to as the cold deck and the hot deck, respectively. In a dual-duct system, separate hot and cold supply ducts are run throughout the building with hot and cold air being supplied to each zone. The hot and cold air is then mixed at each zone by a dual-duct mixing box to provide a volume of the proper temperature air to meet each zone's needs. Traditional dual-duct systems were constant volume. However, dual-duct systems are also used with variable volume systems.

- Blow-thru arrangement
- Normally a constant volume system
- Coil section with cooling and heating coils in parallel
  - Coils referred to as "cold " and "hot" decks, respectively
  - Provide simultaneous cooling and heating
  - May eliminate heating coil and use "neutral" air- becomes F&BP system
  - Separate cooling and heating supply air ducts throughout building
  - Zone mixing boxes blend air from both ducts to meet space requirements
- Fan sized for sum of zone peak cooling loads
- Cooling coil sized for block load (less than sum of peak loads)

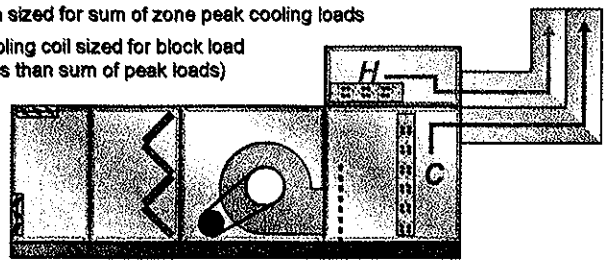


Figure 68

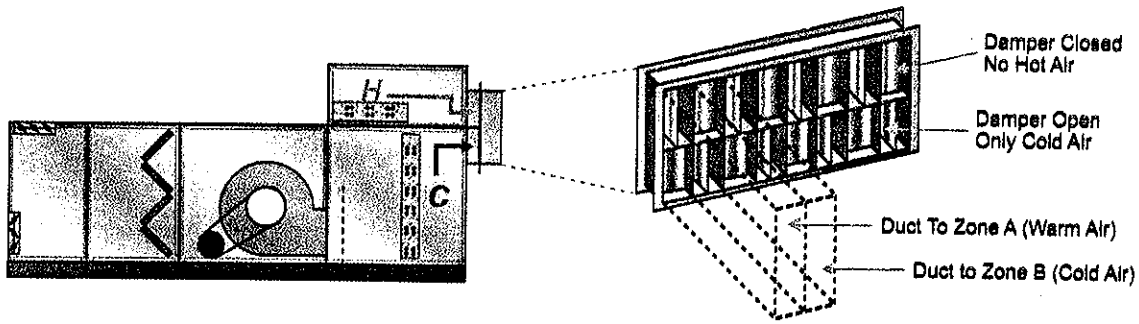
Dual Duct Air Handler

Multizone

A zoning damper section can be installed over the cold and hot deck duct openings of a dual-duct unit. This damper assembly is specially designed with multiple sets of dampers for use in a blow-thru configuration. The heating coil supplies warm air to the upper portion called the hot deck, while at the same time, the cooling coil supplies cool, dehumidified air to the lower portion called the cold deck. Zoning dampers in the heating and cooling decks are 90 degrees opposed to each other on a common shaft so that a nearly constant volume of air is supplied to each zone. The air temperature in each zone may be modulated from 100 percent heating capacity to 100 percent cooling capacity, based on the position of the zone damper. The number of zones available varies with unit size.

The zoning dampers are positioned by damper operators controlled from the various conditioned spaces. For example, shown is a multizone blow-thru unit, with discharge facing the reader. Note that some zones are calling for full cooling, some zones are being provided mixed air from the hot and cold decks for some intermediate load conditions, and some zones are providing heating. It should be noted that even though eight zones are shown here, two or more of the damper sections may be ganged together to provide from one to eight zones, depending on the job requirements and the corresponding air quantity requirement of each zone.

In many cases, the steam or hot water supply to the heating coil is shut off during summer operation (or any time that outside dry bulb temperature is greater than inside design dry bulb temperature), and control is achieved by using the heating section and dampers to bypass the cooling coil at partial load condition. In this mode the blow-thru unit is actually functioning as a zoning face-and-bypass system.



Constant Volume, Blow-thru variation of Dual Duct System

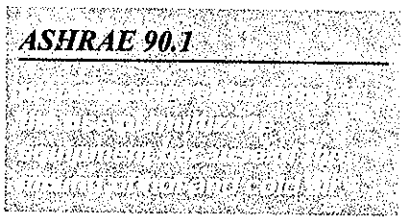
- Zoning dampers mounted on dual coil section in lieu of zone mixing boxes
  - Opposed dampers mix air from cold and hot decks into a common duct
    - When cold deck damper open-hot deck damper is closed to maintain constant volume
  - Each set of dampers supplies a separate zone in building with up to 18 zones/unit
  - Dampers controlled by space temperature
  - Separate supply air ducts to each zone

Figure 69

Multizone Damper Arrangement

The air quantity handled by the fan is essentially constant at all times. The quantity of air handled by the fan is the sum of the dehumidified air quantities required by each zone at the time of its peak load and is called the total air quantity.

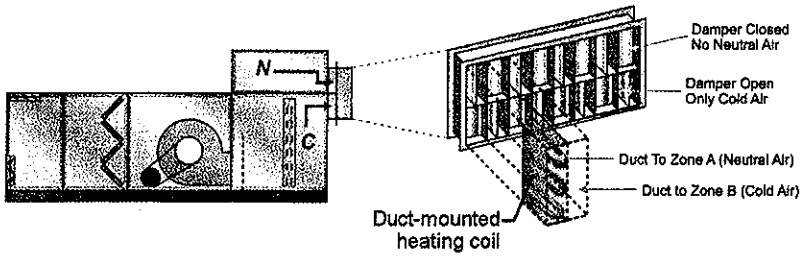
On the other hand, the air passing over the cooling coil varies in quantity and the maximum is established by the instantaneous peak building load or block load of the area served by the multizone unit. This is the dehumidified air quantity and is generally less than the total air quantity handled by the fan.



Therefore, it is important in selecting the blow-thru multizone unit that the dehumidified air quantity is used in determining coil performance and the total air quantity is used in the selection of the fan.

Texas Multizone

Another less common variation of the standard multizone system is called the Texas multizone, shown in Figure 70. The main difference between this and a conventional multizone system is that the air handler will have only cold and neutral decks, no heating. Zone dampers on the air handler will mix cooled and neutral air to maintain space requirements. Any heating is accomplished by individual heating coils located in each zone.



Multizone Variation

- AHU has cold deck and neutral deck-no heating
- AHU zoning dampers mix cold and neutral air to zones
- Heating coils located in separate zones as required
- Sometimes referred to as "Texas" multizone

Figure 70

Texas Multizone Damper Arrangement

Triple-Deck Multizone

In an effort to improve the energy efficiency of the multizone system, a variation called the "triple-deck" multizone has been developed. A triple deck system, as its name implies, adds a third "neutral" deck between the cold and hot decks, eliminating the mixing of cooled and heated air. The neutral deck contains no heating or cooling coil and simply allows a passage for unconditioned air to bypass the cold and hot decks. The zone dampers then mix this "neutral" air with air from either the hot or cold decks, depending upon the needs of each zone.

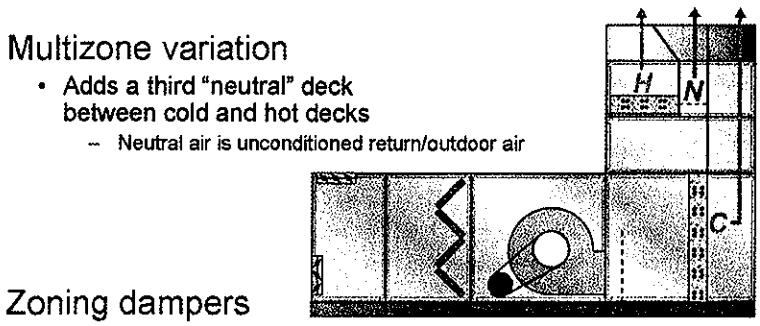


Figure 71

Triple Deck Multizone

Air Handler Selection Example

As mentioned previously, most manufacturers offer selection software for their standard central station product lines to facilitate layout, selection and ordering of the air handler.

The engineer will generally provide a list of the major components and performance required, based on the application and load requirements.

Let's take a look at a typical example. For purposes of this example, we will use the Carrier Air Handler Builder (*AHU*Builder) program, which uses a graphical interface.

Assume that the application requires the air handler to provide the following functions:

- mixing of outdoor and return air
- filter section with MERV 7 efficiency media
- heating section with hot water
- cooling section with chilled water
- fan with constant volume to handle 5000 cfm at 2 in. wg external static pressure

Let's take a look at a typical example.

Step 1:

Select unit size from menu or with sizing tool as shown in Figure 72. Size is based on coil face area.

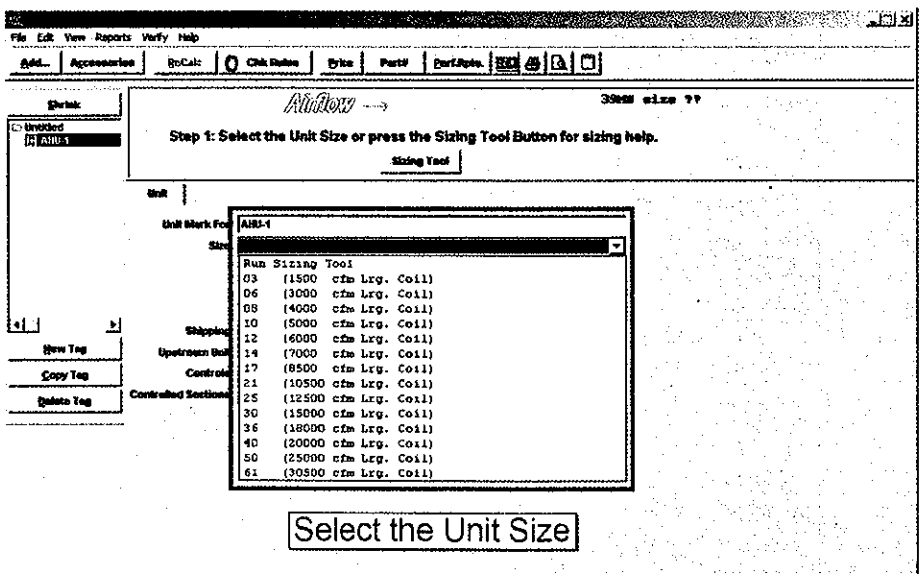


Figure 72

Unit Size Selection

Step 2:

Select the sections required from the list of available sections to meet the job requirements. For this job, we will select mixing box, angle filter, cooling coil, heating coil, and fan sections.

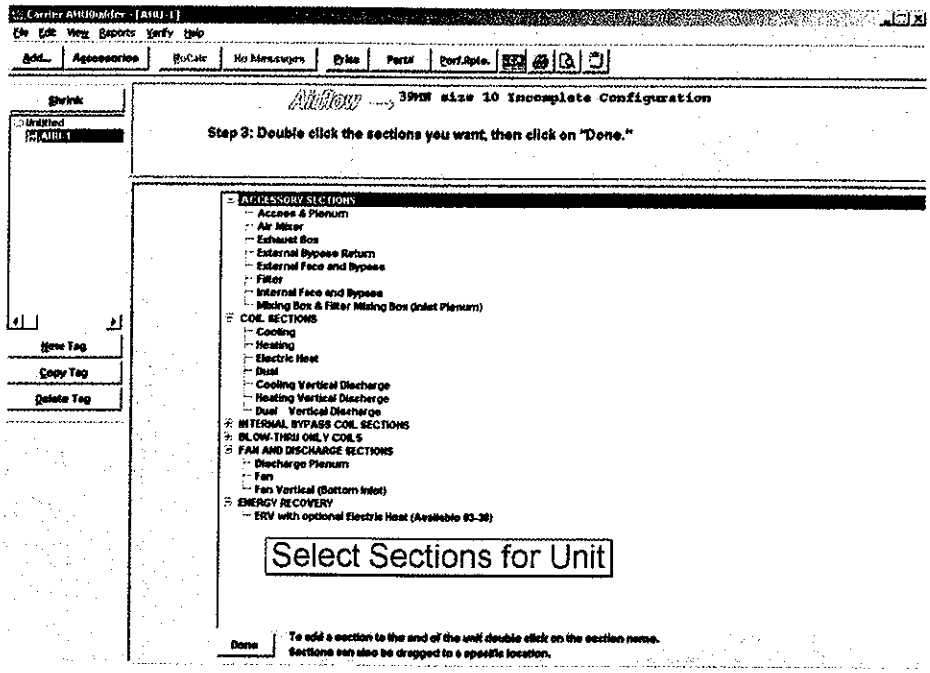


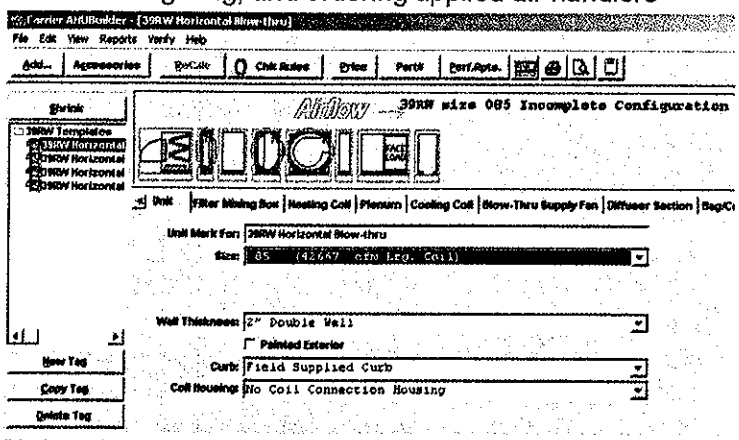
Figure 73

Section Selection

Step 3:

Once sections are selected, complete the configuration details for each section. Specific coil and fans and coils can be evaluated and selected during this process. The program contains configuration rules that will help prevent the user from improperly configuring sections, and offer hints on how to correct errors.

AHUBuilder is a program for selecting, configuring, and ordering applied air handlers



- Builds the desired unit section by section through the use of a graphical icon-driven display
- Contains configuration rules to ensure functional units are constructed
- Engineering selection and performance routines are available for selecting coils and fans based on performance requirements

Figure 74

Configuration Rules

Once a selection is complete, the program will print submittal data, including performance reports, schedules and certified drawings. Figure 75 shows a certified drawing and Figure 76 shows a performance report. The program also allows drawings and schedules to be created in .dxf file format that can be imported directly into engineering CAD drawings.

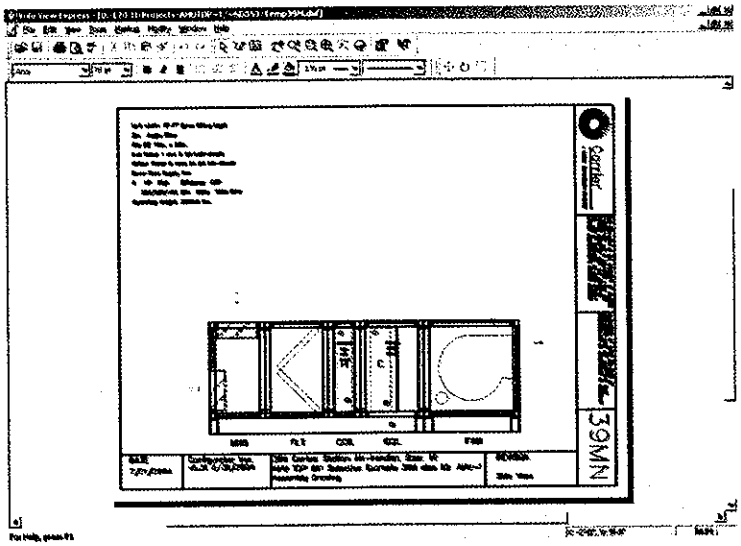
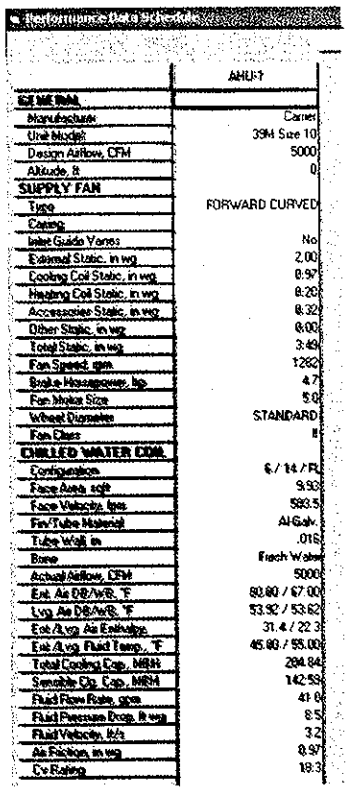


Figure 75

Sample Submittal Drawing



	AHU-1
GENERAL	
Manufacturer	Carrier
Unit Model	39M Size 10
Design Airflow, CFM	5000
Altitude, ft	0
SUPPLY FAN	
Type	FORWARD CURVED
Casing	
Inlet Guide Vanes	No
External Static, in wg	2.00
Cooling Coil Static, in wg	0.97
Heating Coil Static, in wg	0.20
Accessories Static, in wg	0.32
Other Static, in wg	0.00
Total Static, in wg	3.49
Fan Speed, rpm	1282
Brake Horsepower, bp	4.7
Fan Motor Size	5.0
Wheel Diameter	STANDARD
Fan Class	II
CHILLED WATER COIL	
Configuration	6 / 14 / FL
Face Area, sqft	9.93
Face Velocity, fpm	503.5
Fin/Tube Material	Al/Galv.
Tube Wall, in	.016
Rows	Fresh Water
Actual Airflow, CFM	5000
Ent. Air DB/WB, °F	80.80 / 67.00
Lvg. Air DB/WB, °F	53.92 / 53.82
Ent./Lvg. Air Enthalpy	31.4 / 22.3
Ent./Lvg. Fluid Temp., °F	45.00 / 55.00
Total Cooling Cap., MBH	284.84
Sensible Cooling Cap., MBH	142.55
Fluid Flow Rate, gpm	41.0
Fluid Pressure Drop, ft wg	8.5
Fluid Velocity, ft/s	3.2
Air Friction, in wg	0.97
Coil Rating	18.3

Figure 76

Performance Report

Summary

In this presentation on central station air-handling equipment, unit components have been discussed, as well as their arrangement and the accessories that are normally available for these units. Coil performance and some guidelines and recommendations on the application of air-handling units were also presented.

The types, performance and characteristics of fans types were explained. We learned one should not generalize about advantages and disadvantages of one type fan versus another type. Only by analyzing each situation can the designer determine which fan is the most advantageous.

IAQ requirements have affected the design features on modern central station air handlers. Many of the prominent design features discussed in this TDP, like double-wall construction, sloped drain pans, and low leak casings and dampers are all directed at upgrading the IAQ capabilities of the units.

Work Session

Questions may have more than one correct answer. Include all correct answers in your response to the questions below.

1. List the seven functions of a central station air handler as defined by ARI.

   _____ _____
   _____ _____
   _____ _____
   _____

2. True or False? A packaged air handler offers a wider size range with more features and flexibility than a central station air handler. _____

3. The size of a central station air handler is usually based on _____

   a) nominal cooling capacity.
   b) nominal heating capacity.
   c) the airflow through the unit.
   d) cooling coil face area.

4. Name the two most common types of casing construction used in central station air handlers.

   _____
   _____

5. The primary reason for the recent market trend toward double-wall air handlers is: _____

   a) double wall is lower cost than single wall.
   b) increased concern over indoor air quality.
   c) double wall air handlers are perceived as higher quality.
   d) double wall allows the use of anti-microbial coatings.

6. Which of the following features is NOT generally found on an air handler designed for outdoor installation? _____

   a) Sloped roof with standing seams
   b) Air inlet hood or louvers
   c) High-quality gaskets and seals
   d) Single wall, modular construction.

7. True or False? In a draw-thru air handler, the cooling coil is located on the upstream or inlet side of the fan. _____

8. True or False? Forward-curved fans are designed to operate at higher static pressures than airfoil fans. _____

9. Which of the following statements are true regarding AMCA fan class: _____

   a) A Class I fan may be operated under Class II or Class III performance requirements.

   b) A Class II fan may be operated under Class III performance requirements.

   c) A Class III fan may be operated under Class II performance requirements.

   d) All of the above.

10. Under identical operating conditions, _____.

   a) a fan bearing with an L_{50} life of 200,000 hours is equal to a bearing with an L_{10} life of 40,000 hours.

   b) a fan bearing with an L_{10} life of 500,000 hours is equal to a bearing with an L_{10} life of 40,000 hours.

   c) the L_{50} life represents the period of time all bearings of that type or designed will survive without fatigue failure.

   d) None of the above.

11. Which of the following air handler design features would more likely be found on a fully custom outdoor air handler than a standard central station AHU? _____

   a) Sloped roof with standing seams

   b) Outdoor intake hoods

   c) Service and maintenance vestibules

   d) G-90 galvanized casing panels

12. True or False? A filter with a MERV 15 rating is likely to be a 2-inch deep flat panel filter. _____

13. In a variable air volume (VAV) system, which type of fan wheel could be operated without fan volume control (allowed to ride the fan curve)? _____

   a) Forward-curved

   b) Backward-inclined

   c) Airfoil

   d) Plenum

14. Which types of energy recovery devices commonly provide both sensible and latent heat recovery? _____

   a) Energy recovery wheels

   b) Heat pipes

   c) Run-around loops

   d) Fixed-plate heat exchangers

15. True or False? A dual-duct unit is a specific blow-thru configuration that can deliver cold and neutral or hot air simultaneously. _____

16. True or False? In a multizone air handler, the fan and cooling coil are always selected based on the instantaneous "block" load of the area served by the air handler.

_____

17. The best placement of an ultraviolet light germicidal lamp (UV-c) within an air handler is immediately _____.

a) downstream of a cooling coil.
b) upstream of a filter section.
c) downstream of a heating coil.
d) downstream of a mixing box.

18. Face and bypass dampers: _____.

a) may be used as a means of capacity control for both heating and chilled water coils.
b) minimize the possibility of coil freeze-up in cold climates.
c) are not recommended with direct expansion cooling coils.
d) All of the above.

19. True or False? Variable frequency drives (VFDs) have replaced inlet guide vanes (IGVs) as the most common method of fan volume control. _____

20. For a coil constructed of the same materials and face area, rank the following factors in order of their impact on cost, from highest to lowest: fin spacing, circuiting, rows.

_____

_____

_____

Notes

Appendix

Work Session Answers

1. circulating, cleaning, heating, cooling, humidifying, dehumidifying, and mixing air
2. False
3. c and/or d
4. post and panel, structural panel
5. b
6. d
7. True
8. False
9. c
10. d
11. c
12. False
13. a
14. a
15. True
16. False
17. a
18. d
19. True
20. rows, fin spacing, circuiting



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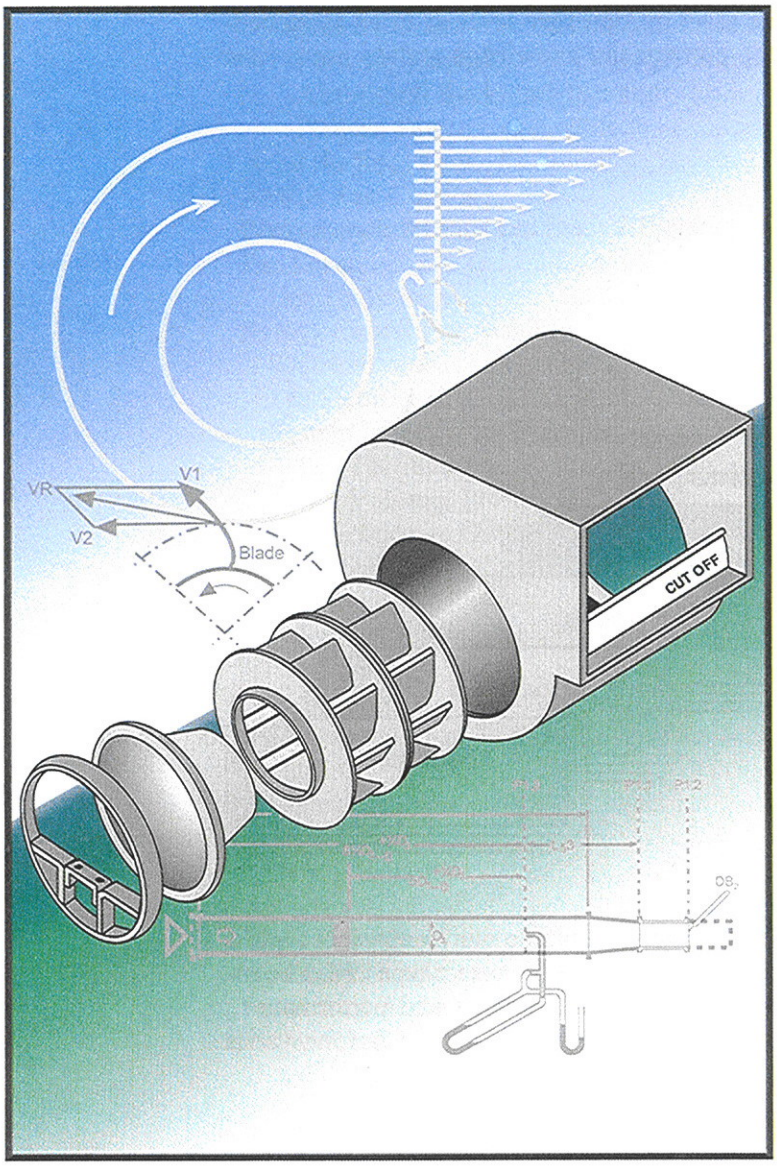


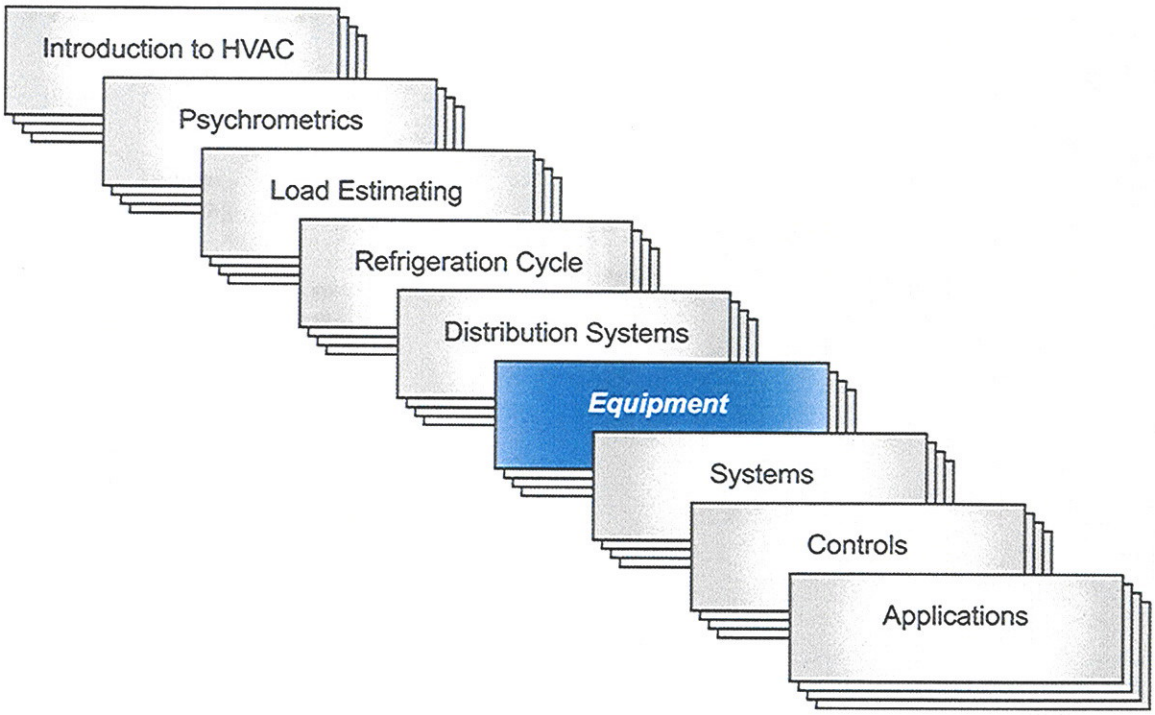
COMMERCIAL HVAC AIR-HANDLING EQUIPMENT

Fans: Features and Analysis

Technical Development Program

Technical Development Programs (TDP) are modules of technical training on HVAC theory, system design, equipment selection and application topics. They are targeted at engineers and designers who wish to develop their knowledge in this field to effectively design, specify, sell or apply HVAC equipment in commercial applications.

Although TDP topics have been developed as stand-alone modules, there are logical groupings of topics. The modules within each group begin at an introductory level and progress to advanced levels. The breadth of this offering allows for customization into a complete HVAC curriculum – from a complete HVAC design course at an introductory-level or to an advanced-level design course. Advanced-level modules assume prerequisite knowledge and do not review basic concepts.



The heart of any air-handling system is the fan. Fans may consume more energy in a typical HVAC system than the compressors! It is extremely important that the correct type of fan be chosen for the application. This TDP module will describe fan characteristics and performance in order to provide designers with the knowledge to select and apply the proper fan for various HVAC situations.

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Introduction

In the HVAC industry, the fan is one of the most important components in the heating and cooling system. It is also one of the easiest components to misapply because of all the types and arrangements available. Fans are important because they can consume more energy than the air conditioning compressors in a building.

The fan itself consists of a rotating impeller and a fan scroll housing to collect and direct the airflow in the direction desired. A fan operates on the same basic principle as a centrifugal pump, converting rotational mechanical energy into fluid or air energy. The energy created by the fan is determined by the total pressure increase (velocity pressure + static pressure) of the air passing through the fan.

The fan industry is based on technology that is, for the most part, not new. The basic fluid mechanics governing fan aerodynamic design and performance have been well known for decades. Standards for the construction, testing, and performance rating of fans are well established and strictly adhered to by most fan manufacturers. Because of this, fans are often treated like commodities instead of important pieces of HVAC equipment that should be carefully and thoughtfully selected.

