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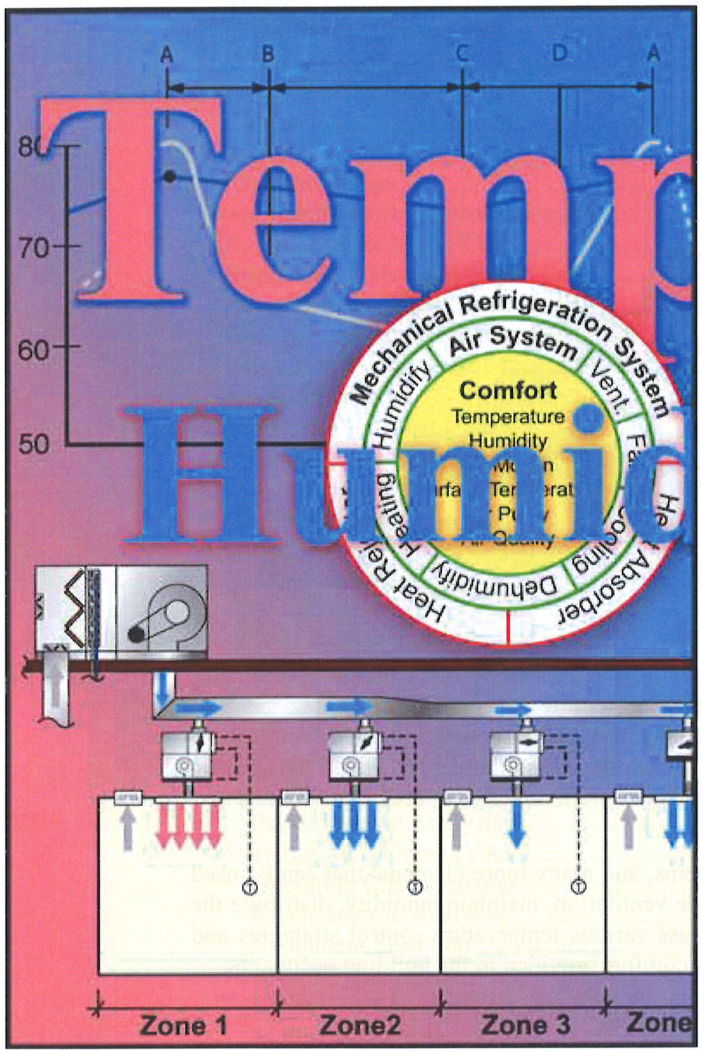


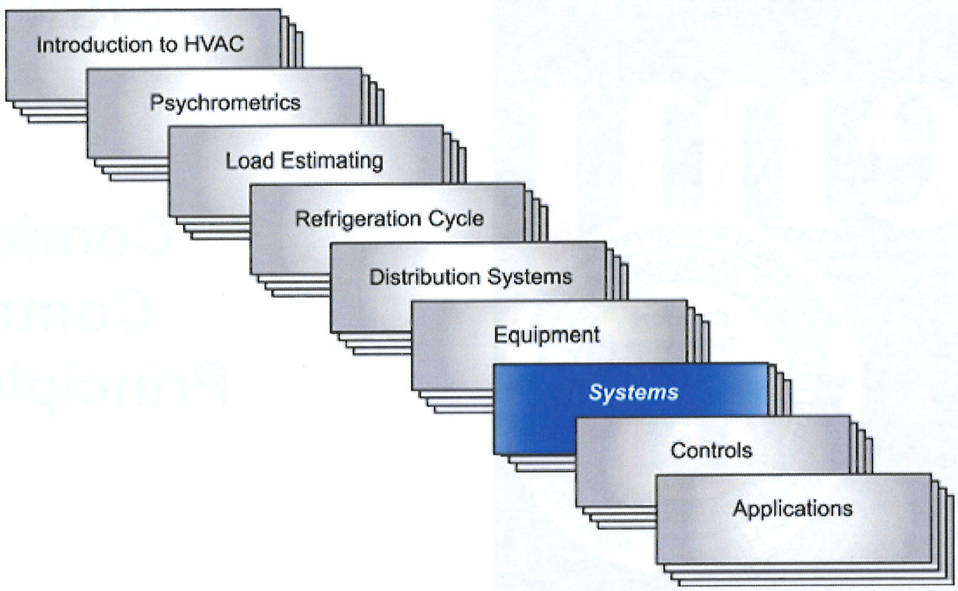
COMMERCIAL HVAC SYSTEMS

Comfort Control Principles

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 systems maintain the desired indoor comfort level, starting with space temperature. Other comfort parameters include maintaining acceptable room humidity, air motion, air quality, and air purity. The relative importance of each system function depends upon the specific project and application. Zoning is required to maximize the number of spaces that are successfully conditioned to the design criteria.

There are many different types of HVAC systems, and many more elements that can be used to achieve the heating and cooling capacity, provide ventilation, maintain humidity, distribute the air within the spaces, etc. This module will discuss various temperature control strategies and HVAC systems that can be employed to maximize comfort provided to the building occupants.

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Printed in Syracuse, NY

CARRIER CORPORATION
Carrier Parkway
Syracuse, NY 13221, U.S.A.

Table of Contents

Introduction 1
Zoning 3
  Operating Schedule Zoning 3
  Air Quality Zoning 4
  Temperature Control Zoning 4
  Multiple Units versus Multizone Systems 5
    Load Diversity and System Capacity 6
    Redundancy 6
Flow Control Devices 6
  Dampers 7
  Valves 7
  Fans and Pumps 7
Space Temperature Control Strategies 7
  Room Sensible Heat (RSH) 8
  Room Latent Heat (RLH) 8
  Room Sensible Heat Factor (RSHF) 8
  Cooling Coil Psychrometrics 9
  Simplified Psychrometric Diagrams 10
    Fan Heat Gain 11
    Return Duct Heat Gain/Loss 12
    Supply Duct Heat Gain/Loss 12
    Duct Leakage 12
  On-Off Control 12
    Operating Characteristics 13
    Psychrometric Analysis 14
  Coil Discharge Temperature Control 15
    Operating Characteristics 16
    Psychrometric Analysis 16
  Reheat Control 17
    Single-Zone Reheat 18
    Terminal Reheat 18
    Operating Characteristics 18
    Psychrometric Analysis 19
  Variable Air Volume (VAV) 20
    Operating Characteristics 21
    Psychrometric Analysis 21
    Equipment Selection 23
  Face and Bypass Control 24
    Single-Zone Face and Bypass 24
    Multizone Face and Bypass 24
    Dual-Duct 25
    Operating Characteristics 25
    Psychrometric Analysis 26
    Equipment Selection 26
  Hot and Cold Air Blending 27
    Operating Characteristics 28
    Psychrometric Analysis 28
    Operational Variations 29

Combinations of Basic Control Strategies 30
Control Strategy Recommendations 30
Humidity Control in Air-Conditioning Systems 31
Cooling Selection and Coil Performance 32
Part Load Operating Conditions 32
Sensible Cooling and Heating Loads 33
Latent Loads 34
Control Strategy Performance 34
Summary 34
Work Session 35
Appendix 37
References 37
Work Session Answers 38

Introduction

The primary function of an air-conditioning system is to maintain the desired indoor air quality, which starts with space temperature. Other comfort parameters include maintaining acceptable levels of such indoor environmental factors as relative humidity, air motion, air quality, and air purity.

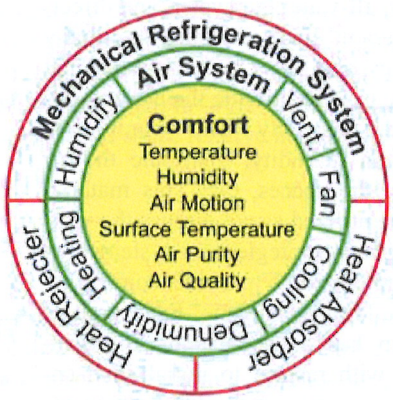

Figure 1

Comfort starts with space temperature.

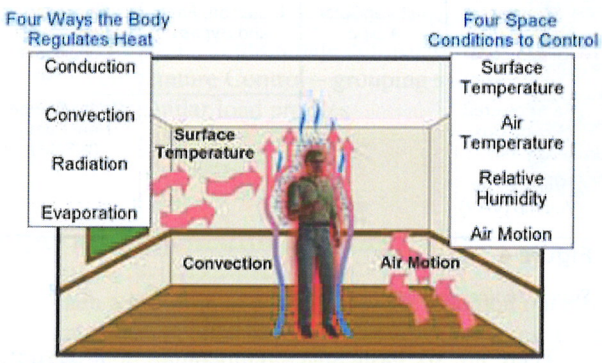

Figure 2

Body temperature regulation and control.

Five of these additional parameters relate to the air system conditioning the space. This is because body heat regulation, key to occupant comfort, relates to space conditions affected by the air system. The relative importance of each parameter depends upon the specific project and application.

In a large building or a building with multiple spaces, or rooms, and differing load patterns (often called load profiles), the space temperatures cannot be consistently controlled unless the HVAC system is properly zoned. The space temperature can only be properly controlled at the location of the temperature sensor or thermostat. The temperature in all other spaces, or all other points in a large room, will be different whenever their load pattern is not exactly the same as that at the location of the temperature sensor. The only solution is to provide a separate temperature control zone for each space with a uniquely different load profile. The system designer must evaluate each project to determine the appropriate temperature control zoning. This step is usually done during heating and cooling load estimating (see TDP-302, Load Estimating, Level 3: Block and Zone Loads for further discussion).

There are many different types of HVAC systems. The basic system categories or types can be classified as all-air, all-water, air-water, and direct-refrigerant. Each category has many systems, and many more elements that can be used to provide the heating and cooling capacity and ventilation, maintain humidity, distribute the air within the spaces, etc. This material will be utilized as we discuss the various control strategies and systems that are employed to provide comfort to the building occupants. Each type of system has advantages or disadvantages with respect to initial cost, energy consumption, building space requirements, etc. Providing proper comfort for the building occupants is only one of the parameters used in system selection, but it is easily the most fundamental and is a basic requirement for all systems.

Objective

The objectives of this module are:

- ***Explain the need for building control zoning***
- ***Present common space temperature control strategies***
- ***Work through typical design and part load performance***
- ***Guide the designer in selecting them for many types of HVAC systems.***

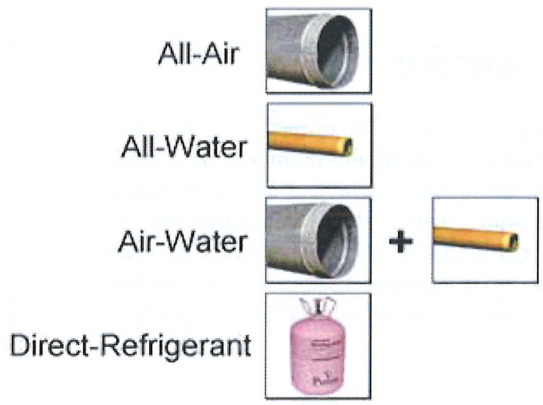

Figure 3

System Types

All-Air	All Water	Hybrid, Air-Water	Direct-Refrigerant
CV, Reheat	Fan Coil, 2-pipe	Induction, 4-pipe	Duct-Free Split
Constant Volume (CV) Single Zone (CVSZ)	Unit Ventilator, 4-pipe	Induction, Face and Bypass	Packaged Terminal Air Conditioner (PTAC)
Fan-Powered Mixing Box (FPMB)			Water Source Heat Pump (WSHP)
Variable Air Volume (VAV)			

Figure 4

System choices for each type.

There are six basic space temperature control strategies, each with advantages and disadvantages.

- On-off control
- Coil discharge temperature control – CAV
- Reheat control
- Variable air volume (VAV) control
- Face and bypass (F&BP) control
- Hot and cold air blending control

They define the operation of an air-conditioning system to adjust the cooling capacity that is supplied to a space so that it equals the actual space cooling load at that moment. Control strategies used to maintain space temperature include such actions as: on-off equipment cycling, airflow (cfm) volume control, and waterflow (gpm) throttling control. Each control strategy performs differently, especially with respect to the resultant relative humidity within the space.

No HVAC system can use all the available control strategies for space temperature control; some systems can use only one type of control, while others can be used with three or four different types.

The comfort performance of each type of HVAC system depends in part upon the type of control strategy that is used and the system part load operating conditions. The HVAC designer needs to first consider the required equipment capacity (do not oversize) and the building load profiles that are likely to occur. Then select a control strategy that will provide reasonable performance during the expected part load conditions. The preferred control strategy may not be available with the preferred type of HVAC system or equipment. In such cases, the designer makes a reasonable compromise of HVAC system type or equipment in order to provide acceptable part load comfort performance, paying particular attention to space relative humidity.

