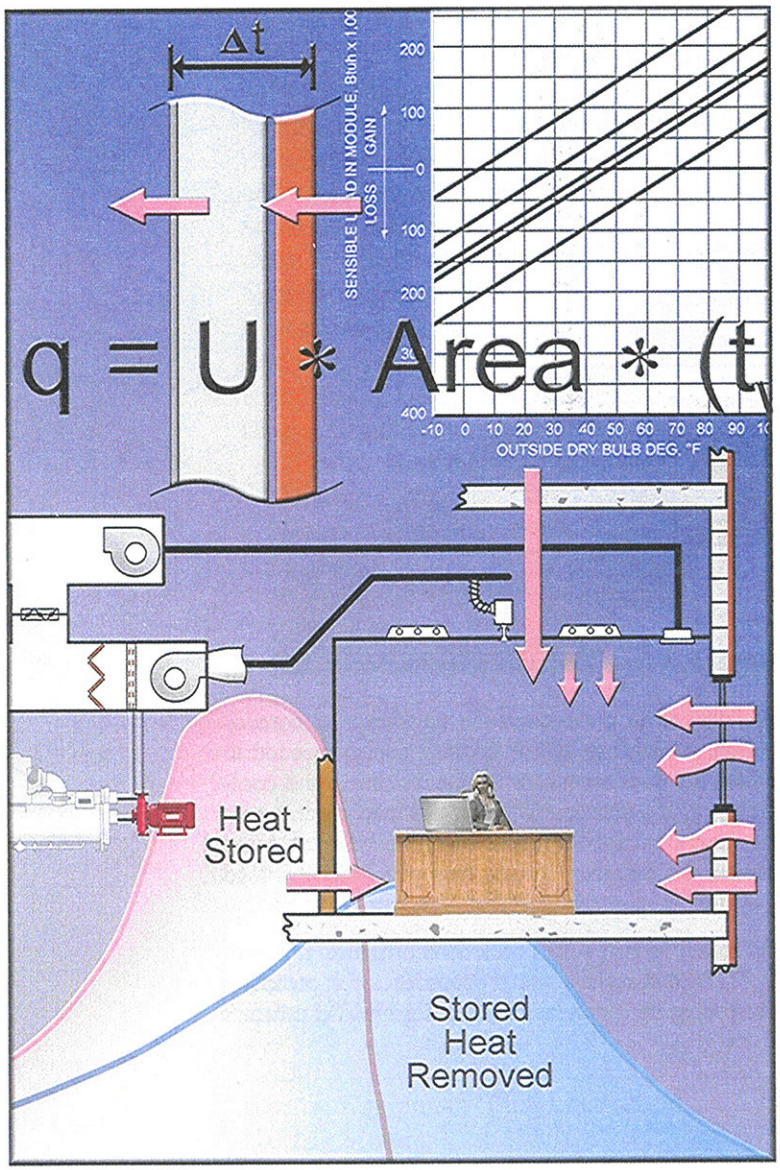
Carrier

Turn to the Experts.



COMMERCIAL LOAD ESTIMATING

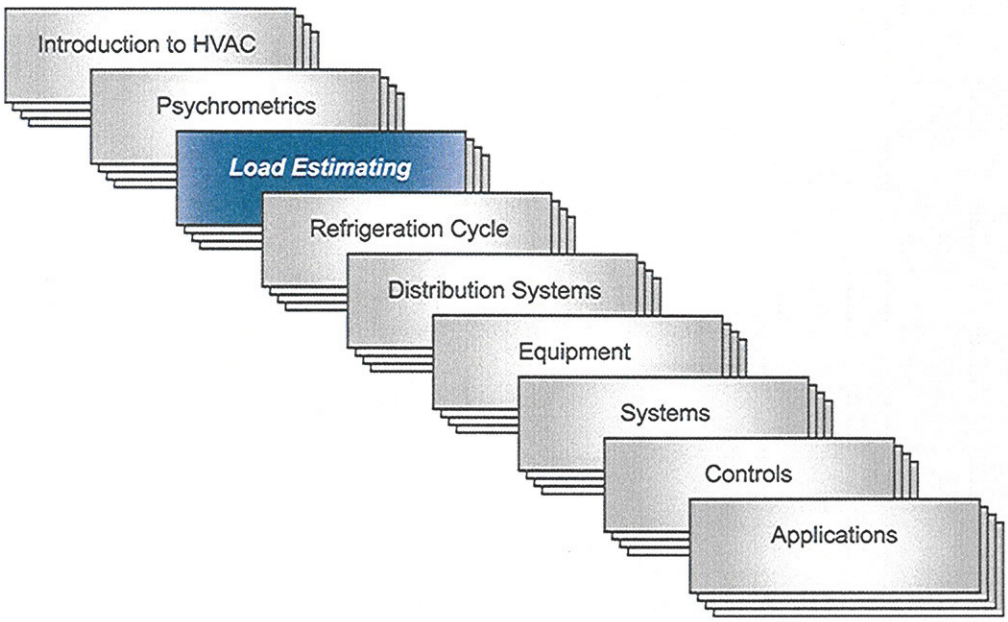
Load Estimating

Level 2: Fundamentals

Technical Development Program

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

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



The fundamentals of commercial load estimating are needed to understand the various load components that go into making a practical estimate of the amount of heating and/or cooling energy needed to condition a building. Done properly, a load estimate provides the data necessary to select heating and cooling equipment that can condition the spaces within a building. If the characteristics of the load components for the building and the HVAC system are known, then an analysis of the application can be used to come up with the correct load and equipment selections to complete the design. Along with psychrometrics, load estimating establishes the foundation upon which HVAC system design and operation occur.

Load Estimating, Level 2: Fundamentals is the second in a four-part series on load estimating. It is preceded by an overview of the topic and followed by two TDPs that review the procedures for completing block and zone load estimates and refinements required for preparing a system-based design load estimate.

© 2005 Carrier Corporation. All rights reserved.

The information in this manual is offered as a general guide for the use of industry and consulting engineers in designing systems. Judgment is required for application of this information to specific installations and design applications. Carrier is not responsible for any uses made of this information and assumes no responsibility for the performance or desirability of any resulting system design.