Centrifugal fans are the most widely used type of fan in the HVAC industry. For that reason, this TDP module is geared primarily towards centrifugal fans.

We will examine fan construction, types of fans, the fan laws that govern centrifugal fan performance, stability factors, and the effects of field application of fans (system effect).

At the end of this TDP, the reader should have an understanding of the technical issues involved to properly select and apply the correct fan for a commercial HVAC system.

Fan Types

A fan is a device used to produce a flow of air. Fans are classified into two general types, centrifugal and axial.

Centrifugal Fans

Centrifugal fans are classified according to impeller (wheel) blade design. The most commonly used impeller designs for centrifugal fans for comfort air conditioning are forward- curved, backward-inclined, and airfoil. Impellers and their applications will be covered in this TDP module.

The air is drawn in through one or both sides of the centrifugal fan impeller and is discharged at a right angle to the fan shaft. A centrifugal fan impeller is usually enclosed in a housing also called a scroll. The air is discharged from the impeller through the outlet in the fan housing. When this housing is mounted inside an insulated cabinet, it comprises the fan section of an air handler. Refer to TDP-611, Central Station Air Handlers for further information.

Air is discharged at a right angle to fan shaft

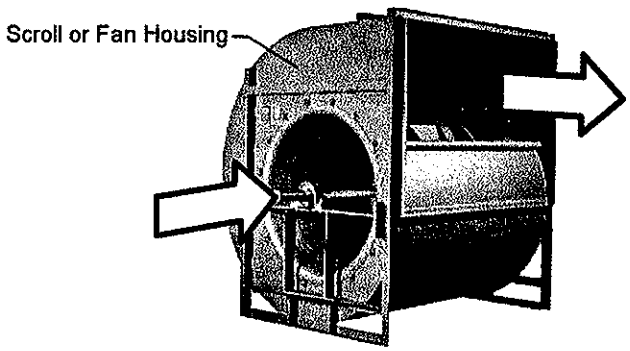


Figure 1

Centrifugal Fan Configuration

Plenum Fans

When centrifugal airfoil impeller is applied without the housing, and is located inside a cabinet, it is called a plenum fan. Plenum fans will be covered also in this TDP module.

Single-width, single-inlet airfoil impeller design, for mounting inside a cabinet

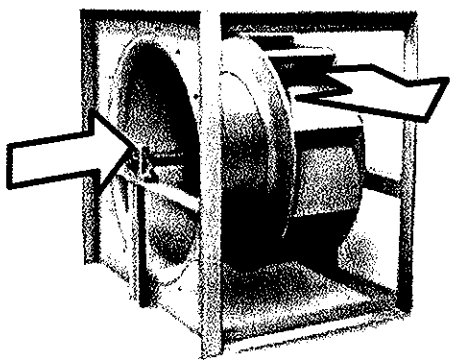


Figure 2

Plenum Fan Configuration

Axial Fans (In-line)

In an axial fan, air flows and is discharged parallel to the fan shaft, not at right angles to the fan shaft as with a centrifugal. Axial fans are classified as propeller, tube axial, and vane axial. These fans (with the exception of the propeller) have a tubular configuration, hence the term "in-line." Vane, or tube axial, fans can be driven with an internal direct connected motor or an external shell mounted motor.

There are several variations on an axial or in-line fan that we will cover in this TDP module. The first utilizes a centrifugal impeller in an in-line cylindrical tube configuration. Air is discharged from the impeller and turns 90 degrees in the shell before flowing through straightening vanes.

The second is a hybrid between a centrifugal and an axial. It is called a mixed flow fan. Air is discharged off a centrifugal type impeller that has angled blades. The air then exits the cylindrical tube that houses the fan.

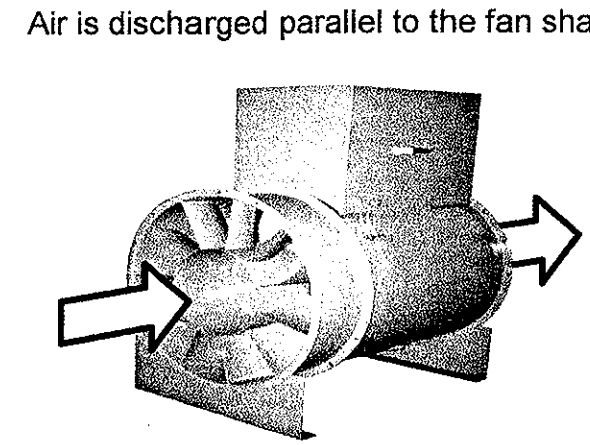


Figure 3

Axial Fan Configuration
Photo courtesy of Barry Blower

Centrifugal Fans

Shown here are the components of a double-width double-inlet (DWDI) fan assembly. This is essentially two single-width fans, side by side, with two inlets and a single outlet or discharge with no partition in the scroll housing. A single-width single-inlet fan (SWSI) would have a single inlet and take up less space from a width standpoint, but would need to be of greater diameter than the DWDI to move the same volume of airflow. SWSI fans are often applied where it is necessary to mount the fan motor out of the air stream, for example corrosive air. DWDI designs are more common in HVAC equipment.

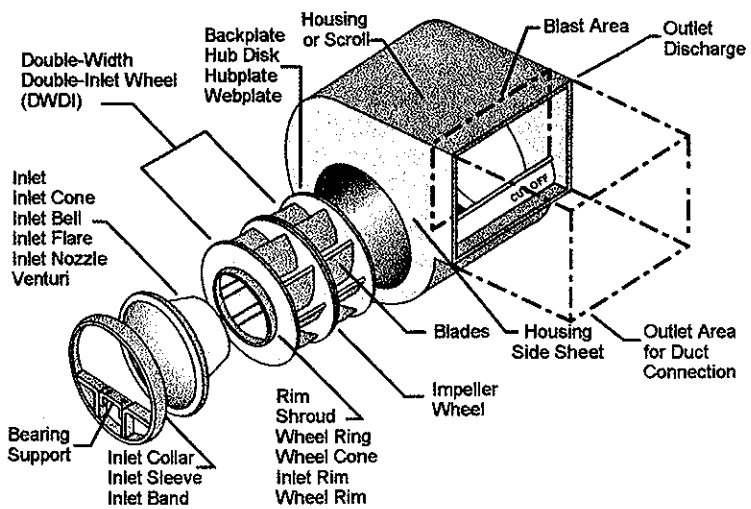


Figure 4

Centrifugal Fan Construction and Terminology (DWDI Fan)

The following are some of the basic components that make up the fan assembly:

- Bearing support – supports the fan shaft bearing on both inlet sides of the scroll housing
- Inlet collar – attaches the bearing support to the fan housing
- Inlet cone – an aerodynamic inlet design used to reduce entrance losses of the fan (used on backward inclined and airfoil fans)
- Impeller – the round assembly containing multiple fan blades that is attached (keyed) to the fan shaft. The impeller (also called the wheel) spins to move the air from inlet to discharge.
- Fan blades – parts of the impeller that are mounted to the hub that force the air to move; the types of blades define the fan capability and application
- Wheel backplate or hub – supports the impeller blades and allows the fan wheel assembly to mount to the fan shaft (not shown)
- Fan shaft – the round piece of precision-ground steel that the wheel is fastened to that in turn is driven by the fan motor normally through pulleys to spin the wheel. The fan shaft may also be direct coupled to the motor. This is called direct drive.
- Fan scroll housing – the fan scroll housing is the sheet metal wrapper that leads the air-stream from the fan wheel inlet to the discharge outlet
- Cut off – a plate that is positioned under the blast area that is designed to give the fan the desired discharge characteristics and performance
- Blast area – the open discharge area of the fan scroll housing, which is above the cutoff
- Fan outlet – the part of the fan scroll housing that will connect to the discharge ductwork

Impeller Design

Shown here is a radial fan impeller (wheel) with straight blades. We have chosen to show this straight impeller design as a way to examine the vectors related to the fan blades. This straight blade designed centrifugal impeller is often used for material handling in industrial plants. Later we will show the curved impeller designs and vector characteristics for the centrifugal fan types used in HVAC applications. In this and other blade vector diagrams we will show later, V1 represents the radial velocity component leaving the wheel, V2 represents the tangential velocity leaving the wheel. V2 is equal to the tip speed of the blade. VR is the velocity resulting from the V1 and V2 vectors and is the velocity relative to the fan scroll housing. The relative length of VR is a function of the blade design and the tip speed

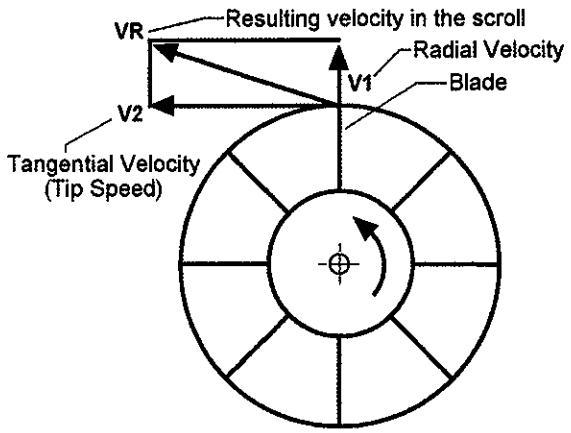


Figure 5

Impeller Velocity Vectors

Tip speed is a function of fan RPM. Certain impeller designs can be spun at lower speed than others to produce the same airflow. As an example, as we will see, a forward-curved impeller produces a large VR relative to tip speed, versus an airfoil. So a fan with a forward-curved impeller can be operated at a lower rpm than the airfoil.

For all fans, the impeller type used develops a total pressure difference over the inlet and outlet air streams. The total pressure (P_T) rise comprises two main components. The first is static pressure (P_S), which depends on the blade profile, number of blades, pitch (angle), and other aerodynamic characteristics of the fan impeller. The second component is velocity (dynamic) pressure (P_V), which develops due to velocity or kinetic energy imparted to the air stream.

Static pressure is the "bursting" pressure in all directions in the ductwork created by the fan. Velocity pressure is the pressure in the direction of airflow.

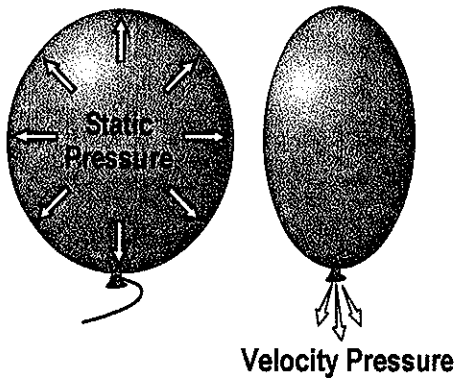


Figure 6

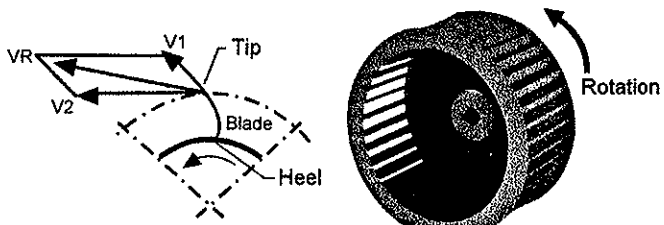
Static and Velocity Pressures

Forward-Curved

On a forward-curved centrifugal fan, the impeller blades are curved as can be seen here. The air leaves the wheel (VR) at a velocity greater than the tip speed (V2) of the blades. Tip speed is a function of wheel rpm. Since this impeller blade design results in such a large VR, the wheel rpm can be reduced and still produce a comparable airflow to other blade designs. Airfoil and backward inclined, which we will discuss, must be rotated at higher speed. At a given airflow capacity, the forward-curved fan impeller can often utilize a smaller diameter wheel.

Speed

Because the forward-curved fan can be rotated at slower speeds and is used for lower static pressures, it is a lightweight design and is therefore less expensive. The fan wheel has 24 to 64 shallow blades with both the heel and the tip of the blade curved forward. This fan is used primarily for low-pressure HVAC applications. Forward-curved fans are best applied operating at static pressures up to 5.0 in. wg.



Characteristics:

- Most commonly used wheel in HVAC
- Light weight – low cost
- Operates at static pressures up to 5 in. wg max
- 24 to 64 blades
- Low rpm (800 to 1200 rpm)

Note

Figure 7

Forward-Curved Wheel Design

Forward-curved wheel designs, like all centrifugal fan wheels, should be used in clean environments. Operating in dusty or dirty environments could result in an unbalanced fan wheel.

Forward-curved centrifugal fans have an overloading horsepower characteristic as the airflow through the fan increases at a constant rpm. This is why forward-curved centrifugal fans are called overloading type fans.

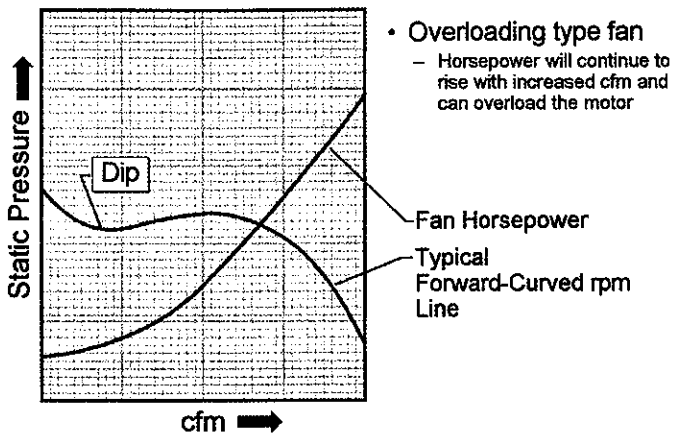


Figure 8

Forward-Curved Centrifugal Fan Characteristics

A typical example of an overloading situation is where a forward-curved centrifugal fan is used for temporary heat duty in an unfinished building. If the ductwork is not completed, the resistance of the duct system may be lower than design, and the fan can deliver more air than required and may eventually overload the motor.

It may be noted that the static pressure-cfm curve of a fan using a forward-curved wheel has a somewhat gradual slope and also contains a "dip." That is how you can recognize a forward-curved application, versus an airfoil or backward-inclined impeller application, which will have a steeper slope and no dip. The dip in the curve of the forward-curved centrifugal fan is to the left of peak pressure. When making fan selection with a forward-curved centrifugal fan, it should be made to the right of the dip to avoid unstable fan operation.

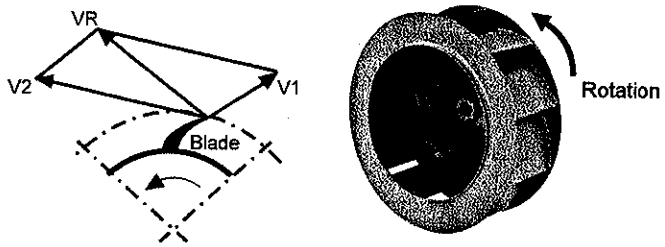
Centrifugal Forward-Curved Housing

The housing is an aerodynamic scroll configuration, which promotes the conversion of velocity pressure from the impeller to static pressure for the duct system. The fan housing width will vary based on whether or not the fan wheel inside is a single width single inlet, or double width double inlet type. With forward-curved fans, the scroll design is critical for the conversion of velocity pressure to static pressure and the inlet design is of secondary importance.

Airfoil and Backward-Inclined

The airfoil impeller is shown below. The airfoil blades have a cross section similar to an airplane wing. Airfoil blades have a thickness that forward-curved and backward-inclined blades do not. A backward-inclined impeller is a thinner (single thickness) bladed airfoil and has an efficiency only slightly less than an airfoil. A backward inclined (BI) impeller will have single thickness blades that are inclined away from the direction of rotation. Fans with airfoil and backward-inclined impellers have the highest efficiency of all centrifugal fans.

Each airfoil and backward inclined impeller uses approximately 8 to 18 blades inclined backward from the direction of rotation. Because of this, the air leaves the wheel (VR) at a velocity less than the blade tip speed (V2). For a given duty, fans with these impellers will have the highest wheel speed. Fans with airfoil impellers are designed to operate, depending on the fan



Characteristics:

- Blades are curved away from direction of rotation
- Static pressure up to 10 in. wg
- 8 to 18 blades
- High rpm (1500 to 3000 rpm)

Figure 9

Airfoil Wheel Design

Backward-inclined and airfoil fan wheels are considered "non-overloading" because they have the characteristic of almost constant power consumption for the same operating speed (rpm). Some engineers like to use airfoil instead of forward-curved centrifugal fans (when the choice exists) for that reason, even though they cost more than forward-curved fans. In those areas of applications where either type of fan could be used, it is prudent to make both selections and compare.

size and the manufacturer, at static pressures up to 10 in. wg or higher. Fans with airfoil impellers are not typically used at the static pressures where forward-curved centrifugal fans are the best choice such as less than approximately 5 in wg.

Typically, fans with airfoil impellers are used primarily in large air handlers for systems having relatively high static pressure requirements. Since they are capable of higher static pressures and operate at higher speeds, they are more ruggedly built, which adds to their cost and weight.

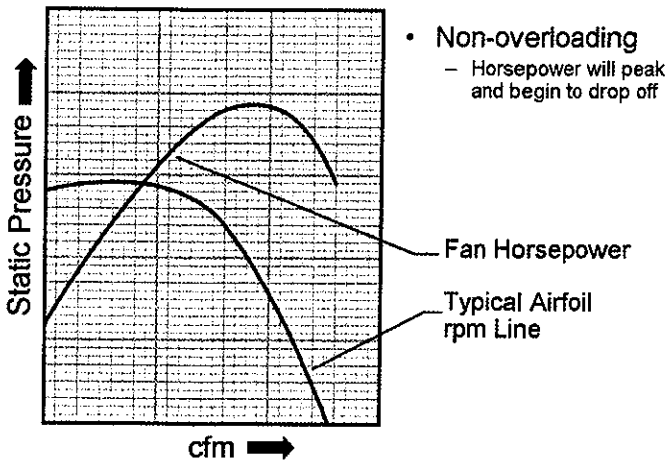


Figure 10

Airfoil Centrifugal Fan Characteristics

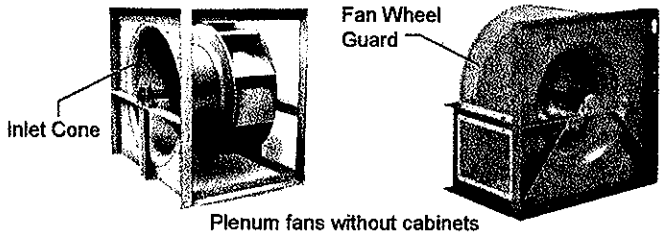
Centrifugal airfoil and backward-inclined housing

The housing design for an airfoil and backward inclined centrifugal fan is similar to the housing for a forward-curved. However it is more critical to maintain close clearance and alignment between the impeller and the inlet in order to maintain the high efficiency.

Plenum Fan

Plenum fans use non-overloading, single-width single inlet (SWSI) centrifugal airfoil impeller designs constructed of heavy gauge steel with each blade continuously welded to the wheel cone. The fan and its motor operate un-housed within a pressurized plenum or cabinet. When this type of fan utilizes a motor external to the plenum, it is called a plug fan. In a central station air handler, the plenum is the unit casing provided by the manufacturer. Ductwork is connected directly to the plenum without an intermediate transition. In essence, plenum fans use their plenum enclosure as a fan scroll.

Plenum fans do not discharge air directly off their impeller and into a discharge duct. The fan pressurizes the plenum it is located in and air is discharged out of the various openings, which are typically field cut into the plenum. For this reason, fan discharge noise is absorbed in the plenum cabinet. This makes the plenum fan ideal for acoustically sensitive fan applications.



Characteristics:

- Single-Width, Single-Inlet (SWSI)
- Operate at static pressures up to 10 in. wg
- Best application with limited space or when multiple duct discharge is desired

Figure 11

Plenum Fan Characteristics
Courtesy of Barry Blower

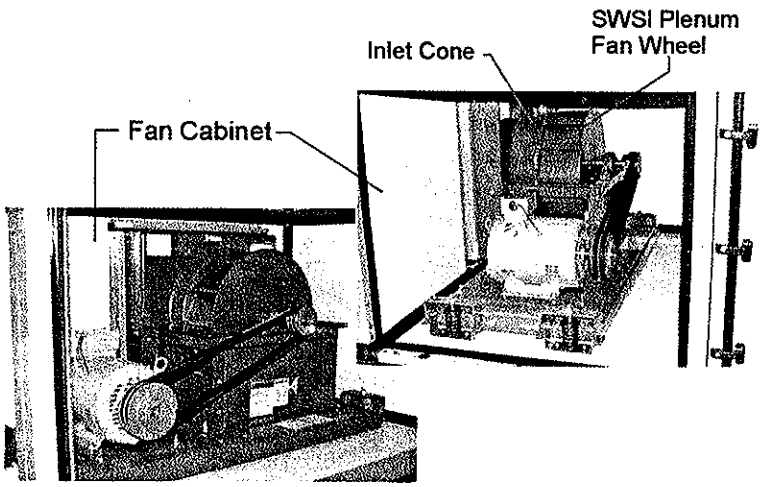


Figure 12

Plenum Fans with Cabinets

Notice the developed inlet cone design to the single inlet airfoil wheel. This allows the fan to efficiently develop static pressure within the wheel.

An important reason that makes plenum fans so popular is that they allow for flexibility in discharge arrangements. The plenum fan may also reduce the space required in the mechanical room for the air-handling unit and the discharge ductwork.

Plenum fans

Axial (In-line) Fans

Axial (also called in-line) fans are often used for high cfm, low to medium-static applications. The design of the in-line fan allows for direct connection to supply or return ductwork, which can save space in the mechanical room. Axial fans are often applied as return fans as part of a supply-return fan system. They are also used for exhaust air applications and can even be fitted into factory fabricated air-handling units for supply duty.

- **Use for high cfm applications**
- **In-line space savers with no cabinet**
- **Often used in industrial AC and ventilation applications**
- **Impeller similar to prop fans but blades are more aerodynamic**
- **Often used for return fans in AC applications**

One major difference from centrifugal fans is that air is discharged parallel to the shaft on an axial fan.

Propeller fans are a type of axial fan that is not typically ducted. They are used for moving high volumes of air at very low static pressures. Propeller fans operate at low rpm and are an inexpensive design.

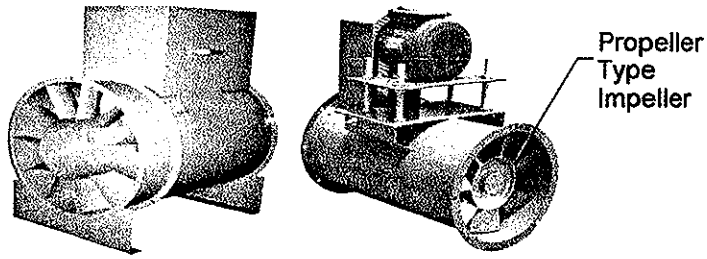
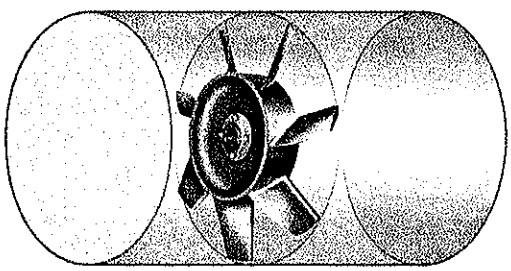


Figure 13

Axial (In-line) Fans
Photo courtesy of Barry Blower

Tube axial fans use a fan design with a propeller type impeller (but with a more aerodynamic configuration) inside a cylindrical tube. They may come with a sound attenuating accessory to help reduce noise levels. Tube axial fans offer a greater efficiency than propeller fans and can be ducted.

Vane axial fan designs are similar to tube axial but incorporate guide (straightening) vanes on the discharge to help redirect the air and improve efficiency. Some vane axial fans have a moveable impeller blade capability. The pitch or angle of the blades can be varied based upon the static pressure and airflow required. The blade angle can be changed manually or automatically.



- **Axial Wheel**
  - **–Air discharged parallel to the shaft**
  - **–Air is often redirected via straightening vanes making the fan a vane axial**

Figure 14

Axial Impeller Design
Photo Courtesy of Barry Blower

The impeller design of an axial fan wheel is similar to a propeller except that the blades are more aerodynamic.

Axial fans are often referred to as in-line or tubular fans. However, not all in-line (or tubular) fans use conventional axial designed impellers.

For instance, another version of an in-line fan actually uses a centrifugal impeller. It is called a tubular centrifugal. Even though this fan uses a centrifugal impeller, its overall tubular configuration resembles that of an axial so we have placed it in the axial section of this TDP module.

A centrifugal impeller is mounted in an in-line (tubular) housing and air is redirected out via straightening vanes just as in a vane axial. A tubular centrifugal takes advantage of the efficiencies of a centrifugal impeller and the space-saving configuration of an in-line design.

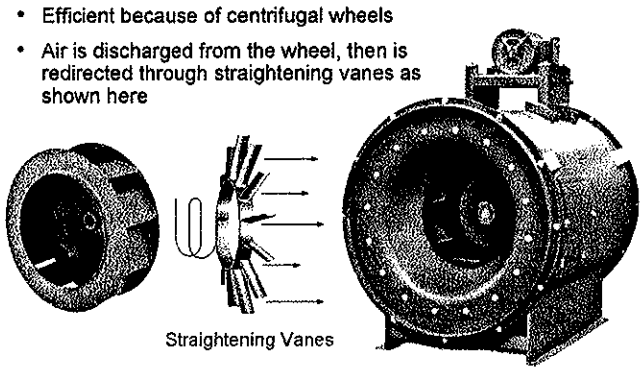


Figure 15

Tubular Centrifugal In-Line Fan
Photo Courtesy of Barry Blower

Mixed Flow Fan

Mixed flow fans can be used for return air, supply air, or general ventilation applications where low sound level and good efficiencies are important.

The mixed flow wheel design combines the working properties of both axial fans and tubular centrifugal fans. Mixed flow fans draw the air in and exhaust it in a more linear fashion, resulting in a more efficient system, which, in turn, reduces motor horsepower requirements.

Another advantage to the mixed flow design is the reduced sound. Mixed flow fans run at lower rpm to deliver the same amount of airflow. As a result of slower wheel speed, sound generation is reduced significantly.

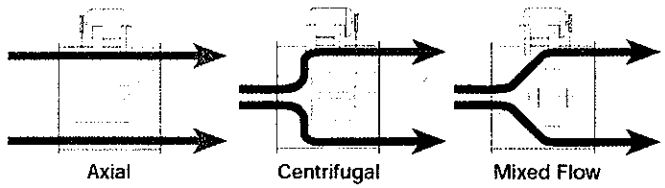


Figure 16

In-line Fan Types
Photo Courtesy of Greenheck.

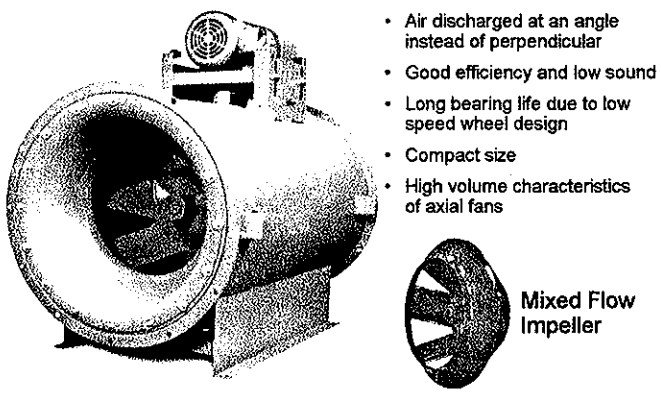


Figure 17

Mixed Flow Fan
Photo Courtesy of Barry Blower

Axial Fan Housing Design

The housing design for axial fans is a cylindrical tube. The tube axial and vane axial fans are built with close tolerances from the blade tips to the sidewall of the tube or shell. On the tubular centrifugal and mixed flow design, the clearance to the wheel is not as critical since the air comes off these impellers and must be turned to exit the shell.

In an axial fan, the motor may be internal to the shell in a direct drive configuration or externally mounted on the shell in a belt drive configuration. The benefits to direct drive are there are fewer components to wear since there are no pulleys and belts. Also, the overall unit can be more compact than the equivalent belt driven model. With direct drive, the motor is in the air stream, which helps the efficiency by cooling the motor.

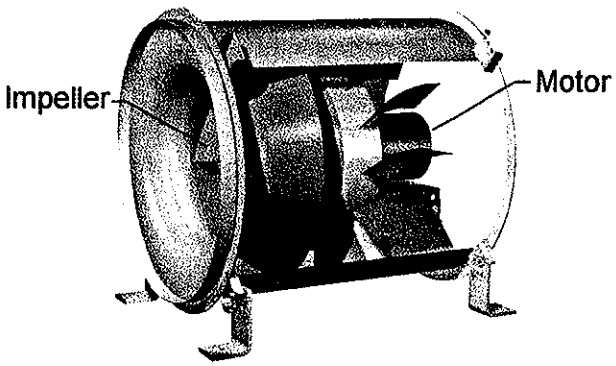


Figure 18

Direct Drive Axial Fan
Photo Courtesy of Greenheck

Belt drive units position the motor out of the air stream for easy access and service. Also, system airflow adjustments can be accomplished by simply changing pulleys. Discharge sound levels are also less with a belt driven model

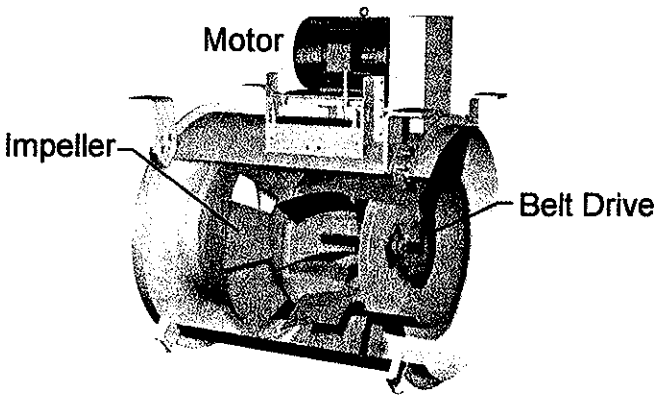


Figure 19

Belt Drive Axial Fan
Photo Courtesy of Greenheck

Vane axial, tube axial, and mixed flow fans are typically controlled by VFDs when used in variable air volume systems. Other means of control include the use of inlet vane dampers and/or control dampers. The use of both of these devices is on the decline in favor of VFDs.