Zoning

Air-conditioning systems can be divided into individual control zones for several reasons. The necessary zoning can be provided by using individual air-conditioning units for each zone, or by using systems that are capable of providing multiple control zones from a common air source. The following basic zoning criteria must be carefully observed when selecting and laying out an HVAC system for a building where multiple zones will be served by a central system.

- Operating Schedule – common occupied/unoccupied cycles
- Air Quality – grouping spaces based on ventilation, contaminate control, or rh
- Temperature Control – grouping spaces with similar load profiles

A control zone

is a building, group of spaces or rooms, single room, or part of a room, with heating or cooling requirements sufficiently similar that comfort conditions can be controlled by a single thermostat or temperature sensor.

Operating Schedule Zoning

When a building is zoned based on common operating schedules, the air-conditioning equipment must be operated whenever any of the zones it serves are in use. Spaces that are occupied differently than the zone schedule cannot be served when the system is turned off, like during nights and weekends, because most of the building is unoccupied. Many buildings have security offices or communication equipment rooms containing electronic equipment that are in use even though the building is unoccupied. It is also common to schedule general office personnel to work overlapping shifts so phone coverage is available 10-14 hours a day. These unique rooms need to be taken off the main zones and given a separate system to match the different schedule.

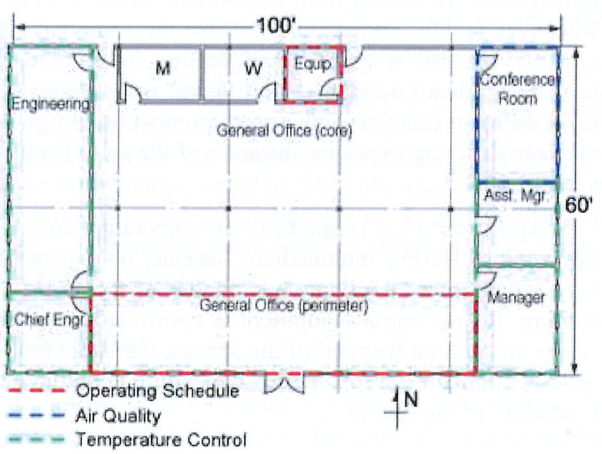

Figure 5

Zoning Possibilities

Air Quality Zoning

The air quality supplied from air-conditioning equipment must be adequate to serve the needs of the most demanding zone. It may not be practical to serve zones with diverse air quality needs from a common air system. For example, a large training area or conference room would not have the same outdoor air ventilation needs as an individual office. Other air quality problems that must be addressed in HVAC system design relate to contaminant control and space relative humidity. Minimum code requirements in these areas are presented in *ASHRAE Standard 62.1, Ventilation for Acceptable Indoor Air Quality.* To meet the basic ventilation requirement, the system outdoor airflow must be sufficient to provide adequate ventilation to the most critical space served by the system.

Using a more sophisticated ventilation control strategy, like demand controlled ventilation, would allow the outdoor air content of the air source to be adjusted to the actual demands of the zones, reducing energy usage and providing variable delivery of ventilation outdoor air to match the zone needs. This control strategy is not covered in depth in this TDP, but discussions can be found in TDP-631, Rooftop Units, Level 1: Constant Volume and TDP-703, Variable Air Volume Systems.

A local air treatment device can be provided to serve a zone with unusual cleanliness requirements if the quality of the central system air is insufficient. A local dehumidifying coil and/or humidifier can be provided to serve a zone that requires supply air at a different dew point than that available from the central system.

Spaces where dangerous or objectionable air contaminants are generated should not be on the same air system as spaces that would otherwise be free of these contaminants. The alternative is to exhaust 100 percent of the air that is supplied to the contaminated spaces plus maintaining a negative pressure to adjoining spaces.

Temperature Control Zoning

The need for temperature control zoning should be investigated whenever several building spaces are to be served from a common HVAC system. Temperature control zoning is necessary whenever building areas do not share the same pattern of variation in cooling and heating loads. The peak design loads may occur at different times of day or in different months. This occurs in most buildings between differing exposures because of the solar load variability.

With large central systems

It is possible to group zones with different operating schedules on the same system by isolating minor schedule groups with air dampers or water valves to avoid energy waste.

Exposure zoning is one form of temperature control zoning. During intermediate seasons, one space may actually need cooling while another space needs heating. If the central equipment is controlled by a temperature sensor located in the common return air mixture from all of the spaces, the temperature in all of the spaces will likely be higher or lower than the control setting, depending upon the relationship of the space load to the average load of all of the spaces.

The need for zoning can be determined by investigating the percent of design load airflow that is required by each zone at various months and times of day. Figure 6 shows the partition arrangement of a typical one-story office building. Figure 7 shows the percentage of airflow for each zone at 9 a.m., 1 p.m., and 4 p.m. in the months of July and October, assuming peak load occupancy and lighting in all zones.

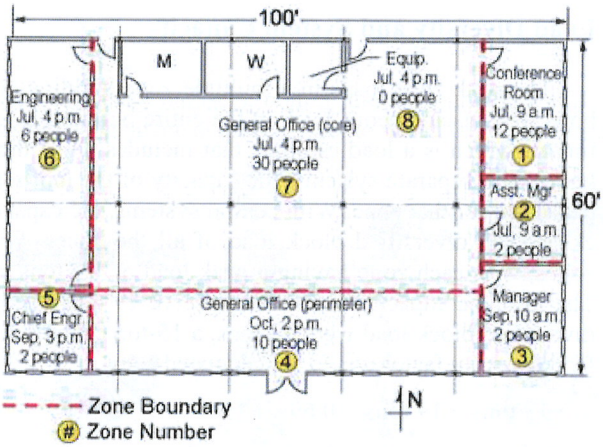

Figure 6

Initial Control Zoning Decisions

With a 20° F supply air temperature rise in the space, a 10 percent difference in airflow between zones on the same thermostat might result in a space temperature difference of about 2° F. Zones 3 and 4 are the only zones in which the required airflow is within about 10 percent at all of the different times and months. All of the other zones must have their own temperature control. Zone 3 is a large corner office, the manager's office. The thermostat would probably be located there. The manager might prefer an unusually cold or warm temperature setting, and the office load would be affected by occasional meetings with three or four additional people. These factors would cause the temperature in Zone 4 to vary more than the expected 10 percent; so good practice would require these two zones to also have independent temperature controls. Individual temperature control is always desirable with private partitioned offices but is seldom provided because of the expense.

Zone	Peak Time	July			October		
		9 a.m.	1 p.m.	4 p.m.	9 a.m.	1 p.m.	4 p.m.
#1 Conference	2 p.m. Jul	91	100	98	70	80	79
#2 East Office	2 p.m. Jul	89	100	96	62	75	72
#3 SE Corner	3 p.m. Oct	54	75	79	58	96	97
#4 South Office	2 p.m. Oct	48	70	74	54	96	96
#5 SW Corner	4 p.m. Sep	49	73	95	42	79	97
#6 West Office	5 p.m. Jul	52	69	96	36	52	76
#7 Interior	4 p.m. Jul	81	97	100	75	91	93

Figure 7

Percent of Peak Design Airflow

Zoning analysis requires consideration of the most likely part load operating conditions in each area. Zones 1 and 2 have a common east exposure and a relatively similar load pattern under design load occupancy and lighting. Zone 1 is identified as a conference room, however. The occupancy load varies from two or three people, to as many as twelve. The solar load will change if the window blinds are closed for projection equipment, and the lights may be dimmed or off. The unpredictability of these loads and the need for disproportionate ventilation require that conference and training rooms be considered for separate control zones.

Multiple Units versus Multizone Systems

The choice of whether to use a separate unitary system for each zone or to serve many separate zones from a single larger central system is beyond the scope of this text. However, we will discuss several considerations that affect this choice. A more detailed discussion is presented in TDP-701, System Features and Selection Criteria.

Load Diversity and System Capacity

The peak zone loads in a multiple zone building almost never occur at the same month and hour. The instantaneous load on the entire building is found by calculating a peak block load estimate, which is a load estimate that includes all of the individual zones at the same month and hour. With separate systems, the capacity of the unit serving each zone must equal or exceed the peak load for that zone. With central systems, the capacity of the central equipment must equal or exceed the diversified block load of all the zones. For example, a building might include ten zones, with each zone having a peak load of 1.9 tons and a unit selected with 2.0 tons capacity. The installed capacity is 20-tons to serve 19 tons of loads which occur at different times. If the diversified block load were 13 tons, a 15-ton central unit would be adequate. The resulting block load diversity factor would be calculated as:

13 tons ÷ 19 tons = 0.69

and the equipment capacity diversity factor would be:

15 tons ÷ 20 tons = 0.75

Redundancy

Redundancy is defined as the ability of a system to continue to serve its loads during a failure. A unitary system with a separate air-conditioning unit to serve each zone is considered to have good redundancy because the failure of a single unit does not affect the operation of any of the other zones. For example, 90 percent of the building would not be affected by the failure of one unit in a building with ten units. A central system can have more or less redundancy than a unitary system. For example, the entire building is out of service during a failure if the entire building is served by a single central unit. However, multiple central units can be provided. Two or three units can be provided with a total capacity equal to the building peak load, and manifolded so that the central units serve all zones equally. Alternatively, three units can be provided with each unit having a capacity of 50 percent of the building load, so that all of the zones can be served at full capacity during a failure of any one of the central units.

Flow Control Devices

Whether it is air, water, or DX (direct expansion refrigerant), control of the flow of some type of fluid is necessary to bring about a capacity change in the output of the HVAC system or one of its components. The task is accomplished by a controlled device responding to a controller output that is executing an algorithm that was chosen to implement a particular control strategy. The fundamentals of controls is covered in more detail in TDP-801, Controls, Level 1: Fundamentals. The basic devices follow.

Controlling Flow

Even though most system elements receive electric control signals (relays or DDC), the sequence of control ends up changing some fluid flow.

Primary Air and Zone Air	Common Primary Elements	Common Secondary Elements
Constant Air Volume/Constant Air Temperature	Heating Source	Air Handler
	Cooling Source	Waterside Conduits
Constant Air Volume/Variable Air Temperature	Ventilation Source	Zoning
	Filtration Source	Room Air Distribution

Figure 8

System elements are controlled.

Dampers

All-air and air-water (or hybrid) systems use many dampers, both at the air source and at the zones, to adjust the airflow (cfm) as a means of changing the output capacity to match the need. Large dampers are found in the mixing box (return, exhaust/relief and outdoor air), and coil (face and bypass) sections of the air source (rooftop unit, central station, or packaged). Out at the zone level, dampers are again found in the air terminals or zone equipment (VAV and FPMB terminals, or fan coil and unit ventilators).

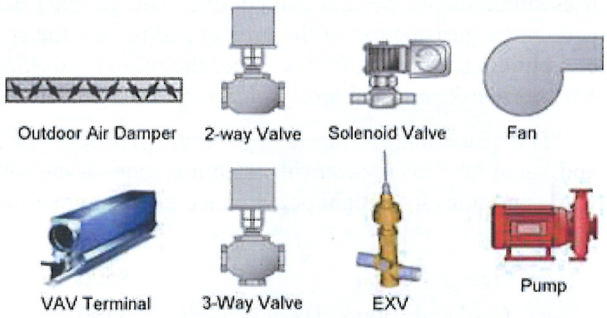

Figure 9

Common Controlled Devices ≠ Control Device.

Valves

With all-water systems, it is most common to use two-position and modulating valves to control the waterflow, again changing the output capacity of the air source, air terminal or zone equipment heating and/or cooling coils. Piping details can be reviewed in detail in TDP-502, Water Piping and Pumps. For DX systems, both two-position solenoid valves and modulating expansion valves (TXV-thermal or EXV-electric), are used to control the flow of liquid refrigerant to the cooling coil.

Fans and Pumps

The most basic flow control devices change output capacity by regulating the speed of the device or simply turn it on and off. These devices include system fans and pumps, along with the zone pumps and dedicated system equipment (single zone units and terminal equipment) fans. This control is usually accomplished using relays, two-speed motors and variable frequency drives.

Space Temperature Control Strategies

There are six basic strategies for adjusting the space cooling capacity of an air-conditioning system to equal the space sensible cooling load. Some types of HVAC systems switch from one control strategy to another if a desired space parameter is exceeded, such as switching from coil temperature control to reheat control on a rise in space humidity. Other types use two control strategies in sequence, such as first using variable air volume to reduce the airflow to 50 percent and then using reheat to keep the space temperature from dropping on a further drop in space load.

The basic space temperature control strategies are:

- On-off control
- Coil discharge temperature control
- Reheat control
- Variable air volume (VAV) control
- Face and bypass (F&BP) control
- Hot and cold air blending control

Only One Comfort Parameter!

Space temperature control strategies only respond to the sensible loads in the zone.