The information in this publication is subject to change without notice. No part of this publication may be reproduced or transmitted in any form or by any means, electronic or mechanical, for any purpose, without the express written permission of Carrier Corporation.

Printed in Syracuse, NY

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

Table of Contents

Introduction 1
Importance of Load Estimating 1
Load Estimate versus Load Calculation 1
Types of Load Estimating 3
Rules of Thumb 3
Manual vs. Software Generated 3
Space, Zone, and Block Loads for Equipment Sizing 4
Safety Factors 6
Loads Run for HVAC System Modeling 7
Energy Code Requirements 7
Load Estimating Methods 8
Storage Load Factor/ETD Method 9
Cooling Load Temperature Differential/Cooling Load Factor (CLTD/CLF) Methods 10
Transfer Function Method 10
Radiant Time Series Method 12
Choosing Which Method to Use 12
External Space Loads 13
Design Conditions 13
Climatic Design Conditions 13
Cooling 14
Heating 15
Solar 15
Window and Glazing Types 16
Coatings 17
Shading - Internal, External, and Adjacent Buildings 17
Transmission 18
Wall and Door 18
Roof 18
Windows and Skylights 19
On Grade and Below Grade 19
Infiltration 19
Wind and Stack Effects 20
Sensible and Latent Components 20
Partitions, Floors, and Ceilings 21
Internal Space Loads 21
People 21
Lighting 22
Electric Equipment (Plug Load) 23
Electric Motors 23
Gas-Fired Equipment 24
Piping, Tanks, and Evaporation 24
Bare and Insulated Piping 24
Steam Leaks, Absorption, and Evaporation 25

System Loads 25
Ventilation Outdoor Air 26
Sensible and Latent Loads 26
Duct Heat Transfer and Airflow Leakage 26
Duct Heat Gain 27
Duct Leakage Loss 27
Fan Horsepower and Motor Heat Gains 28
Bypassed Outdoor Air 29
Plenums 29
Ceiling Plenums 29
Underfloor Plenum 30
Load Component Profile 31
Building the Profile 31
Balance Points 32
Plotting on the Psychrometric Chart 32
State Points 33
Room Sensible Heat Factor (RSHF) Line 33
System Lines 34
Heating Load Estimate 34
Design Conditions 35
Transmission 35
Floor and Basement Heat Losses 35
Infiltration 37
No Credits 38
Warm-Up Safety Factor 38
Summary 38
Work Session 1 39
Appendix 42
Glossary 43
Cooling Check Figures 45
Approximate Design Air Distribution cfm (Cooling) 46
Air Conditioning Load Estimate Form 47
ASHRAE Standard 90.1-2001 48
Table 2 – Corrections in Outdoor Design Temperatures for Time of Day 50
Table 3 – Corrections in Outdoor Design Conditions for Time of Year 50
Table 59 – Heat Gain from Air Conditioning Fan Horsepower, Draw-Thru System 51
Heating Load Estimate Form 52
References 53
Work Session Answers 53

Introduction

This Technical Development Program (TDP) on commercial load estimating presents the fundamentals needed to understand the various ***load components*** that go into making a practical estimate of the amount of ***heating*** and/or cooling energy needed to provide comfort conditioning of a building. Once a project begins to take shape, the job requirements (a scope of work) can be tabulated for the HVAC portion. Given this data, it is possible to perform a ***load estimate*** for the selection of the heating and cooling equipment that will condition the various building spaces. In the earliest stages, the load estimate will tell the designer how big the job is, either in terms of cooling capacity, expressed as ***tons of refrigeration*** or MBtuh (1000s of ***Btuh***, or in terms of airflow in cfm. ***Airflow*** refers to how much conditioned air needs to be supplied to the various ***rooms***, or ***spaces***, to maintain the temperature, humidity, or CO_2 ***set points***.

If the designer knows the elements of the loads for the building and the features of the HVAC system, then the application can be analyzed regardless of the calculation method used, and correct equipment selections can be made to complete the design.

If a simple, ***rule of thumb*** load estimate has been used during preliminary design, then later on in the design process the building and HVAC loads should be calculated in detail. When completed, all the necessary parameters for sizing the selected ***packaged equipment*** (rooftops units and split systems) or ***applied equipment*** (chillers, boilers, and air handlers), along with terminals and air diffusion devices will be available to the designer.

Importance of Load Estimating

An accurate load estimate is needed to get the process of designing, installing, and operating a project off to a good start. The load estimate numbers provide the data for a host of subsequent calculations, selections, and decisions. Among these items are: HVAC system selection; equipment selections for fans, coils, and pumps; duct and electrical feeder sizing; and water piping design.

An accurate estimate will provide the correct cooling and heating requirements, offer options for load reductions at the least incremental cost, provide properly sized equipment, and yield efficient air, water, and electrical distribution designs.

Load Estimate versus Load Calculation

One often hears load estimate and ***load calculation*** used synonymously. A load estimate is an approximation gained from methods that are not detailed, such as a rule of thumb, ***check figures***, or quick hand calculation based on assumptions. Check figures and rules of thumb have been derived for common applications by HVAC designers and are often complied for ease of use. Some common check figures represent cooling in ft^2/ton and people density as ft^2/person.

A load calculation is a more detailed analysis of load components based on actual building design knowledge and is usually performed by computer software spreadsheets and programs. Even though a load calculation is considered more accurate, not all the details of the inputs required by the software are known. The user must rely on good judgment, so the word "estimate" is still appropriate for the results.

Current calculation models have increased the accuracy of software programs. However, simplifying assumptions are a part of these methods too, so as far as trying to approach the reality of nature, it is still an estimate, but on increasingly higher levels.

Proper load estimating requires gathering enough details to obtain the level of accuracy required. Preliminary HVAC load estimates can be gained from minimal project knowledge by use of rules of thumb and check figures, since order of magnitude for preliminary budgets and rough schematics are the goals.