AMCA Fan Classes

AMCA (Air Movement and Control Association) is an international, non-profit organization, dedicated to the certification of performance ratings on fans, louvers, dampers, and other air-handling equipment. AMCA provides fan manufacturers an independent third party verification of their performance ratings.

There are eight certified programs covered by AMCA. For the purposes of this TDP module, the most important programs are Air Performance and Sound Performance.



Figure 20

AMCA

AMCA Class	Maximum System Static Pressure
I	4 in. wg
II	7 in. wg
III	12 in. wg

Figure 21

AMCA Fan Classes

AMCA categorizes centrifugal fans into three performance/construction classes (Class I, II, and III) based on certain defined operating criteria. Each different class corresponds to a certain maximum total pressure at which the fan will operate. This chart shows the maximum pressure limits for each fan class.

Fan construction class ratings are based on the outlet velocity from the fan discharge and the total system static pressure. Most fan discharge velocities are designed around 2500-3000 fpm.

To go to a higher class, manufacturers may use different methods. Some may increase metal gauge, shaft diameter, add tip material, change to a higher strength material, etc. The bottom line is that the added loads of the higher speeds must be accommodated in the design.

If you run a Class II wheel in a Class I condition, it should last longer than a Class I wheel in the Class II conditions.

A Class II wheel running in Class II conditions will not necessarily last longer than a Class I wheel in Class I conditions.

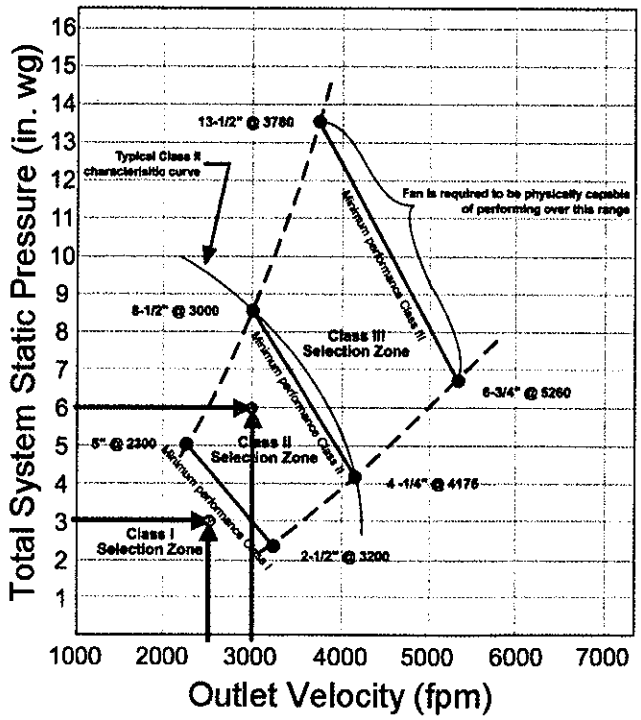


Figure 22

AMCA Centrifugal Fan Construction Class

The cost of Class III construction is usually prohibitive to be used for Class I conditions.

Here are two examples of how to determine fan class using the chart in Figure 22. If the fan discharge velocity is 3000 fpm and the total system static pressure is 6 in. wg, the operating conditions fall within the AMCA Class II range and a Class II fan should be considered for this application. If the fan discharge velocity is 2500 fpm and the total system static pressure is 3 in. wg, the operating conditions fall within the AMCA Class I range and a Class I fan could be used for this application.

Performance Ratings and Static Efficiency

One of the early methods for rating centrifugal fans for HVAC applications was a multi-rating table. A typical multi-rating table shows the wheel speed (rpm) and brake horsepower (bhp), of the fan for various combinations of air flow rate (cfm), outlet velocity (fpm) and static pressure (sp). The disadvantage of multi-rating tables was that interpolation was frequently necessary to obtain specific values such as fan speed. Tabular ratings of this type do not give the user a graphical representation of the fan characteristics, performance, or efficiency.

VOL	VEL	1/4" SP		3/8" SP		1/2" SP		5/8" SP		3/
CFM	FPM	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM
6032	800	391	0.36	431	0.48	471	0.60	518	0.76	557
7450	1000	448	0.52	484	0.68	517	0.82	549	0.98	580
9048	1200	508	0.74	541	0.92	571	1.11	600	1.29	627
10556	1400	573	1.02	601	1.23	629	1.44	655	1.66	681
12064	1600	639	1.38	664	1.61	689	1.85	713	2.10	736
13572	1800	706	1.82	729	2.08	752	2.35	774	2.62	795
15080	2000	774	2.36	796	2.65	816	2.94	836	3.24	856
16588	2200	843	3.01	863	3.33	882	3.65	901	3.97	919
18096	2400	913	3.78	931	4.12	949	4.47	966	4.82	984
[illegible]	2600	982	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1033	[illegible]	[illegible]

Figure 23

Centrifugal Fan Multi-Rating Table

Preferred alternatives to multi-rating tables are fan curves. Figure 24 shows an example of a fan curve from years ago. The cfm was plotted on the horizontal axis, with static pressure (in.wg) plotted on the vertical axis. This happens to be a high static capable airfoil type fan curve. At the intersection of these two required values, the fan speed may be read from the family of speed curves.

As an example, a selection at 26,000 cfm and 6 in. wg static pressure requires a fan rpm of 1800.

The bhp was often represented on yet another curve (not shown here).

There are several advantages to selecting with curves. Static efficiency lines (SE) may be provided as shown. Maximum static efficiency (MSE) is measured in percentages; MSE defines the most efficient operating range of the fan, but does not define an actual value of efficiency. Below curve C, in this example, the actual static efficiency drops to less acceptable performance limits.

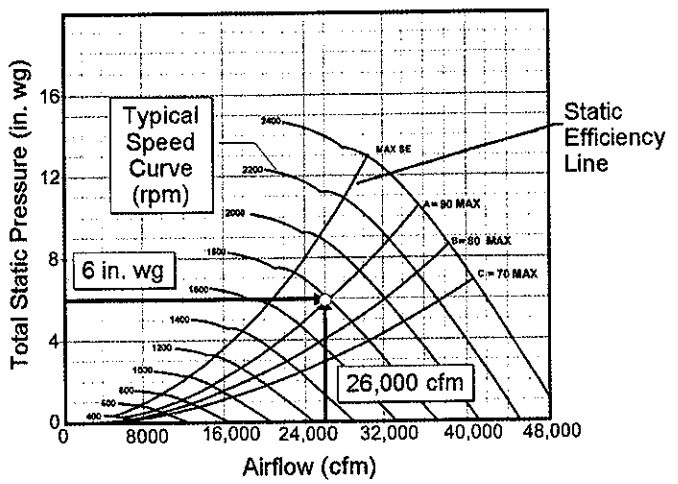


Figure 24

Centrifugal Fan Curve Example

Our example point at 26,000 cfm and 6 in. wg fell on the 90% SE line, so that is an efficient operating point.

When selecting the type of fan in an air conditioning system, the goal is to keep the energy input low, while having a stable selection. For centrifugal fans, the forward-curved impeller is the lowest in static efficiency at approximately 65-70%. The backward-inclined is a higher efficiency fan at 75 to 80% static efficiency. The airfoil impeller, which is a refinement of the backward-inclined design, is the most efficient at approximately 80-85% static efficiency.

The best fan selections

are those whose point of operation (rated point) ends up near (preferably slightly to the right) of the MSE (maximum static efficiency) line.

Fan Laws

Fan laws are a series of equations that can predict the performance of fans at any operating condition. However, to use the fan laws, a known condition of operation is required as a starting point.

Fan laws predict the airflow (cfm), static, velocity or total pressure and required brake horsepower (bhp) at varying fan speeds (rpm) and air densities. Designers of HVAC systems are usually interested in knowing the behavior of a given fan operating within a given duct system. Under these circumstances the following fan laws are applicable.

The most commonly used fan laws in simplified form are:

cfm varies DIRECTLY with rpm

P_S varies with the SQUARE of the rpm

bhp varies with the CUBE of the rpm

Figure 25

The Three Main Fan Laws

Here are the three most widely used fan laws. Others involving density and air temperature changes from standard are listed in the Appendix.

First Fan Law:

- cfm varies directly with the fan speed.

$$\frac{cfm_1}{cfm_2} = \frac{rpm_1}{rpm_2} \quad \text{and} \quad rpm_2 = rpm_1 * \left[\frac{cfm_2}{cfm_1}\right]$$

Second Fan Law:

- Total system static pressure or system resistance varies as the square of the fan speed.

$$\frac{P_{S1}}{P_{S2}} = \left[\frac{cfm_1}{cfm_2}\right]^2 = \left[\frac{rpm_1}{rpm_2}\right]^2 \quad \text{and} \quad P_{S2} = P_{S1} * \left[\frac{rpm_2}{rpm_1}\right]^2$$

Third Fan Law

- Brake horsepower varies as the cube of the fan speed.

$$\frac{bhp_1}{bhp_2} = \left[\frac{cfm_1}{cfm_2}\right]^3 = \left[\frac{rpm_1}{rpm_2}\right]^3 \quad \text{and} \quad bhp_2 = bhp_1 * \left[\frac{rpm_2}{rpm_1}\right]^3$$

Density Effects

At a steady fan rotation (rpm), a fan is a constant displacement device. It will move the same airflow (cfm) regardless of the density of the air being handled. Fans are rated based on standard air conditions. Standard air has a density of 0.075 lb/ft^3. This density is the same as that of dry air at 69.8° F and a barometric pressure of 29.92 in. Hg or 14.696 psia.

For example, if a fan can move 5000 cfm at standard air conditions, at the same speed, it will move 5000 cfm of air at 200° F. However, the density of air at 200° F is 80 percent of the standard air density at 69.8° F, therefore only 80 percent of the horsepower is required to move it.

Air Density Factors

Altitude (ft.)	Temperature			
	70	100	200	300
0	1.000	.946	.803	.697
1000	.964	.912	.774	.672
2000	.930	.880	.747	.648
3000	.896	.848	.720	.624
4000	.864	.818	.694	.604
5000	.832	.787	.668	.580
6000	.801	.758	.643	.558
7000	.772	.730	.620	.538
				.518

Figure 26

Fan Laws — Air Density Factors

Because the mass flow of air at 200° F is only 80 percent (0.803 from table) of the mass flow at 69.8° F, the fan will create only 80 percent of the velocity and static pressures. The reduction in static pressure will be proportional to the horsepower; therefore, the static efficiency of the fan will remain unchanged. For example as shown above, at 6000 feet above sea level, the density of air at 69.8° F is approximately 80 percent (0.801 from table) of standard air density. At this elevation, the fan would perform in the same as described when handling air at 200° F at sea level.

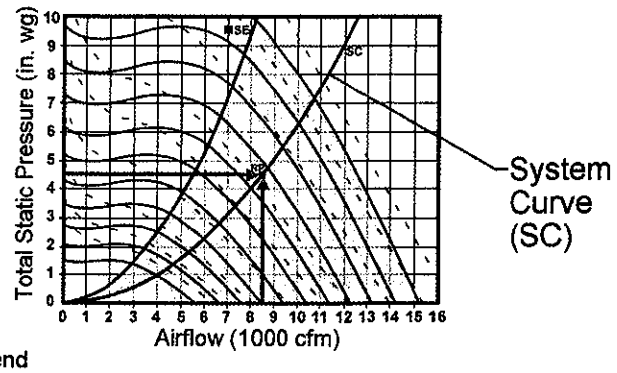
Other Fan Laws

When the density of the air due to altitude or temperature is considered, there are additional fan laws that apply. These are shown in the Appendix section.

It is important that the system designer understand the basic characteristics of fans. Once the required airflow is known (recirculation or exhaust), the best way to evaluate the system is at standard air conditions. Adjustments can be made in the system to ensure the desired mass flow is being provided when the density of the air handled by the fan differs significantly from standard air density fan ratings.

Example: Using the Fan Laws

An engineer estimates that his duct static resistance will be 2.5 in. wg for an 8500 cfm air-handling system installed in a nursing home. Filter resistance is 0.50 in. wg, cooling coil resistance is 1.1 in. wg, and heating coil resistance is 0.4 in. wg for a system static pressure of 4.5 in. wg. An airfoil centrifugal fan is selected and submitted with the following fan curve. The resulting rpm is 2539 and the bhp is 11.2.



Legend

rpm bhp **MSE** - Max. Static Eff. **SC** -System Curve **RP** - Rated Point
rpm = 2539 bhp = 11.2 Class II Max. rpm = 2950
rpms (*100, L to R): 12 14 16 18 20 22 24 26 28 30 32
bhps (L to R): 1.5 2 3 5 7.5 10 15 20 25
Note: Shaded Area -- Recommended Operating Range

Figure 27

Example: Using the Fan Laws

During the actual installation of the duct system, the architect decided to change the ceiling from a flat suspended ceiling to a more "aesthetically pleasing" tray ceiling. This requires four additional elbows and other duct changes to be added to the supply and return ductwork, which raised the supply duct static resistance another 0.75 in. wg by the engineer's calculation. The new total static pressure will be 5.25 inches. Using the fan laws, what will the new fan rpm and motor horsepower be?

Using the second fan law to solve for the new rpm:

$$\frac{P_{S1}}{P_{S2}} = \left[\frac{rpm_1}{rpm_2}\right]^2 \text{ or } \sqrt{\frac{P_{S1}}{P_{S2}}} = \frac{rpm_1}{rpm_2}$$

$$rpm_2 = rpm_1 * \sqrt{\frac{P_{S2}}{P_{S1}}}$$

$$rpm_2 = 2539 * \sqrt{\frac{5.25}{4.5}}$$

$$rpm_2 = 2742 \ rpm$$

Where:

P_S = total system static pressure
Condition 1 = known condition
Condition 2 = new condition

Using the third fan law to solve for the new brake horsepower:

$$\frac{bhp_1}{bhp_2} = \left[\frac{rpm_1}{rpm_2}\right]^3$$

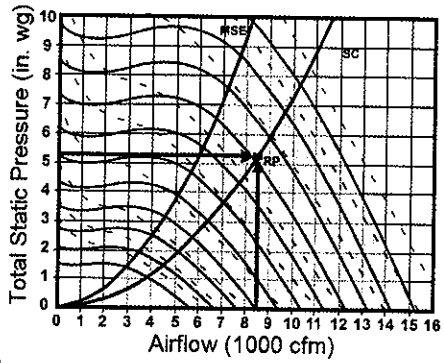
$$bhp_2 = bhp_1 * \left[\frac{rpm_2}{rpm_1}\right]^3$$

$$bhp_2 = 11.2\,hp * \left[\frac{2742}{2539}\right]^3$$

$$bhp_2 = 14.1\,bhp$$

Where:

bhp = fan brake horsepower
Condition 1 = known condition
Condition 2 = new condition



Legend
rpm bhp **MSE** - Max. Static Eff. **SC** -System Curve **RP** - Rated Point
rpm = 2539 bhp = 11.2 Class II Max. rpm = 2950
rpms (*100, L to R): 12 14 16 18 20 22 24 26 28 30 32
bhps (L to R): 1.5 2 3 5 7.5 10 15 20 25
Note: Shaded Area – Recommended Operating Range

Figure 28

Using the Third Fan Law

After calculating the new rpm and horsepower, we need to make sure that we have not exceeded the fan or motor's capability. A quick check of the maximum fan rpm shows us that we have not (Class II max rpm 2950). The original fan selection required 11.2 bhp so a 15 hp motor was selected. The new horsepower requirement is 14.1 hp, so the motor will not need to be changed.

Analyzing this one step further, what would be the additional energy cost because of the architect's change? Given the following:

Operating hours per day	18 hours
Days of operation	365 days
Additional motor horsepower	2.9 bhp
Motor efficiency	92%
Electrical rate	$0.10435 per kWh

$$= \frac{18\ hrs * 365 \times 2.9\ bhp * 0.746 kW / bhp * \$0.10435 / kWh}{.92}$$

Additional energy cost per year = $1,612

System Curve, Fan Stability, System Effect

System Curve

An air system may consist simply of a fan with ductwork connected to the inlet or the discharge, or both, as in an exhaust system. A more complex system could include a fan, supply and return ductwork, cooling and/or heating coils, filters, air mixers, mixing boxes, diffusers, zoning terminals, dampers, sound attenuators, etc. The function of the fan is to provide the required energy to the airstream to overcome the resistance to flow imposed by all the system components.

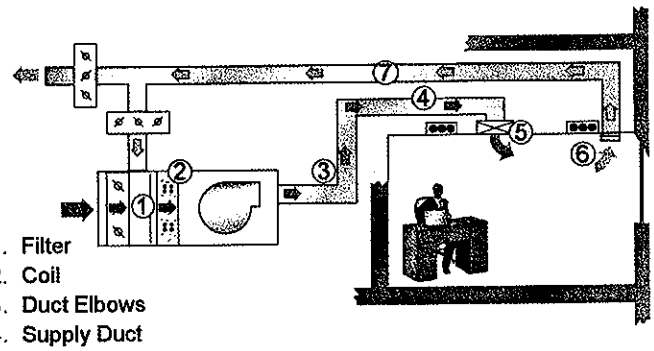


Figure 29

System Resistance Components

The component manufacturer usually provides the pressure loss or flow resistance for individual components. In addition, the pressure losses for the duct system must be determined. The procedure for determining duct resistance is discussed in TDP-504, Duct Design Fundamentals. Later in this module, the effects of field connections will be discussed to assist the designer in evaluating the effect of these items on final fan performance. The summation of all these resistances establishes the required fan total static pressure.

The system curve defines the volume flow rate versus pressure characteristics of the duct system in which a fan will be installed. For most applications, the volume flow rate to pressure relationship of a system is governed by the following equation, often called the "duct law." Notice it is the second fan law.

$$\frac{P_{S1}}{P_{S2}} = \left[\frac{cfm_1}{cfm_2}\right]^2 = \left[\frac{rpm_1}{rpm_2}\right]^2 \quad \text{or} \quad P_{S2} = P_{S1} * \left[\frac{cfm_2}{cfm_1}\right]^2$$

Once the system designer has determined the total system static pressure loss (P_S) for one airflow (cfm), it is very easy to calculate the corresponding pressure loss for any other flow rate. The system curve is not included on the fan performance curve when it is issued from the fan manufacturer since its determination must be left to the system designer.

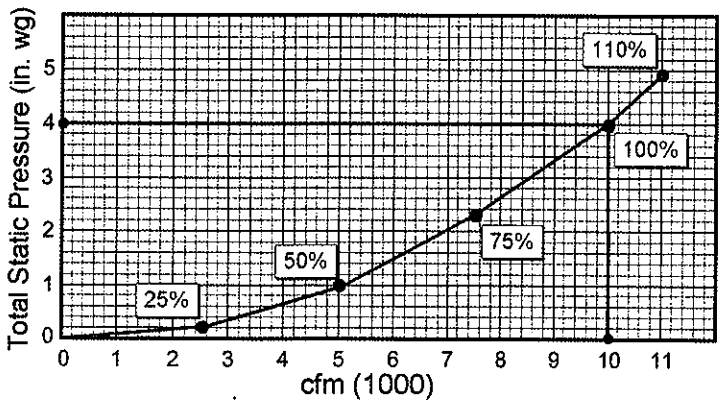
A fan running at a particular speed can have an infinite number of operating points all along its system curve. The fan rpm line will intersect the system curve to produce a single operating point. There can be only one operating point at the intersection of the system curve and the fan curve.

Example: System Curve

Assume that a fan delivers 10,000 cfm against a total static pressure of 4.0 in. wg.

Find the duct system static pressure resistance to flow at 110%, 75%, 50% and 25% airflow:

$$P_{S2} = P_{S1} * \left[\frac{cfm_2}{cfm_1}\right]^2$$



Known: Fan delivers 10,000 cfm at 4 in. wg total static pressure

Figure 30

System Curve

at 110% (11,000 cfm): $P_{S2} = 4.0 * \left[\frac{11,000}{10,000}\right]^2 = 4.0 * (1.21) = 4.84 \ \ in.\ wg$

at 75% (7500 cfm): $P_{S2} = 4.0 * \left[\frac{7500}{10,000}\right]^2 = 4.0 * (0.5625) = 2.25 \ \ in.\ wg$

at 50% (5000 cfm): $P_{S2} = 4.0 * \left[\frac{5000}{10,000}\right]^2 = 4.0 * (0.25) = 1.0 \ \ in.\ wg$

at 25% (2500 cfm): $P_{S2} = 4.0 * \left[\frac{2500}{10,000}\right]^2 = 4.0 * (0.0625) = 0.25 \ \ in.\ wg$

No two system curves are alike unless constructed identically and handling the same airflow.

System curves

look a lot like static efficiency lines, but as we can now see, they are two completely different concepts. Both of them are instrumental in conducting a fan system analysis.

Using the System Curve

The intersection of the calculated system curve and the fan pressure airflow curve (rpm line) is the rated point (RP). The resulting fan cfm and pressure can then be read.

However, in reality the as built duct system and other factors may result in a system curve with less resistance or greater resistance (dotted lines) than was estimated.

Figure 31 shows a situation where the duct system has more resistance to flow than was estimated. The calculated operating point is point ①. However, at the same fan speed and higher static pressure, the fan will operate at point ②. To get the design airflow at a higher static pressure, it is necessary to increase the speed of the fan so it will operate at point ③. Assuming the air quantity at ② is 90 percent of design, it will be necessary to increase fan speed by 10 percent. This will result in a large increase in fan horsepower, based on the third fan law. If the fan motor is already operating near its nominal horsepower rating, it will be necessary to replace it with a larger motor.

Quite often, system designers add a safety factor to compensate for an inaccurate or incomplete estimate of system static pressure losses. If the resulting system resistance is less than estimated, the fan will operate at point ④. There is no advantage to operating at this point – the fan may operate at a lower efficiency and may require more horsepower than at design flow. In this case, the fan speed should be reduced so that the fan will operate at point ⑤.

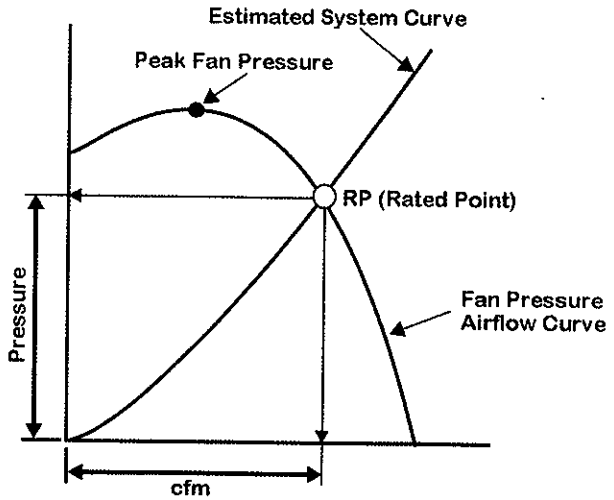


Figure 31

Intersection of System Curve and Fan rpm

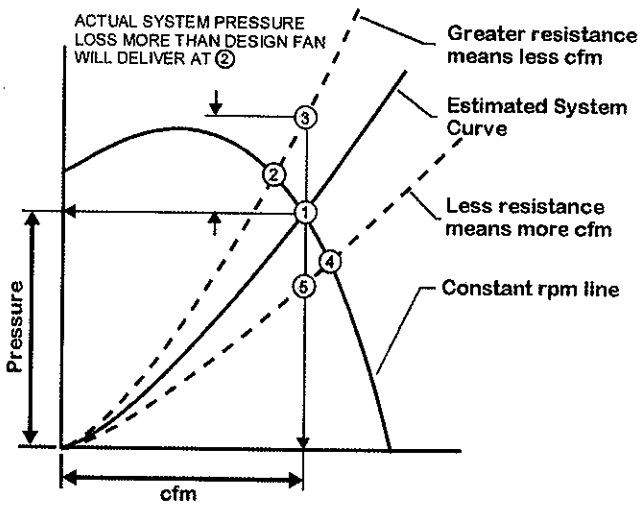


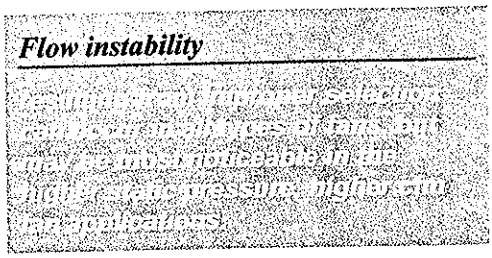
Figure 32

Variation from Estimated System Curve

Fan Stability

To learn about fan stability, we should discuss those factors that lead to fan instability. Fan instability occurs when the airflow through the fan surges or pulses due to turbulent airflow conditions. There are two causes of turbulent airflow through the fan. Incorrect duct connection at the discharge of the fan is the first cause. Turbulence in the area of the fan's cut-off plate can result.

Selecting the fan outside of its natural stability region is the second cause. If a fan is operated too far to the left of the maximum static efficiency line, uneven flow through the fan blades can result. The fan may not be able to maintain stable laminar flow under these low cfm and high static pressure conditions; turbulence will exist in part of the blade passage.



To avoid instability, care must be taken to select fans operating in a constant volume air system near their maximum efficiency point. When operating in a variable air volume (VAV) system, fans should ideally be selected to the right of the maximum efficiency curve for the design operating point. Because fans in VAV systems spend most of their operating hours at part load, this approach optimizes the efficiency and helps ensure that the fan operation does not drift too far to the left into the naturally unstable region.

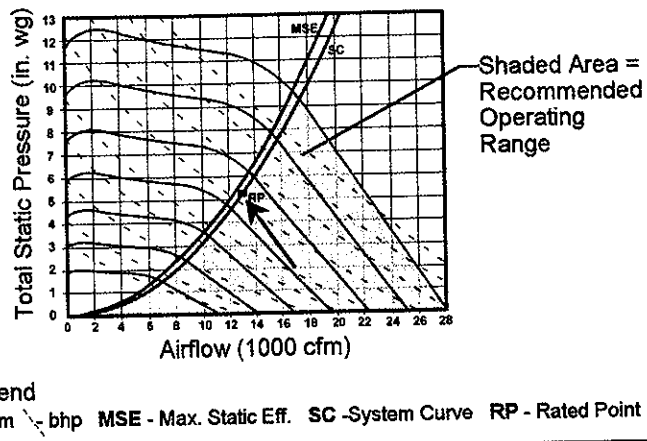


Figure 33

Fan Stability — Good Selection

Improper selection and installation can result in noise and vibration from the fan in the air handler. Here is an example of a potentially unstable fan selection because it was selected to the left of the maximum static efficiency line, well outside of the recommended area of operation. For a complete discussion on part load stability, refer to TDP-613, Fans in VAV Systems.

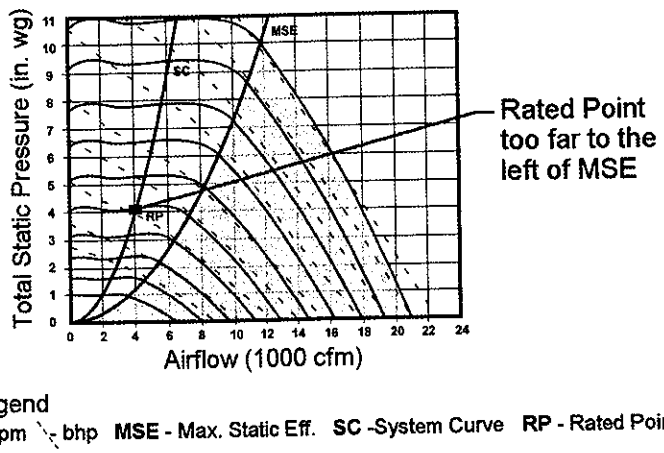


Figure 34

Fan Stability — Poor Selection

System Effect, with Example

If there is a difference between what the fan ratings say on paper and what is actually happening in the field, the explanation may be attributable to system effect. System effect refers to the conditions in a duct system that affect fan performance and related testing, adjusting, and balancing work. The subject of system effect usually arises after no allowance or consideration has been made for the effect of the duct system connection on the manufacturer's fan performance.

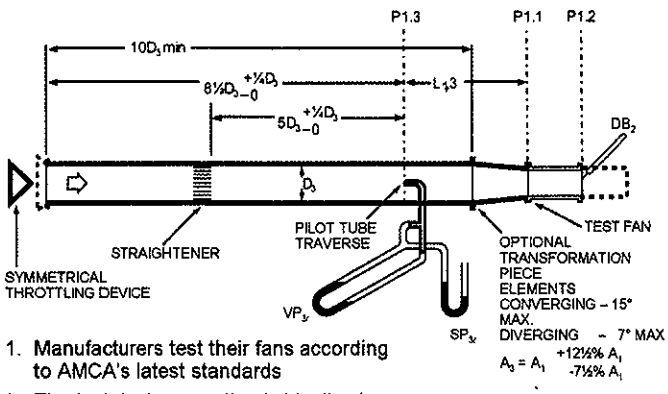
Estimating the impact of system effect is needed in order to arrive at a satisfactory fan selection.

Fan Test Station

Most manufacturers in the USA and Canada rate fan performance from tests made in accordance with the latest AMCA Standard Test Code for Air Moving Devices.

AMCA defines exact test procedures and conditions of fan testing so that fan ratings provided by various manufacturers are all on the same basis and can be compared.

In general, a fan is installed in an inlet duct test setup as shown in Figure 34. Centrifugal fans are tested with a discharge duct that is specified by AMCA. The duct connection to the fan is idealized to insure accuracy, consistency, and maximum fan performance. Any fan installation that deviates from this "idealized" inlet duct connection will not be able to deliver rated performance. The impact of system effect reduces fan performance.