All of these strategies can be used during either the cooling mode or the heating mode. This discussion is focused upon the cooling mode.

The International Energy Conservation Code (IECC) that affects building system design throughout the United States has strict requirements relating to using any strategy that in effect uses simultaneous heating and cooling. The designer needs to consult the enforced version of the code in the jurisdiction of the project and review the code in detail. Building design for commercial buildings needs to follow either the referenced *ASHRAE 90.1 Energy Standard for Buildings Except Low-Rise Residential Buildings*, or the related chapter in the IECC.

The space temperature control strategy discussions that follow are based on room sensible and latent heat loads occurring within a zone on the following design day conditions, and at part load conditions that might occur on a cloudy summer morning.

	RSH	RLH	RSHF
Design Load	8500	1500	0.85
Cool and Cloudy Day	3400	1500	0.69

Outside air conditions of 80° F dry bulb (t_{oa}), 73° F wet bulb (t'_{oa}), 111 grains/pound (W_{oa}), and a space design relative humidity (rh) of 50 percent. The ventilation outdoor airflow rate is 15 percent and the cooling coil bypass factor (bf) is 0.10. The performance of hot and cold air blending control is based on t_{oa} of 65° F db, and t'_{oa} of 64.5° F wb, because the heating coil would not normally be active at an outside temperature of 80° F db.

Room Sensible Heat (RSH)

The RSH includes all sensible cooling loads which occur within the space and are sensed by the space temperature control, such as: solar gain, transmission through walls and windows, lights, equipment and the sensible portion of the heat gain from the occupants.

Room Latent Heat (RLH)

The RLH includes all of the latent heat gains that occur within the space, such as from the space occupants or cooking.

Room Sensible Heat Factor (RSHF)

The RSHF is the ratio of the RSH divided by the room total heat (RTH = RSH + RLH). A RSHF of 0.8 means that 80 percent of the load is sensible cooling and 20 percent is due to moisture removal.

Cooling Coil Psychrometrics

Before discussing the individual control strategies, we first need to explain the simple basic cooling coil diagram, the labels we will use to identify the various points on the psychrometric diagrams throughout this publication, and the optional application details such as fan and duct heat gain. Psychrometric diagrams are more complex if all of these details of the specific coil application are accurately represented on the diagram. The diagrams that are shown for the various control strategies do not include such application details, with the objective of keeping the diagrams as simple as possible.

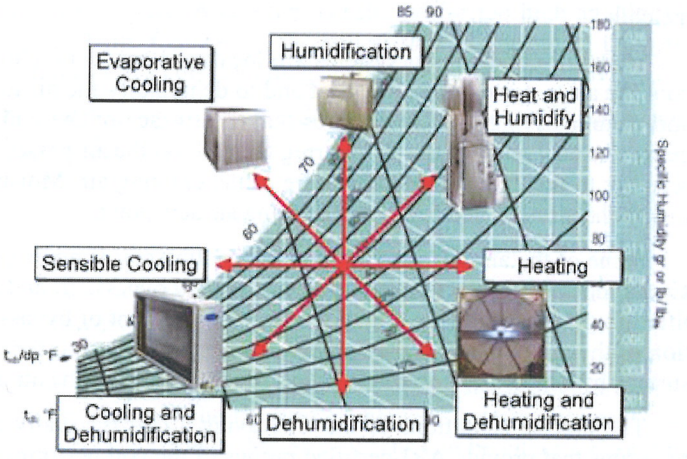

Figure 10

Psychrometric Process Lines

Figure 10 shows the eight basic psychrometric process lines, including the typical cooling and dehumidification system process. The psychrometric process discussed here for comfort cooling is similar to those involved in the other processes. A typical cooling and dehumidification process is shown in Figure 11. The various state points are labeled with descriptors such as RM for room and OA for outdoor air. If desired, temperatures can be labeled as t with a subscript descriptor to identify the point. The moisture content at each point can be labeled as W, for specific humidity (grains per pound of dry air, gr/lb_{da}), with the same subscript. The wet bulb and relative humidity can also be shown with the same subscripts.

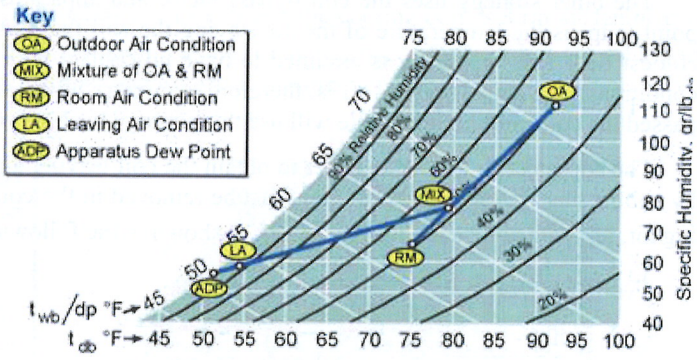

Figure 11

Psychrometric Plots, Cooling and Dehumidification Process

Simplified Psychrometric Diagrams

The simplified diagram usually begins at the room conditions (RM). The return air (RA) is assumed to be at the same conditions as the room air. The mixture (MIX) of ventilation outdoor air (OA) and RA is on a straight line between the two points. The location of the MIX point is the same proportion of the distance between the RA and OA points as the proportion of outdoor airflow to the total mixture airflow. This MIX state point is the condition of the entering air (EA) to the coil.

Many air-conditioning systems are located in moderate climates. The cooling coil serves to cool the air to remove sensible heat and to dehumidify the air to remove latent heat. The cooling and dehumidifying process requires that the surface of the coil be maintained at a temperature below the dew point of the air entering the coil. As the air passes through the coil, it is first cooled sensibly to the dew point temperature of the entering air. Moisture then condenses onto the coil fins as the air is cooled below the entering air dew point.

Some of the air that passes through a cooling coil is not exposed to the coldest coil fins. The air that does not contact the coldest fins is called bypass air, and the percentage of this air is identified as the bypass factor (bf) of the coil. The amount of bypass air depends upon the configuration of the coil, including the number of rows and fins, and the degree of enhancement of the fin surface by bending or corrugating the surface to increase the air turbulence.

The precise conditions of the air leaving a coil is usually calculated by complex computer programs that provide ARI-certified ratings of the coil performance. Several simplified strategies are used to represent and understand basic coil performance. One strategy is to assume that air will leave the coil at a constant relative humidity (rh), which depends upon the coil configuration. Air may leave a coil with relatively few rows and fins/inch at 80 percent rh, whereas it may leave a coil with more rows and fins/inch at 90 percent or 95 percent rh.

The other strategy uses the coil bypass factor and apparatus dew point. The apparatus dew point (adp) is the temperature of the air leaving the coil that has been in direct contact with the coldest fins. The bypass air is assumed to have no contact with the fins, so it leaves at the EA conditions. The actual leaving air is, therefore, a mixture of the air at the adp and the EA that bypassed the coil fins and tubes. We will use this strategy to show coil performance in this study.

The adp must be known in order to obtain the coil leaving air conditions. It can be calculated by subtracting the enthalpy (Δh) that must be removed in the coil from the enthalpy of the entering air, $h_{la,}$ with allowance for the coil bf, as shown in the following formula:

$$h_{coil} = GTH_{coil} \div (cfm * 4.5)$$

$$h_{adp} = \frac{h_{ea} - h_{coil}}{1 - bf}$$

The coil adp is at the intersection of h_{adp} with the saturation curve on the psychrometric chart. The air will leave the coil on the mixture line between the entering air (EA or MIX) and the adp:

$$t_{la} = adp + bf * (t_{mix} - adp)$$

The air leaving the cooling coil (LA) is assumed the same air that is supplied to the room (SA). The capacity of the supply air to satisfy the RSH and RLH loads is:

$$RSH = cfm_{coil} * 1.10 * (t_{rm} - t_{sa})$$

$$RLH = cfm_{coil} * 0.69 * (gr_{rm} - gr_{sa})$$

If the coil selection does not satisfy the proper proportions of RSH and RLH, it will either over-dehumidify or under-dehumidify the space. A coil that is selected for too high an adp will result in the space humidity being too high. Remember, the coil airflow is dependent upon the selected adp, so a change in adp will result in a corresponding change in airflow.

Several adjustments can be shown on a psychrometric diagram to include additional system details of a specific application (Figure 12).

Fan Heat Gain

The work of the fan to create the system airflow is dissipated as heat into the airstream, which warms the airstream from 0.5° F (for a very low pressure drop return air fan) to as much as 3° F (for a high pressure drop supply fan). The amount of heat varies with the fan pressure and efficiency. Fan heat also includes the heat dissipated from the motor windings and the belt drive loss if these items are located inside the fan plenum. The fan heat is shown as a temperature rise between the two points representing the fan inlet and outlet. Supply fans can be either a draw-thru or blow-thru arrangement, described by whether the supply fan is located downstream of the coil (draw-thru) and draws the air through the cooling coil into the fan, or whether the supply fan is upstream of the coil (blow-throw) and blows the air through the coil. Draw-thru fan heat is represented by the line from the point leaving the cooling coil, t_{la}, to_{tsa}, which is the supply air to the space. Blow-thru fan heat is represented by the line from the point representing the mixture of return and outdoor air, t_{mix}, to t_{ea}, which is the air entering the cooling coil. Return air fan heat is usually represented by the line from the point leaving the space, t_{rm}, to t_{ra}, which is the return air condition at the connection to the outdoor and return air mixing plenum.

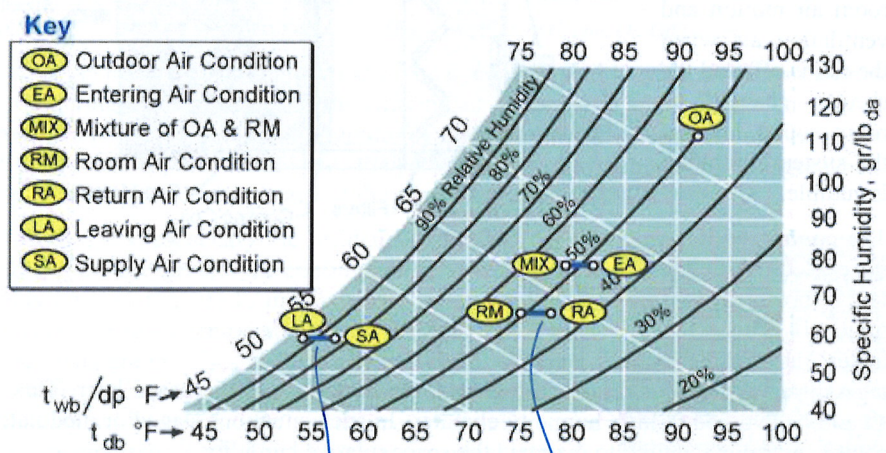

Figure 12

Psychrometric Plots, Fan and Duct Heat Gains

Return Duct Heat Gain/Loss

Return duct heat gain is represented in the same manner as return air fan heat, as a sensible temperature rise from the room temperature t_{rm}. A return duct heat loss is represented as a temperature drop from the same point.

Supply Duct Heat Gain/Loss

Supply duct heat gain and loss are represented in the same manner as draw-thru supply air fan heat, as a sensible temperature rise (or fall) from the coil leaving temperature t_{la}.

Duct Leakage

Inward leakage of air into a duct at negative pressure is a mixing process, represented as a point on a straight line between the air conditions inside and outside the duct, with the location of the point determined by the percentage of air leakage. Outward leakage from a positively pressurized duct cannot be represented on the psychrometric diagram because the temperature and moisture condition of the air remaining in the duct is not affected by the leakage.

On-Off Control

This type of control consists of cycling the equipment. Figure 13 shows a typical air-handling unit that could have either chilled water (shown) or a DX cooling coil. There are two options with on-off control:

- The entire unit can be cycled. Disadvantages of this option include the lack of room air motion and ventilation air when the fan is off, and the abrupt change in room sound level if the system operation is audible.
- The cooling coil can be cycled while the fan is operated continuously. This eliminates the disadvantages listed above, but results in higher space humidity, as discussed later. This option should be avoided in high humidity climates. In such situations staged or modulating capacity control is required to maintain the space relative humidity.

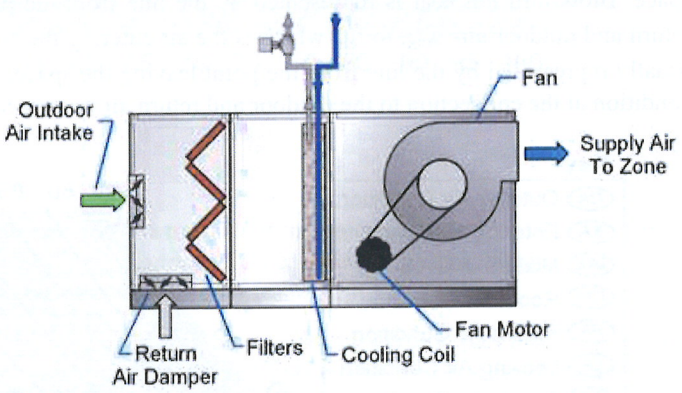

Figure 13

Air Source Components, On-Off Control

Regardless of which option is used, on-off control has the advantage of being simple and inexpensive.