As a project progresses in the design process, more information becomes available and the load estimates must be refined. The methodologies used for the detailed sizing calculations or system modeling often will dictate the type and depth of information that is needed for input into various software programs. These more in-depth subjects will be covered in the two remaining load estimating TDPs.

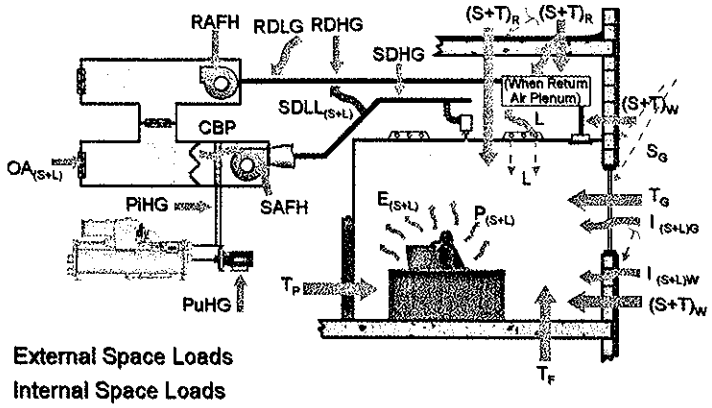


Figure 1

Cooling Estimate Load Components

Load estimates are the summation of heat transfer elements into (gains) or out of (losses) the spaces of a building. Each heat transfer element is called a load component. Load components represent sensible and latent heat transfers (in Btuh) from a multitude of sources, but all can be assembled into one of three basic groups, external space loads, internal space loads, and system loads. Each will be covered in detail in a separate section of this TDP. The Appendix contains a terms list for the abbreviations found in the figure.

To properly understand the workings of the various external, internal and ***system load components***, details regarding the following items will need to be gathered from a set of plans, existing building surveys, or occupant interviews:

- Occupancy of the entire building (office, health care, retail store, educational, etc.)
- Orientation of the building (sun effects on surfaces)
- Weather data (design conditions, heat transfer Δt)
- Use of the spaces within the building (offices, conference room, lab, data center)
- Hours of operation (occupied and unoccupied)
- Thermostat set points (main comfort parameter)
- Dimensions of walls, roofs, windows, and doors
- Construction materials (gather densities, external color, and ***U-factors***, or describe material type layer-by-layer (***R-value***s)
- Stairways and elevators (floor-to-floor openings)
- People occupancy and activity, and when they are present
- Lighting intensity and hours used
- Motor and appliance sizes or kW, and times they are used
- Ventilation needs (IAQ and exhaust makeup)

While many of these items will be explored as this module progresses, the Load Estimating TDP on Block and Zone Loads needs to be consulted for a full ***building load takeoff*** example.

Types of Load Estimating

There are many types of load estimating and calculation procedures. They range from fast and easy hand calculations to detailed descriptions for advanced computational procedures and methods. You might want to think of these like golf clubs or fishing rods. You may need to have more than one type on hand depending on what you need to do.

Rules of Thumb

There are tables of rules of thumb available from various industry sources. One such compilation is found in the Carrier *Commercial Systems Quick Reference* (CSQR). These are reproduced in the Appendix of this manual. The Cooling Check Figures allow for approximation of total cooling needs (building gross area * cooling ft^2/ton), electrical space load component (building gross area * lights + equipment Watts/ft^2), and outdoor air ventilation requirements (building gross area ÷ people ft^2/person * ventilation ***cfm***/person). The Approximate Design Air Distribution cfm (Cooling) table allows early approximation of the ***zone*** primary supply air needs for terminal selection and ductwork space evaluation.

Rules of Thumb

These ballpark numbers are very useful in the preliminary design phases and are good for comparing with the results of more detailed procedures to see if any gross input errors were made.

Manual vs. Software Generated

Going beyond the rules of thumb method, the designer now has to do a load takeoff and provide a level of detail on the project that allows individual computations for the major and hopefully important minor load components.

A manual method can use a tabular form like the Carrier E-20A, Air Conditioning Load Estimate form (see Appendix), or a computer spreadsheet like Excel or Lotus. In one, a pencil and a calculator are used to run a load estimate for a simple building or to study unusual space or zone requirements. In the other, input is still manual, but the computer spreadsheet tabulates the many calculations required for larger multi-room systems with extensive ***zoning*** or multiple central equipment selections to be sized.

Manual methods still have a place, especially in the early stages of design, and when a quick calculation is needed to check sizing requirements or the magnitude of load change created by a change in a takeoff value. Larger commercial jobs, especially ones with unusual load components, should have the cooling and heating loads estimated using one of the many software load estimating programs available to designers. This method proves to be efficient whenever a change in the takeoff at the later stages of design requires a recompilation of the load estimate.

The computer printed outputs generally organize the data well (Figure 2), and in some cases can provide graphical outputs of the detailed calculations, such as a psychrometric plot of the system design (Figure 3). Software-generated load estimates run the gambit from simple load estimating to a full computer ***energy simulation*** derived from detailed inputs that include full system and component descriptions. The energy calculation modeling is described in the energy estimating TDP.

Air System Design Load Summary for Top Floor

Project Name: Chicago Office Building (TDP-301)
09/09/2004 08:40AM
Prepared by:

	DESIGN COOLING			DESIGN HEATING		
	COOLING DATA AT Jul 1500 COOLING OA DB / WB 91.0 °F / 74.0 °F			HEATING DATA AT DES HTG HEATING OA DB / WB -6.0 °F / -7.2 °F		
ZONE LOADS	Details	Sensible (BTU/hr)	Latent (BTU/hr)	Details	Sensible (BTU/hr)	Latent (BTU/hr)
Window & Skylight Solar Loads	2200 ft²	69444	-	2200 ft²	-	-
Wall Transmission	4400 ft²	12367	-	4400 ft²	60557	-
Roof Transmission	14814 ft²	31416	-	14814 ft²	101100	-
Window Transmission	2200 ft²	14972	-	2200 ft²	91111	-
Skylight Transmission	0 ft²	0	-	0 ft²	0	-
Door Loads	0 ft²	0	-	0 ft²	0	-
Floor Transmission	0 ft²	0	-	0 ft²	0	-
Partitions	0 ft²	0	-	0 ft²	0	-
Ceiling	0 ft²	0	-	0 ft²	0	-
Overhead Lighting	23628 W	67014	-	0	0	-
Task Lighting	7876 W	24354	-	0	0	-
Electric Equipment	12759 W	40257	-	0	0	-
People	79	14557	16146	0	0	0
Infiltration	-	0	0	-	0	0
Miscellaneous	-	0	0	-	0	0
Safety Factor	0% / 0%	0	0	10%	25277	0
= Total Zone Loads	-	274381	16146	-	278045	0
Zone Conditioning	-	303189	16146	-	268466	0
Plenum Wall Load	0%	0	-	0	0	-
Plenum Roof Load	0%	0	-	0	0	-
Plenum Lighting Load	0%	0	-	0	0	-
Return Fan Load	12900 CFM	0	-	1213 CFM	0	-
Ventilation Load	1710 CFM	25596	53044	1607 CFM	124115	0
Supply Fan Load	13294 CFM	38601	-	1607 CFM	-2886	-
Space Fan Coil Fans	-	0	-	-	-1570	-
Duct Heat Gain / Loss	2%	5488	-	2%	5561	-
= Total System Loads	-	372874	69190	-	393685	0
Central Cooling Coil	-	372874	69198	-	0	0
Preheat Coil	-	0	-	-	98260	-
Terminal Reheat Coils	-	0	-	-	295386	-
= Total Conditioning	-	372874	69198	-	393645	0
Key:	Positive values are clg loads Negative values are htg loads			Positive values are htg loads Negative values are clg loads		

Figure 2

System Block Load

Space, Zone, and Block Loads for Equipment Sizing

Each of the load types, ***space loads***, ***zone loads*** and ***block loads*** are all used for equipment sizing, and can be obtained from almost any load estimating software or hand calculation method. The purpose is to get the Btuh and fluid quantities (cfm and gpm) for selecting diffusers, sizing fans, and determining the cooling and heating capacities for selecting the zone and system HVAC equipment. A total building or project block load is useful for sizing central cooling and heating plant equipment such as chillers and boilers. A system level block load (Figure 2) provides fan and coil sizing details for the central air-handling units.

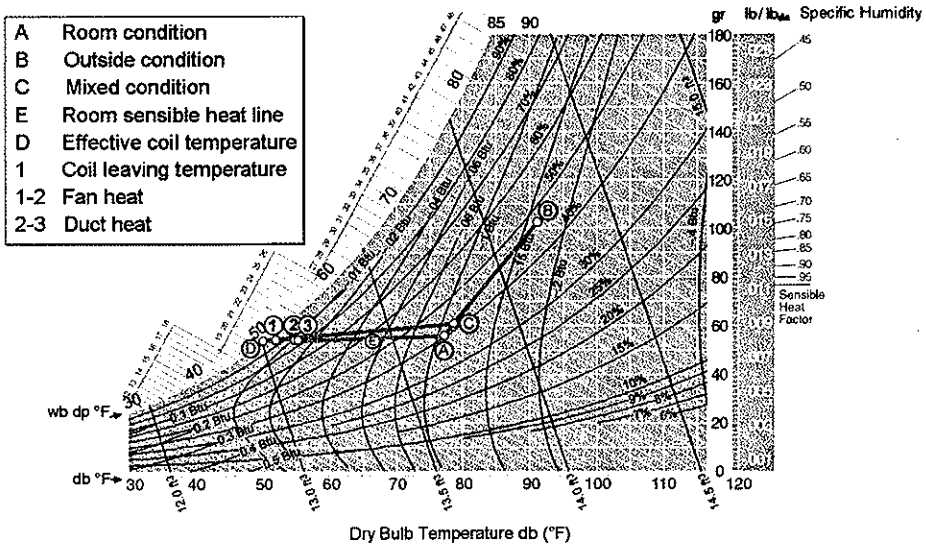


Figure 3

Psychrometric Plot of an HVAC System

A zone load provides the information to select such items as CV and VAV terminals and terminal heating coils (Figure 4), and room air diffusers (Figure 5). The peak block load and zone loads may occur at different times of the day and at different months, so each is usually run. The Carrier *Block Load* program and the *Hourly Analysis Program (HAP)* are examples of this type of software.

One item of importance not listed previously is the type of system equipment that is chosen for the project. Some procedures will have an input for cooling coil ***bypass factor*** or some other input regarding the efficiency of the cooling coil. A typical 20-ton rooftop HVAC unit will have a bypass factor of about 0.08. This will be discussed in later in the module.

Even more important is the cooling supply air temperature used in the calculations. This is needed to select the equipment that will meet the load requirements, and perform as expected on startup. Applied equipment has many more choices than packaged equipment for selecting coil rows, fins, and circuiting; so a 55° F supply air temperature for various system applications can be easily achieved. The lower supply air set point gives better part load dehumidification, very important in limiting room IAQ issues around moisture damage and mold growth.

Zone Sizing Summary for Top Floor

Project Name: Chicago Office Building (TOP-301)
09/09/2004
Prepared by: 08:40AM

Air System Information

Air System Name ..... Top Floor
Equipment Class ..... CW AHU
Air System Type ..... VAV
Number of zones 9
Floor Area ..... 15752.0 ft²
Location ..... Chicago IAP, Illinois

Sizing Calculation Information

Zone and Space Sizing Method:
Zone CFM ..... Peak zone sensible load
Space CFM ..... Individual peak space loads
Calculation Months ..... Jan to Dec
Sizing Data ..... Calculated