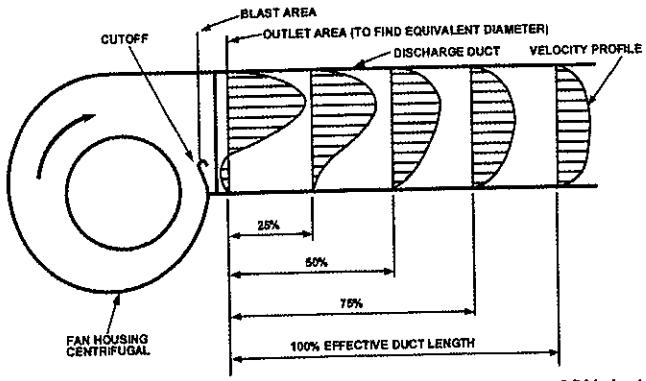
1. Manufacturers test their fans according to AMCA's latest standards
2. The test duct connection is idealized
3. Installations not meeting this ideal connection will have lower fan performance

Figure 35

Idealized Fan Test Station

Fan Velocity Profile

The velocity profile at the outlet of a fan is not uniform. The air tends to hug the outer portion of the fan housing – providing higher velocities at the top of the blast area and lower velocities at the bottom of the blast area. A straight section of duct at the fan discharge is required in order to establish a more uniform air velocity profile. To calculate 100% effective duct length, assume a minimum of 2.5 duct diameters. In the diagram this uniform velocity profile is established at the 100% "effective duct length" and mirrors that of the AMCA standard test setup. Failure to provide this straight section of discharge duct as shown will result in air turbulence and loss of fan performance.



To calculate 100% effective duct length, assume a minimum of 2½ duct diameters, for 2500 fpm or less. Add 1 duct diameter for each additional 1000 fpm.

Example: 5000 fpm = 5 equivalent duct diameters.

If duct is rectangular with side dimensions a and b, the equivalent duct diameter is equal to $\sqrt{\frac{4ab}{\pi}}$

Figure 36

Fan Discharge Velocity Profile

Transition to Outlet Ducts

The first section of ductwork is required to transition to the main duct size. This transition must follow AMCA rules in order to minimize fan losses. The outlet duct area is to be no greater than 107.5 percent or less than 87.5 percent of the fan discharge outlet area. Further, the transition slope is not be to more than 15 degrees for a converging duct or more than 7 degrees for diverging duct.

For 100 percent effective diffusion to a uniform duct velocity the initial transition plus portion of the main duct must extend in a straight line for at least two and one-half equivalent duct diameters for duct velocities of 2500 fpm or less based on the duct width and height.

An additional duct diameter must be added for each additional 1000 fpm. An equivalent duct diameter for a rectangular duct is equal to the square root of the quantity: [(4 * width * height)/pi].

Losses-Outlet Ducts

Because it is virtually impossible to design a duct system identical to that used to test the fan, a system effect factor must be determined and added to the expected system resistance losses.

Example: Determination of System Effect for Outlet Duct on SWSI Fan.

Step 1: Find the fan's blast area to outlet area ratio.

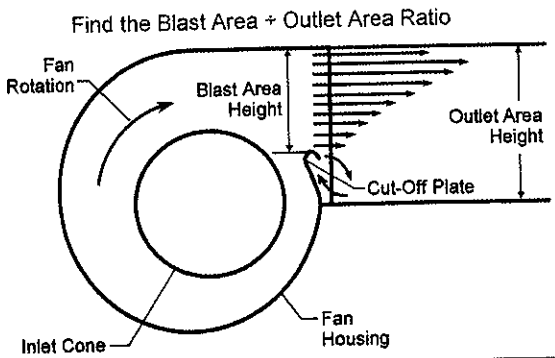


Figure 37

Step 1—Determine Fan Outlet Arrangement

Using the ratio of the fan's blast area/outlet area, find the system effect factor in the table (Figure 37). If this is not readily available, assume a ratio of 0.6.

Step 2: Use the Outlet Duct Table to find the system effect factor.

To determine the system effect factor, use the table shown.

You will need to establish the percent effective duct length that applies to your situation. Enter the table and determine the system factor curve(s) to use.

	No Duct	12% Effective Duct	25% Effective Duct	50% Effective Duct	100% Effective Duct
Pressure Recovery	0%	50%	80%	90%	100%
Blast Area / Outlet Area	System Effect Curve				
0.4	P	R-S		W	–
0.5	P	R-S		W	–
0.6	R-S	S-T	U-V	W-X	–
0.7	S	U	W-X	–	–
0.8	T-U	V-W	X	–	–
0.9	V-W	W-X	–	–	–
1.0	–	–	–	–	–

Determining system effect

- Find blast area/outlet area from Step 1 or use 0.6 if not known
- Determine effective duct length
- Enter table above to find appropriate letter for system effect
- Example: 0.6 and 25% effective duct (use curve U or V)

Figure 38

Step 2 — Losses - Outlet Duct Factors

Step 3: Use the system effect curves to find system effect factor.

Calculate the outlet duct velocity in fpm. Then enter the system effect curves and find the system effect factor.

With an outlet duct velocity of 2500 fpm, enter the curves below. Follow the 2500 fpm line until the U curve is reached. Read to the left and find the system effect factor to be 0.15 in. wg.

This is an additional resistance penalty the fan will operate against and must be added to the total system static pressure before selecting the fan.

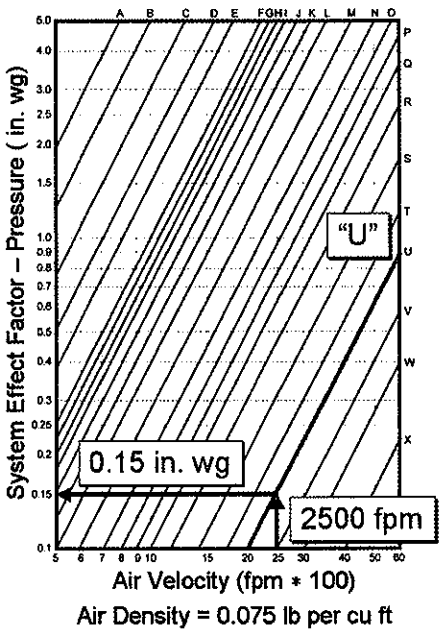


Figure 39

Step 3 — System Effect Curves Pressure Add

Discharge Elbows

Published or calculated values for elbow pressure losses according to SMACNA or ASHRAE are based on a uniform air velocity profile entering the elbow. Locate the elbow at least as far away from the fan discharge as the 100% equivalent length. If this is not possible, elbow losses will be greater than expected.

If the elbow is located closer than the 100% effective length, do not use turning vanes in the elbow. The turning vanes tend to continue the non-uniform velocity profile beyond the elbow.

Let's continue our example problem with an elbow "Position B" located at 25% equivalent length. What is the pressure loss of the elbow?

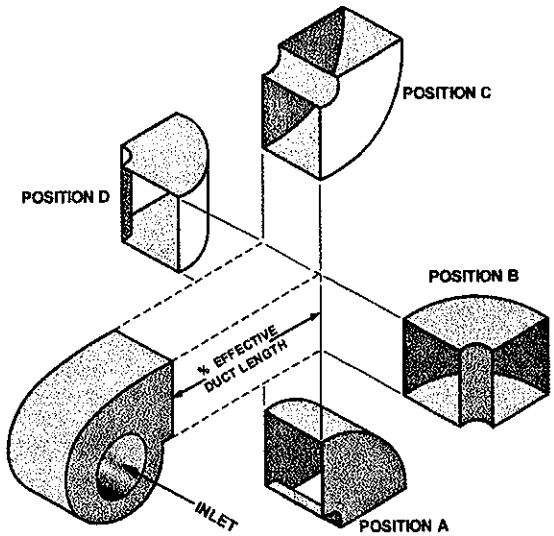


Figure 40

Discharge Elbows

System Effect-Discharge Elbow

The table provides the appropriate system effect curve to be used to determine the added pressure loss imposed by the location and bend of an elbow at the fan discharge.

For the example problem, we will use a "B" type elbow located at 25% effective length with a ratio of 0.6. This results in the use of curve "R" to establish added elbow losses.

System Effect Factor Curves for SWSI fans

Impact of elbows:

- Enter table at 0.6 blast area ratio
- Enter at 25% effective duct
- With elbow "B" find curve "R"
- Now go to system effect curves to find loss

Blast Area / Outlet Area	Outlet Elbow Position	No Outlet Duct	12% Effective Duct	25% Effective Duct	50% Effective Duct	100% Effective Duct
0.4	A B C D	N M L-M L-M	O M-N M M		S R Q Q	NO SYSTEM EFFECT FACTOR
0.5	A B C D	P N-O M-N M-N	Q O-P N-O N-O		T S R-S R-S	
0.6	A (B) C D	Q P N-O O	Q-R Q O-P P	R-S (R) P-Q Q-R	U T S S-T	
0.7	A B C D	S-T R-S Q-R R	T S R R-S	U T S S-T	W V U-V U-V	
0.8	A B C D	S R Q Q-R	S-T R-S Q-R R	T-U S-T R-S S	V-W U-V U U-V	
0.9	A B C D	S-T R-S R R-S	T S R-S S	U T S-T T	W V U-V V	
1.0	A B C D	R-S S-T R-S R-S	S T S S	T U T T	V W V V	

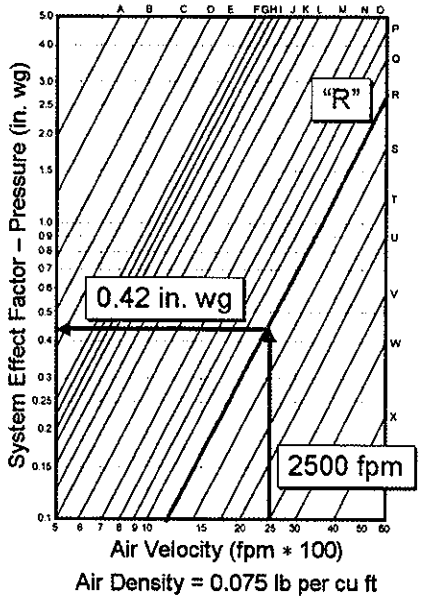
Multipliers For DWDI Fans
Elbow Position B = ΔP_S * 1.25
Elbow Position D = ΔP_S * 0.85
Elbow Positions A and C = ΔP_S * 1.00

Figure 41

System Effect Factors for Outlet Elbows

Elbow Loss

Knowing the curve ("R") from the previous page and the duct velocity of 2500 fpm, the added impact of the "Position B" elbow can be determined to be 0.42 in. wg.



Elbow "B"

added pressure loss

Figure 42

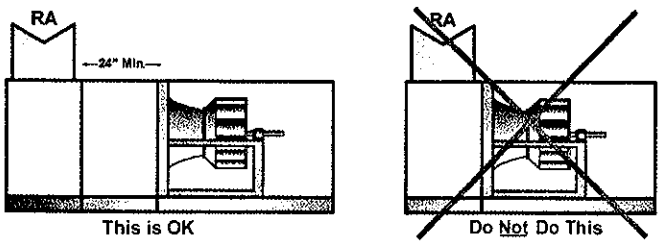
Elbow Loss

System Effect Conclusion

As we can see, the easiest way to reduce or eliminate system effect is to avoid discharge duct situations that would create non-uniform airflows, such as elbows too close to the discharge.

The same principle applies to inlet airflow arrangements. There are system effect charts that are available to be used to determine inlet system effect just like we did for the discharge. They are available from sources like AMCA.

Here is an example of non-uniform inlet flow created by ducting the return too close to the suction of a plenum fan.



System effect caused by non-uniform airflow into the vortex of the plenum fan

Figure 43

System Effect Plenum Fan Inlet

Note

The easiest way to eliminate the possibility of a system effect problem is to avoid the situations that create non-uniform airflows entering the fan section in the first place.

The discussion of system effect in this TDP module has been limited to the effects of some of the more common arrangements influencing fan performance. For a more comprehensive treatment, reference should be made to SMACNA or AMCA.

Miscellaneous Fan Topics

Bearings

Fan manufacturers use several different types of bearings in their product line. That is because the bearings on a small gas furnace are subject to a different loading than those on a large central station air-handling unit. The important terms that one should understand apply to many bearing types. Fan manufacturers work with bearing suppliers to establish a level of quality and assure the bearing life expectance required by the HVAC industry.

Bearing Life

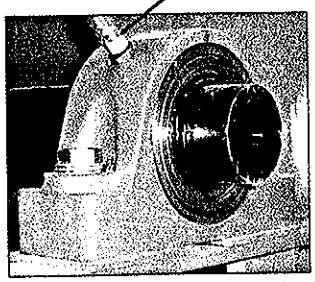
The life of a bearing is a function of the number of revolutions it experiences before developing evidence of fatigue in the moving elements. The terms that have been used in the industry are B_{10}, L_{10} and B_{50} or L_{50}. The terms B_{10} and L_{10} mean the same thing, as do B_{50} and L_{50}. The current terms to be used are L_{10} and L_{50}.

The American Bearing Manufacturer's Association (ABMA) defines L_{10} as the bearing life associated with a 90 percent reliability rate when operating under normal conditions. Normal operation means the bearing was kept clean, properly lubricated, operated at a reasonable temperature, and free of dust and debris with perfect alignment. In reality, this may not be the case, so the actual life of the bearing can be shortened based on the application conditions. However, following the manufacturer's installation and maintenance requirements will help extend the life to the manufacturer's specified values.

The designation L_{50} indicates the duration in hours that one half (50 percent) of the bearing can be expected to survive without showing evidence of failure. Conversely, it is the life at which one half of the bearings can be expected to fail. Thus a bearing with a longer L_{50} life rating for a given application can be expected to perform more reliably than another bearing with a shorter L_{50} life rating. L_{50} life equals five times the L_{10} life.

To get a L_{50} life equal to a L_{10} 100,000 life, you must specify the L_{50} life to be 500,000 hours.

Bearing life is useful when specifying a level of bearing construction. When required to provide a given life such as L_{10} all equipment manufacturers must supply the same capability bearing for the same given application. A 100,000 hour L_{10} bearing will have a life over twice as long as 40,000 hour L_{10} bearing and hence should last longer on a similar field application.



Typical Pillow Block Bearing

Hours and Years			
How long is 200,000 hours? The following table converts hours to years based on different daily usage.			
Hours	YEARS		
	8 hours per day	16 hours per day	Continuous duty
40,000	13.7	6.8	4.6
100,000	34.2	17.1	11.4
200,000	68.4	34.2	22.8
400,000	137	68.4	45.8
500,000	171	85.6	57.0
1,000,000	342	171	114

Figure 44

Bearing Life

Bearing Selection

Most manufacturers select their bearings as an integral part of the air-handling unit fan design. Some of the main selection criteria include shaft diameter and weight, motor horsepower range, weight and location on the shaft, maximum fan speed, fan wheel weight, and the direction of belt pull.

Ball bearings with stamped steel housings are well suited for applications with light loads, as in smaller equipment. The use of these bearings is limited to fan products with ¾ inch and smaller diameter shafts, and one horsepower and smaller motors, such as small fan units.

Air-handling units will tend to use ball, spherical, or tapered roller pillow block or flange-mount bearings. Once the application exceeds the speed limit for the contact seal and lubrication capabilities of the solid housing, a pillow block bearing is typically specified. The pillow block design incorporates a friction-free seal and a larger grease cavity. Higher speeds can then be attained and the rollers become the limiting factor instead of the seal.

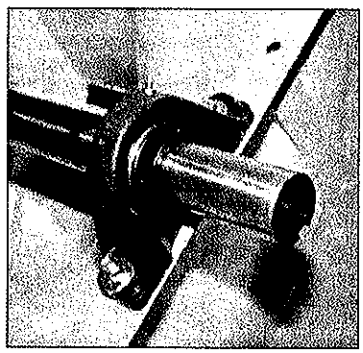


Figure 45

Bearing life is affected by several variables.

To enhance accessibility, it is often desirable to extend the bearing lubrication lines to the drive side of the fan. In some cases customers want the lubrication lines and fittings extended to the cabinet exterior so that bearing lubrication can be performed without stopping the unit. But, customers should also consider the downside of extended lube lines. Bearings should be inspected at the time of lubrication to look for improper operating conditions or signs of failure. If lube lines fail or vibrate loose, lubricating grease may never reach the bearing, creating an ideal condition for premature bearing failure. Also, bearings can be over-lubricated, in which case seals are dislodged, allowing the surplus lubricant to escape.

Motors

HVAC Fan motors typically have two types of enclosures: open drip proof (ODP), and totally enclosed fan cooled (TEFC). These two names refer to the method used to cool the motor windings and describe the type of motor enclosure and internal construction.

Electricity flowing through motor windings develops heat due to the resistance of the windings. This heat is developed continuously and there

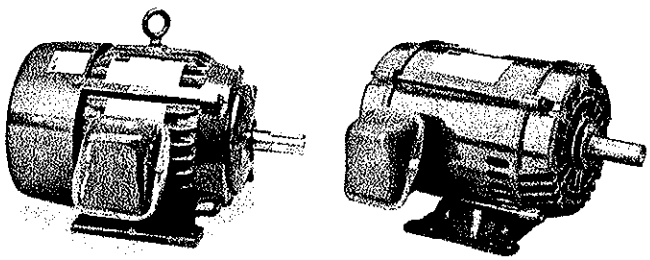


Figure 46

Common HVAC Motor Types

fore must be removed continuously or the temperature of the windings would rise until the winding insulation burns out. ODP and TEFC motors use different methods to remove heat from the windings.

ODP motors have an internally mounted fan pulling ambient air from intake vents in one end of the motor, through the windings, then out of the other end of the motor enclosure through exhaust vents. These vents are placed to prevent falling rain from directly entering the motor enclosure.

ODP motors are advantageous because of their low price, availability, and resistance to runaway heating. However, in ODP motors, air is moved directly through the windings, which leaves deposits on the windings from airborne contaminants such as dust, aerosols, and moisture. Also, splash and wind-driven rain, and even insects and vermin, can enter the motor.

TEFC motors have an externally mounted fan covered by a shroud blowing ambient air across the surface of the motor enclosure. Heat developed in the windings moves by conduction outward through the motor case then into the air moving along the surface of the motor case. The motor case is a heat sink drawing heat from the motor interior to the outside. TEFC motors may have fins on the motor case enhancing this heat transfer into the air.

TEFC motors are advantageous because air is not drawn into the motor for cooling and therefore the windings stay clean and dry. The windings are protected against direct entry of wind driven rain, directed spray, and splash from the ground. Also, insects and vermin cannot enter the motor. TEFC motors protect the single-phase switch keeping it clean and dry.

Some single-phase motors have a switch mechanism located next to the windings, which operates the start capacitors and windings. This switch is easily affected by dust, sand, dirt, and corrosion, and is the largest cause of problems on single-phase motors. ODP motors constantly pull contaminated air over this switch. TEFC motors keep the single-phase switch clean and dry, and therefore single-phase TEFC motors have fewer problems than single-phase ODP motors.

Convection through a motor enclosure (TEFC) is less efficient than directly cooling the windings with air (ODP). This makes TEFC motors more expensive to build.

Some of the construction differences that make TEFC motors more expensive are:

- The fan shroud and a higher-grade winding insulation are used to withstand higher temperatures.
- TEFC motor enclosures are often physically larger than ODP motors
- Finned motor enclosures cost more

It is important to note that TEFC motors should never be thought of as "sealed" or "wash down" duty motors, which they are not. TEFC motors are resistant to directed spray, but TEFC motors are definitely not intended to withstand directed sprays or washing. Air that is heavily laden with caustic or oxidizing vapors can enter a TEFC motor, but more slowly than an ODP motor.

Drives

Most fan drive systems are based on the standard "V" drive belt, which is most commonly used and is relatively efficient. The use of a belt drive allows fan rpm to be easily selected through a combination of motor rpm and drive pulley ratios. Multiple belts and multiple-groove sheaves are required to meet higher horsepower requirements. Drive ratio is defined as follows:

$$Drive\ Ratio = \frac{motor\ rpm}{desired\ fan\ rpm}$$

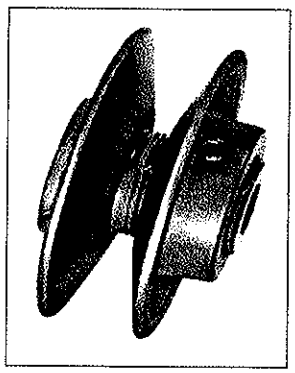


Figure 47

Variable Sheave

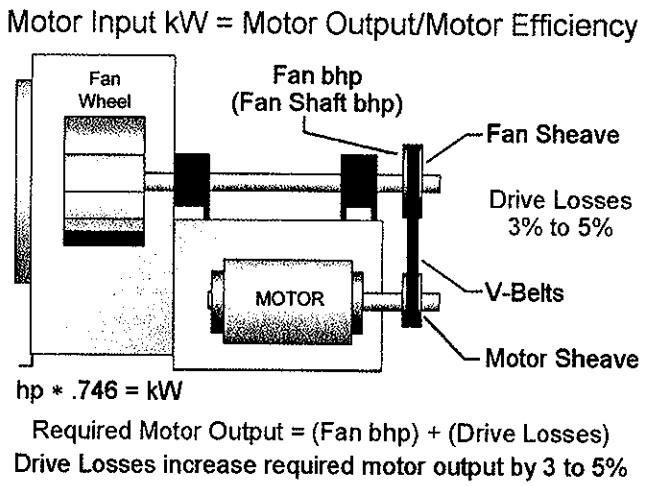


Figure 48

Motor and Drive Terminology

The fan drives are either fixed drive or adjustable drive. When a unit is furnished with an adjustable drive, the fan sheave diameter can be changed to fine tune the fan speed and performance.

Drive losses refer to the inefficiencies resulting from the frictional effects of pulley and belt assemblies between the motor and the fan wheel.

Higher belt speeds tend to have higher losses than lower belt speeds at the same horsepower.

Drive losses are based on the conventional V-belt, which has been the most commonly used drive in the industry for several decades.

For example, fan brake horsepower output is determined to be 17.1 bhp. What is the required motor output horsepower?

The belts are V-types, the drive loss is 5%.

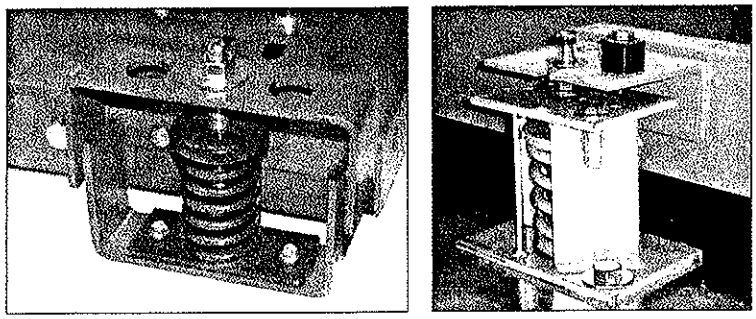
Drive loss = 0.05 x 17.1 hp = 0.86 hp

Motor power output = 17.1 bhp + 0.86 bhp = 18 bhp

Spring Isolation

The presence of vibration is not desirable in any piece of mechanical equipment, and fans are no exception. Excessive fan vibration can cause premature failure of critical parts that may result in high maintenance cost and downtime. Consequently, it is common to find a "vibration clause" written into many specifications.

The causes of vibration may be vibration that is a result of an unbalanced fan wheel or vibration caused from drive misalignment, belt tension, bent fan shaft, etc.



Standard 2-inch Steel Spring Isolator

2-inch Seismic Rated Isolator

Figure 49

Fan Spring Isolation

To alleviate problems caused by vibration, manufacturers may supply internal spring isolation as part of the fan assembly. Most fan assemblies are dynamically balanced before they are installed in the fan cabinet. This ensures that the assembly does not suffer from rotating part unbalance.

Summary

The objective of this module has been to familiarize the designer with fans and how they represent a very important segment of the typical air conditioning system. A clear understanding of how they operate is an essential part of being able to design a good system.

System effect exists on most projects as a result of the fan being installed somewhat differently than laboratory test conditions. Since we cannot always prevent these differing conditions, we must account for their system effect.

An examination of the types of fans available and their performance prepares a system designer to evaluate fan system performance. While it is important that system designers understand the intricacies of fans and fan selection, there is software available to aid in the determination of what fan should be used in a specific application. It is also important to understand motors, drives, and bearings, which are an important component of any fan selection and HVAC system.

Work Session

1. What type of fan would be considered as an overloading type and why?

2. What are the two major types of fans used in the HVAC industry today?

   _____, _____

3. What are the construction differences between a forward-curved and airfoil type fan wheel?

4. How long should the straight length of supply duct be prior to installing an elbow?

   _____

5. What is system effect and how does it effect fan operation? Can it be prevented?

6. Explain the difference between the L_{10} and L_{50} designations for fan bearings?

7. State the three fan laws.

8. Explain how a plenum fan works.

9. Name two reasons a system curve might change over time.

10. What is the difference between a tube axial fan and a vane axial fan? Which is more efficient?

Notes:

Notes:

Notes:

Appendix

Fan Law Equations

Fan Laws for Constant Mass Flow – Capacity, speed and pressure vary inversely as the air density, that is, inversely as the barometric pressure and directly as the absolute temperature.

a. $$cfm_2 = cfm_1 * \left[\frac{DENSITY_{STD}}{DENSITY_{ACT}}\right] \text{ or } cfm_1 * \left[\frac{T_{ACT}}{T_{STD}}\right]$$

b. $$rpm_2 = rpm_1 * \left[\frac{DENSITY_{STD}}{DENSITY_{ACT}}\right] \text{ or } rpm_1 * \left[\frac{T_{ACT}}{T_{STD}}\right]$$

c. $$P_{S_2} = P_{S_1} * \left[\frac{DENSITY_{STD}}{DENSITY_{ACT}}\right] \text{ or } P_{S_1} * \left[\frac{T_{ACT}}{T_{STD}}\right]$$

d. $$bhp_2 = bhp_1 * \left[\frac{DENSITY_{STD}}{DENSITY_{ACT}}\right] \text{ or } bhp_1 * \left[\frac{T_{ACT}}{T_{STD}}\right]$$

Fan Laws for Constant Volume (cfm) and Fan Speed – Horsepower and pressure vary directly with the air density, that is directly as the barometric pressure, and inversely as the absolute temperature.

a. $$bhp_2 = bhp_1 * \left[\frac{DENSITY_{ACT}}{DENSITY_{STD}}\right] \text{ or } bhp_1 * \left[\frac{T_{STD}}{T_{ACT}}\right]$$

b. $$P_{S_2} = P_{S_1} * \left[\frac{DENSITY_{ACT}}{DENSITY_{STD}}\right] \text{ or } P_{S_1} * \left[\frac{T_{STD}}{T_{ACT}}\right]$$

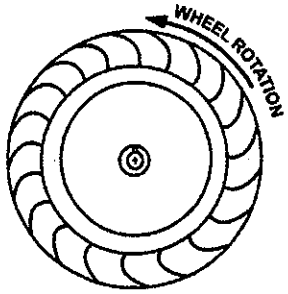
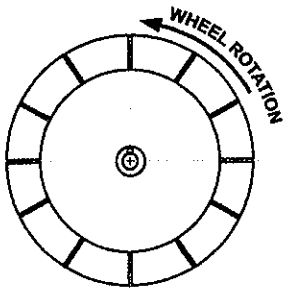
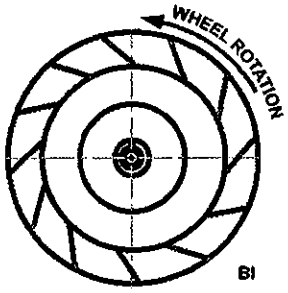
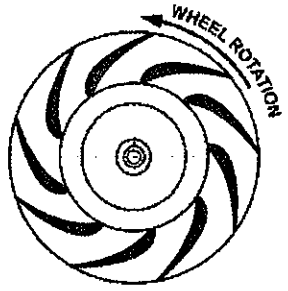
Fan Laws for Constant Static Pressure – Speed, volume flow (cfm) and horsepower vary inversely as the square root of the density, that is, inversely as the square root of the barometric pressure and directly as the square root of the absolute temperature.

a. $$rpm_2 = rpm_1 * \sqrt{\frac{DENSITY_{STD}}{DENSITY_{ACT}}} \text{ or } rpm_1 * \sqrt{\frac{T_{ACT}}{T_{STD}}}$$

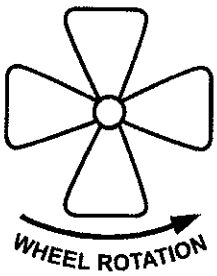
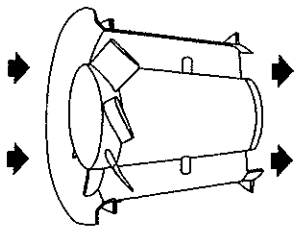
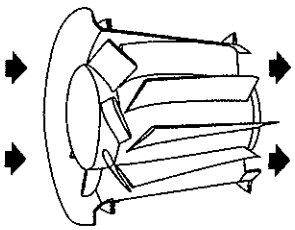
b. $$cfm_2 = cfm_1 * \sqrt{\frac{DENSITY_{STD}}{DENSITY_{ACT}}} \text{ or } cfm_1 * \sqrt{\frac{T_{ACT}}{T_{STD}}}$$

c. $$bhp_2 = bhp_1 * \sqrt{\frac{DENSITY_{STD}}{DENSITY_{ACT}}} \text{ or } bhp_1 * \sqrt{\frac{T_{ACT}}{T_{STD}}}$$

Centrifugal Fans: Impeller Comparisons

TYPE	CHARACTERISTICS	APPLICATION
Forward-Curved (FC) 	1. Best efficiency at low or medium pressure (approximately 0 - 5 in. wg). 2. Horsepower increases continuously with increase in air quality (overloads) as static pressure decreases. 3. Less expensive than BI, or AF fans. 4. Runs at relatively low speed, typically 800 - 1200 rpm. 5. Blades curve toward direction of rotation.	For low- to medium-pressure air-handling applications.
Radial (RA) 	1. Self-cleaning blades; dirt and dust do not deposit. 2. Horsepower increases with increase in air quantity (overloads) while static pressure decreases. 3. Operates at high speed and pressure, typically 2000-3000 rpm. 4. Blades radiate from center along radius of fan.	Dust or particle movement in areas such as woodworking shops. Ventilation or dusty environment.
Backward-Inclined (BI) 	1. Best efficiency at medium pressure. 2. Horsepower increases with increase in air quantity but peaks at a higher cfm capacity (non-overloading). 3. More expensive than FC fan. 4. Runs at high speeds, typically 1200-2400 rpm, about double that of FC fan for similar air quantity. 5. Blades curve away or incline from direction of rotation.	For medium-pressure air-handling applications.
Airfoil (AF) 	1. Best efficiency in high capacity and high-pressure applications (4-10 in. wg). 2. Horsepower peaks at high capacities. 3. Most expensive of centrifugal fans. 4. Operates at high speeds, typically 1500-3000 rpm. About double the speed of FC fan for similar air quantity. 5. Blades have aerodynamic shape similar to airplane wing and are backwardly curved.	For medium to high air capacity and pressure applications.