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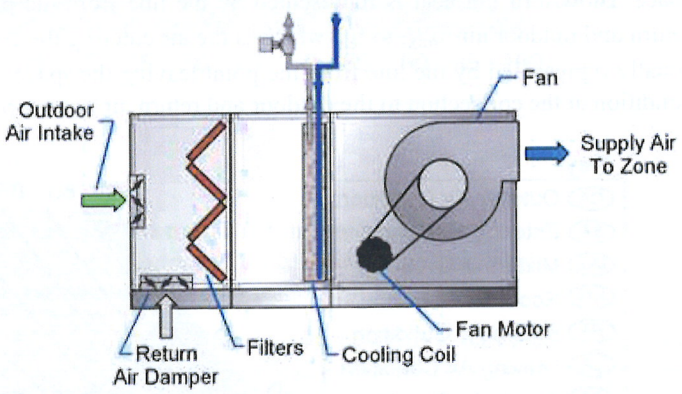

Figure 13

Air Source Components, On-Off Control

Regardless of which option is used, on-off control has the advantage of being simple and inexpensive.

Operating Characteristics

The cooling load is usually less than the cooling capacity of the equipment. On-off control systems balance the sensible cooling load to the equipment capacity by cycling the equipment. During normal operation, the space temperature gradually drops toward the off setting of the space temperature control. When it reaches this point, the control stops the equipment until the space temperature rises to the on setting of the space temperature control, which restarts the equipment. The length of the on and off cycles is determined by the space load, the equipment capacity, and the temperature differential between the on and off settings. The temperature difference of the space control settings must be large enough to prevent short cycling, which can be damaging to the equipment. An alternative is to provide a timed cycling control to limit the frequency of control cycles.

A simplistic view of on-off control is that the cooling coil is always at design cooling temperature during the on cycle, and that no cooling is provided during the off cycle. The percent operating time is then equal to the RSH (room sensible heat) load divided by the sensible cooling capacity delivered to the space while the equipment is on.

In reality, the supply air dry bulb temperature is a periodic curve, as shown in Figure 14. The system starts at point A and begins to lower the temperature of the air leaving the coil. No dehumidification occurs until the coil surface temperature drops below the entering air dew point. The coil leaving air temperature continues to drop until point B, where the equipment capacity is in balance with the load imposed by the coil entering and leaving air conditions. A-B is the pulldown portion of the operating cycle. The pulldown cycle can be relatively short if the coil did not warm up during the previous off cycle, and quite long if the coil is a chilled water coil with a much greater mass of coil and water to be cooled.

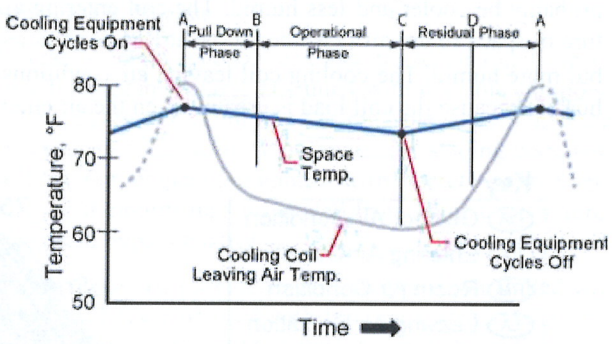

Figure 14

Time-Temperature Graph, On-Off Control

B-C is the normal operating portion of the cycle. The coil leaving air temperature will gradually drop during this stage because the entering air temperature and moisture content will drop as the space temperature and humidity gradually drop. Stage B-C continues until the space temperature drops to the off point of the space temperature control, which will take longer when the space cooling load is a greater percentage of the equipment capacity.

Most systems, especially the residential type, can be arranged to cycle the fan and the cooling coil in unison, with a fan delay on the off cycle to improve the system efficiency. In this case, the space temperature control simply waits until the space temperature rises to start another cycle. Commercial systems are usually arranged to operate the supply fan continuously with the outdoor air damper remaining open while the compressor is off. This results in a residual stage C-D of the cycle. The cold coil surface is warmed by the flow of return and outdoor air across the coil. The moisture that had condensed on the coil surface, but not dropped into the

Residual Phase Effects

If the outdoor air is warmer and moister than the space conditions, the outdoor ventilation air adds both heat and moisture to the space during stages C-D and D-A.

drain pan, is re-evaporated into the warm airstream and returns to the space where it increases the space humidity. Once the moisture has re-evaporated, stage D-A is the remainder of the off cycle, which lasts until the space temperature rises to the on point of the space temperature control.

Operating the fan while the cooling coil is inactive is beneficial in some respects, but harmful in others. Constant room air motion and sound level are an advantage. In addition, the outdoor air damper can remain open to provide code-required constant ventilation. The main disadvantage is the impact on space humidity, which is usually 5 to 10 percent higher if the fan is operated continuously, due to the wet coil condensate re-evaporating into the space.

On-off control is attractive for many reasons. It is simple, inexpensive, and thermally efficient. Other than the poor humidity control discussed above, the disadvantages include the wide temperature differential that may be necessary to minimize short cycling of equipment, intermittent sound level, and lack of ventilation and room air motion if the fan is also cycled.

Psychrometric Analysis

A typical psychrometric diagram plot of on-off control performance is shown here. The peak load system design diagram is shown dotted. The part load diagram lines are solid. At part load, the room air conditions (RM) will become more humid. The outdoor air conditions (OA) will probably be cooler and less humid. The coil entering air conditions (EA), t_{edb}/t_{ewb}, being a mixture of space return air (assumed to be at RM conditions) and OA entering the coil, will be cooler but more humid. The cooling coil leaving air conditions (LA), t_{ldb}/t_{lwb}, will be warmer and more humid, because the coil load is greater when the air entering the coil is more humid.

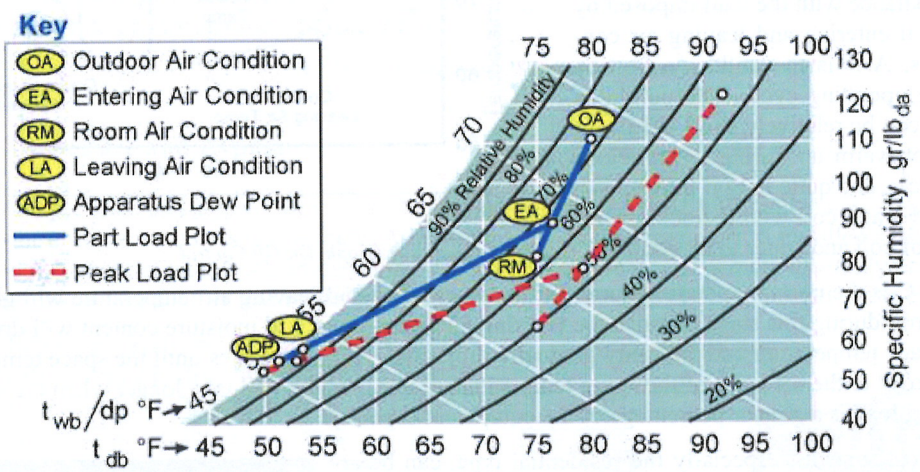

Figure 15

Psychrometric Plot, On-Off Control

valve open - dehumidication

For the time being, consider the cooling coil LA to be the same as the supply air conditions (SA) into the room or space. The room sensible and latent loads are absorbed as follows:

$$RSH = 1.10 * cfm_{sa} * (t_{rm} - t_{sa}) * \text{percent OT}$$

$$RLH = 0.69 * cfm_{sa} * (W_{rm} - W_{sa}) * \text{percent OT}$$

Where:

t_{rm} = Room temperature, °F

t_{sa} = Supply air temperature, °F

W_{rm} = Room moisture content, grains/lb

W_{sa} = Supply air moisture content, grains/lb

percent OT = Operating time percentage (room sensible heat load ÷ room sensible cooling capacity of the equipment while it is operating)

The diagram is in balance when the air leaving the coil will satisfy the room sensible and latent loads at part load conditions, and the equipment will produce the capacity to satisfy these loads at the same operating conditions. A theoretical balance point at any part load condition can be found by using a spreadsheet to equate the loads and the equipment capacity, and iterating the space condition until the loads and equipment are in balance.

On-off control is preferred for small capacity single zone DX equipment and for small capacity chilled water fan coil units. Large capacity constant volume DX units with multiple compressors and multiple refrigerant circuits are more likely to be controlled in multiple steps. When the coil remains active while the compressor capacity is reduced by unloading or staging compressors, the control system behaves like a coil leaving temperature control with discrete steps. Some DX units have multiple face-split coil refrigerant circuits that are cycled from space temperature. This control behaves like a multiple step face and bypass control.

Room-type chilled water fan coil units can easily be provided with a two-position chilled water valve. Modulating chilled water valves should never be used with fan coil systems unless the system includes a separate dehumidified ventilation air system to control the space humidity.

Coil Discharge Temperature Control

This type of control consists of varying the air temperature leaving the cooling coil, while maintaining constant system airflow. A modulating coil valve is used with chilled water coils, as shown. A step control arrangement is used for DX systems with multiple stages of refrigeration capacity connected to a common evaporator coil

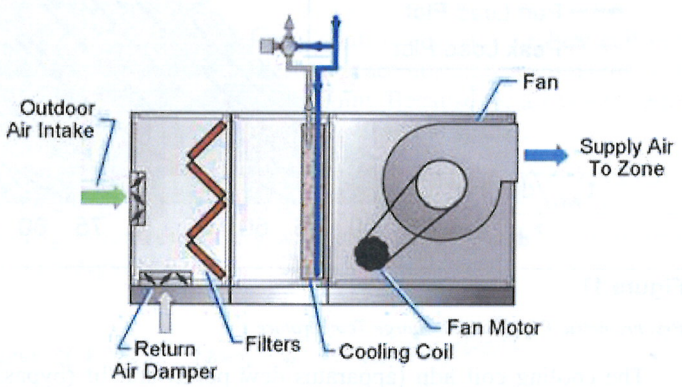

Figure 16

Air Source Components, Coil Discharge Temperature Control

The primary advantage of this control method is constant airflow, which provides constant outdoor air ventilation and air distribution. The room temperature can be very precisely controlled if a modulating valve is used with a chilled water coil. The most serious disadvantage is the loss of dehumidification, which results when the coil temperature becomes warmer at part load.

Operating Characteristics

This strategy is easily applied to constant volume air systems using chilled water coils. The space temperature control positions a modulating chilled water control valve at the chilled water coil to obtain the supply air temperature required by the current room sensible heat load. Each increment of decrease in RSH at part load results in a corresponding increase in the temperature leaving the cooling coil.

Advantages of this control strategy include the following:

- Modulating the coil discharge air temperature provides precise control of the space dry bulb temperature.
- Low energy consumption (compared to reheat or hot and cold air blending).
- No additional space for the air source equipment (compared to face and bypass or hot and cold air blending).

Psychrometric Analysis

Assuming constant coil airflow, the dry bulb temperature leaving the cooling coil can be calculated as follows:

$$t_{ldb} = t_{rm} - (RSH \div 1.10 * cfm)$$

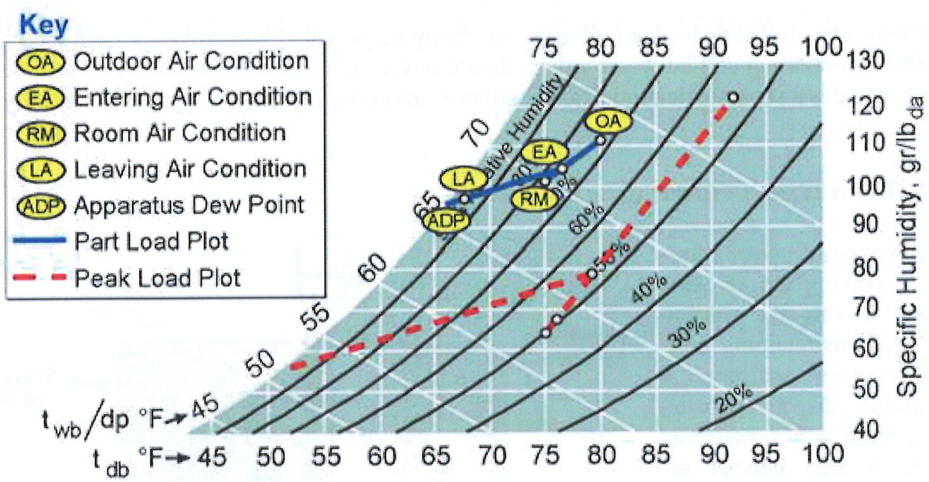

Figure 17

Psychrometric Plot, Coil Discharge Temperature

The cooling coil adp (apparatus dew point) and bf (bypass factor) determine the moisture content of the air leaving the coil, as shown by point LA on the psychrometric diagram.