Zone Sizing Data

Zone Name	Maximum Cooling Sensible (MBH)	Design Air Flow (CFM)	Minimum Air Flow (CFM)	Time of Peak Load	Maximum Heating Load (MBH)	Zone Floor Area (ft²)	Zone CFM/ft²
North West Corner Top	8.1	385	16	Jul 1600	8.8	156.3	2.46
North East Corner Top	6.9	329	16	Jul 0800	8.8	156.3	2.10
South East Corner Top	8.1	384	16	Sep 0900	8.8	156.3	2.46
South West Corner Top	9.1	434	16	Sep 1400	8.8	156.3	2.78
North Side Top	39.0	1852	188	Jun 1700	59.6	1875.6	0.99
North West Corner Top	36.4	1728	94	Jul 0800	29.8	937.8	1.84
East Side Top	72.9	3456	188	Jul 0800	59.6	1875.6	1.84
West Side Top	39.4	1868	94	Jul 1600	22.8	937.8	1.99
Interior Top	112.0	5314	950	Jul1700	71.3	9500.0	0.56

Zone Terminal Sizing Data

Zone Name	Reheat Coil Load (MBH)	Reheat Coil Water gpm @ 40.0 °F	Zone Htg Coil Load (MBH)	Zone Htg Water gpm @ 40.0 °F	Mixing Box Fan Airflow (CFM)
North West Corner Top	9.0	0.45	0.0	0.0	166
North East Corner Top	9.0	0.45	0.0	0.0	166
South East Corner Top	9.0	0.45	0.0	0.0	166
South West Corner Top	9.0	0.45	0.0	0.0	166
North Side Top	62.5	3.13	0.0	0.0	1131
North West Corner Top	31.3	1.56	0.0	0.0	565
East Side Top	62.5	3.13	0.0	0.0	1131
West Side Top	24.2	1.21	0.0	0.0	432
Interior Top	86.3	4.32	0.0	0.0	0

Figure 4

Zone Sizing Summary

Zone Sizing Summary for Top Floor

Project Name: Chicago Office Building (TOP-301)
09/09/2004
Prepared by: 08:40AM

Air System Information

Air System Name ..... Top Floor
Equipment Class ..... CW AHU
Air System Type ..... VAV
Number of zones 9
Floor Area ..... 15752.0 ft²
Location ..... Chicago IAP, Illinois

Sizing Calculation Information

Zone and Space Sizing Method:
Zone CFM ..... Peak zone sensible load
Space CFM ..... Individual peak space loads
Calculation Months ..... Jan to Dec
Sizing Data ..... Calculated

Space Loads and Airflows

Zone Name / Space Name	Mult.	Cooling Sensible (MBH)	Time of Load	Air Flow (CFM)	Heating Load (MBH)	Floor Area (ft²)	Space CFM/ft²
North West Corner Top							
North West Corner Top	1	8.1	Jul 1600	385	8.8	156.3	2.46
North East Corner Top							
North East Corner Top	1	6.9	Jul 0800	329	8.8	156.3	2.10
South East Corner Top							
South West Corner Top	1	8.1	Sep 0900	384	8.8	156.3	2.46
South West Corner Top							
South West Corner Top	1	9.1	Sep 1400	434	8.8	156.3	2.78
North Side Top							
North Typical Top	12	3.3	Jun 1700	154	5.0	156.3	0.99
East Side Top							
East Typical Top	6	6.1	Jul 0800	288	5.0	156.3	1.84
South Side Top							
South Typical Top	12	6.1	Jul 0800	288	5.0	156.3	1.84
West Side Top							
West Typical Top	6	6.6	Jul 1600	311	3.8	156.3	1.99
Interior Top							
Interior Typical Top	4	28.0	Jul 1700	1329	17.8	2375.0	0.56

Figure 5

Space Sizing Summary

Note:

Most packaged equipment and room fan coils have limited coil circuitry options and as a general guideline, deliver a supply air temperature of about 58° F to 59° F.

Safety Factors

A ***safety factor*** should be a function of the designer's confidence in his or her load survey and the details gathered on the project's construction materials.

There is usually a built in creeping safety factor as the HVAC system is designed. A creeping safety factor occurs as additive items of capacity and safety are added on top of each other due to the nature of the estimating method and equipment sizing procedures. For instance, assume the room airflow requirement is 108 cfm of conditioned supply air. This value was then increased to 113 cfm based on a 5 percent sensible load safety factor. The equipment selected to deliver air to the room would probably then be rounded up and selected for 125 cfm. This is a 10 percent increase on top of the original 5 percent calculation safety factor.

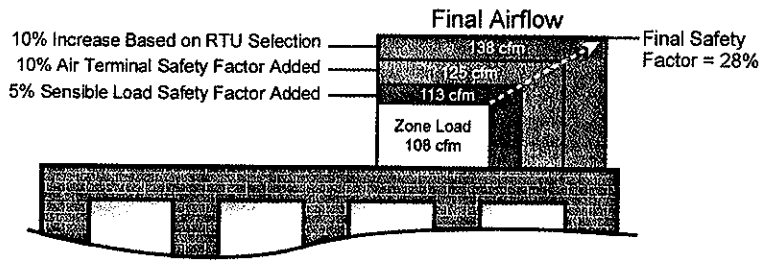


Figure 6

Creeping Safety Factor

Then the total load is calculated for the cooling equipment at 105,445 Btuh (8.79 tons), again based on a safety factor. However, the piece of equipment selected to do the job only comes in discreet sizes such as 5, 7½ and 10 tons. So a 10-ton unit is installed, which is over capacity by 10 percent on a load that already contained multiple safety factors. In fact, for more favorable outside conditions than the standard rating values (80° F db/67° F wb), the unit capacity can be even more oversized.

Minimal Safety Factors

The use of safety factors in the sizing of heating and cooling equipment is discouraged. Anyone who has taken off and double-checked a set of plans knows that there are many factors in addition to the mathematical calculations that affect the heating and cooling loads.

This over-conservative approach to cooling sizing causes over-sized equipment to be installed. This results in less efficient operation and difficulty in controlling room conditions, especially relative humidity and the associated IAQ (***indoor air quality***) problems.

Designers should examine the results of the calculations, understand where the numbers come from, and satisfy themselves that the resulting numbers make sense. Because beginners may be unsure of the calculation methods and output values, the tendency is to use safety factors as insurance. If one feels that this insurance is needed, a factor of a few percent, not to exceed 5 percent, is suggested for cooling. As you will see latter in the heating section, a single safety factor is recommended, often quite large (up to 25 percent) in cold climates.