Axial Fans: Impeller Comparisons

TYPE	CHARACTERISTICS	APPLICATION
Propeller (Prop)  	1. High efficiency at free delivery (0 to ½ - in. wg). 2. Air delivery decreases with increase in air resistance. 3. Inexpensive. 4. Operates at relatively low speeds, typically 900-1800 rpm. 5. Blade rotation is perpendicular to direction of airflow.	Used on no-duct systems or low resistance systems.
Tube Axial Fan 	1. More efficient than propeller at high air volume. 2. Similar to propeller fan but blades may have aerodynamic configuration. 3. May require sound attenuation. 4. Operates at high speeds, typically 2000-3000 rpm. 5. Axial fans may be constructed to be overloading type or non-overloading. Non-overloading type is more common.	In-line or duct mounting for high air volume applications. May require sound attenuators to reduce noise levels.
Vane Axial Fan 	1. Similar to tube axial fan but has guide vanes on discharge side to improve efficiency. Vanes aid in redirecting air-leaving blades. 2. More costly than tube axial fan.	Same application as tube axial fan but with improved efficiency.

Work Session Answers

1. A forward-curved fan because the airflow through the fan increases at a constant rpm.

2. centrifugal, and axial

3. A forward-curved fan wheel is fabricated of lightweight and low cost materials, and has 24-64 shallow blades with both the heel and the tip of the blade curved forward. The airfoil blades are more ruggedly built, adding to their weight, and are curved backward with 8-18 blades.

4. 2.5 equivalent duct diameters.

5. System effect refers to the conditions in an actual duct system that affect fan performance and related testing, adjusting, and balancing work. It can be prevented to a degree, but must be accounted for.

6. L_{10} rated bearings have a 90% reliability of their stated amount of time, generally expressed in hours. That is, 90% of the bearings with L_{10} ratings will not have developed metal fatigue after their designated life span. L_{50} rated bearings have an 50% reliability of their stated amount of time; only 50% of a group of identical bearings with an L_{50} rating will not yet have developed metal fatigue.

7. cfm varies directly with fan rpm

   static pressure varies with the square of the fan rpm

   brake horsepower varies with the cube of the fan rpm

8. A plenum fan builds up static pressure in the plenum. It does not propel the air down a single duct opening. The contractor cuts discharge openings in the plenum and the air exits under the static pressure developed by the fan.

9. The filters load up (collect dirt) increasing pressure drop, which changes the resistance of the total system. Someone may change the position of a balancing damper, which changes the system curve.

10. A vane axial fan incorporates a straightening vane assembly. This helps to make it more efficient than the tube axial fan.

Prerequisites:

Form No.	Book Cat. No.	Instructor Presentation Cat. No.	Title
TDP-102	796-026	797-026	ABCs of Comfort
TDP-103	796-027	797-027	Concepts of Air Conditioning

Learning Objectives:

After reading this module, participants will be able to:

- Identify fans types that are used in the HVAC industry, their operating characteristics, and basic construction.
- Understand the application limitations for types of fan impellers.
- Utilize the fan laws to construct a system curve for a typical system.
- Identify stable fan selections using fan curves.
- Calculate the system effect for an example fan operating condition.
- Understand fan bearing life, fan drives, and fan isolation techniques.

Supplemental Material:

Form No.	Book Cat. No.	Instructor Presentation Cat. No.	Title
TDP-623	796-055	797-055	Water-Cooled Chillers
TDP-705	796-070	797-070	Chilled-Water Systems
TDP-504	796-045	797-045	Duct Design, Level 1: Fundamentals

Instructor Information

Each TDP topic is supported with a number of different items to meet the specific needs of the user. Instructor materials consist of a CD-ROM disk that includes a PowerPoint™ presentation with convenient links to all required support materials required for the topic. This always includes: slides, presenter notes, text file including work sessions and work session solutions, quiz and quiz answers. Depending upon the topic, the instructor CD may also include sound, video, spreadsheets, forms, or other material required to present a complete class. Self-study or student material consists of a text including work sessions and work session answers, and may also include forms, worksheets, calculators, etc.



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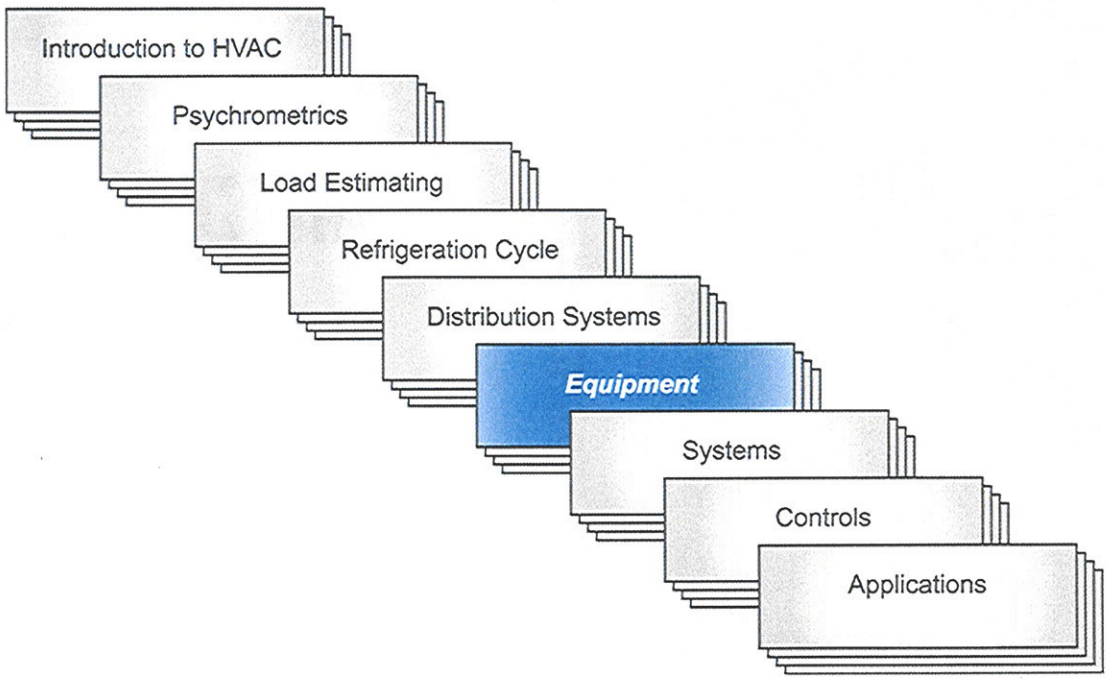
COMMERCIAL HVAC CHILLER EQUIPMENT

Air-Cooled Chillers

Technical Development Program

Technical Development Programs (TDP) are modules of technical training on HVAC theory, system design, equipment selection and application topics. They are targeted at engineers and designers who wish to develop their knowledge in this field to effectively design, specify, sell or apply HVAC equipment in commercial applications.

Although TDP topics have been developed as stand-alone modules, there are logical groupings of topics. The modules within each group begin at an introductory level and progress to advanced levels. The breadth of this offering allows for customization into a complete HVAC curriculum – from a complete HVAC design course at an introductory-level or to an advanced-level design course. Advanced-level modules assume prerequisite knowledge and do not review basic concepts.



Chillers are used in a variety of air-conditioning and process cooling applications. Air-cooled chillers can be used as a single piece unit or a split in various configurations. This flexibility has contributed to their overall popularity among designers of chilled-water systems. Air-cooled chillers range in size from small capacity models to several hundred-ton models that are utilized to cool large commercial buildings. This TDP module will cover both packaged single piece air-cooled chillers as well as split system types. This TDP module will also cover the available options and accessories, the applications, as well as criteria for selecting an air-cooled chiller.

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Introduction

Air-cooled chillers utilize the mechanical refrigeration cycle to produce chilled water or a chilled water and antifreeze mixture. They reject the building heat to the ambient with an air-cooled condensing coil. Chillers are the heart of the chilled-water air-conditioning system since they serve the pivotal function of creating the cooling effect required to maintain comfort conditions.

Chillers are used in a variety of comfort air-conditioning and process cooling applications. The chilled liquid is transported by pumps and pipes that can be connected to literally hundreds of room fan coils and terminals. This allows chillers to be applied on applications requiring many zones of control.

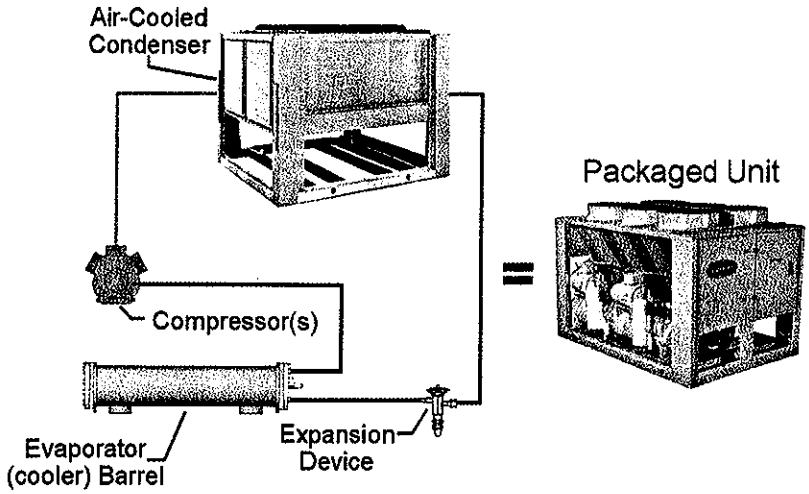


Figure 1

Air-Cooled Chiller Package

Air-cooled chillers are common in modern systems and have been growing in popularity since the 1980s, nearly doubling in the last ten years. Today, air-cooled chillers are applied on small to large commercial jobs and can be used in multiples to form systems reaching several thousand tons of installed capacity.

The popularity is primarily due to the all-inclusive nature of air-cooled chillers and the reduction of costs associated with eliminating cooling tower. On some models, even the hydronic accessories, such as the pump and expansion tank, may be included, assembled, and tested from the factory ready to begin operation.

Typical air-cooled chiller applications include schools, hospitals, retail environment, and offices. Additionally, air-cooled chillers are popular for cooling process or manufacturing operations.

This TDP will cover packaged single-piece, as well as split system air-cooled chillers. To learn more about water-cooled chillers, refer to TDP-623, Water-Cooled Chillers.

Air-Cooled versus Water-Cooled Chillers

A differentiating feature of the types of chillers is the method used to condense the refrigerant as it leaves the compressor. The two methods involve using either air-cooled or water-cooled condensers. Air-cooled condensers employ ambient air as the condensing medium and use a fan to move the air over the coil. For a given surface and airflow rate, the capacity of an air-cooled condenser varies with the refrigerant condensing temperature, which is a function of the entering dry-bulb temperature. Shown in Figure 2 is a typical air-cooled condensing temperature based on 95° F dry bulb ambient air.

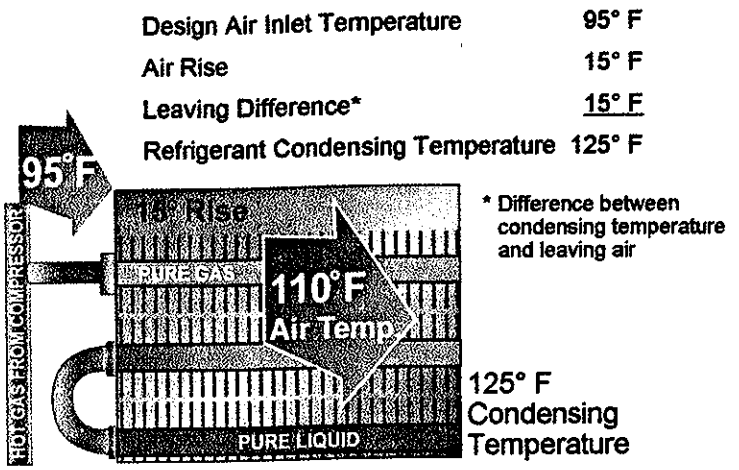


Figure 2

Air-Cooled Condensing Temperature

Water-cooled condensers employ water as the condensing medium and use a pump to circulate the water through the condenser and out to a cooling tower that rejects the heat to the atmosphere.

Operating cost is one of the primary factors when deciding between air-cooled or water-cooled chillers. Air-cooled chiller systems typically have a lower first and maintenance cost since they do not require a cooling tower, condenser water pumps, and associated condenser water chemical treatment. Operating costs, however, generally favor water-cooled chillers. This is because water-cooled chillers can take advantage of lower condensing temperatures than air-cooled chillers.

Air-cooled chillers have a full load kW/ton of approximately 1.25 while water-cooled chillers have a full load kW/ton of between 0.55 and 0.8 kW/ton. The kW draw of the cooling tower fans and condenser water pump should be added to the water-cooled chiller kW/ton for an even comparison. Even after accounting for this added auxiliary energy draw, water-cooled chilled-water systems normally have an efficiency advantage over air-cooled.

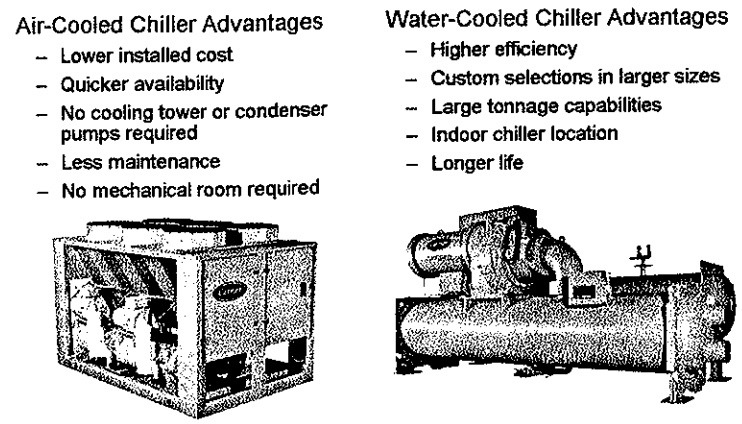


Figure 3

Air-Cooled vs. Water-Cooled Chiller Benefits

Air-cooled chillers

are typically available as a package ranging up to 500 tons, while packaged water-cooled chillers are typically available up to 3,000 tons. Some custom water-cooled chillers go even higher in capacity.

Air-cooled chillers eliminate the concerns and maintenance requirements associated with condenser water treatment, condenser-tube cleaning, cooling tower service, tower freeze protection, and the availability and quality of makeup water. This reduced maintenance requirement is particularly attractive to building owners because it can reduce overall costs. Water-cooled chillers must have a condenser water treatment program to eliminate contaminants such as bacteria and algae growth. Fouled condenser tubes, not uncommon with water-cooled chilled-water systems, can also reduce chiller efficiency.

Air-cooled chillers are often selected for use in systems that require year-round mechanical cooling requirements. Air-cooled condensers have the ability to operate in below-freezing weather, and can do so without the freeze protection issues associated with operating the cooling tower in these conditions. Cooling towers often require a basin heater, or even an indoor sump, for safe operation in freezing weather. For process applications, such as computer centers that require cooling year-round, air-cooled chillers have a distinct advantage over their water-cooled counterpart.

Note:

The primary convenience of an air-cooled chiller is that it does not require a cooling tower, condenser water pump, and the associated maintenance.

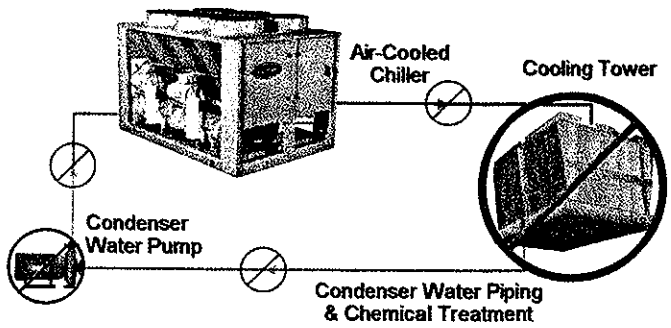


Figure 4

Condenser water systems are not required with air-cooled chillers.

Basic Refrigeration Cycle for Air-Cooled Chillers

The refrigeration cycle of an air-cooled chiller includes two important processes:

1. The evaporation of the liquid refrigerant in the evaporator, which absorbs heat and lowers the temperature of the chilled-water system
2. The condensation of the refrigerant vapor in the air-cooled condenser and rejection of heat to the atmosphere

In the air-cooled chiller refrigeration cycle, water enters the evaporator (also known as the cooler) and is cooled by the colder refrigerant flowing through the other circuit inside the evaporator. The chilled water is pumped from the chiller to the building coils to provide cooling. In the evaporator, the chilled water cools the building or process load and the cycle is completed when warmer water flows back to the evaporator. A mixture of cold liquid refrigerant and flash gas passes through the evaporator circuit opposite the water to be chilled.

The refrigerant in the evaporator absorbs heat from the warmer return water, evaporates to a vapor, and finally exits the evaporator as a superheated vapor. The superheated refrigerant vapor then enters the suction inlet of the compressor. In the compressor, the refrigerant is compressed, raising its pressure and temperature. High pressure and temperature refrigerant gas exits the compressor, passes through the discharge line and enters the condenser. While in the air-cooled condenser coil, the hot gas condenses to liquid inside the tubes as it gives up heat to the cooler outside air being drawn across the condenser coil by the condenser fans.

The refrigerant

is on the shell side of some evaporator designs and on the tube side of others. When the refrigerant is on the tube side, the evaporator is called a DX evaporator. When on the shell side, it is a flooded cooler.

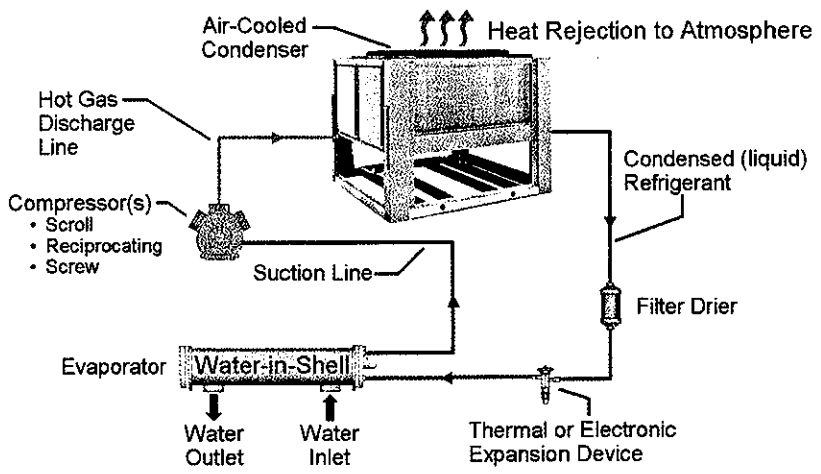


Figure 5

Refrigeration Cycle Components of an Air-Cooled Chiller

The condensed liquid refrigerant then leaves the condenser and enters the expansion device. As the refrigerant passes through the expansion device, its pressure and temperature is decreased to the point that some of the liquid flashes to vapor. The expansion device controls the amount of flashing in order to maintain a certain superheat to ensure no liquid droplets enter into the compressor suction. After leaving the expansion device, the refrigerant enters the evaporator and the cycle is repeated.

Air-Cooled Chiller Components

All air-cooled chillers are comprised of the following components:

- Evaporator (cooler)
- Condenser
- Compressor
- Expansion Device

Let's examine these components individually and the role of each in the air-cooled chiller.

Evaporator

Three types of evaporators are used in air-cooled chillers:

1. Brazed-plate
2. Direct expansion (DX) shell-and-tube
3. Flooded shell-and-tube

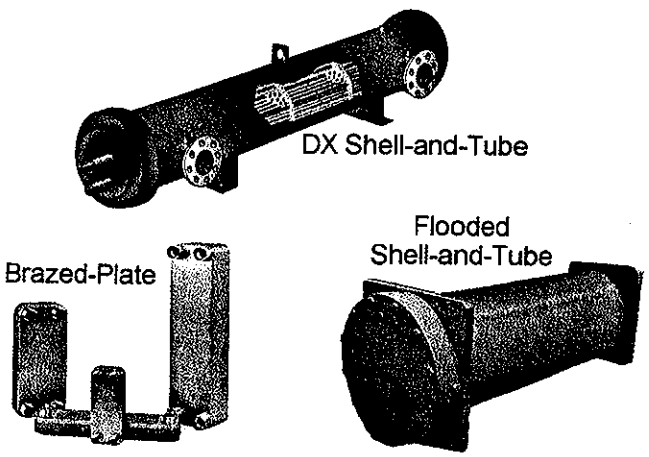


Figure 6

Three Types of Evaporators
Photos: Brazed-plate courtesy of API Heat Transfer; DX Shell-and-Tube

The most common choice of evaporator design for larger screw and reciprocating chillers has been the shell-and-tube types while smaller chillers utilize brazed-plate. Shell-and-tube evaporators are widely used and they are larger and heavier than brazed-plate heat exchangers. The tubes for some evaporator types may utilize internal and/or external enhancements to promote turbulence and create increased heat transfer between the fluid and the refrigerant. Turbulence actually increases the heat transfer process by keeping more of the fluid in contact with the heat exchanger surface area. On the refrigerant side and water side, more contact means greater heat transfer and a higher efficiency rating. Enhancement of tubes typically is achieved through a rifling or finning process during manufacturing.

Brazed-Plate

Brazed-plate evaporators are used on chillers up to approximately 60 tons and consist of a series of plates brazed together with every second plate turned 180 degrees. This design creates two highly turbulent fluid channels that flow in opposite directions over a surface area with a high heat transfer coefficient and good performance characteristics. The plates are stacked so they form a multi-layered alternating path for each fluid to travel through. Each layer or circuit is linked to an inlet and outlet via a manifold at either end.

A brazed-plate evaporator offers good heat transfer for the cost. In the multiple layers, or circuits, the fluid comes into contact with the heat exchanger surface over a very large area resulting in fast heat dispersion. The smaller cross sectional area and fluted construction also adds turbulence to the fluid, further increasing the efficiency of heat transfer. The fluid turbulence is also present at low heat exchanger loadings, making the brazed-plate heat exchanger ideal for part-load applications. Brazed-plate heat exchangers are much smaller than their shell-and-tube counterpart. They may be as little as 25 percent of the size of an equivalent shell-and-tube heat exchanger.

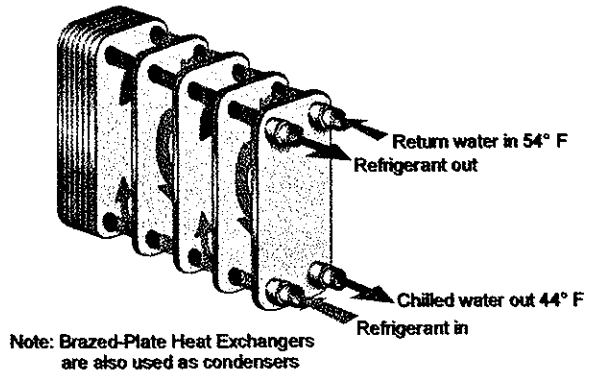


Figure 7

Brazed-Plate Evaporator

Plate type heat exchangers *are also used on some air-cooled chiller designs as a method of increasing performance via an economizer circuit. The purpose is to increase subcooling and thus increase the overall capacity of the refrigeration system.*

This smaller size of brazed-plate heat exchangers results in a smaller overall footprint for the chiller, a benefit for both shipping and installing the chiller. Additionally, the refrigerant circuits can be designed to require considerably less refrigerant than shell-and-tube systems. Because the circuitry of a brazed-plate heat exchanger system holds less volume of refrigerant, fewer pounds will be needed to charge the chiller.

Given the small areas of the multiple circuits, brazed-plate heat exchangers are more susceptible to being plugged. A factory-supplied or field-supplied strainer should be used to remove particles and contaminants from entering the water circuit of the heat exchangers. The brazed-plate design is not mechanically cleanable inside.

Direct Expansion (DX) Shell-and-Tube

In the direct expansion evaporator, the refrigerant flows through the tubes and the water flows around the tubes in the shell section. As heat is transferred to the colder refrigerant from the warmer water or water / anti-freeze mixture, the refrigerant in the tubes evaporates and the water is cooled. Baffles within the shell direct the water flow path over the tubes creating the turbulence that improves heat transfer.

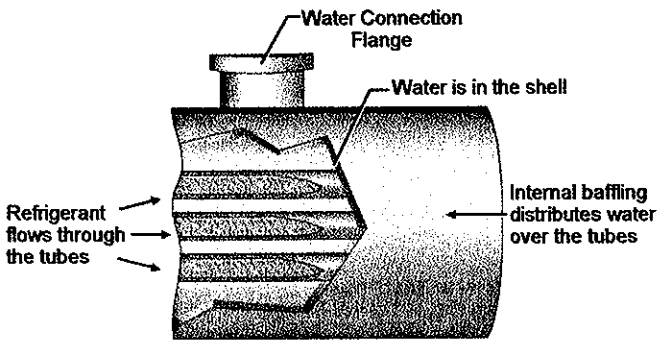


Figure 8

Direct Expansion Shell-and-Tube Evaporator

Air-cooled chillers with reciprocating and scroll compressors typically utilize a DX evaporator design. The direct expansion type evaporator allows the oil to be swept along with the refrigerant inside the tubes to a location on the chiller where it can be separated and returned to the oil pump thus accomplishing the required lubrication these compressors require.

The waterside

of a direct expansion evaporator cannot be chemically cleaned since mechanical cleaning of the shell is not possible due to the internal baffling.

Flooded Shell-and-Tube

Flooded shell-and-tube evaporators are also called "flooded coolers." They have water or a water/antifreeze mixture flowing through the tubes while the refrigerant is on the shell side. The liquid refrigerant is uniformly distributed along the bottom of the cooler.

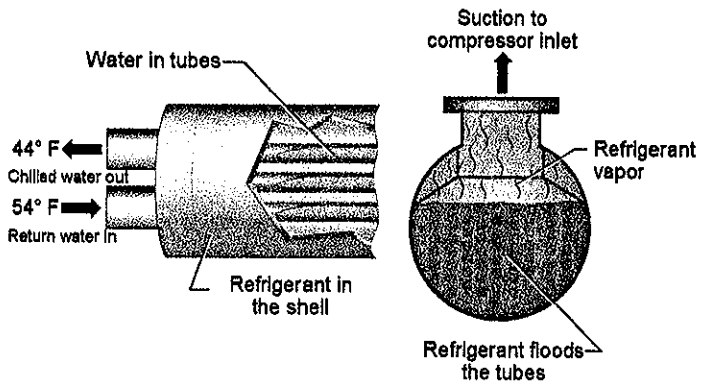


Figure 9

Flooded Shell-and Tube Evaporator

Additionally, the refrigerant charge does not have to be removed during tube cleaning. Flooded evaporators are typically no more efficient a design than DX or brazed-plate, but the chiller types they are used on tend to have a more efficient compressor, such as a centrifugal.

On air-cooled chillers that use flooded shell-and-tube evaporators, the number of tube passes within the vessel may be changed to further increase the efficiency by increasing the tube velocity and turbulence. Or, they may reduce the number of passes to decrease the pressure drop by reducing the tube velocity. Although higher pass arrangements tend to improve heat transfer, they also increase pumping horsepower for the chilled-water circuit because of the higher pressure drop. Thus, an economic balance must be struck between higher pumping horsepower and lower compressor power.

Flooded Evaporator Cleaning:

Since the water flows through the tubes of a flooded evaporator, the waterside can be mechanically cleaned. Brushing is a common method. The ability to mechanically clean the waterside of a tube shell and tube evaporator is a major advantage of this design.

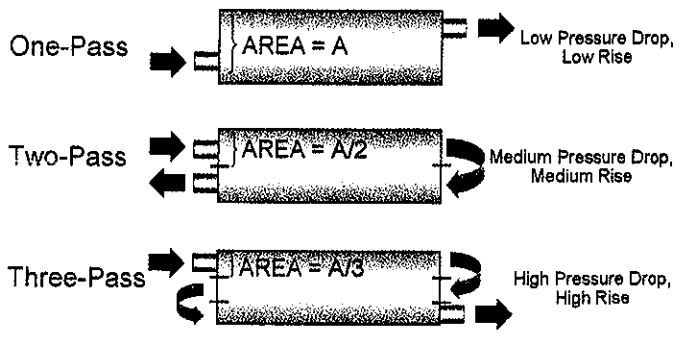


Figure 10

Flooded Evaporator Pass Arrangements

On DX evaporators, there are no "pass" arrangement options because the water is on the shell side. Internal to the shell, changes to the baffling are made to accommodate greater or less water flows.

Evaporator Passes

The method most utilized in selection of heat exchanger passes is to specify reasonable pressure drop limits and allow the manufacturer to recommend the optimal pass arrangement for the load and application under consideration.