The space moisture content is calculated by adding the moisture content that is represented by the room latent heat (RLH) load to the moisture content of the air leaving the coil:

$W_{rm} - W_{la} + (RLH \div 0.69 * 1.10)$

The space humidity of 77 percent is determined from the psychrometric chart, using the calculated moisture content at the 75° F space temperature.

The space humidity is high because the moisture content of the air leaving the cooling coil was increased when the coil valve operated to raise the dry bulb temperature leaving the coil. The humidity performance of coil temperature control is so poor that it should not be used as the primary means of space temperature control in a humid climate. A high humidity override control can be provided to add reheat, which will require the coil to operate at a cooler and drier leaving air temperature. A better solution is to choose a different type of control.

For room fan coil systems, the recommended solutions are to provide a separate dehumidified ventilation air system (dedicated air system) and to arrange the fan coil unit control valve for two-position on-off cycling control. Face and bypass control is a preferred solution for constant volume air-handling units and unit ventilators. VAV control is the recommended solution wherever a constant flow system is not required.

The coils of large systems with multiple coils can be controlled in sequence using modulating chilled water valves. Each active coil will be a variable temperature control, but the inactive coils will function as an air bypass control. The result is space humidity between that of coil discharge air temperature control and coil face and bypass control.

This strategy is not generally applicable to DX coils. The refrigerant pressure (evaporating temperature) in the coil can be modulated using a pneumatic signal to a refrigerant hot gas bypass valve or a refrigerant suction line pressure regulator, though this is not done very often in comfort applications. DX systems with multiple steps of compressor capacity serving a common coil provide increments of coil discharge air temperature control. The space humidity will generally be somewhat better than for a modulating coil control because the coil leaving air temperature will be colder for a few minutes of each cycle, thus providing a short period of better dehumidification.

load reduced will affect humidity.
∴ need additional heat.

Reheat Control

Reheat control starts with the coil discharge temperature control and adds a reheat coil(s) to the system. The LA set point is established based on the desired space humidity, cooling the air to a lower dew point than obtainable with space temperature set point. Because the air would be too cool for most RSH loads, reheat is needed to warm up the supply air to the spaces.

Reheat control provides maximum dehumidification capacity to the space at all times but is wasteful of both cooling and heating energy. Its use is, therefore, restricted by most energy codes. Site-recovered waste heat from refrigeration condensers or other sources should be investigated as a code-compliant source of reheat energy.

valve open, temp ↓ ∴ need reheat.

Single-Zone Reheat

This type of control is arranged as shown. A heating coil (reheat coil) is installed downstream of the cooling coil. The space temperature control adjusts the temperature of the air leaving the reheat coil. The temperature leaving the cooling coil is either uncontrolled or controlled by a sensor in the air leaving the cooling coil. The system has a single reheat coil to provide one large temperature control zone.

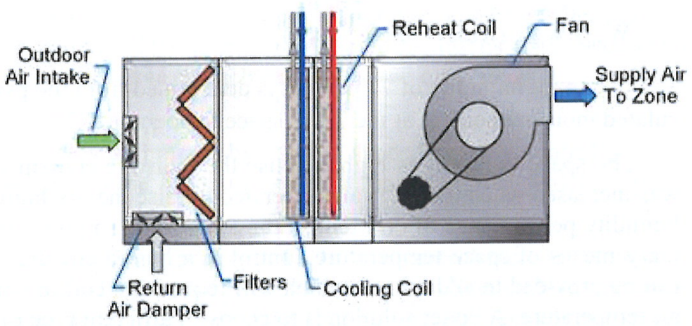

Figure 18

Air Source Components, Single-Zone Reheat Control

Terminal Reheat (not used because not energy efficient)

Another reheat strategy uses a multitude of small zone reheat coils to provide multiple control zones from a single large cooling unit, as shown.

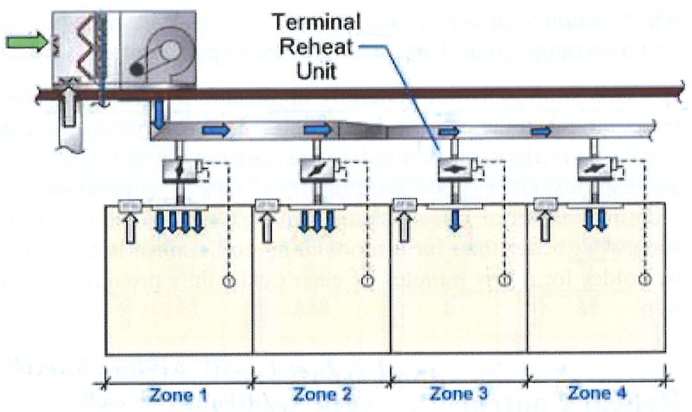

Figure 19

HVAC System Components, Terminal Reheat Control

Operating Characteristics

Reheat control can be provided on almost any air-conditioning system by installing a heating coil downstream of the cooling coil, either inside the air-handling unit or in the supply duct leaving the air handler. It cannot be applied with ductless systems unless the unit is designed to accept a heating coil in the reheat position.

The space temperature control modulates the output of the reheat coil located downstream of the cooling coil to prevent the space temperature from dropping when the actual RSH is less than the design RSH. Reheat control provides excellent control of space humidity throughout the entire load range from 100 percent RSH to near zero percent RSH.

Psychrometric Analysis

Figure 20 shows the typical system performance at part load. If the dry bulb temperature of the air leaving the cooling coil (state point LA) is the same as at design load, the space relative humidity remains at 50 percent at 40 percent RSH and 100 percent RLH load.

The amount of reheat is:

Reheat = design RSH - actual RSH

The supply air temperature leaving the reheat coil, t_{sa}, is:

$t_{sa} = t_{rm}$ - (actual RSH ÷ 1.10 * cfm)

The dry bulb temperature leaving the cooling coil, t_{ldb}, is:

$t_{ldb} = t_{sa}$ - (reheat ÷ 1.10 * cfm)

The moisture content of the air leaving the cooling coil is found from the psychrometric chart, with allowance for coil bypass factor. The actual room moisture content is calculated, from which the room relative humidity can be obtained by use of the psychrometric chart:

$W_{rm} = W_{la} + (RLH \div 0.69 * 1.10)$

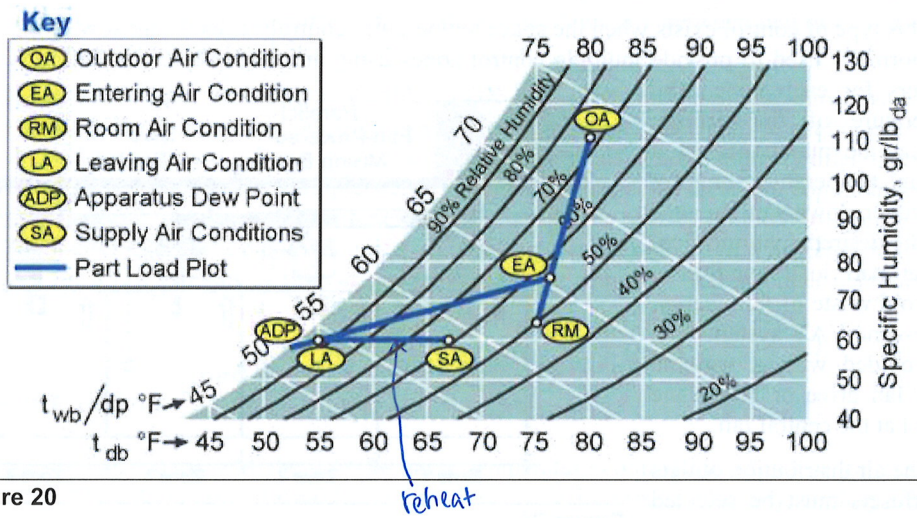

Figure 20

Psychrometric Plot, Reheat Control

The cooling coil entering air conditions are found by mixing the proportions of outdoor and return air at their actual part load conditions. As mentioned earlier, the solution is an iterative process, because the assumed space humidity affects the mixed air moisture content entering the cooling coil, which affects the moisture content of the air leaving the cooling coil, which in turn affects the humidity in the space.

Contrary to some expectations, the cooling coil can experience a large reduction in load at part load. Any reduction in room latent load causes a drop in space humidity, and a corresponding drop in the moisture content of the mixed air entering the coil. Any drop in outdoor air temperature or moisture content is also reflected in a reduction in heat content of the air entering the coil.

Chilled water systems result in less reheat energy penalty than DX systems, because the cooling coil LA can be more precisely controlled. With DX systems, a reduction in entering air temperature or moisture content results in a corresponding drop in leaving air conditions because the refrigeration capacity is relatively constant. This is true at all load conditions for DX systems without unloaders and is true for the range between each capacity stage on systems with multiple steps of capacity control.

The energy waste of reheat control is minimized by using a space humidity control to reset the cooling coil leaving air temperature to the highest possible temperature that will satisfy the space humidity.

Reheat control provides the best functional performance of any control method. However, ASHRAE Standard 90.1 and the state and local building codes that refer to Standard 90.1 prohibit the use of reheat control for applications that can be satisfied with more efficient control strategies. The energy waste may be permissible or required in critical applications, such as laboratories that may require a constant flow of 100 percent outdoor air for makeup from exhaust hoods, and in other specialized applications that require humidity control, such as museums and computer rooms.

Variable Air Volume (VAV)

This type of control exists when the space temperature control varies the airflow to the space. It is normally used to provide multiple control zones using automatically controlled air volume dampers for each zone. The temperature of the cooling coil can be maintained at a constant temperature, or reset upward or downward to provide the desired dehumidification at part load conditions. To avoid waste of fan energy, single-zone VAV systems can be provided with a variable speed fan drive or inlet vane control at the central fan.

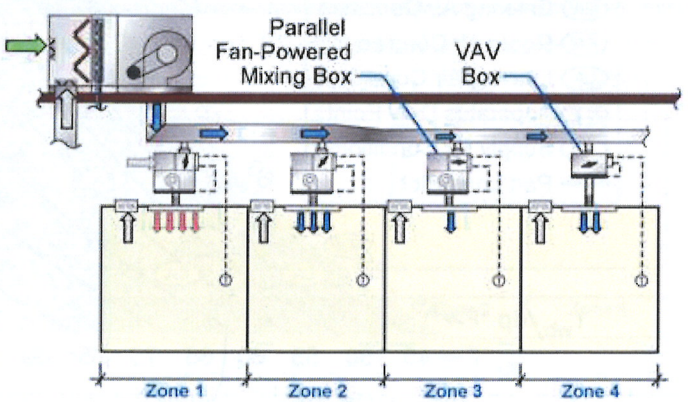

Figure 21

HVAC System Components, VAV Control

The air distribution outlets or diffusers must be selected for proper performance at the minimum expected airflows. These problems are overcome by using demand controlled ventilation (DCV) and limited-range VAV (minimum cfm terminal set point) in conjunction with another control strategy such as reheat when the room load does not create sufficient airflow for proper VAV system performance.

Other types of VAV systems have local fans, such as series and parallel fan-powered mixing boxes (FPMB), that induce air from the space or ceiling plenum to create adequate supply airflow at low loads. A variable volume and temperature or VVT system is a type of VAV that varies both the supply air volume and temperature on a time/demand-weighted control strategy that allows use of smaller packaged CV equipment to be used in a multizone VAV layout.

Operating Characteristics

VAV control modulates the supply airflow at each zone to balance the room sensible cooling loads. The reduced airflow requires less fan power than constant volume systems. There is no reheat penalty as in reheat or hot and cold air blending systems. VAV systems are sized for the system block load, not the sum of the zone peak loads; so whatever load diversity exists can be used to downsize the ductwork sections feeding multiple zones. Any capacity (airflow) that is not required in one zone is available for any other nearby zone. This can be viewed as a built-in safety factor or allowance for future loads in sizing the system. When fed from a self-balancing ductwork design, damper modulation is quiet and unobtrusive within the spaces. FPMB systems coupled with DCV eliminate the IAQ issues of traditional VAV, and use of high efficiency ECM motors reduces the energy penalty of the zone-level fans.

Psychrometric Analysis

Typical VAV system performance is shown at part load. In this case, the coil LA is assumed constant and controlled at the design load set point. At 40 percent RSH and 100 percent RLH loads, the supply airflow was reduced to 40 percent of the design flow. The space relative humidity increased to 58 percent.