Loads Run for HVAC System Modeling

There are software applications that can model and size different kinds of HVAC systems and components in more detail than basic block and zone load software. This modeling software allows the designer to do system-based design because the inputs describe schedules, set points and equipment operation unique to the selected HVAC system. For programs like Carrier's *HAP*, by building on the information already used for the detailed load modeling, an annual energy analysis simulation can also be done. Additional items needed beyond the load calculation inputs include hourly weather data for the entire year, comprised of 8760 instances of dry bulb, wet bulb, and solar data, electric and fuel rate costs, and equipment efficiencies. With this degree of input data and program computational methods, detailed ***life cycle costs*** for alternative HVAC designs can easily be compared.

Energy Code Requirements

Now that the various model building codes across the United States have been unified into the International Codes Council family of codes, it is easy to discuss energy code requirements by referring to the International Energy Conservation Code 2003 requirements. As the design progresses, the designer should make sure that the following items are completed:

- Read over the Administrative and Definitions chapters to coordinate those requirements with the specific project being worked on.
- Determine exterior design conditions (dry and wet bulb temperatures, and degree days) from ASHRAE (American Society of Heating, Refrigerating, and Air-Conditioning Engineers) and NOAA (National Oceanic and Atmospheric Administration) references.
- By direct reference, all commercial buildings follow *ASHRAE Standard 90.1-2001, Energy Standard for Buildings Except for Low-Rise Residential Buildings*. While most items in Section 6 (Heating, Ventilating, and Air-Conditioning) of ASHRAE 90.1 relate to system design and operating efficiency (see 90.1 summary in Appendix), paragraph 6.2.2 requires running heating and cooling load estimates before selecting equipment.
- Chapter 8, which provides an alternative to ASHRAE 90.1 for architects who want to follow "acceptable practice" tables for building exterior envelope heat transfer *U*-factors and *R*-values.
- Model Codes can always be superseded by the AHJ (local Authority Having Jurisdiction, or code official). The designer needs to review critical issues (e.g. will the simplified approach option, paragraph 6.1.3, be allowed for your project?) with local standards and/or ask for interpretation from the AHJ directly.

Load Estimating Methods

Over time, load estimating has become more sophisticated. The early methods, like instantaneous method and storage load factor/ETD method, use hand calculations that were not quick and easy, even with the simplifying assumptions and look-up tables. Doing an entire building makes for lengthy calculations. Working up to the current methods, which try to track actual heat transfer within a building, only the use of computer software allows executing the millions of calculations required to complete a load estimate. Load estimating today can still be done manually using hand calculations or software spreadsheets, but these methods are usually reserved for early design rough approximations, or studies of special zones or material options effects. Running a final load for a large multizone building requires doing away with as many simplifying assumptions as possible, and using the HVAC system modeling techniques and refinement of heat transfer calculations of the transfer function and radiant time series methods.

Figure 7, plots relative accuracy vs. complexity for many common methods. The two extreme methods on the ends are instantaneous and heat balance. Instantaneous is the fastest and easiest to use, however, it is now only used for heating load calculations; it is too simple in the assumptions used to be used for cooling loads. The method does not take into account the storage and release of heat over time within the building material and furnishings. The heat balance method at the other end of the chart uses the least amount of simplifying assumptions, making it the most accurate method, but it must be done using computer software. All the load components (solar, wall, roofs, people, lights etc) interact dynamically with each other. This method calculates these interactions simultaneously and yields very precise results. Since the load components are calculated interactively, the method cannot provide output on how much of the load comes from walls, windows, roofs, people, lights etc. Since one of the main uses of load calculations is for designers to determine the individual load component contributions, this method is not used very much for actual load estimates. The heat balance method is very good for doing energy simulations, which is a topic for the Energy Estimating TDP.

1. Instantaneous, q = U * A * Δt
2. Storage Load Factor/ETD, Carrier E20
3. CLTD/CLF, Cooling Load Temperature Differential/ Cooling Load Factor, ASHRAE
4. Radiant Time Series, ASHRAE
5. Transfer Function, ASHRAE
6. Heat Balance, ASHRAE

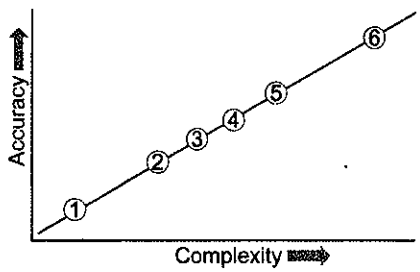


Figure 7

Accuracy vs. Complexity

Carrier's SDM-1 storage load factors/ETD manual load estimating method is still available, along with Block Load and Hourly Analysis Program computer software programs, both of which currently use the transfer function method. Plans are underway to convert these programs over to the current ASHRAE radiant time series method.

Storage Load Factor/ETD Method

The Carrier E20 method (storage load factor/ ***equivalent temperature difference*** method) was developed before ASHRAE became the industry source. In this method, a way for hand calculating the effects of storage and release of heat over time was put into tables to determine the appropriate storage factor. The method is covered in complete detail in Carrier's System Design Manual, Part 1, and two User Guides that cover the use of the E-10A (heating) and E-20A (cooling) forms for load estimating and coil performance.