Evaporator Types Pros & Cons

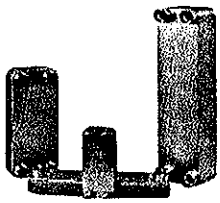
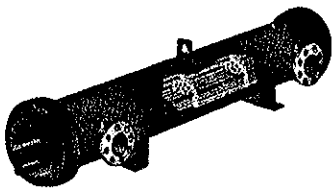
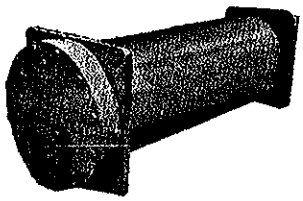
Brazed-Plate Evaporator** 	Pro	Smaller footprint
	Pro	Lightweight
	Pro	Cost effective for smaller sizes
	Con	Sensitive to contamination of the water circuit
	Con	Not mechanically cleanable
DX Shell-and-Tube Evaporator* 	Pro	Cost effective for small and medium sizes
	Pro	Familiarity with customers/engineers
	Pro	Design can accommodate heater cable on the shell to prevent freezing of water
	Con	Larger footprint
	Con	Not mechanically cleanable
Flooded Shell-and-Tube Evaporator 	Pro	High efficiency
	Pro	Mechanically cleanable
	Con	Large footprint
	Con	Design cannot accommodate heater cable on the shell to prevent freezing of water in tubes

**Photo courtesy of Standard Refrigeration*
***Photo courtesy of API Heat Transfer*

Condenser

The condenser in an air-cooled chiller is a heat exchanger that condenses the hot refrigerant gas by using cooler outside air passing over a coil. In a typical air-cooled chiller, propeller-type fans are used to draw the outdoor air over the coil. The hot, high-pressure refrigerant vapor flows through the coil tubes and is condensed to a liquid. In the last portion of condenser tubing, the condensed liquid refrigerant is then subcooled. A final portion of the condenser is used to lower the refrigerant temperature below the saturated condensing temperature. This is called subcooling. This increases the refrigeration effect and increases system capacity by about 1 percent for each degree of subcooling provided.

Altitude Adjustment

The airflow over the condenser coil is fixed, but the density of the air is affected by altitude, which affects condenser performance. Chiller selection software will ask for project elevation to adjust the capacity accordingly.

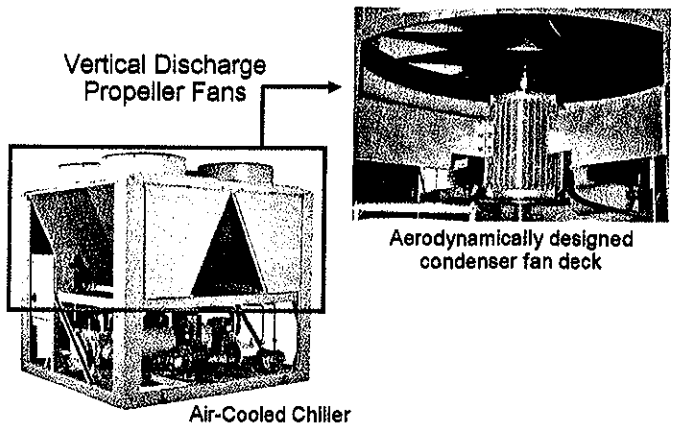


Figure 11

Condenser Design of an Air-Cooled Chiller

Compressor

There are three types of compressors utilized for air-cooled chillers. They are reciprocating, scroll, and screw.

Reciprocating

Reciprocating compressors, like a reciprocating engine, have pistons, rods, and discharge and intake valves. The valves operate on suction and discharge pressure. Compression is achieved by trapping a fixed amount of refrigerant gas into a chamber. For this reason, reciprocating compressors are positive displacement type compressors as are scrolls and screws. Reciprocating compressors were the workhorse of the small chiller market for decades and are still used in many designs. Multiple reciprocating compressors were often installed in a

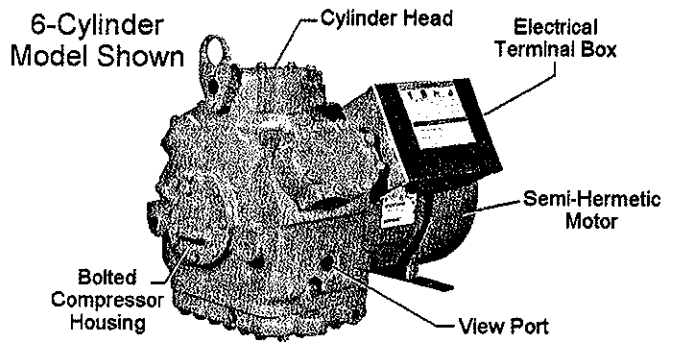


Figure 12

Semi-Hermetic Reciprocating Compressor

single chiller to provide chiller capacities of up to 400 tons. The reciprocating compressor is available open, semi-hermetic, or hermetic. Open compressors extend the shaft through a seal in the crankcase for use with an external driver. Hermetic compressors have the motor and compressor contained in the same housing, typically welded and sealed. Semi-hermetic compressors utilize a bolted housing, instead of a welded one, to facilitate repair in the field.

Scroll

Scroll compressors have a unique compression process. A fixed scroll coupled with the movement of an orbiting scroll, ingests suction gas into pockets. As the orbiting scroll moves, the pockets of gas are compressed to an intermediate pressure. In a final orbit, the pockets reach discharge pressure and exit through the discharge port.

Scroll compressors have emerged as a popular alternative to reciprocating compressors and are available in hermetic configurations in capacities up to 25 tons each for use in air-cooled chillers. As with reciprocating compressors, multiple scroll compressors are often used in a single chiller design to meet larger capacities. Scroll compressors are 10 to 15 percent more efficient than reciprocating compressors and have proven to be very reliable, primarily because they have approximately 60 percent fewer moving parts. Reciprocating and scroll compressors are typically used singly or in multiples to provide an air-cooled chiller range from approximately 10 to 400 tons.

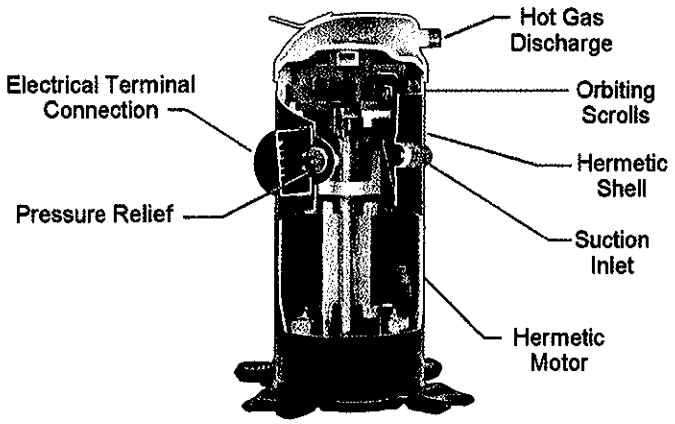


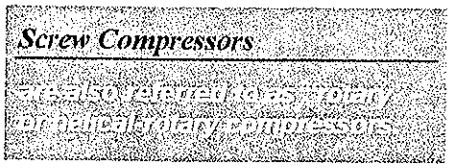
Figure 13

Scroll Compressor

Screw

Screw compressors are widely used in medium-sized water chillers, either singly or in multiples to accomplish a chiller range from about 70 to 500 tons. As the screw rotors turn, the suction gas is compressed. Screw compressors have one or more rotors to accomplish the compression. Like scroll compressors, screw compressors have a reliability advantage over reciprocating due to fewer moving parts. Screw compressors, as well as reciprocating and scroll compressors are positive displacement type compressors, which means they compress a fixed amount of gas in a compression chamber. Screw compressors have become popular because of excellent reliability and good efficiency, especially at part loads.

There are several designs of screw compressors featuring single or multiple rotors. Screw compressors unload by slide valve or solenoid valve mechanism that changes the amount of gas volume being compressed, thus reducing the capacity.



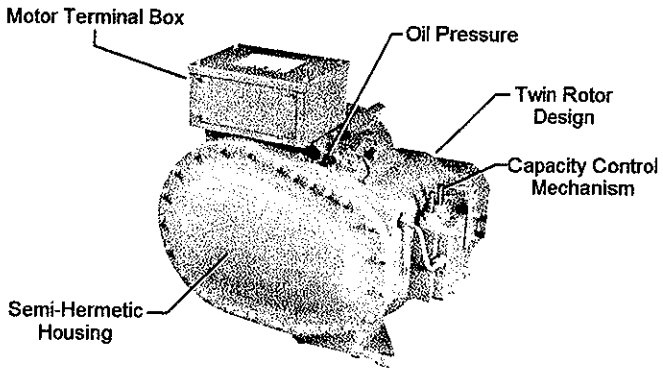


Figure 14

Double-Rotor Screw Compressor

Expansion Device

An expansion device is used to maintain the pressure difference between the high-pressure (condenser) and low-pressure (evaporator) sides of the refrigeration system. This pressure difference allows the evaporator temperature to be low enough to absorb heat from the water being cooled while also allowing the refrigerant to be at a high enough temperature in the condenser to reject heat to the outside air. High-pressure liquid refrigerant flows from the condenser through the expansion device, where it undergoes a pressure drop that reduces the refrigerant pressure to that of the evaporator. This pressure reduction causes a small portion of the liquid to boil off, or flash, cooling the remaining refrigerant to the desired evaporator temperature. In addition to maintaining the high-to-low side pressure differential, the expansion device maintains the proper suction superheat of the vapor entering the compressor, which enables all of the liquid to be vaporized in the evaporator. Heat can still be added to the vapor after it has completely boiled in the evaporator. This additional heat insures no liquid will enter the compression suction. This is referred to as superheat.

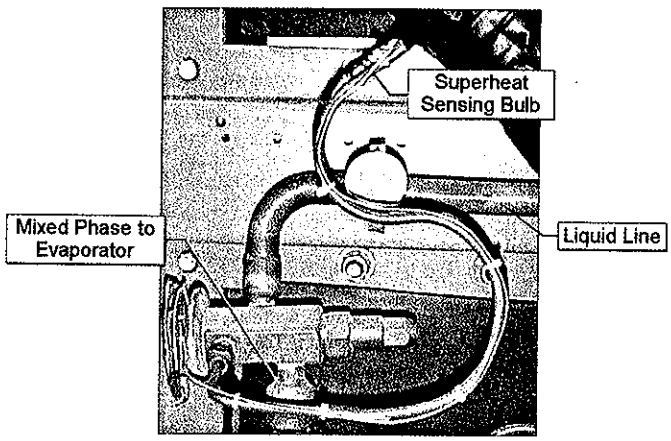


Figure 15

TXV Expansion Device Mounted Horizontally

Air-cooled chillers typically use either thermostatic expansion valves (TXV) or electronic expansion valves (EXV). A TXV consists of a regulating valve and connected sensing bulb. The sensing bulb is placed on the suction line leaving the evaporator. The TXV senses the refrigerant temperature in the evaporator and the refrigerant-filled bulb senses the temperature of the superheated refrigerant leaving the evaporator. This allows the TXV to regulate flow through the evaporator to maintain a set amount of superheat at the exit of the evaporator.

An EXV utilizes external temperature and pressure sensors to measure the superheat and then adjust the flow through the valve using a finely controlled stepper motor. Although the costs are greater for an EXV than a TXV, the precise refrigerant flow that they afford and their extremely rapid reaction to changing conditions allow for lower compressor head pressure resulting in greater chiller efficiencies.

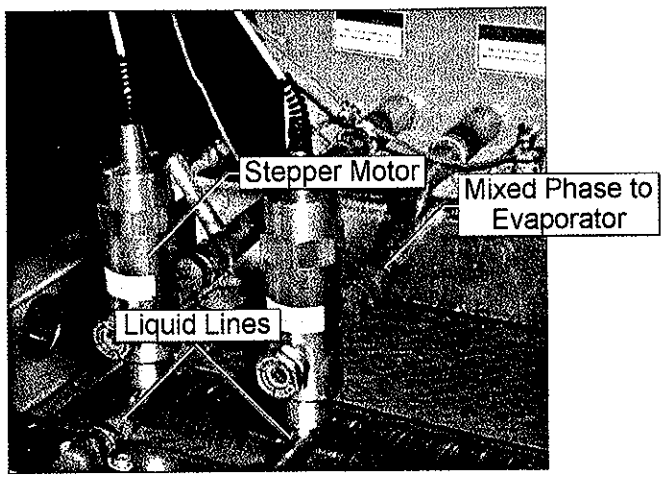


Figure 16

EXV Expansion Device

Some advantages of an EXV are:

- Creates low head pressures (increase part load efficiency)
- Has great range and flexibility of operating conditions
- Provides precise capacity control
- Uses predictive algorithms that can avoid potential operational problems

As an example, an air-cooled chiller applied on a comfort cooling application, would not be allowed to produce low leaving chilled-water temperatures that could cause potential freeze-up.

Note:

For a detailed discussion of TXVs and EXVs, refer to "IDP-103: Expansion Devices and Refrigeration Specialties"

Chiller Controls

All chillers need to be properly controlled for safety and efficient operation. Today's chiller equipment is making extensive use of technology to provide this control. A chiller today uses microprocessors, electronic sensors, and digital user interfaces that have changed the look of chiller equipment control systems. This technology is referred to a Direct Digital Control (DDC).

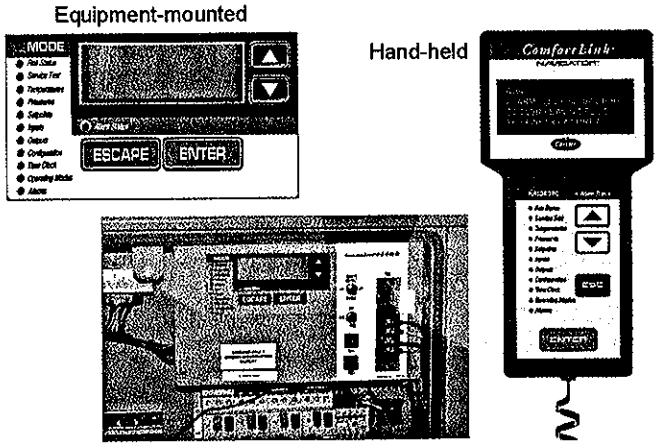


Figure 17

Air-Cooled Chiller Control Panel User Interfaces

DDC system user interfaces give owner-operators and service technicians the ability to do diagnostic evaluation of the operation of the equipment using either equipment-mounted or handheld displays. In DDC systems, information is digital and can be shared by other equipment to function as a system. Most DDC systems will have a communication terminal that allows for network communications or the attachment of a PC for more detailed diagnostic capabilities. DDC systems are capable of sharing information on a communication bus that can be viewed worldwide.

Compressor Starting Methods

A compressor motor draws high starting current called "inrush" or "locked rotor current" when started because the motor rotor is not yet rotating. This high current is required to generate the magnetic field to start the rotor turning and is directly dependent upon both the efficiency and horsepower of the motor. High current draw during a compressor start can cause the line voltage to dip and that may adversely affect other equipment within the facility. This is related to the size and efficiency of the motor. Current draw restrictions may require that larger motors start with reduced inrush current. This is called soft start.

Across-the-Line Starting

Across-the-line starters

On multi-compressor chillers, across-the-line starters can be used because the inrush for compressors is manageable and the compressors are not started all at once.

The simplest and least expensive form of motor starter is an across-the-line starter, which consists of a contactor and an overload protection relay. The contactor consists of a coil and contacts that close the circuit thus applying power to the compressor motor. The overload protection relay opens the circuit to the motor if excessive current is drawn while the motor is running. Upon start, the contactor is closed, applying full line voltage to the motor windings. The motor will draw a very high locked rotor inrush current for a very short time. As the motor accelerates, the current will begin to drop, but not significantly until the motor is about 85 percent of full speed. The actual starting current curve is a function of the motor design and the terminal voltage, but is independent of the motor load. The motor load will affect the time taken for the motor to accelerate to full speed and, therefore, the duration of the high starting current, but not the magnitude of the starting current. For a typical chiller compressor motor, the inrush or locked rotor current with across-the-line starting will be approximately 600 percent of the motor rated load amperage (RLA) rating.

Advantages of across-the-line starting are lower costs and simplified controls. Motors using across-the-line starting will usually be up to speed quickly. However, high current inrush can affect other equipment in the plant.

Part-Winding Starters

A part wind start is a type of soft starting. An electrical engineer typically determines if soft starting is required based on an electrical system study that calculates voltages drop in the system when a load is started. If the power supply is weak or the system is heavily loaded, the system is

more likely to be sensitive to motor starting and soft starters are more likely to be required. 208V systems are more likely to require soft starters than 460V systems because of the higher amp draw.

Part-winding starters are a cost effective means of providing a reduced current start where smaller horsepower motors are utilized like on many air-cooled chiller designs using multiple compressors.

Part-winding starters can only be used with part-winding motors that have two sets of identical windings that are intended to be operated in parallel. During a part-winding start, only one winding is energized, reducing the inrush current to 60-70 percent of normal starting values with both windings energized. With only one winding in the circuit, a typical part-winding motor will not accelerate to more than half of the motor rated speed. Because the motor is operated with one winding only during the initial acceleration period, after the transition from start to run mode, the current draw of the motor will be closer to the rated load amps. A part-winding starter essentially provides a two-step start.

Wye-Delta Starting

Wye-delta starters are also used on applications that require reduced inrush current as described above. Wye-delta starters may be applied in applications where larger horsepower motors are utilized, such as a larger screw compressor air-cooled chiller.

Wye-delta motors can only be used with wye-delta starters. These motors have six leads allowing for the motor windings to be connected in either a wye, where one end of each of three windings is attached to a common point and the other end of each is attached to the line, or delta, where one end of each of two windings is attached to each line connection. During start-up, the windings are connected in the wye configuration that reduces inrush current to 33 percent of the delta-connected values. After a set time delay, the motor leads are reconfigured to the delta connection for normal run operation. When using a wye-delta starter, in the delta connection the running current is shared between two contactors.

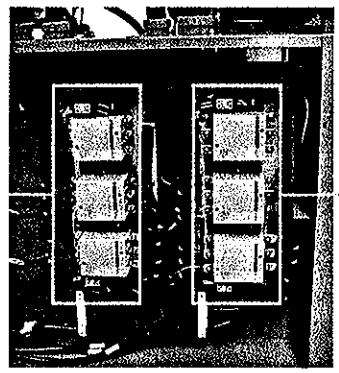


Figure 18

Wye-Delta Starters on a Two-Compressor Screw Chiller

Starting Method	Motor Starting Current as a % of	
	Locked Rotor Current	Full Load Current
Across-the-Line	100	600
Part Wind	65	390
Wye-Delta	33	200

Figure 19

Starter Summary Information

Capacity Control

Like most chillers, air-cooled chillers operate occasionally at full capacity conditions, due to building load variations. Therefore, it is important to understand the three primary methods of capacity control:

- Unloading the compressors
- Cycling the compressor(s) on and off (single or multiple compressors)
- Hot gas bypass

Unloading the compressors is possible with reciprocating and screw compressor types. Scroll compressors do not unload; they are either on or off. Reciprocating compressors utilize unloaders that lift the suction gas valve so that the pistons do not compress the gas. Unloaders are a cost-effective method of reducing capacity, but because the pistons continue to operate at all times, the energy efficiency using unloaders is not as high as with the cycling method. Screw compressors utilize a slide valve to cover the suction inlet ports, effectively shortening the length of the screws and reducing the resulting amount of compressed refrigerant vapor. Capacity control is not continuous with slide valves as the valves cover or uncover the suction ports similar to staging the compressor.

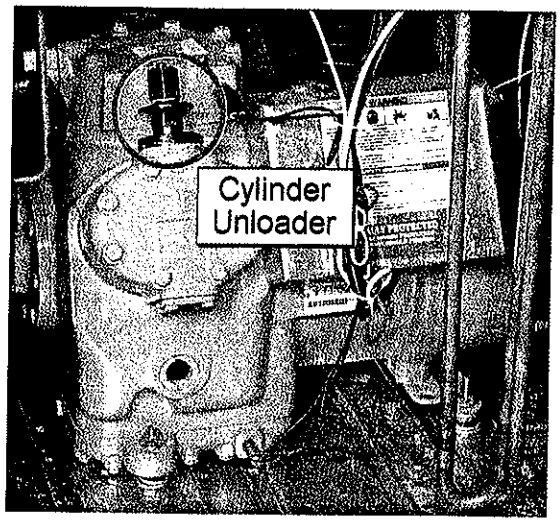


Figure 20

Unloading Reciprocating Compressor

Cycling the compressor(s) is a cost-effective and energy-efficient method for reducing capacity, especially when multiple compressors are used in the system. However, cycling the compressors too frequently may result in compressor motor damage and failure, thus controls are built into the chiller to control the time between cycles. Larger chiller designs that use multiple compressors combined with multiple refrigeration circuits can successfully cycle the compressors to achieve part-load capacity control. During peak loads, all compressors and circuits are operational. During lighter loads, individual compressors on one circuit may be cycled off thus providing an acceptable part-load profile.

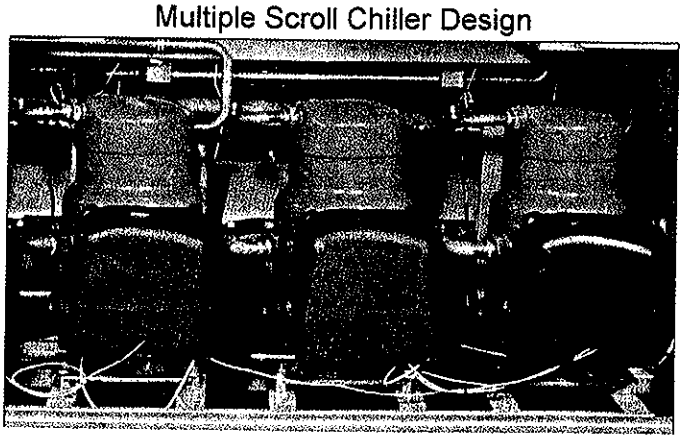


Figure 21

Capacity Control – Cycling Compressors

Hot gas bypass is a method that bypasses some of the high-pressure refrigerant gas (hot gas) discharged from the compressor back to the evaporator line without going through the condenser. Naturally, this affects the system capacity, as some of the refrigeration is effectively "lost" from the cycle. The capacity of the chiller on hot gas bypass will be somewhat lower than the capacity at the lowest step of unloading. However, use of hot gas bypass is not considered efficient since the compressor continues to operate while a significant portion of the system does not provide any cooling.

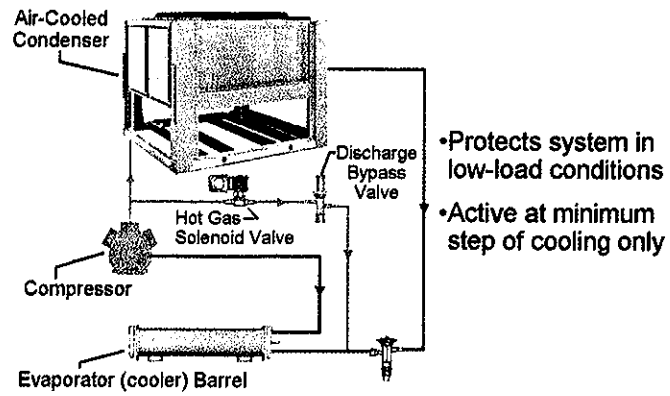


Figure 22

Hot Gas Bypass

Hot gas bypass is often used on a chiller with limited unloading capability.

Energy Management

Use of energy management practices can significantly reduce operating costs, especially during off-peak modes of operation. Chilled-water temperature reset and demand limiting are two techniques for accomplishing efficient energy management.

Chilled-Water Reset

Chilled-water reset means to change the chilled water temperature leaving the chiller based on some parameter. Increasing the leaving chilled water temperature reduces compressor power usage by reducing lift at both full and part-load conditions. However, at part-load conditions, design chilled water temperature may not be necessary, especially for comfort cooling applications, so reset is possible. Increasing the chilled water temperature may result in greater humidity levels in the conditioned spaces. Higher coil temperatures resulting from the increased chilled water temperature will reduce the latent heat capacity of the coils and the ability of the air distribution system to remove space humidity.

Chilled water temperature can be reset as a function of:

- Ambient air temperature (most common) – used when the ambient temperature is the best indication of load
- Return chilled water temperature – used when return water temperature is the best indication of load. This method is typically used when it is desired to maintain a fixed Δt in the chiller plant.

Note:

Lift refers to the difference between compressor suction and condensing temperature. Reducing lift increases efficiency.

- Temperature within the building – used when space temperature is the best indication of load on the chiller. This is used when there are critical areas in a building such as labs that might require a specific temperature be maintained.

Reset doesn't always mean an increase in water temperature. For example space temperature reset could actually lower the chilled water supply temperature, not raise it, in order to maintain conditions.

Demand Limit and Duty Cycling

Demand limit is a feature that limits the unit capacity and power draw during periods of peak energy usage. When a utility company's demand for electricity exceeds a certain level, users are charged extra money called demand charges. To avoid these charges, loads in the building are limited to keep demand below a prescribed maximum level. Demand limiting may be desirable on hot days when air conditioning is most needed. Demand may be limited on the chiller by increasing the chilled-water temperature or by unloading the chiller to a predetermined percentage of the full current draw which is an indication of load. This may result in temporary increased building or process temperatures if the actual load is greater than the demand limited capacity that the chiller can provide.

Demand

is measured in kW; energy charges are measured in kWh. Together, the two form a typical user's electrical bill.

Duty cycling will cycle selected electrical equipment in an installation (building, factory, etc.) at regular intervals to limit the electrical demand, thereby lowering demand charges. However, duty cycling is not recommended because constant cycling will cause increased stress and damage to the motor windings, bearing, and controls. If demand must be controlled, the demand limit sequence available from the manufacturer is recommended.

Air-Cooled Chiller Configurations

There are several configuration options for designing an air-cooled chiller system. The design can choose to use a packaged system (all refrigeration components in one unit) or a split system (refrigeration components separated and located in different areas). Packaged systems are simpler to design, install and maintain. However, due to cold weather conditions, it may be more desirable to locate the evaporator inside the building to prevent freezing. Air-cooled chillers offer the flexibility of separating the components. This flexibility allows the system design engineer to place the components where they best fit the available space, while maintaining the acoustic and performance requirements of the customer.

Single Piece Units

A packaged air-cooled chiller has all of the refrigeration components located outdoors in a single chassis. A major advantage of this configuration is factory assembly and testing of all components, including the wiring, refrigerant piping, and controls. This eliminates field design and labor while often resulting in faster installation and improved system reliability. Additionally,

all noise-generating components (compressors and condenser fans) are located outdoors, easing indoor noise concerns. Finally, the MER (mechanical equipment room) space requirements are minimized. With all of the components located outdoors, below-freezing ambient temperature conditions must be considered for certain climates. Many engineers incorporate a glycol concentration in the chilled water for freeze protection for outdoor chillers. Adding a glycol concentration will derate the performance of a chiller compared to using fresh water. The capacity derate depends on concentration and chiller type. See the freeze protection methods described in detail later in this TDP.

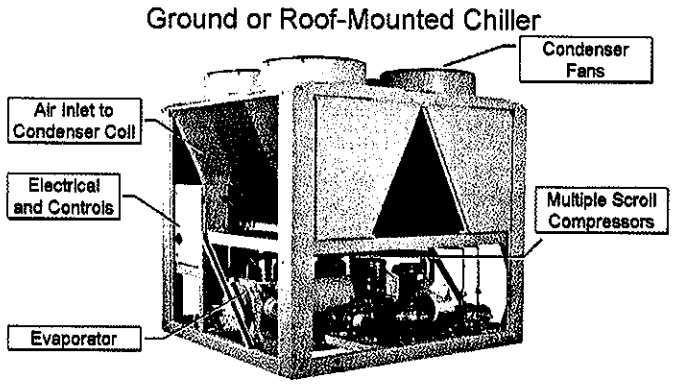


Figure 23

Single Unit Configuration

Split Systems

An alternative to the packaged air-cooled chiller is to use a split system. There are several variations of the split system.

Remote Evaporator

One configuration of the split system is to have the compressor and condenser located outside the building while the evaporator is remotely mounted in an indoor location. The indoor and outdoor components are then connected with field-installed refrigerant piping. This configuration locates the part of the system that is susceptible to freezing (evaporator) indoors and the noise-generating components (compressor and condenser fans) outdoors. This configuration usually eliminates any further requirements to protect the chilled-water loop from freezing during cold weather.

However, the evaporator will take up some indoor space. Additionally, as there is now refrigerant piped inside the building, ASHRAE 15 Standard for mechanical equipment rooms must now be considered. See a discussion of ASHRAE Standard 15 in the Codes and Standards section of this TDP. Performance of the chiller will be slightly affected because of refrigerant line losses between the compressor/condenser and the evaporator.

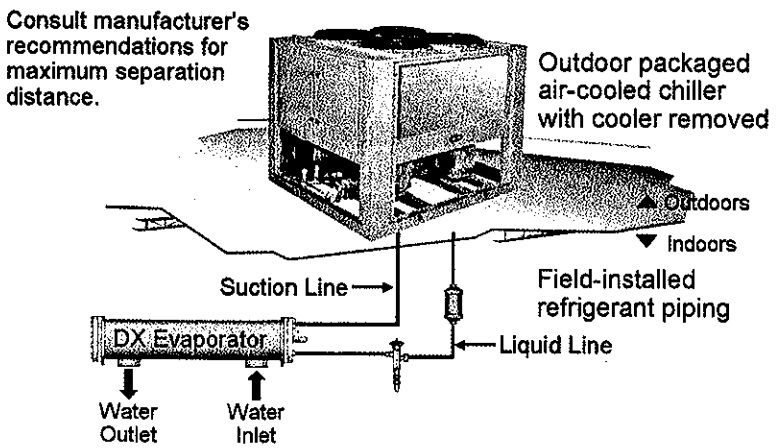


Figure 24

Remote Evaporator

There are greater chances for refrigerant leaks with this configuration as the interconnecting piping is field-installed. Also, within the long piping runs to the remote evaporator, proper oil return to the compressor must be assured by sizing the lines correctly. For more information on sizing split system piping, consult TDP-501, Refrigerant Piping.