The space airflow, cfm_{sa}, was calculated as follows:

$$cfm_{sa} = RSH \div 1.10 * (t_{rm} - t_{sa})$$

The moisture content at the space (state point RM) can then be calculated as follows:

$$W_{rm} = W_{la} + (RLH \div 0.69 * 1.10)$$

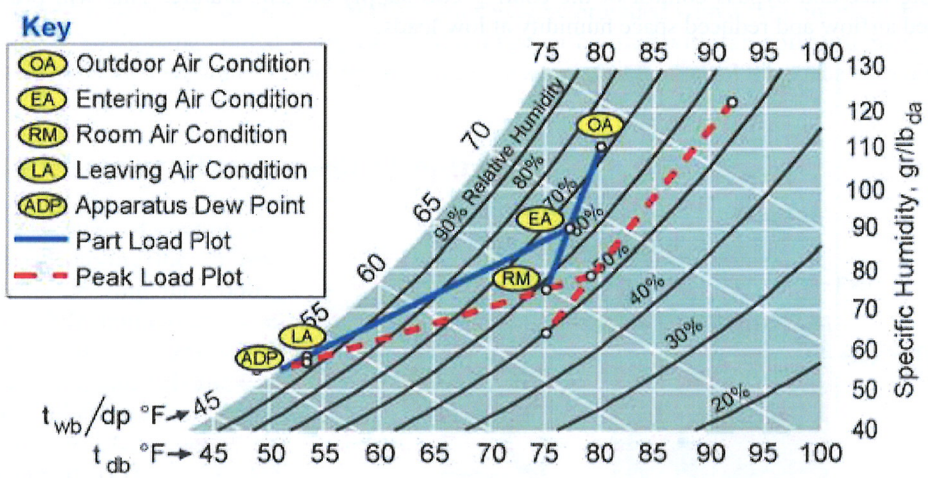

Figure 22

Psychrometric Plot, VAV Control

air volume controlled.

Point EA, the condition entering the cooling coil, is found by mixing the proportions of outdoor air and return air at their actual part load conditions. If the ventilation air quantity is held constant, the percentage of outdoor air will increase as the supply airflow is reduced.

In a VAV system, the outdoor airflow rate will be variable unless a positive flow control is provided. In this case, we assumed that the outdoor airflow was constant at the design flow rate. Without control, the outdoor airflow would be somewhat proportional to the supply flow, but would be also influenced by the building exhaust systems and infiltration.

VAV control is limited by the ventilation air requirements in each zone. It is also limited by the performance of the air distribution devices. Most ceiling diffusers provide reasonable comfort with up to a 40 percent volume reduction. A few linear diffusers of superior design provide reasonable space coverage without drafts at up to a 75 percent volume reduction.

In dry climates, the useful volume reduction range of a VAV system can be extended by resetting the supply air temperature upwards as the system airflow is reduced.

In most climates, however, resetting the supply air temperature upwards will result in unacceptable space humidity at part load. In humid climates, the part load humidity performance is improved by letting the cooling coil leaving air temperature drop several degrees as the airflow is reduced. A potential problem is that the supply airflow will be further reduced by the colder supply air temperature, thus reducing the useful volume reduction range of the system.

Outdoor Air cfm

The amount of ventilation air required at the central system intake, for all HVAC systems, is stipulated in* ASHRAE Standard 62.1 Ventilation for Acceptable Indoor Air Quality. *The 2004 release of Standard 62.1 revises the entire ventilation rate procedure in many key areas, making it critical that designers review these new requirements.

An alternative to improve both humidity and space airflow is to provide upward supply air reset, using face and bypass control of the cooling coil supply air temperature. This will provide increased airflow and reduced space humidity at low loads.

Equipment Selection

VAV control cannot be used unless the HVAC equipment fans and cooling system are designed to operate satisfactorily throughout the necessary range of supply airflows and cooling loads.

Small DX packaged units can be used successfully with special VAV systems such as the Carrier VVT (variable volume and temperature) system, which limits the range of volume reduction that can occur.

For general use with VAV control, DX equipment must be provided with an adequate number of stages of refrigeration capacity and controlled to provide a reasonable constant supply air temperature. Special versions of most types of packaged equipment are designed for VAV applications.

Chilled water equipment is simple to apply to VAV control because any chilled water coil can be easily arranged to control the air temperature leaving the cooling coil.

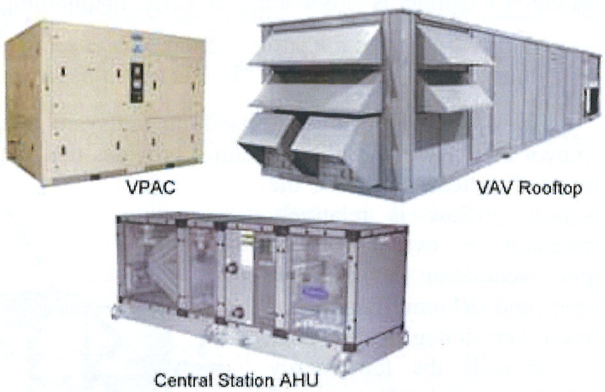

Figure 23

VAV Equipment Options

For all VAV equipment, the fans must be capable of operating satisfactorily throughout the expected range of airflows and operating pressures. Small capacity forward-curved fans with discharge dampers are used on many lower pressure class systems (<2 in. wg static pressure). Variable inlet vanes provide better operation with higher pressure class systems and those using backward-curved or airfoil wheel fans. Electronic variable speed control provides the best control and efficiency for all types of fans.

Return air and/or relief air fans are more difficult to apply with VAV systems, because they must be arranged and controlled to operate in conjunction with the supply fan without over-pressurization or under-pressurization of the building. A building pressurization analysis is necessary to determine whether such fans are needed. Return and relief air fans are usually not necessary, unless the system has extensive return air ductwork, and is provided with an outdoor air economizer that is sized for 100 percent of the system peak load airflow. Then the large amounts of outdoor air that are brought into the building to cool the building during mild weather, plus the return ductwork static pressure, usually require removal by fans to avoid over-pressurization of the building.

Face and Bypass Control (not used in Sg).

This type of control consists of a face damper that controls the airflow through the cooling coil, and a bypass damper that permits air to bypass around the cooling coil. The two dampers are usually cross-linked and operated by a common damper actuator, so that the bypass damper opens as the face damper is closed and vice versa, maintaining the constant airflow through the unit.

Single-Zone Face and Bypass

Draw-thru single-zone air-handling units are arranged with face and bypass dampers as shown here. Face and bypass control provides the best overall functional performance for commercial systems. Because the supply airflow is relatively constant, it overcomes the poor ventilation, air distribution, and dehumidification of the other strategies, but does not provide the fan energy savings of VAV control. Chilled water coils are usually provided with control valves to shut off the chilled water flow through the coils to prevent overcooling of the space due to air leakage through the coil face damper at low cooling load.

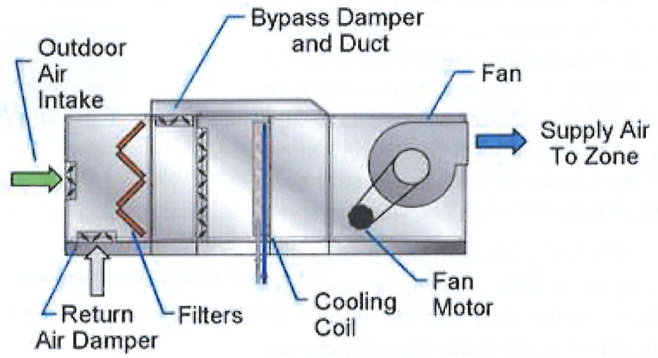

Figure 24

Air Source Components, Single-Zone Face and Bypass Control

Multizone Face and Bypass

Multiple face and bypass zones can be provided from a single air supply unit. A generic blow-thru multizone unit is shown here. A multiple zone damper assembly is installed at the cold air and bypass air plenum connections at the discharge of the unit. The damper assembly provides a cross-linked face damper and bypass damper for each zone that is to be connected to the unit.

Multizone systems are physically limited by the number of control zones that can be installed at the discharge of an air-handling unit and by the duct space. This duct space is necessary to install a separate supply duct from each zone damper connection to the zone which it serves out in the building.

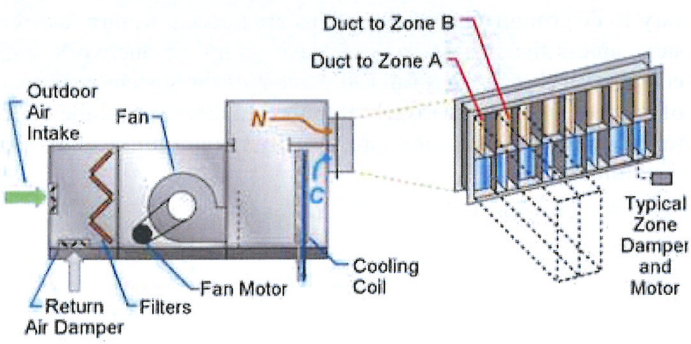

Figure 25

Air Source Components, Multizone Face and Bypass Control

Dual-Duct

Dual-duct systems overcome these limitations by installing higher pressure/velocity class ducts throughout the building from the cold air and bypass air plenum connections of the multizone unit, as shown here. A dual-duct mixing box is installed at each control zone. It mixes the cold air and bypassed neutral air. Dual-duct systems are limited by the space and cost of the supply ducts. Portions of the cold ductwork can be sized based on the diversified peak block cooling airflows of all the downstream zones. The bypass duct can provide warm air to heat the building during cold weather, as described later under blending control. It is usually sized for 60 to 75 percent of the cold airflow.

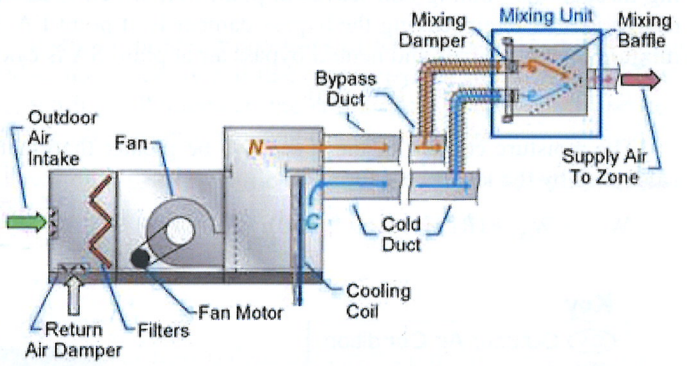

Figure 26

HVAC System Components, Dual-Duct Face and Bypass Control

Operating Characteristics

Face and bypass control maintains humidity like VAV control, with the air distribution and ventilation characteristics of all constant volume control strategies. Blow-thru multizone and dual-duct air blending systems provide face and bypass control when their heating coils are inactive.

The space temperature control modulates the amount of fan discharge air that passes through and around the cooling coil. The coil airflow is roughly proportional to the percentage of RSH load. The bypass airflow percentage is the inverse of the coil airflow, so the fan airflow is roughly constant. Most bypass dampers have a leakage rate of 5 to 10 percent, so the fan design airflow must be that much greater than the design coil airflow.

Psychrometric Analysis

The part load psychrometric performance is shown here. The mixture of room air and outdoor air entering the cooling coil at point EA is the same as for other systems. The portion of air passing through the cooling coil leaves at point LA, which is assumed to be at the same adp as the design load. The air leaving the bypass damper is at point EA. The temperature of the supply air mixture of cool coil air and neutral bypass air at point SA is calculated as follows:

$t_{sa} = t_{rm} - (RSH \div 1.10 * cfm)$

The moisture content of the space will be greater than that of the supply air mixture, and is calculated by the latent heat formula:

$W_{rm} = W_{la} + (RLH \div 0.69 * 1.10)$

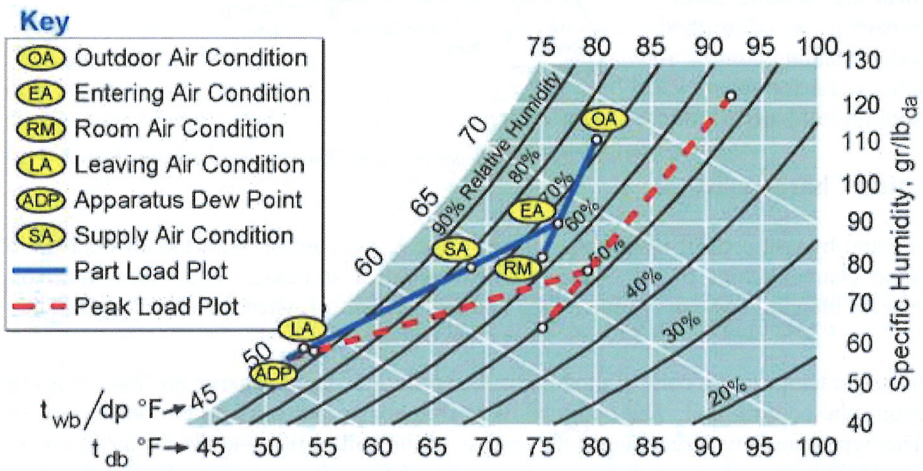

Figure 27

Psychrometric Plot, Face and Bypass Control

The space humidity is approximately 64 percent rh at the part load condition. This is considerably better than the 77 percent rh, which would result if coil discharge temperature control were used, but not as good as the 58 percent rh, which would result with VAV control.