TABLE 6—PEAK SOLAR HEAT GAIN THRU ORDINARY GLASS*
Btu/(hr)(sq ft)

NORTH LAT.	MONTH	EXPOSURE NORTH LATITUDE N†	NE	E	SE	S	SW	W	NW	Horiz	MONTH	SOUTH LAT.
0°	June	59	156	147	42	14	42	147	156	226	Dec	
	July & May	48	153	152	52	14	52	152	153	233	Nov & Jan	
	Aug & April	25	141	163	79	14	79	163	141	245	Oct & Feb	
	Sept & March	10	118	167	116	14	118	167	118	250	Sept & March	0°
	Oct & Feb	10	79	163	141	34	141	163	79	245	Aug & April	
	Nov & Jan	10	52	152	153	69	153	152	52	230	July & May	
	Dec	10	42	147	156	82	156	147	42	212	Oct & Feb	
10°	June	40	153	153				135	39	179	Sept & March	30°
	July & May	30			162	159	162	116	16	165	Aug & April	
	Aug & April	12	12	105	162	165	162	105	12	151	July & May	
											June	
40°	June	17	133	162	111	54	111	162	133	237	Dec	
	July & May	15	127	164	125	69	125	164	127	232	Nov & Jan	
	Aug & April	11	102	162	146	102	146	162	102	214	Oct & Feb	
	Sept & March	9	58	149	162	140	162	149	58	183	Sept & March	40°
	Oct & Feb	7	35	122	163	162	163	122	35	129	Aug & April	
	Nov & Jan	5	12	100	156	166	156	100	12	103	July & May	
	Dec	5	10	86	148	165	148	86	10	85	June	
50°	June	16	126	164	135	93	135	164	126	220	Dec	
	July & May	14	117	163	143	106	143	163	117	211	Nov & Jan	
	Aug & April	11	94	158	157	138	157	158	94	185	Oct & Feb	
	Sept & March	8	58	138	163	158	163	138	58	148	Sept & March	50°
	Oct & Feb	5	29	105	157	167	157	105	29	94	Aug & April	
	Nov & Jan	4	9	64	127	153	127	64	9	53	July & May	
	Dec	3	7	47	116	141	116	47	7	40	June	
		S	SE	E	NE	N	NW	W	SW	Horiz		
		EXPOSURE SOUTH LATITUDE										

Solar Gain Correction	Steel Sash or No Sash	Haze	Altitude	Dewpoint	Dewpoint	South Lat
	×1/.85 or 1.17	−15% (Max)	+07% per 1000 ft	Above 67 F −7% per 10 F	Below 67 F +7% per 10 F	Dec or Jan +7%

*Abstracted from Table 15, page 43.

†Solar heat gain on North exposure (in North latitudes) or on South exposure (in South latitudes) consists primarily of diffuse radiation which is essentially constant throughout the day. The solar heat gain values for this exposure are the average for the 12 hr period (6 a.m. to 6 p.m.). The storage factors in Tables 7 thru 11 assume that the solar heat gain on the North (or South) exposure is constant.

Figure 8

Peak Solar Gain through Glass (from Carrier SDM-1, Chapter 3)

When determining actual cooling loads, Table 6 from SDM-1 is used to determine the peak solar heat gain figures. This table represents the monthly maximums.

The variation of solar intensity and movement of heat into and out of the building mass is taken care of through the storage load factor tables. To demonstrate some of load estimating procedures we will begin creating an example building, and continue with its use throughout the remaining load estimating TDPs. The example building, the Chicago Office Building, has a 12-hour schedule of occupied operation. Figure 9 shows a portion of SDM-1 Table 11, which would be used to determine the appropriate storage factors. Other tables deal with different equipment operating hours.

TABLE 11—STORAGE LOAD FACTORS, SOLAR HEAT GAIN THRU GLASS
12 Hour Operation, Constant Space Temperature†

EXPOSURE (North Lat)	WEIGHT§ (lb per sq ft of floor area)	INTERNAL SHADE* SUN TIME AM 6	7	8	9	10	11	12	PM 1	2	3	4	5	BARE GLASS OR EXTERNAL SHADE‡ AM 6	7	8	9	10	11	12	PM 1	2	3	4	5	EXPOSURE (South Lat)
Northeast	150 & over	.59	.67	.62	.49	.33	.27	.25	.24	.22	.21	.20	.17	.34	.42	.47	.45	.42	.39	.36	.33	.30	.29	.26	.25	
	100	.59	.68	.64	.52	.35	.29	.24	.23	.20	.19	.17	.15	.35	.45	.50	.49	.45	.42	.34	.30	.27	.26	.23	.20	Southeast
	30	.62	.80	.75	.60	.37	.25	.19	.17	.15	.13	.12	.11	.40	.62	.69	.64	.48	.34	.27	.22	.18	.16	.14	.12	
East	150 & over	.51	.66	.71	.67	.57	.40	.29	.26	.25	.23	.21	.19	.36	.44	.50	.53	.53	.50	.44	.39	.36	.34	.30	.28	
	100	.52	.67	.73	.70	.58	.40	.29	.26	.24	.21	.19	.16	.34	.44	.54	.58	.57	.51	.44	.39	.34	.31	.28	.24	East
	30	.53	.74	.82	.81	.65	.43	.25	.19	.16	.14	.11	.09	.36	.56	.71	.76	.70	.54	.39	.28	.23	.18	.15	.12	
Southeast	150 & over	.20	.42	.59	.70	.74	.71	.61	.48	.33	.30	.26	.24	.34	.37	.43	.50	.54	.58	.57	.55	.50	.45	.41	.37	
	100	.18	.40	.57	.70	.75	.72	.63	.49	.34	.28	.25	.21	.29	.33	.41	.51	.58	.61	.61	.56					
	30	.09	.35	.61	.78	.86	.82	.69	.50	.30	.20	.17	.13	.14	.27			.64	.75	.79						
South	150 & over	.28	.25	.40	.53	.64																				
	100	.26	.22	.38	.51																					
	30																									

Figure 9

Solar Glazing Storage Load Factors (from Carrier SDM-1, Chapter 3)

As an example, lets look at a typical east-facing window. Chicago is approximately 40° N latitude, so looking at July, Table 6 (Figure 8) indicates a peak solar heat gain of 164 Btuh per ft^2. Going to Table 11 (Figure 9), and using the east exposure with a building weight of 100 pounds per ft^2 and internal shading, the peak storage load factor would be 0.73 at 8 a.m., reducing the solar heat gain to 164 * 0.73 = 120 Btuh/ft^2. By contrast, if there was no shading, the load factor would drop to a peak of 0.58 at 9 a.m., and the load profile is more leveled, demonstrating the heat storage effect.

Cooling Load Temperature Differential/Cooling Load Factor (CLTD/CLF) Methods

In the 1970s, ASHRAE published the CLTD/CLF Methods. These hand calculation methods were derived from the dynamic heat balance method using transfer functions on computer runs to derive location-specific tables that could be used in hand calculations.