Condenserless Chiller

Another configuration is to place only the air-cooled condensers outdoors. The compressors, controls, and evaporator remain indoors in what is called a condenserless chiller. Scrolls, reciprocating, and screw chillers are all available in condenserless split system configurations. These split system chiller configurations start about 15 tons and go over 260 tons per chiller. A large benefit for mounting the compressors, evaporator, and controls indoors is for ease of maintenance and troubleshooting in inclement weather. Some other benefits of this split system chiller configuration are:

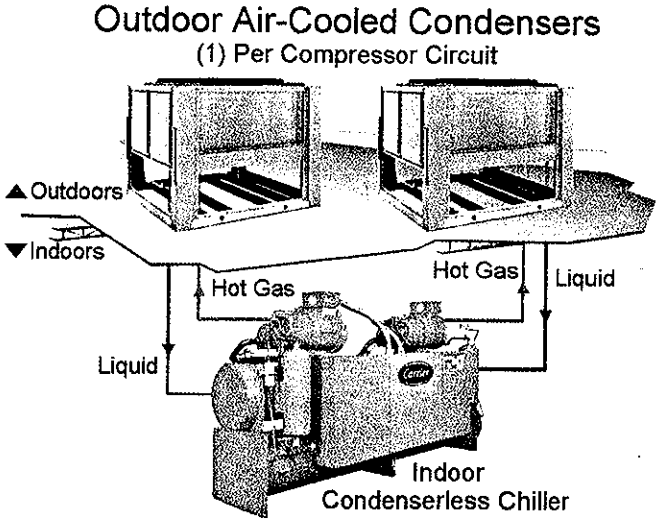


Figure 25

Condenserless Split System Chiller

- The roof-mounted condenser(s) do not weigh as much as the compressor section so the structure need not support the single piece air-cooled chiller
- The main electrical users (compressors) are closer to the electrical panel (which is typically near the mechanical equipment room), resulting in lower electrical wiring costs.
- There are more options for sound and noise control with indoor compressors.

This configuration is particularly popular in schools and other institutional applications, primarily due to reduced seasonal maintenance for freeze protection.

The maximum allowable refrigerant line length is less than 100 feet for condenserless air-cooled chiller installations. This is dependent upon the manufacturer, however.

As with the remote evaporator design, there is now a refrigerant vessel inside the building and ASHRAE 15 requirements should be followed. In most all cases with an indoor chiller and outdoor condenser, the chiller must be housed in a mechanical equipment room. Capacity of the split system chiller will be slightly less than a fully packaged outdoor unit due to refrigerant line losses.

Split Systems

A drawback of splitting the components is the requirement for field-installed refrigerant piping. Additionally, longer time is generally required for the proper selection, sizing, and installation.

Indoor Centrifugal Condenser

The next split system configuration includes the condenserless chiller located in an indoor equipment room connected to an *indoor* air-cooled condenser. The air used for condensing is ducted from outdoors through the condenser coil and rejected either outdoors or inside the building as a means for heat recovery. Indoor condensers use a centrifugal fan to overcome the duct static-pressure losses, rather than the propeller fans used in conventional outdoor air-cooled condensers. The components are connected with field-installed piping. This configuration is used where an outdoor condenser is architecturally undesirable, where the system is located on a middle floor of a multistory building, or where vandalism to exterior equipment is a problem. A disadvantage of this configuration is that increased condenser fan energy is required to overcome the duct resistance compared to a conventional outdoor propeller fan air-cooled condenser that has no ductwork to overcome. As with all split systems incorporating field-installed piping, there is a greater chance of refrigerant leaks and due to the increased volume of piping, more refrigerant will be required. Additionally, ASHRAE 15 Standards are applicable. Additional consideration should be given to the indoor condenser fans with respect to generation of sound.

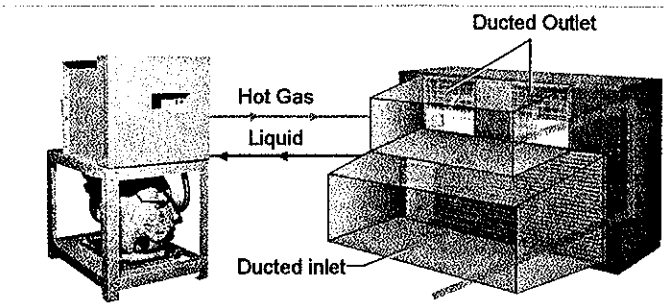


Figure 26

Indoor Chiller, Indoor Centrifugal Chiller Condenser

Subcooling

Chiller manufacturers often require a specified amount of subcooling at the evaporative condenser's. The chiller performance will be adversely affected if the chiller requires subcooling and the evaporative condenser manufacturer does not include it.

Evaporative Condenser

The last application for condenserless chillers is to match them with an evaporative condenser. Evaporative condensers use a spray on the coil to provide lower condensing temperatures due to the evaporative effect. This increases split system efficiency. Because a manufacturer other than the chiller manufacturer typically supplies the evaporative condenser, so care must be taken in the selection. For complete information on evaporative condensers, see TDP-641, Condensers and Cooling Towers.

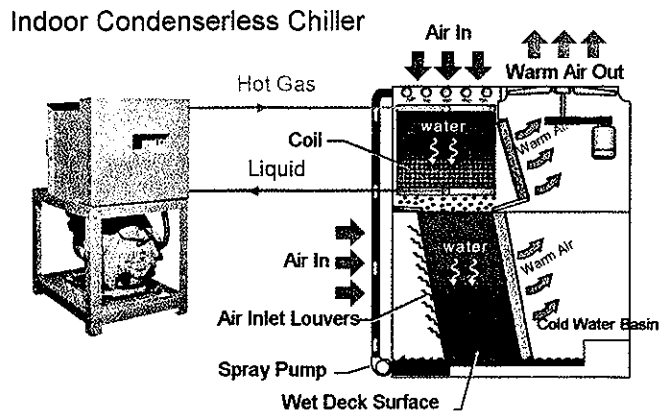


Figure 27

Indoor Chiller, Evaporative Condenser

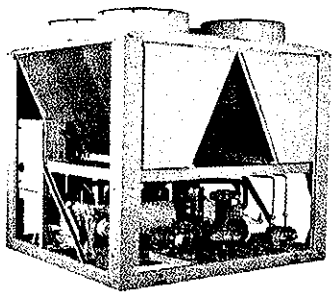
Application Topics

Typical Operating Limits

There are a number of parameters that define the operating envelope for an air-cooled chiller. Typical representative values are shown in Figure 28 and may vary slightly from manufacturer to manufacturer.

Operation to –20° F will require the addition of a low ambient head pressure control accessory. Operation to 15° F leaving water (brine) temperature will require freeze protection. See the freeze protection methods section in this TDP module.

Check with manufacturer for unit specifics



- Minimum Outdoor Temperature -20° F (Standard unit 0° to 32° F)
- Maximum Outdoor Temperature 125° F
- Minimum Leaving Fluid Temperature 15° F (Standard unit 40° F)
- Maximum Leaving Fluid Temperature 60° to 70° F
- Maximum Return Water Temperature 85° F

Figure 28

Typical Operating Limits

Effects of Outside Air Temperature

Air-cooled chillers utilize ambient air to condense the refrigerant. However, as the outside air temperature varies, so does the efficiency and performance of the chiller. As the chilled water leaving the chiller is set to maintain a constant temperature, increases in the ambient temperature result in a higher condensing temperature, larger temperature differentials and greater lift across the compressor. Greater lift requires more work and results in lower efficiency and performance from the chiller.

The climactic design dry bulb temperature is 95° F for most of North America. The resulting design condensing temperature is approximately 125° F. Compressor efficiency and capacity declines as ambient air temperature rises.

Typical Air-Cooled Chiller Full Load

Entering Air °F	Tons	kW/Ton	% Change Capacity	% Change Efficiency
75° F	112.1	0.952	+15	+20
85° F	105.5	1.069	+8	+11
95° F	97.5	1.208	–	–
105° F	89.5	1.384	-8	-15
115° F	81.2	1.621	-17	-35
125° F	73.1	1.923	-25	-60

Figure 29

Example: Air-cooled chiller effect of ambient at full load and 44° F leaving chilled-water temperature.

Effects of Leaving Chilled-Water Temperature

As a general statement, it is economical for the chiller to produce chilled water no colder than is required to maintain space relative humidity conditions. For comfort cooling applications, 44° F or 42° F leaving chilled water is normally adequate. Hospitals often use 42° F because of high outdoor air loads and a greater need to dehumidify. Large chilled-water plants with extensive piping runs have used leaving water temperatures lower than 42° F to minimize pumping costs and offset pipe heat gain. However, these larger plants are typically not the ones that would tend to use air-cooled chillers. Water-cooled centrifugals would be used most often in large chilled-water plants.

Typical Air-Cooled Chiller at 95° F Ambient

Leaving Chilled Water Temperature °F	kW / Ton	% Change Efficiency per °F	% Change Efficiency From 44° F
44	1.237	–	–
43	1.253	-1.1	-1.1
42	1.266	-1.0	-2.2
41	1.280	-1.1	-3.5
40	1.295	-1.2	-4.6

Figure 30

Effects of Lowering the Leaving Chilled-Water Temperature

Shown in Figure 30 is an air-cooled chiller model of slightly different full load efficiency from the one used in Figure 29. From Figure 30, we can see the effects of producing colder leaving chilled water.

The question sometimes arises whether or not there is an advantage to produce and circulate colder water than the 44° F to 42° F range. For colder water designs to make sense, the chiller penalty must be more than offset with chilled water pumping cost reductions resulting from lower flows and the higher rise in the water.

As a general rule of thumb

for each degree lower leaving chilled-water temperature produced, there is between a 1 - 2% efficiency loss of the chiller.

Variable Flow Operation, Minimum and Maximum Flow Rates

Variable flow operation refers to varying the chilled-water flow rate through the chiller evaporator based on the cooling load in order to minimize water-pumping costs. In effect, the chilled-water flow rate is reduced when cooling loads are lower and increased when the loads are greater. The maximum gpm would tend to correspond to the gpm at design load.

Modern electronic chiller controls have allowed for variable evaporator flow applications because of the ability of the controls to respond to changing conditions. However, there are both minimum and maximum values for water flow through the evaporator that apply. If the flow rate through the evaporator becomes too low, the temperature differential between the entering and leaving water becomes too large and may result in leaving water temperatures approaching freezing. Additionally, too low of a flow rate may result in a loss of turbulence that would significantly reduce the heat transfer.

In flooded shell-and-tube evaporators

a typical minimum flow is that which corresponds to 3 feet per second water velocity in the tubes. This is obtainable from the air-cooled chiller selection software. For DX evaporators, the minimum flow will often be published in the product data, and is based on 1 fps water velocity in the shell.

Conversely, flow rates that are too high result in pressure losses within the evaporator that are too great. Maximum cooler flow is often approximated at 5 gpm/ton. Another way to determine maximum cooler flow is that flow resulting in a 5° F Δt in the chilled water. Ratings for air-cooled chillers are based on a 10° F Δt and are generally suitable for a range of 5 to 15° F Δt without adjustment. To accommodate special high flows, some manufacturers offer a low pass option to keep the waterside pressure drop manageable.

For variable flow applications, the rate of change of the chilled-water flow is very important. A maximum rate of change of flow of 10 percent of full flow, per minute, is a suggestion. Always consult the manufacturer's selection software or literature for minimum gpm for variable flow system application.

Recirculation and Bypass Line Piping

When the system water flow is out of the range of the chiller, two piping arrangements can be used. The first piping arrangement recirculates excess flow from the evaporator if the flow rate required by the chiller is too great for the system. The second piping arrangement utilizes a bypass flow around the evaporator of the chiller if the flow required in the system is greater than the normal flow through the chiller.

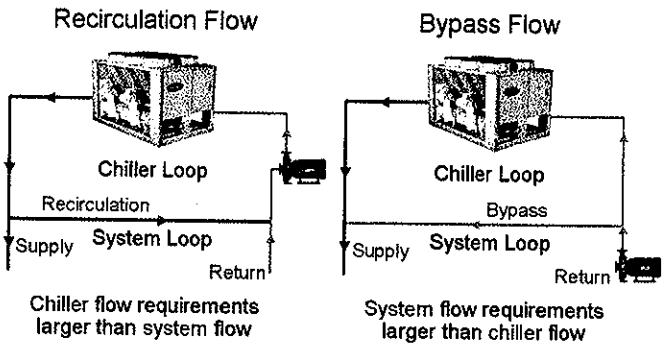
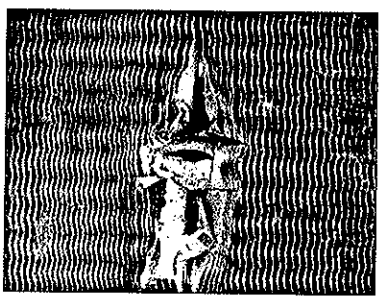


Figure 31

Recirculation and Bypass Line

Freeze Protection Methods

Because air-cooled chillers are usually located outside, evaporators require some procedures to prevent freezing. In areas where the ambient temperature will drop below 32° F, any unprotected water will freeze. Lack of freeze protection can result in consequences ranging from a single burst tube in an air handling unit coil piped to the chiller, to replacement of an entire chiller due to damaged components like evaporator tubes or compressors. Freezing can lead to extensive downtime and cleanup.



Burst Coil

Cracked Housing



Figure 32

Freeze Damage

Here are the suggested freeze protection methods.

Drain the Chiller Loop

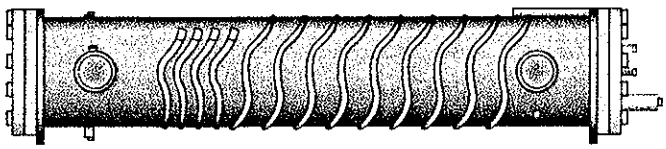
If the unit will not be operational, water can be drained from the exposed piping and evaporator during the off-season. Draining of the chiller loop is limited to applications where requirements for mechanical cooling do not exist while ambient is below freezing for extended periods of time. With increasing use of free outside air-cooling, this is a plausible approach. However, as not all of the water is guaranteed to be completely drained, an antifreeze solution should still be added to prevent freezing of the residual water left in the evaporator.

Use Evaporator Barrel Heaters

Electric heaters installed on exposed pipes and the evaporator barrel is a cost effective solution that requires little maintenance. Chiller manufacturers usually supply evaporator barrel heater options, but the chilled-water piping to the evaporator must be taped with heat trace in the field. However, dependence on electrical power is the key drawback. If an ice storm or a circuit breaker interrupts power to the unit for a long enough period of time, freezing can result. This method is often combined with continuous water circulation.

Evaporator barrel heaters

are usually supplied with a separate power supply and disconnect in order to stay operational when the power to the chiller is cut via the main chiller disconnect such as when service is performed on the chiller.



- **Refrigerant in tubes**
- **Water (glycol) in shell**
- **Heater cable wrapped around the DX Evaporator shell**
- **Factory or field-installed**

Figure 33

Evaporator Barrel Heater

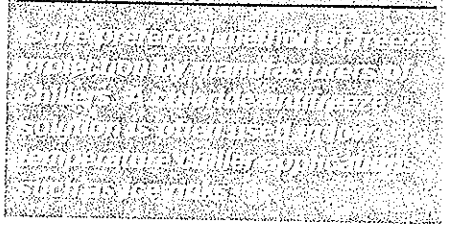
Circulate Water Continuously

In this case, the chilled-water pump is energized to circulate the water in the loop to prevent it from freezing. The idea is, moving water will not freeze as quickly. The pump is activated when the cooler temperature gets near or below freezing. This method depends on pump operation and on electrical power being available. Water circulation is often used in conjunction with barrel heaters.

Use an Antifreeze (Glycol/Water Solution)

Glycol is often added to the chilled-water loop and is the preferred method of freeze protection by chiller manufacturers for comfort air-conditioning applications and some low temperature designs also. It is the most reliable system protection, as it does not require electrical power to be effective. It also protects chilled-water coils and other piping that might be exposed to temperatures below freezing. However, adding a glycol solution to the chilled-water system will decrease performance of the chiller because the glycol increases fouling, compressor lift, and pumping power. The quantity of glycol used in the solution should be designed for a temperature 15° F below the minimum anticipated temperature to which the chiller will be exposed. The minimum glycol concentration suggested should be approximately 20-25 percent, but lower concentrations are also used. This is the decision of the local design engineer.

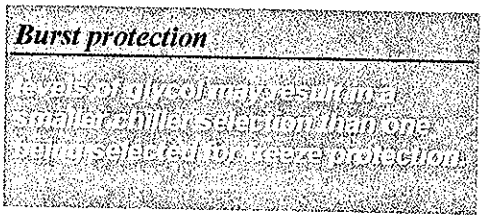
The use of a glycol/water solution



Glycol/water antifreeze solutions are capable of providing two levels of protection for the chilled-water system: burst protection and freeze protection. Burst protection allows for the fluid in the chilled-water loop to become slushy, but never to fully freeze solid to the point of rupturing or collapsing tubing. Freeze protection requires the glycol concentration be greater than for burst protection so the chilled fluid constantly remains in a liquid state. In essence, if the unit will not be operational during the cold weather, burst protection may be adequate, as the glycol solution slush will not be pumped throughout the system. However, if the chiller is to be operational during cold weather, freeze protection is necessary because the pumps require the solution to be in a liquid state.

To provide burst protection, the antifreeze concentration only needs to be great enough to prevent the mechanical damage that would happen if the fluid were to solidify. The formation of crystals is not a problem because the fluid is not being pumped through the system when burst protection measures are taken. Burst protection requires less antifreeze than freeze protection. Burst protection is suitable for chilled-water systems that are dormant in the winter. The lower concentration of glycol for burst protection results in lower costs for initial fill of the system and lower system operating costs due to improved heat transfer in the heat exchanger.

Burst protection



Propylene Glycol		
Protection Temperature (°F)	For FREEZE protection ** % by weight	For BURST protection ** % by weight
+20	18	12
+10	29	20
0	35	25
-10	41	29
-20	45	30
-30	48	32
-40	52	34

**Confirm with the local glycol supplier

Figure 34

Freeze versus Burst Protection

As previously stated, the formation of ice crystals is not desirable when the pump or pumps are circulating the fluid at low fluid temperatures. Examples of such applications are ice storage and process cooling systems. The formation of crystals in fluids being pumped could result in erosion of the pump, piping, valves, etc. Typically, these systems comprise the minority of air-cooled chiller installations.

The local antifreeze manufacturer should be consulted for final recommendations based on their formulation of the heat transfer antifreeze fluid. Each antifreeze solution has physical characteristics that determine the percentage of antifreeze in the solution at which the freezing point or first crystals point occurs. This can be found in reference books such as the Carrier System Design Manual Part 4 and publications from antifreeze manufacturers like Dow Chemical Company, etc. Refer to those publications for the freezing point or first crystals point. Antifreeze also has a temperature at which the solution becomes a solid. This information can be found in publications from the antifreeze manufacturers. The temperature at which the antifreeze/water solution becomes a solid is the lowest temperature for burst protection. Above that temperature, the antifreeze/water solution is still slush (a combination of fluid and crystals) and will not burst or collapse the tubes or shell of the heat exchangers.

Refrigerants Used in Air-Cooled Chillers

Currently manufacturers of air-cooled chillers utilize R-22 refrigerant in some of their models. R-22 has been the refrigerant of choice for many years. However, this situation is changing. R-22 contains chlorine which, when released to the atmosphere, has detrimental environmental effects. In addition to R-22, newer chiller designs utilize R-134a, which is used in larger centrifugal chillers and has no chlorine. Also, new blends have been developed for use in air-cooled chillers, like R-410A (Puron™). R-407c is also used.

By the year 2010, R-22 will no longer be shipped in chiller product.

With the year 2010 in mind, manufacturers will be modifying their products in phases to the new environmentally sound refrigerants mentioned.

Note:
The phase-out date for R-22 is 2010 when used in brand new equipment and 2020 for service.

It should be understood that it is not possible to simply replace R-22 in an existing chiller with one of the new blends. The new refrigerant may have far different pressure – temperature and heat of vaporization characteristics than R-22, and major modifications to the chiller would be required and is not practical.

Please consult TDP-401, Refrigerants for a full discussion concerning refrigerants.

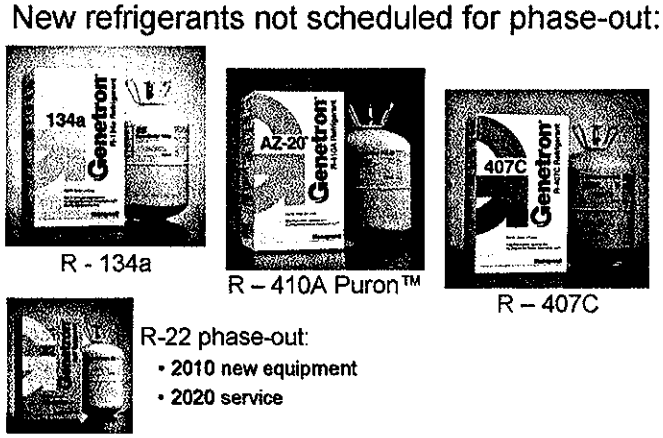


Figure 35

Refrigerants Used in Air-Cooled Chillers

Chiller Sizing

Over sizing chillers by more than approximately 15 percent at design conditions must be avoided, as the system operating efficiency will be adversely affected (resulting in greater and/or excessive electrical demand and cycling of compressors). When future expansion of equipment is anticipated, install a single chiller to meet present load requirements, and install a second chiller to meet the additional load demand.

Oversizing
is especially not recommended if the chiller does not have multiple compressors or some unloading capability.

It is also recommended that the installation of two smaller chillers be considered where operation at minimum load is critical. The operation of two small chillers at higher loading is preferred to operating single chiller at or near its minimum recommended value.

Minimum Chilled-Water System Volume

It is important to understand the relationship between loop volume of the chilled-water system and chiller operational stability. Loop volume is the amount of fluid in the cooler, chilled-water piping, cooling coils and optional storage tank that remains in circulation at all times. If the loop volume is too small, rapid changes in water temperature and short cycling of the compressor can occur. In effect, adequate loop volume acts as a "flywheel" so that the chiller cannot cycle off too quickly after cooling down the water. To minimize cycling, the minimum loop volume should be at least three gallons per nominal ton chiller capacity for normal comfort cooling duty applications. Always follow individual manufacturer guidelines for minimum loop volumes. For batch, process, and loads that operate under variable flow conditions, the minimum loop volume should be from six to ten gallons per ton of installed chiller. Applications that operate under low ambient temperatures or use brine typically also require six to ten gallons per nominal ton. Again, all recommendations are manufacturer dependent.

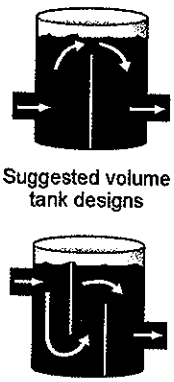


Figure 36

Minimum Chilled-Water System Volume

A method to increase the loop volume of a chilled-water system is the addition of a volume tank.

For example, suppose a chiller installation with a 200-ton capacity chiller has a system loop volume of 400 gallons. The recommended minimum loop volume would be:

(200 nominal tons) * (3 gallons/nominal ton)= 600 gallons

If the loop volume was calculated to be 400 gallons and the minimum required loop volume 600 gallons, a volume tank holding a minimum of 200 gallons would be required. Alternatively, pipe size may be increased to satisfy the loop volume requirements.

Parallel and Series Chillers

Where chiller capacities greater than can be supplied by a single chiller are required, or where stand-by capability is desired, chillers may be installed in parallel. Units may be of the same or different sizes. Usually, equally sized chillers are utilized to accomplish commonality of parts and maintain simplicity. If unequal sized chillers are used, cooler flow rates must be balanced to ensure proper flow to each chiller. Software is available from the chiller manufacturer that automatically stages multiple chillers of equal or unequal size.

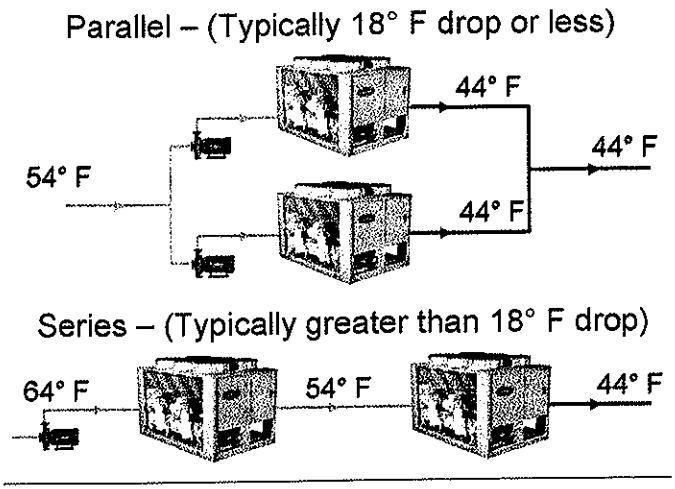


Figure 37

Parallel and Series Chillers

Future Expansion Arrangements

Parallel chiller arrangement lends itself to future expansion because another chiller can simply be added to the "ladder". Two chillers in a series cannot accommodate future expansion since pressure drop in the evaporator circuit would become too great with three chillers.

Where a large temperature drop (greater than 18° F) is desired, chillers may have to be installed in series. This is a rule of thumb. Actual chiller selections should be made to confirm when series chillers are needed. The cooler minimum entering fluid temperature limitations should be considered for the chiller located downstream of other chiller. Use of reduced pass configuration may be required to keep waterside pressure drop at an acceptable level.

Clearances and Installation

There are several design considerations when planning to install an air-cooled chiller. First, the chiller must have adequate clearances on all sides to accommodate the necessary airflow. All chiller manufacturers specify the minimum acceptable clearances in order for the chiller to operate correctly and allow for maintenance. Additionally, the National Electric Code (NEC) has specifications regarding the minimum clearance to electrical panels. Also, external piping should not be placed where it could interfere with the maintenance of the chiller.

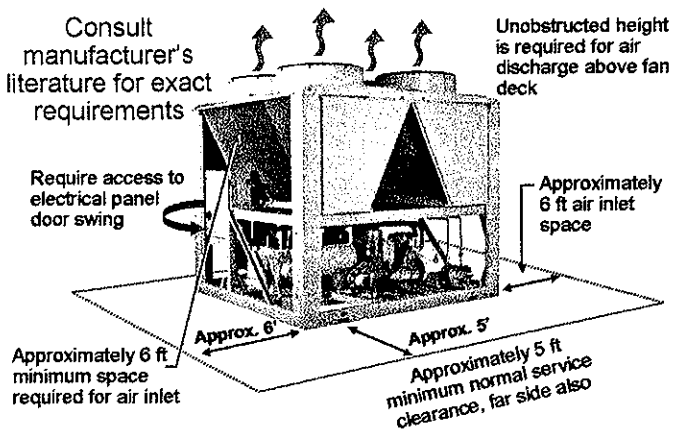


Figure 38

Typical Clearances – Air-Cooled Chiller

Air-cooled chillers generate noise from the condenser fans and compressors that may be an issue in certain environments, such as schools. In these cases, placement behind sound-blocking fences or walls would be required. Sound reduction accessories, as well as security covers for the condensers grilles, may also be used in sensitive areas.

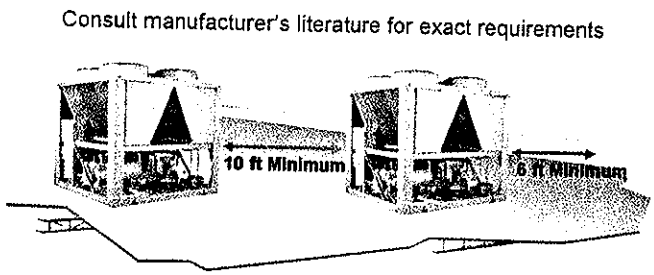


Figure 39

Multiple Unit Separation

Options & Accessories

Some of the options and accessories we will discuss for air-cooled chillers include low-ambient operation, low chilled-water temperature option, condenser coil corrosion protection, and a hydronics package. Typically, options are factory-installed additions to the chiller while accessories are field-installed. Following is a discussion of the more common ones.

Low-Ambient Operation

If the chiller is required to produce chilled water while operating in low ambient environments, head pressure control is required. Air-cooled chillers typically maintain head pressure control in cold weather by cycling and modulating condenser fans. As fans are cycled off due to a reduction in load or ambient temperature or both, the airflow rate through the condenser decreases resulting in elevation of condensing temperatures to correct levels. Depending on the load and unit design, cycling of fans will allow the chiller to operate satisfactorily down to approximately 35° F ambient temperature. If the chiller must operate at ambients below that temperature, then a factory-installed or field-installed option can be used to modulate the speed on the last operating condenser fan. This variable fan speed system allows for stable operation of the chiller for ambient temperatures as low as –20° F depending upon the manufacturer.

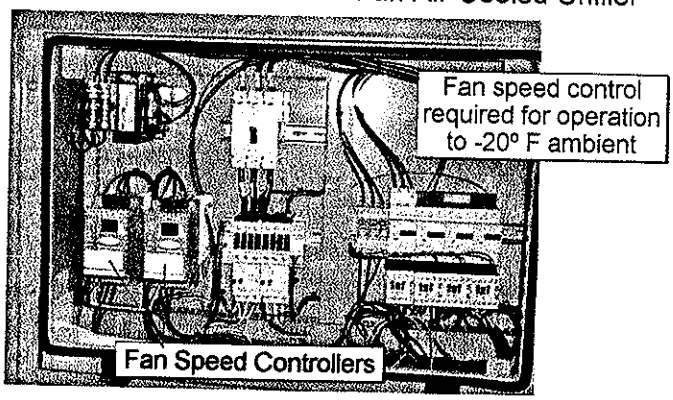


Figure 40

Low-Ambient Operation

Head pressure control on an air-cooled chiller is the regulation of the condenser fans to maintain correct condensing temperature as ambient temperature changes.

In addition, wind baffles are required for chillers operating in ambients below approximately 32° F. Wind baffles are field-fabricated and installed sheet metal covers placed over the condenser coils that allow air movement through the coil while preventing prevailing winds from affecting the chiller's condensing temperature.

Low Chilled-Water Temperature Option

Some typical applications require chilled water well below 40° F, such as the creation of ice for ice rinks. Many of the same procedures for freeze protection discussed earlier also apply to chillers intended for low chilled-water temperature operation. Typically, a antifreeze solution is used in the chilled-water system and some form of condenser head pressure control is used, generally through condenser fan speed reduction. Chilled-water temperatures leaving the cooler can be reduced to as low as 15° F through the use of an antifreeze solution. If the chiller is required to produce leaving fluid temperature as low as 15° F, a factory modification to the unit is required.