In most climates, resetting the cooling coil t_{la} (and the adp) upwards at part load will result in unacceptable space humidity. In humid climates, the part load performance is improved by letting the adp drop several degrees as the bypass damper is opened.

Equipment Selection

Most face and bypass systems are arranged to take bypass air from the outdoor and return air mixing plenum. This bypasses some of the outdoor air around the coil, resulting in higher space humidity than would otherwise occur. The air handler can sometimes be arranged to bypass 100 percent return air, but this requires a separate bypass air filter bank with the associated cost and space.

Fan Motor Horsepower

The fan motors of forward-curved fans with face and bypass dampers should be sized to handle the maximum airflow condition, which results in considerable increase in fan motor brake horsepower.

The airflow resistance of the coil and bypass paths will be approximately 25 percent of their design values when half of the air is flowing through each path. Because of this variable resistance, the fan airflow will also vary unless automatic airflow control is provided. The fan airflow will be at design value when the bypass damper is closed, increasing to about 115 percent of design when the dampers are at midpoint, and decreasing back to approximately the design value when the coil face dampers are closed.

To prevent ice buildup, the cooling coils of DX equipment with face and bypass control must be provided with multiple steps of capacity control and proper refrigerant circuiting, similar to that required for DX-VAV cooling coils due to the variable airflow through the cooling coil. As mentioned previously, air-handling units with multiple cooling coils can be controlled so that one cooling coil is completely inactive before the capacity of the next cooling coil stage is reduced. The result is similar to that of single-coil face and bypass control with discrete steps of refrigeration capacity.

Most small DX packaged equipment is not designed for use with face and bypass control. Central station air-handling units are usually available with an optional face and bypass zoning damper assembly to provide a multizone system. The same unit without the zoning damper can serve a dual-duct system, or two separate single-zone air-handling units can be provided to supply the cold and neutral air ductwork.

Hot and Cold Air Blending

This type of control exists when a hot and a cold airstream are blended to form the supply airstream. For hot and cold air blending to occur, both streams must be significantly cooler or warmer than the space temperature. Otherwise, face and bypass cooling will occur when the heating coil is inactive, and face and bypass heating will occur when the cooling coil is inactive.

A single heating coil could be installed in the bypass plenum. With both cooling and heating coils energized, each of the zones will position its zone dampers to blend the required amounts of cold and hot supply air to satisfy the zone load.

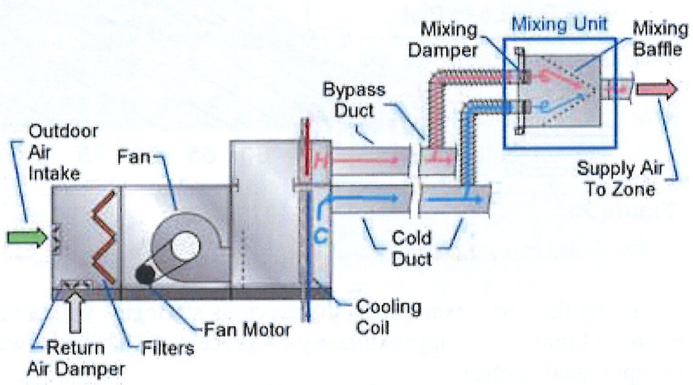

Figure 28

HVAC System Components, Dual-Duct Air Blending Control

Energy waste is the major disadvantage of this control. Any part load condition of either cooling or heating cannot be satisfied without wasting energy of the opposite type. A 90 percent cooling load is handled by reducing the cold airflow and adding some neutral air. A 90 percent heating load is handled by reducing the neutral airflow and adding some cold air.

Operating Characteristics

This control strategy is similar to face and bypass control, except that a heating coil is installed in the bypass around the cooling coil, so that the cold air is cold with respect to the space and the hot air is hot with respect to the space. Face and bypass occurs when one of the two airstreams is neutral with respect to the space.

Psychrometric Analysis

The heating coil is usually inactive during the peak cooling load, so the design load performance is the same as that for face and bypass systems. The typical part load performance shown here is for an outside weather condition in which the heating coil is being operated to provide humidity control for a critical application. The mixture of outdoor and return air at EA is the same as with other systems. The airflow splits at this point, with one portion being cooled and dehumidified to a condition leaving the cooling coil at point CSA (cold supply air), and the other portion being heated sensibly to a condition at point HSA (hot supply air). The proper proportions of these two airstreams are mixed to obtain point SA, which is the supply air condition that will absorb the RSH and RLH to maintain the required space condition at point RM.

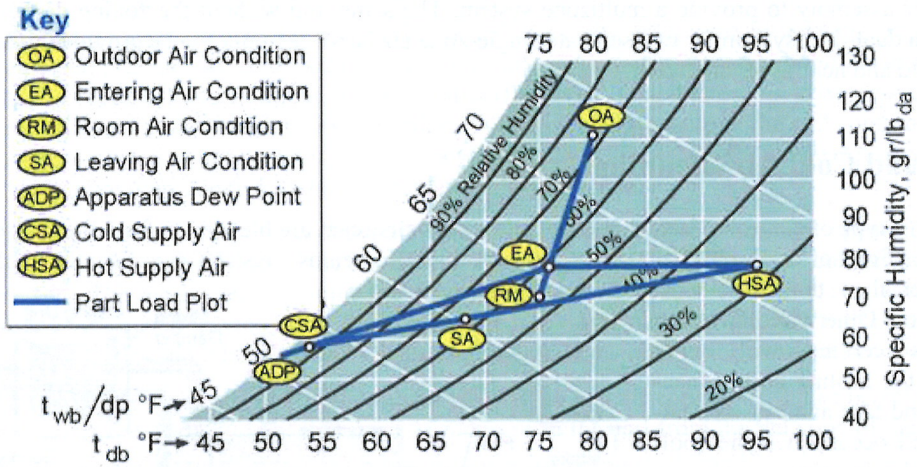

Figure 29

Psychrometric Plot, Air Blending Control

Using the same example as the previous strategies with a hot air temperature of 95° F results in space humidity of approximately 54 percent rh. This is lower than any other control strategy except reheat control.

Resetting the temperature of the cold airstream at point CSA affects the space humidity, with a lower temperature producing lower space humidity. The temperature setting of the hot airstream at point HSA also affects the space humidity by changing the proportion of cooled and dehumidified airflow. A warmer setting results in greater dehumidification, but more heating energy waste.

Like face and bypass control, there is a tendency for the total system airflow to increase when the airflow is divided through both cooling and heating coil paths. The fan motor must be selected for the maximum flow condition unless automatic air volume control is provided. In addition, like face and bypass control, most systems are arranged for both cooling and heating coils to be supplied from a common outdoor air and return air mixing plenum. This results in some of the high humidity outdoor air passing through the heating coil into the space, thus increasing the space humidity compared to what would occur if all of the outdoor air was directed through the cooling coil. Custom air-handling units are usually arranged for 100 percent return airflow to the heating coil, with all of the ventilation outdoor air directed through the cooling coil.

DX cooling coil limitations are similar to those for face and bypass control.

Operational Variations

Most hot and cold blending systems, both multizone and dual-duct type, are arranged with a constant volume supply fan discharging into a plenum from which the air can enter either the cooling coil or the heating coil. With hot and cold air blending control, it is generally not desirable to provide an outdoor air economizer cycle because most hot and cold mixing systems use a common blow-thru fan to serve both ducts. The increase in heating load is usually greater than the decrease in cooling load.

Economizers

An outdoor air economizer cycle is not appropriate with this arrangement. Opening the outdoor air damper to provide economizer cooling results in cold air entering the heating coil as well, greatly increasing heating load.

A waterside economizer, in which the cold outdoor air is used to provide chilled water for the cooling coil, is a much more effective type of economizer. A waterside economizer will provide chilled water to cool the cold duct air without increasing the heating load for the hot air duct. An even better alternative is to operate the refrigeration plant to satisfy the cooling coil load, with a heat recovery system to divert as much of the heat rejection as necessary to satisfy the heating coil load.

Hot and cold air blending is more energy efficient than constant volume reheat, but less efficient than any of the other control strategies. Therefore, its use is restricted by energy codes to critical applications. Systems with heat recovery as described above are usually considered acceptable for almost any application.

Combinations of Basic Control Strategies

Each of the control strategies described above has unique advantages, but none of them, when used alone, are perfect for all applications. A combination of several strategies can usually provide a system that will serve the needs of a project better than any one method.

For example, the VAV control strategy is the most energy efficient option because it is the only one that provides fan horsepower savings. VAV can be combined with face and bypass when using a multizone or dual-duct system, or with reheat when using a single-duct multiple-zone system. The result is a system that is more energy efficient and self-balancing than either constant volume face and bypass or reheat, but without the air distribution and ventilation problems that can occur with full-range VAV.

Reheat control can be combined with either face and bypass or coil discharge air temperature control to create a system that will provide constant airflow and positive humidity control without as much energy waste as a constant volume reheat system.

The blow-thru multizone system shown in Figure 25 uses face and bypass of the cooling coil, which provides good performance when all of the zones require cooling. However, many old multizone systems have a single heating coil located in the bypass plenum that is activated whenever any of the zones require heating. The system operates with hot and cold mixing control during this mode, which is very wasteful of cooling and heating energy. Modern multizone systems include a separate heating coil installed in the supply duct to each zone, sequenced with the zone damper so that all of the air that is to be heated will bypass the cooling coil. Operation of the heating coil for one zone has no effect upon the air supplied to the other zones.

Control Strategy Recommendations

As we have discussed, there are six fundamental space temperature control strategies. All types of HVAC systems use one or more of these strategies. With the same HVAC system, the control strategy can change from one type to another depending upon whether the heating coil is active, or whether the space humidity is outside of the acceptable envelope. The control strategy must be defined in the control system sequences of operation. All of this is part of the iterative process of selecting an HVAC system for the design project, which includes assessing the impacts of using particular space temperature control strategies. While most control strategy impacts focus on comfort, both first cost and operating cost considerations include the impacts of controls.

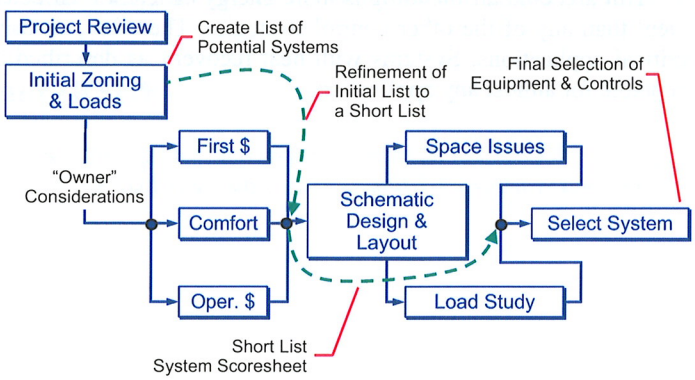

Figure 30

System Selection Process, Controls an Important Priority

The availability of a particular control strategy depends upon the design of the HVAC equipment, in particular whether space is available at the proper location within the equipment for the necessary control dampers and heating coils. These components may or may not be available as accessory items for a particular item of equipment. It also depends upon code requirements regarding energy usage and ventilation.

In comparison with the others, each control strategy has advantages and disadvantages with respect to the responsiveness of temperature control, the space humidity that is likely to occur during part load conditions, the amount of energy used by the operation of the control strategies, and the cost of the necessary control components and equipment arrangements.

These general recommendations are appropriate for climates where dehumidification of outdoor ventilation air is necessary:

- On-off control is best for small packaged DX units and for chilled water fan coil units.
- Coil discharge air temperature control is best for room-type chilled water fan coil units that are provided with a separate preconditioned ventilation air system to maintain the space humidity independently of the fan coil unit operation.
- Reheat control is best for code-acceptable applications where the part load humidity cannot be adequately controlled by other strategies. Examples include process, healthcare, and laboratories with specific relative humidity requirement, or large outdoor air quantities for ventilation or exhaust hood makeup.
- VAV control is best for most office applications, especially with demand controlled ventilation, and reheat where code-acceptable.
- Face and bypass control is best where constant airflow and ventilation are important, such as meeting rooms and training rooms.
- Hot and cold air blending control is an excellent alternative to reheat control for applications that cannot be served adequately by the other control strategies.