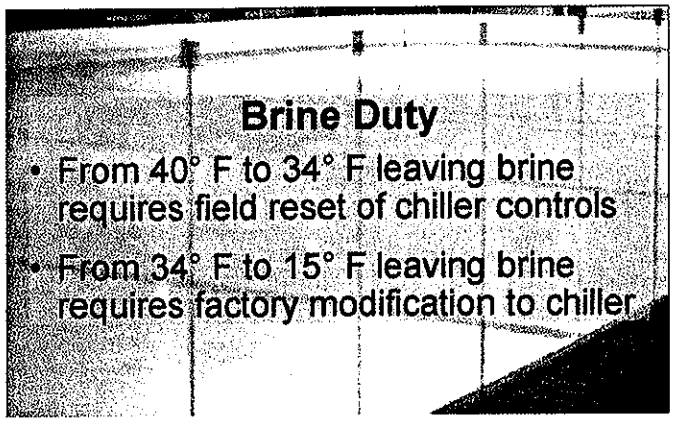


Figure 41

Low Chilled-Water Temperature Option

The term "brine chiller" is often used to designate chillers that are modified for low chilled-water temperature operation. The term "brine" dates back to the early days of fluid cooling when either a sodium chloride/water solution or calcium chloride/water solution was used as the antifreeze or heat transfer solutions. Because both of these were salt solutions, they came to be referred to as brine. That term has continued to be used today for sodium chloride/water solutions, calcium chloride/water solutions, and for glycol solutions even though glycol is technically not a brine.

Condenser Coil Corrosion Protection

By definition, corrosion is the destruction or deterioration of a metal or alloy through chemical, physical, or electrochemical reaction with the environment. As applied to air-cooled chillers, environmental exposure can lead to localized and/or general corrosion of the condenser coil. Improperly protected coils in corrosive environments can lead to premature performance degradation, unsightly surface conditions and, under the most severe conditions, equipment failure. This can result in increased operating, maintenance, and service costs. Fortunately, corrosion can be minimized in most cases with proper coil protection.

Measures can be taken to diagnose potentially corrosive environments prior to equipment selection. With this information, the best-suited coil protection can be applied to minimize the harmful effects of corrosive atmospheres and ensure long equipment life.

Potentially corrosive outdoor environments include areas near the seacoast, industrial sites, heavily populated urban areas, some rural locations, or combinations of any of these environments. Local environments must also be considered, such as close proximity to laundry facilities, diesel-burning devices, exhaust piping, sewer vents, traffic, etc.

The following list shows coil construction along with materials commonly used to combat the effects of corrosive atmospheres for condenser coils on air-cooled chillers.

- Standard coil (copper tubes, aluminum fins)
- Pre-coated coil (copper tubes, aluminum fins)
- All copper coil (copper tubes, copper fins)
- Electro-coated coil (copper tubes, aluminum fins)

Standard Coil Construction

The standard coil construction for air-cooled chillers is copper tubes with aluminum fins. This generally provides the highest performance for non-corrosive environments (e.g., non-polluted rural environments). Application of this coil in any environment containing corrosive constituents is not recommended due to the likelihood of visible deterioration resulting from corrosion. However, standard coil construction meets the requirements for the majority of applications.

Standard coil construction for air-cooled chillers is copper tubes with aluminum fins.

Pre-Coated Aluminum Fin Coils

Pre-coated aluminum fin coils have a durable epoxy coating factory-applied to the fin. This design offers protection in mildly corrosive coastal environments, but is not recommended in severe industrial or severe coastal environments. Aluminum fin stock is coated with a baked-on epoxy coating prior to the fin stamping process. Coating of the fin material prior to the fin stamping process is known as "pre-coating." The pre-coated fin material is then stamped to form a wavy fin pattern for optimum thermal performance. The dissimilar metals of the coil are insulated from one another by a thin layer of inert epoxy pre-coating material. As a result, the electrical connection between the copper and aluminum is broken, thus preventing galvanic action that leads to corrosion.

Copper Fin Coils

Copper fin coils eliminate the bi-metallic bond found on standard coil construction. A copper wavy fin pattern is mechanically bonded to the standard copper tube. A protective Mylar strip installed between the coil assembly and sheet metal coil support pan further protects the coil from galvanic corrosion. Durability in a coastal unpolluted marine environment can be substantially improved over the standard or pre-coated coil construction, since the bi-metallic construction is not present.

Copper tube/copper fin coils are used for corrosion resistance in coastal areas.

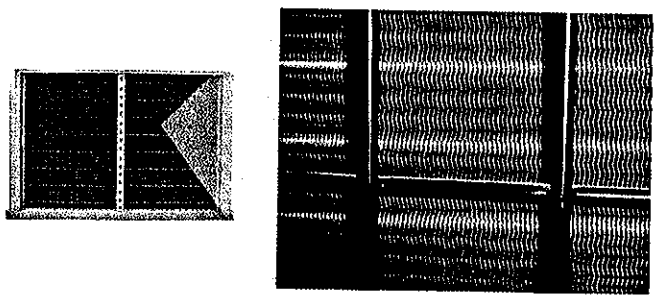


Figure 42

Copper Fin Coil Option

Copper is generally resistant to unpolluted coastal environments, since a natural protective film is formed to passivate the copper surfaces. Furthermore, a mono-metal bond exists between the tube and the fin. Uncoated copper coils are not suitable for polluted coastal applications or industrial applications, because many pollutants attack copper.

Copper-fin coils

provide increased corrosion resistance in coastal environments where pollution sources are not present.

Electro-Coated Aluminum-Fin Coils

Electro-coated coils provide superior protection against many corrosive atmospheres with the exception of formic acid and nitric acid environments. Electro-coated aluminum-fin coils have an extremely durable and flexible epoxy coating uniformly applied over all coil surfaces for protection from exposure to the contaminated environment. A consistent coating is achieved through a precisely controlled electro-coating process that bonds a thin impermeable epoxy coating on the specially prepared coil surfaces.

Electro-coating

The fundamental principle of electro-coating is that the materials with opposite electrical charges attract each other.

An electro-coating system applies a DC charge to the coil immersed in a bath of oppositely charged epoxy molecules. The molecules are drawn to the metal, forming an even, continuous film over the entire surface. At a certain point, the coating film insulates the metal, stopping the attraction, and preventing further coating deposition. This is a self-limiting nature of the coating process. A precisely controlled oven bake cures the coating uniformly to ensure consistent adhesion on all coil surfaces. Finally, a UV protective topcoat is applied to shield the finish from ultraviolet degradation and to ensure coating durability and long life. This creates a smooth, consistent and flexible coating that penetrates deep into all coil cavities and covers the entire coil assembly including the fin edges. This process, in conjunction with the coating material, results in a less brittle, more resilient and more durable coating without bridging between adjacent fins when compared to other coating procedures.

This method is superior to conventional phenolic coatings, which are applied manually by dipping and baking. Electro-coating is a more durable, evenly coated, non-brittle, non-flaking coil protection product than phenolic coatings.

Hydronics Packages

Some models of air-cooled chillers are available utilizing factory-installed and tested hydronics packages. The hydronics package can include a factory-supplied chilled-water pump, strainer, expansion tank, volume tank (not shown in figure), valve, and piping – basically items that traditionally would have to be supplied and installed by the contractor. In the description that follows,

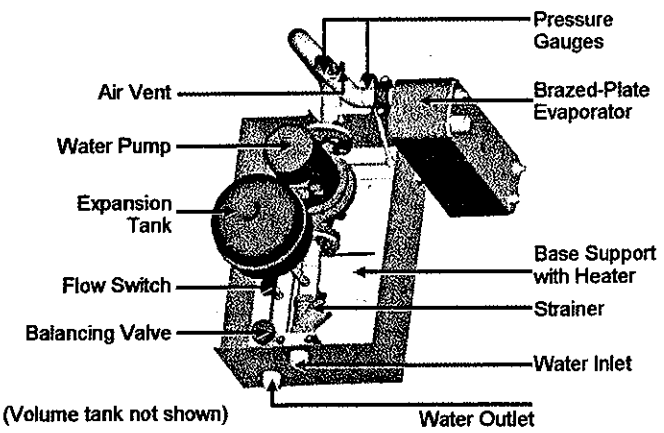


Figure 43

Hydronics Package – Factory-Installed in an Air-Cooled Chiller

keep in mind these hydronic components are necessary on all air-cooled chiller system installations whether or not they come factory-installed as part of the chiller or are provided by the contractor. The volume tank may not be required on all projects if installed loop volume meets or exceeds minimum requirements. For a detailed discussion of all hydronic components in piping systems, refer to TDP-502, Water Piping and Pumps.

Pump

Pumps are an integral part of the chilled-water system and should be selected based on their design, size, serviceability and performance. In a hydronics package, the pump is factory selected. In terms of performance, the pump is selected to provide the required chiller flow rate at the design total head while achieving the lowest possible horsepower.

If the pump is not included as part of a chiller hydronics package, pump selection must be made by the engineer.

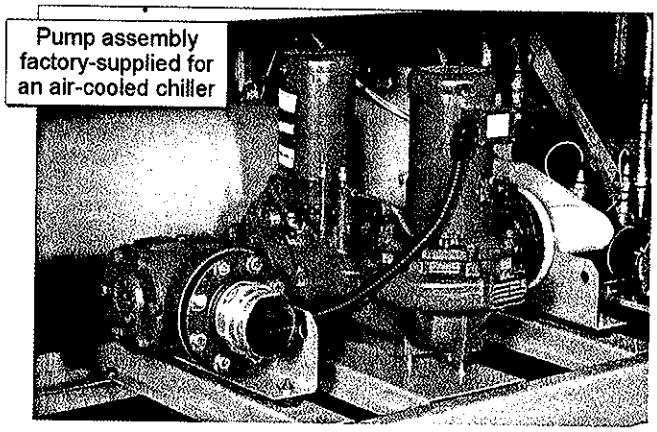


Figure 44

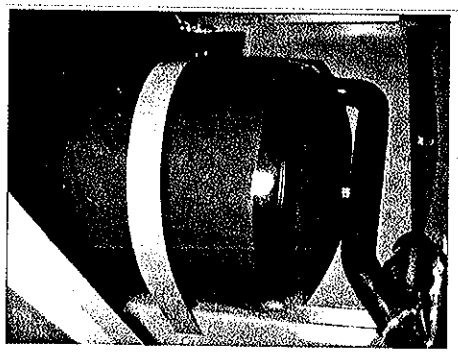
Pump Included in Hydronics Package

Expansion Tank

Temperature changes cause the water to expand or contract within a closed loop piping system. Expansion tanks support the expansion or contraction of water within the system.

Expansion of System Volume

Practically speaking, the change in water system volume is about 1 percent for chilled-water systems.



The change in water system volume is about 1% for chilled-water systems

Figure 45

Expansion Tank in Hydronics Package

Volume Tank

As discussed previously, all chiller manufacturers require a minimum loop volume for proper operation of their chillers to prevent rapid changes in water temperature and short cycling of the compressor. See the minimum chiller water system volume section in this TDP module.

A volume tank is often missed and system problems can occur. That is why this feature is now made available as a factory-supplied and installed option on some chiller designs.

Volume Tank Option

The volume tank is fitted under the chiller when this factory-supplied option is selected on an air-cooled chiller.

Air Separator (Field-Supplied)

Air separators are not included in the hydronics package, but deserve a discussion at this point. Air separators eliminate entrained air from the system. As an example, 50° F water at 30 psig can contain up to 9 percent dissolved air. Circulation of the water through an air separator can remove a large percentage of this air and will improve the overall efficiency of the system because air acts as an insulator. It also reduces the corrosion effect caused by oxygen.

Codes and Standards

There are a number of codes and standards applicable to air-cooled chillers that are important to understand. They have been divided into performance and safety related categories in this section.

Performance Related

ARI

ARI (Air Conditioning and Refrigeration Institute) is a trade association for the industry that has established a chiller certification program with defined testing procedures and tolerances. Manufacturers whose equipment bears the ARI label participate in a program of random audit certification testing. The testing is to check that the "as built" chiller performs per the manufacturer's ratings as represented in product rating literature such as selection software and catalogues. The ARI standard for electric chillers is ARI 550/590, "Water-chilling Packages Using The Vapor-Compression Cycle." To verify performance, ARI at any time may randomly select from a manufacturer's inventory listed units. These units are sent to an independent laboratory for performance testing. The equipment performance must match the listed values within a tolerance dependent upon the application temperature rise of the cooler fluid. The tolerance is 5 percent at standard full-load ARI conditions. Part-load tolerances are generally greater.



Figure 46

Codes and Standards

Here are the ARI Standardized Conditions for air-cooled chillers:

- Leaving chilled water = 44° F
- Chilled-water flow rate = 2.4 gpm / ton
- Cooler fouling factor = 0.0001 hr * °F * ft^2 / Btu
- Entering air temperature = 95° F

Following is a discussion of performance-related terms used by ARI for air-cooled chillers.

Efficiency

EER is measure of a chiller's efficiency at full load based upon the rating condition. COP and kW/ton are other methods of depicting the efficiency.

EER = Output, Cooler Tons (Btuh) / Input, Total Chiller (Watts)
COP = Output, Cooler Tons (Btuh) / Input, Total Chiller (Btu/hr)
kW/ton = Input, Total Chiller (kW) / Output, Cooler (tons)
COP = EER / 3.414
EER = 12 / (kW/ton)

IPLV

Integrated Part-Load Value (IPLV) is used to evaluate the efficiency of a chiller operating in the cooling mode over a range of points. IPLV is a weighted average of the EER calculated at 100, 75, 50, and 25 percent of chiller capacity. The IPLV conditions may not represent any specific project but are considered typical. The conditions and equation for IPLV are listed in the table below:

Conditions for Calculating IPLV

% Load	Entering Air Dry Bulb Temperature (°F)	Weighting
100	95	1%
75	80	42%
50	65	45%
25	55	12%

$$IPLV = 0.01A + 0.42B + 0.45C + 0.12D$$

Where:

A = [EER or COP] at 100% load
B = [EER or COP] at 75% load
C = [EER or COP] at 50% load
D = [EER or COP] at 25% load

Selection Software

Manufacturer's air-cooled chiller selection software automatically calculates IPLV or NPLV when a selection is made.

IPLV is commonly expressed as either an kW/ton value or in unites of EER. There is a fixed relationship between kW/ton and EER.

EER = 12 (kW/ton)

This relationship shows that EER increases as kW/ton decreases, and vice versa. Therefore, a "better" IPLV is shown as a lower value when the units are kW/ton, and, due to the relationship just described, a "better" IPLV is a higher value when the units are expressed in terms of EER.

NPLV

Non-Standard Part-Load Value is similar to IPLV, except NPLV is the term used if the chiller selection conditions are not exactly at the ARI conditions specified earlier.

ASHRAE 90.1

ASHRAE (American Society of Heating, Refrigerating, and Air-Conditioning Engineers) is an organization that establishes and maintains standards for the heating, ventilation, and air-conditioning industry. At the basic level, you may consider ASHRAE Standard 90.1 as defining minimum energy efficiency standards for a variety of building components, including air-conditioning equipment. This standard applies to air-cooled chillers and it defines the minimum EER, IPLV, & COP of the chiller. At the time of printing this TDP, here are the air-cooled chilled efficiency requirements.

Air-Cooled Chillers

Equipment Type	Size Category	Minimum Efficiency kW/Ton
Air-cooled with condenser	< 150 tons ≥ 150 tons	1.255 full load 1.153 IPLV
Air-cooled without condenser	All	1.134 full load 1.019 IPLV

Figure 47

ASHRAE 90.1 Efficiency Requirements

Safety Related

ASME

ASME (American Society of Mechanical Engineers) establishes and maintains rigorous safety related design construction and inspection standards. ASME is responsible for pressure vessel standards as defined in Section VIII Division I of the current ASME Code. The requirement for an ASME stamp is dependent on the physical size and the refrigerant pressure rating on the chiller. For example, evaporator vessels whose size falls under the minimum 6 inch inside diameter size may be constructed to comply with another nationally recognized testing laboratory other than ASME. This may vary from manufacturer to manufacturer.

UL / CSA & ETL

Several safety standards apply to chillers. There are also several widely accepted agencies such as Underwriters Laboratories, Inc., Canadian Standards Association, and ETL Testing Laboratories, which test for compliance to those standards. When a chiller is approved as meeting the required testing, it will bear a mark or label from the certifying agency.

UL (Underwriters' Laboratories, Inc.) is an independent nonprofit organization that tests products for safety and certifies them. The Canadian Standards Association (CSA) is a non-profit association serving business, industry, government, and consumers in Canada. Among many other activities, CSA develops standards that enhance public safety. For heating, ventilation, and air conditioning, UL/CSA Standard UL 1995 / CSA C22, Heating and Cooling Equipment applies.

ETL Testing Laboratories, like UL, conducts electrical performance and reliability testing. OSHA (Occupation Safety and Health Administration) recognizes ETL as a nationally Recognized Testing Laboratory as is Underwriters Laboratories. The ETL Listed Mark and Canadian-

ETL Listed Mark are accepted throughout the United States and Canada compliance with nationally recognized standards such as ANSI (American National Standards Institute), UL, and CSA. This certification mark indicates that the product has been tested to and has met the minimum requirements of a widely recognized U.S. product safety standard, that the manufacturing site has been audited, and that the applicant has agreed to a program of periodic factory follow-up inspections to verify continued conformance. If the mark includes a small "US" and/or "C," it follows product safety standards of United States and/or Canada respectively.

NEC / CEC

The NEC (National Electric Code) and CEC (Canadian Electric Code) are published by organizations whose members come from various government agencies, power utilities, insurance companies, electrical manufacturers, and other organizations. The NEC and CEC address proper design and installation of electrical systems and equipment to protect people and property from hazards arising from the use of electricity in building systems. Wiring design and installation in air-cooled chillers follows NEC guidelines.

ASHRAE 15

ASHRAE 15, Safety Standard for Refrigeration Systems, is a Mechanical Safety Standard relating to the design of systems containing refrigerants within buildings. Its primary goal is to mitigate safety risks to the environment, mechanical equipment room operators, and the general public from any harmful effects of refrigerants. Because air-cooled chillers are mounted outdoors, ASHRAE 15 is not usually applicable. However, if a split system with an indoor condenserless chiller or a remote indoor cooler barrel is used, ASHRAE 15 must be considered because if a refrigerant leak developed, refrigerant could enter the building.

ASHRAE 15

An additional benefit of single piece outdoor air-cooled chillers is that the extra expenses of complying with ASHRAE 15 do not apply. Those expenses include a refrigerant monitor, alarm system, relief piping and a ventilation system.

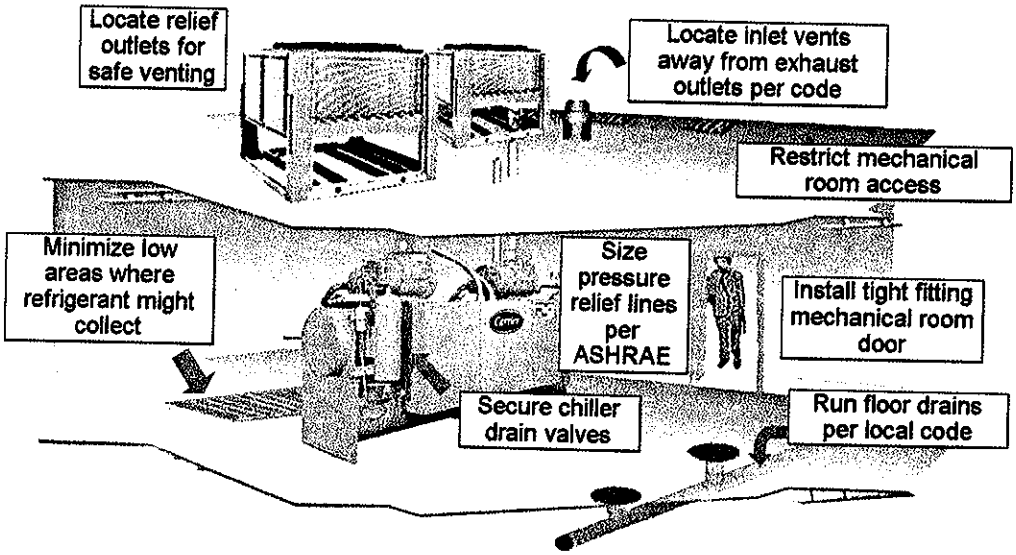


Figure 48

ASHRAE Standard 15 and Split System Air-Cooled Chiller

Selection Criteria

Input

The primary inputs for selecting an air-cooled chiller are listed below along with a typical chiller selection software program's screen capture:

Capacity/Cooling Load (tons)
Fouling Factor for the cooler
Water Type
(fresh, brine concentration, etc.)
Glycol Percentage by weight
Entering Water Temperature or
gpm or gpm/ton
Leaving Water Temperature
Ambient Air Temperature
Altitude (air density for condenser)
Voltage
Condenser Type

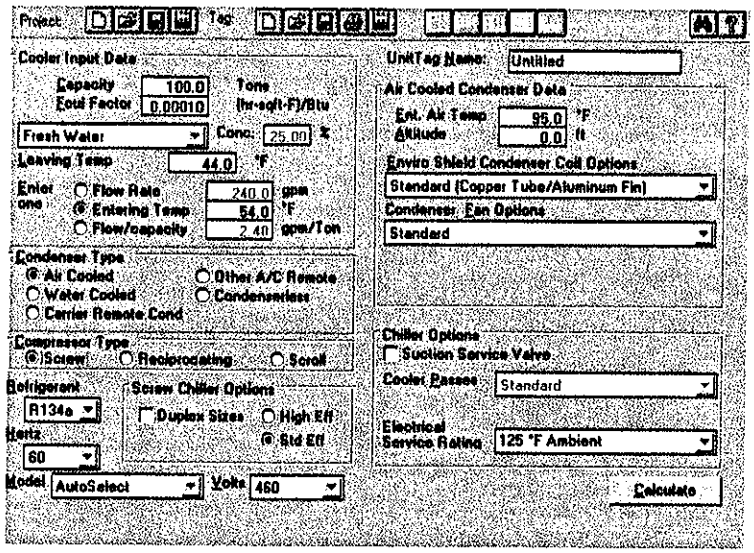


Figure 49

Selection Input

Output

The following is typical output from a chiller selection program:

Unit Size	30GXR114
Cooling Capacity	106.0 Tons
Compressor Input Power	120.5 kW
Unit Input Power	130.1 kW
Minimum Outdoor Operating Temp	-20°F
Capacity control steps	6
Minimum Capacity	20.0%
Input kw/Ton	1.227
Unit EER	9.78
Efficiency Level	Standard
Refrigerant	R134a
Cooler Data	
Fluid Type	Fresh Water
Fluid Entering Temperature	54.0° F
Fluid Leaving Temperature	44.0° F
Fluid Flow Rate	254.2 gpm
Fluid Pressure Drop	13.8 ft wg
Fluid Velocity	5.6 ft/s
Fouling Factor	0.0001 (hr-sq ft-°F)/Btu
SST	
Circuit A	43.1° F
Circuit B	42.5° F

Condenser Data	
Entering Air Temp	95.0° F
Air Flow Rate	68,400 cfm
Number of Fans	6
Fan Speed	1,140 rpm
Altitude	0 ft
SDT	
Circuit A	130.9° F
Circuit B	124.4° F
SCT	
Circuit A	128.8° F
Circuit B	123.0° F
Subcooling	
Circuit A	22.0° F
Circuit B	19.2° F
Heat Rejection	
Circuit A	80.5 Tons
Circuit B	54.6 Tons
Nameplate Voltage	460 Volts
Elec. Power Frequency	60 Hertz
Power Supply to control circuit control circuit	115 Volts
Power Supply to control circuit Phases	1
Minimum Circuit Amps Circuit 1	234
MOCP - Circuit 1	300
Max Instant Current Flow	354
Control Circuit Fuse Amps	300
Electrical Service Rating	125° F

Summary

In this TDP module, we first discussed the differences between air-cooled chillers and water-cooled chillers. The major difference revolves around the convenience of utilizing a single piece outdoor chiller versus an indoor water-cooled design that requires a separate cooling tower and condenser water system. However, as we learned, the full load kW per ton favors the water-cooled chiller.

Air-cooled chillers can be used as a single piece unit or a split system in various configurations. This flexibility has contributed to their overall popularity among designers of chilled-water systems.

There are a myriad of different application recommendations for air-cooled chillers. Variables like ambient temperature, leaving chilled-water temperature, variable flow in the evaporator, freeze protection methods and system water volume, all have an affect on the operation of an air-cooled chiller. Each of these was discussed in this TDP module.

Manufacturers offer different options to fit the application desired and it is important to apply them correctly. Many items, such as hydronic accessories, are now incorporated in some air-cooled chiller models from the factory.

The selection of an air-cooled chiller is performed on selection software furnished by the chiller manufacturer. All the necessary items that affect the performance of the chiller are included in the input data set.

The popularity of air-cooled chillers is increasing as a result of their features and their convenience of use. The ability of designers to use multiple air-cooled chillers together to achieve even the largest of installation capacities is an added attraction for designers of chilled-water systems.

Notes:

Work Session

1. What are the four major components of the air-cooled chiller refrigeration cycle?

   _____
   _____
   _____
   _____

2. List the three types of compressors used in air-cooled chillers.

   _____
   _____
   _____

3. List the three types of evaporators used in air-cooled chillers.

   _____
   _____
   _____

4. What are five configurations of air-cooled chillers?

   1. _____
   2. _____
   3. _____
   4. _____
   5. _____

5. What are the three methods of capacity control used in air-cooled chillers?

   _____
   _____
   _____

6. What can happen if a chiller is improperly sized?

   _____
   _____
   _____

7. List three types of motor starters for air-cooled chillers.

_____

_____

_____

8. List four methods of freeze protection for air-cooled chiller evaporators.

_____

_____

_____

_____

9. List three methods of increasing the corrosion protection for condenser coils.

_____

_____

_____

10. Which type of starter is most likely to be used on a 208V reciprocating air-cooled chiller compressor? _____

11. If the refrigerant is on the shell side of an evaporator, what type of evaporator is that?

_____

12. What is a typical kW/ton at full load of an air-cooled chiller? _____

13. What are the 4 dry bulb temperatures and the corresponding percent of full load used by ARI to calculate the IPLV of an air-cooled chiller?

Percent Load	Entering Dry Bulb Temperature

14. What is the rule of thumb for chilled-water system volume for comfort air-conditioning applications? _____

15. What is the rule of thumb for chilled-water system volume for process cooling applications?

_____

Notes:

Appendix

References

API Heat Transfer, Buffalo, NY. http://www.apiheattransfer.com/

ARI Standard 550/590, "Water Chilling Packages Using the Vapor Compression Guide." http://www.ari.org/std

ASHRAE Standard 15, "Safety Standard for Refrigeration Systems."

ASHRAE Standard 90.1, "Energy Efficient Design of New Buildings Except Low-Rise Residential Buildings"

Carrier Corporation, Syracuse, NY. Packaged Chiller Selection Program, Version 2.8. http://www.carrier.com/

Honeywell International, LLC, Morristown, NJ. http://www.honeywell.com/

Standard Refrigeration, Melrose Park, IL, http://www.stanref.com

Notes:

Work Session Answers

1. compressor, condenser, expansion device, and cooler or evaporator

2. reciprocating, screw, and scroll

3. flooded shell-and-tube, direct expansion shell-and-tube, and brazed plate

4. single unit, remote evaporator, condenserless chiller, indoor centrifugal condenser, evaporative condenser

5. unloading compressors, cycling compressors, and hot gas bypass

6. If the chiller is undersized, it will not be able to sufficiently cool the load under full load operation. However, if the chiller oversized, the reliability of the system, especially the compressors, may be reduced due to continuous cycling or operating at a very low capacity. As the highest stress on motors is greatest during start-ups, decreasing the number of start-ups from cycling is a significant factor in maintaining compressor reliability, efficiency, and limiting electrical demand charges.

7. across-the-line, wye-delta, and part-winding

8. draining the water loop, adding heaters to the cooler and water piping, circulating water constantly throughout the system, and adding antifreeze to the water loop

9. pre-coating; copper fin, copper tube; electro-coating

10. part-winding

11. flooded

12. 1.25 kW/ton

13.

Percent Load	Entering Dry Bulb Temperature
100	95
75	80
50	65
25	55

14. 3 gallons of water in the chilled-water piping loop per ton of installed chiller

15. 6-10 gallons of water in the chilled-water piping loop per ton of installed chiller.

Prerequisites:

Form No.	Book Cat. No.	Instructor Presentation Cat. No.	Title
TDP-102	796-026	797-026	ABCs of Comfort
TDP-103	796-027	797-027	Concepts of Air Conditioning
TDP-201	796-030	797-030	Psychrometrics, Level 1: Introduction
TDP-301	796-034	797-034	Load Estimating, Level 2: Fundamentals
TDP-400	796-037	797-037	Mechanical Refrigeration, Level 1: Introduction

Learning Objectives:

After reading this module, participants will be able to:

- List the various types and tonnages of packaged air-cooled chillers and their operational characteristics.
- Describe the correct application of air cooled-chillers versus water-cooled chillers.
- Identify the options and accessories available for air-cooled chillers.
- Use the selection criteria required to make efficient air-cooled chiller selections.

Supplemental Material:

Form No.	Book Cat. No.	Instructor Presentation Cat. No.	Title
TDP-623	796-055	797-055	Water-Cooled Chillers
TDP-703	796-068	797-068	Chilled-Water Systems

Instructor Information

Each TDP topic is supported with a number of different items to meet the specific needs of the user. Instructor materials consist of a CD-ROM disk that includes a PowerPoint™ presentation with convenient links to all required support materials required for the topic. This always includes: slides, presenter notes, text file including work sessions and work session solutions, quiz and quiz answers. Depending upon the topic, the instructor CD may also include sound, video, spreadsheets, forms, or other material required to present a complete class. Self-study or student material consists of a text including work sessions and work session answers, and may also include forms, worksheets, calculators, etc.



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