Finally, the availability of DDC (Direct Digital Control) microprocessor-based control systems makes it relatively easy to combine several of the basic control strategies to form a hybrid control sequence that will better serve the needs of a specific project.

Humidity Control in Air-Conditioning Systems

Most air-conditioning systems provide dehumidification by cooling the supply air to a dew point that is lower than the desired space dew point. Exceptions are systems with desiccant cooling and those located in desert climates where dehumidification is not necessary. Information from the cooling load estimate is used to calculate the system airflows and the supply air temperature leaving the cooling coil.

Cooling Selection and Coil Performance

The combination of cooling coil airflow rate, cfm_{coil}, and leaving air temperature, t_{la}, must provide both the required RSH and RLH capacity. A colder temperature leaving the cooling coil will provide more dehumidification, usually with greater energy consumption to produce the colder coil. A warmer temperature leaving the cooling coil may provide inadequate dehumidification, resulting in less comfort and possible indoor air quality problems. Most designers use a trial-and-error process for coil selection, first assuming a coil t_{la}, and then calculating the airflow that is necessary to satisfy the RSH. Using this airflow and the moisture content at the assumed coil t_{la}, they calculate the RLH cooling capacity and/or the specific humidity (W_{rm}) that would result from the assumed coil t_{la}. The entire process is repeated with a new assumed coil t_{la} if the original calculated latent performance is not acceptable. Figure 31 shows possible coil performance/room resultant relative humidity for several assumed coil t_{la}.

	A	B	C	D	E
RSH (Sensible)	8000	8000	8000	8000	8000
Room t_{db}	75	75	75	75	75
Coil t_{ldb}	50	52	54	56	58
Room Δt	25	23	21	19	17
Required cfm (1)	291	316	346	383	428
RLH (Latent)	2000	2000	2000	2000	2000
gr/lb_{da} lvg coil	51	55	59	63	68
gr/lb_{da} Diff. to rm (2)	9.8	9.2	8.4	7.6	6.8
Room gr/lb_{da}	60.9	64.1	67.3	70.5	74.8
Room %rh	47	50	52	55	58

(1) = RSH ÷ (1.10 * Temperature Difference)
(2) = RLH ÷ (0.69 * cfm)

Figure 31
Possible Coil Selections

Supply air temp ↑, air volume ↑. RH ↑

The same technique can be used to analyze the part load performance of a system. The apparatus dew point (adp) process is a direct approach. The RSHF is calculated and used with the psychrometric chart or tables to determine the coil leaving air temperature that will exactly satisfy both sensible and latent cooling loads simultaneously. The introductory cooling and dehumidification process psychrometric plot, Figure 11, shows the relationship of adp to the coil t_{la}.

Part Load Operating Conditions

The system designer must select equipment with sufficient capacity to handle the design loads. Many air-conditioning systems are designed with safety factors for reasonable pulldown time and potential future loads. The result is that most systems never operate at their peak cooling or heating capacity, except during pulldown after extended night or weekend unoccupied periods.

The load variation for a given space throughout the day and the season depends upon many factors:

- Time of day and solar orientation of the space
- Outside weather conditions and solar cloud cover
- Occupancy of the space
- Usage of lights and internal equipment
- Scheduling of control set points

Sensible Cooling and Heating Loads

Figure 32 shows a profile typical RSH load components in an interior office on an intermediate floor of a multi-story building at varying outside temperatures. If the surrounding offices are maintained at proper temperature, this office is not affected by outside weather changes and heating will never be required. Depending upon how much lighting, equipment, and people are present, however, the space temperature control strategy must be able to function properly at any load from 0 to 100 percent of peak RSH.

The same office on a top floor will be different because of the addition of the roof load, which may be either a cooling or heating load, depending upon the outside conditions. Top floor interior zones may require some heating throughout the day, depending upon the presence of a ceiling return plenum, a well-insulated roof, and a large, constant lighting load.

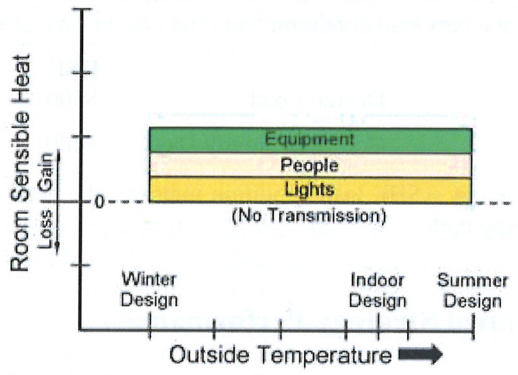

Figure 32

Interior Office Design Load

Figure 33 shows the RSH load profile for a perimeter office. The transmission loss through the exterior walls and windows results in a possible need for heating when the outside temperature drops more than a few degrees below space temperature. The solar load is a highly variable load, being influenced by building orientation, time of day, cloud cover, and season of year. Shadows from trees or adjacent buildings can result in one office being in full sun while the adjacent office on the same exposure is in full shade. A perimeter space temperature control may have to operate at any point within the load profile, shifting from cooling to heating within a short time.

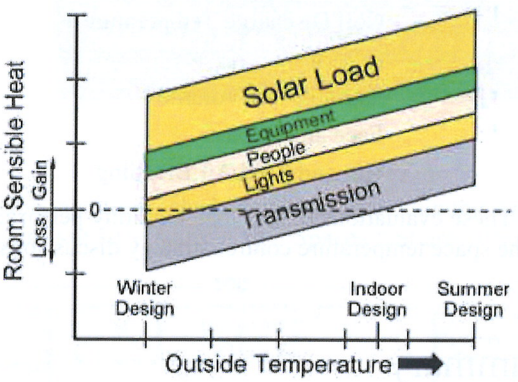

Figure 33

Perimeter Office Design Load Profile

Latent Loads

The RLH load is due to occupants and to activities within the space such as cooking. These latent loads are equally possible regardless of outside weather. The result is that the RSHF can become very low at part load conditions in exterior zones, even though it is quite high at peak load. This table shows the room sensible heat ratio for a typical perimeter office, at design load, and at a part load condition that might occur on a cloudy summer morning.

	RSH	RLH	RSHF
Design Load	8500	1500	0.85
Cool and Cloudy Day	3400	1500	0.69

A low RSHF load condition indicates the need for a cold dew point temperature, t_{adp}, at the cooling coil.

Control Strategy Performance

For the same space, at the same part load condition, each space temperature control strategy will result in a different space rh. The control strategies will generally perform as follows at 40 percent RSH part load:

Control Strategy	Part load rh
On-Off	64%
Coil Discharge Temperature	77%
Reheat	50%
Variable Air Volume	58%
Face and Bypass	64%
Hot and Cold Air Blending	54%

These evaluations of part load humidity performance were based on the same conditions used in the space temperature control strategy discussions.

Summary

Comfort is the primary design parameter for air-conditioning systems. Since space temperature is the foundation for determining occupant comfort satisfaction, control strategies for HVAC systems should first focus on proper control of space temperature. Space relative humidity, especially under part load conditions, must be carefully reviewed when selecting control strategies for the HVAC system design. Energy codes will limit the use of strategies that employ simultaneous heating and cooling. Always utilize psychrometric plots of system part load performance to evaluate the use of various central equipment and zone control strategy options.

Work Session

1. What is the most common comfort parameter used to control HVAC systems?

   _____

2. Zoning is used to group spaces with similar load characteristics. This maximizes comfort without providing a thermostat for every space. What are three common zoning methods?

   a) _____

   b) _____

   c) _____

3. HVAC equipment capacity is modulated, or controlled, to affect a change in room temperature by controlling one of the three fluids listed below. After each, write in at least two controlled devices that would be used in an HVAC system to implement the control strategy.

   a) Air: _____

   b) Water: _____

   c) Refrigerant: _____

4. List the six basic space temperature control strategies.

   a) _____

   b) _____

   c) _____

   d) _____

   e) _____

   f) _____

5. Which one of the eight psychrometric processes represents the actions of the space temperature control strategies presented in this TDP?

   _____

6. Simple on-off control has many disadvantages created by the cycling of the fan; list the one major benefit.

   _____

   _____

7. Each space temperature control strategy has distinct advantages and disadvantages. List the primary advantage (think feature) of the following strategies:

   a) Coil discharge temperature control:

   _____

   _____

   b) Reheat control:

   _____

   _____

   c) Variable air volume (VAV) control:

   _____

   _____

   d) Face and bypass control:

   _____

   _____

8. Simultaneous heating and cooling strategies, best represented by reheat are not permitted by Energy Conservation Codes in most applications. Name two or three applications that are given exceptions, and for what reasons.

   _____

   _____

   _____

9. True or False? Cooling coil selection parameters, like leaving air temperature and/or airflow, can be adjusted to meet both the room sensible heat and room latent heat loads.

   _____

10. Part load operation occurs on many projects due to variation in load components. Name five conditions that would affect load variations in the conditioned spaces.

    a) _____

    b) _____

    c) _____

    d) _____

    e) _____

Appendix

References

Carrier Corp.

Variable Air Volume Systems. 2005.
TDP-703 Book, Cat. No. 796-068
TDP-703 Instructor Presentation, Cat. No. 797-068

Rooftops, Level 1: Constant Volume. 2005.
TDP-631 Book, Cat. No. 796-056.
TDP-631 Instructor Presentation, Cat. No. 797-056

Demand Controlled Ventilation System Design. 2001
Cat. No. 811-10088.

ASHRAE

ASHRAE Standard 62.1 - 2004, "Ventilation for Acceptable Indoor Air Quality"

ASHRAE Standard 90.1 - 2004, "Energy Efficient Design of New Buildings Except Low-Rise Residential Buildings"

Work Session Answers

1. Space dry bulb temperature.

2. Three common zoning methods are:
   a) Operating schedule zoning
   b) Air quality zoning
   c) Temperature control zoning

3. Controlled devices used in an HVAC system to implement the control strategy are:
   a) Air: fan, outdoor air damper or VAV terminal.
   b) Water: pump, 2-way valve or 3-way valve.
   c) Refrigerant: solenoid valve or electronic expansion valve (EXV).

4. Six basic space temperature control strategies are:
   a) On-off control
   b) Coil discharge temperature control
   c) Reheat control
   d) Variable air volume (VAV) control
   e) Face and bypass control
   f) Hot and cold air blending control

5. Cooling and dehumidification

6. When the fan turns off after satisfying the cooling load, the wetted coil surface does not re-evaporate the condensate into the supply air, raising space relative humidity.

7. Primary advantages:
   a) Coil discharge temperature control: precise control of the space dry bulb temperature using a modulating chilled water coil control valve.
   b) Reheat control: maximum dehumidification capacity to the space at all times.
   c) Variable air volume (VAV) control: variable airflow to multiple zones from a single piece of central cooling equipment.
   d) Face and bypass control: improved dehumidification performance from the central cooling equipment during part load conditions.

8. Laboratories, healthcare, or process, for special pressurization relationships, cross-contamination requirements, or relative humidity needs.

9. True. A colder temperature leaving the cooling coil will provide more dehumidification, usually with greater energy consumption to produce the colder coil.

10. Conditions affecting load variations in conditioned spaces are:
    a) Time of day and solar orientation of the space.
    b) Outside weather conditions and solar cloud cover.
    c) Occupancy of the spaces.
    d) Usage of lights and internal equipment.
    e) Scheduling of control set points.

Prerequisites:

An understanding of human comfort parameters, air-conditioning equipment and systems, along with fundamentals of load estimating and psychrometric properties of air-water mixtures. This knowledge can be gained from TDP-102 ABCs of Comfort, TDP-103 Concepts of Air Conditioning, TDP-301 Load Estimating Level 2: Fundamentals, TDP-302 Load Estimating Level 3: Block and Zone Loads, and TDP-201 Psychrometrics Level 1: Introduction.

Learning Objectives:

In this module, participants will learn the skills and knowledge necessary to:

- Identify the different space temperature and relative humidity control strategies.
- Compare typical peak design operation and part load operation of the strategies.
- Categorize control strategy availability for many types of HVAC systems.
- Select the appropriate control strategy, considering the building, application, and trade-offs.

Supplemental Material:

Those who wish to build their knowledge in control of HVAC systems should consider the following related publications for their library:

TDP No.	Book Cat. No.	Instructor CD Cat. No.	Title
TDP-701	796-066	797-066	System Features and Selection Criteria
TDP-801	796-074	797-074	Controls, Level 1 Fundamentals
TDP-631	796-056	797-056	Rooftop Units, Level 1: Constant Volume Units
TDP-632	796-057	797-057	Rooftop Units, Level 2: Variable Volume Units
TDP-703	796-068	797-068	Variable Air Volume Systems
TDP-704	796-069	797-069	Variable Volume and Temperature 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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