As hand calculation procedures, both methods (CLTD and CLF) did a good job of balancing complexity (and therefore effort) with accuracy, however, both methods lacked flexibility. Building loads are affected by a wide variety of factors involving design, construction, environment, and building use. Table-based hand calculation methods typically deal with a fixed set of basic conditions (such as envelope loads for July 40 degrees North latitude) and then attempt to handle other conditions via correction factors. Ultimately, this approach introduces error and reduces accuracy when compared with methods that are more detailed. Some way of calculating loads specific to each design application was needed, which lead to the use of later methods that can best be done or only be done on a computer. Neither method is used currently since the transition from manual calculations to software-based load estimating has been successfully completed.

Transfer Function Method

The transfer function method is a simplification of the heat balance method that requires computer software to use. The transfer function method uses mathematical Laplace transformations to simplify the heat balance solution process, thereby yielding calculation times that are faster than those of the heat balance method without sacrificing too much of its accuracy.

The dynamics of calculating heat gain over time are retained, but individual load components can be calculated and then summed to get total loads. This allows the designer to gain element contribution values. For example, in Figure 10, the lights are turned on for 5 hours. The heat from the lights is split between convective (instantaneous) and radiant (heat storage) portions. The radiant portion is stored and released over time, just like solar load coming through a window. Therefore, as the lights remain on, the resultant load is

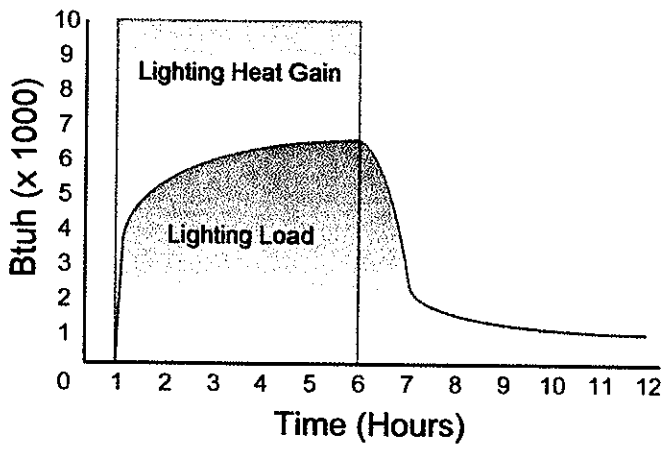


Figure 10

Lighting Load Transfer Function

a mixture of convective from instantaneous light heat and released radiant heat from storage. Note that after the lights are off, there is still the residual stored heat to be removed.

In the transfer function method, loads are calculated over a 24-hour period to allow for the heat storage dynamics. In that way, the transfer function method accounts for loads that occur during the unoccupied period which affect the loads during the occupied period.

The temperature of the room air is tracked for both occupied and unoccupied periods, and the effects of set point changes, and the magnitude of thermostat response to offset from set point are considered.

Looking at Figure 11, there are three items graphed: cooling loads, ***heat extraction rate***, and zone temperature. The first step of the transfer function procedure is to use the ***conduction*** and room transfer equations to calculate the cooling load, assuming a constant room air temperature over 24 hours. The next step uses the equipment sizing derived in the first step, along with the control system characteristics to simulate the way the room would be conditioned. From this, the true amount of heat that needs be extracted to maintain the room air temperature set point is determined. These calculations take into account the temperature during the unoccupied time. The graph shows that the system starts up early in the morning. The ***pulldown load*** on the air conditioning equipment can be readily seen. The system then operates during the course of the occupied period until it shuts off late in the evening. The zone air temperature achieved is also plotted. During the occupied time, the thermostat set point is traditionally about 75° F, and the unoccupied set point will often climb up to around 85° F.

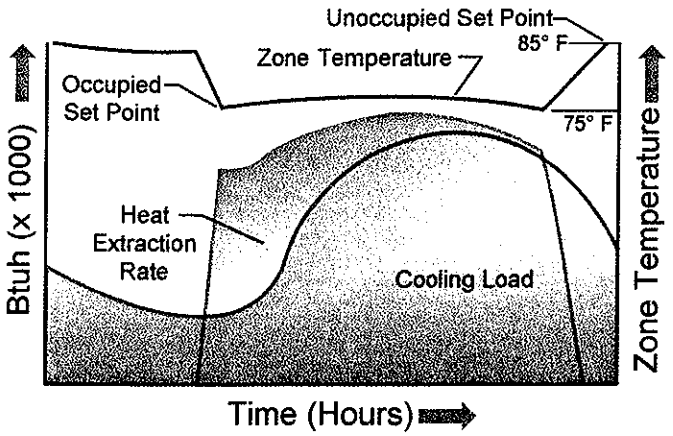


Figure 11

Transfer Function Elements

Because the transfer function method retains the dynamics of the heat balance method, it can be used for energy analysis as well.

Radiant Time Series Method

The radiant time series method, as you may expect by now, is a further simplification of the heat balance method, reducing the computation burden of the transfer function method. The term "radiant time series" refers to a set of factors that define the fraction of a heat gain converted to cooling load during a series of hours. The results from this method are similar to the first step of the transfer function method. It calculates a load based on a constant zone air temperature over 24-hours. It is not applicable for doing energy analysis calculations, but is easier to interpret pure load calculation results.

For walls and roofs, a solar air temperature is derived. This is a temperature that has the combined effects of solar intensity, external color, and actual dry bulb temperature. It represents the effective temperature on the outside of the wall. Then, each load element for each of 24-hours is split into convective (instantaneous) CTS (Conduction Time Series) and radiant RTS (Radiant Time Series) portions. A sample of the radiant and convective splits used by some components is listed in Figure 12. The items are then summed up for each hour to get a total for any hour of the day.

Table 19 Convective and Radiant Percentages of Total Sensible Heat Gain

Heat Gain Source	Radiant Heat %	Convective Heat %
Transmitted solar, no inside shade	100	0
Window solar, with inside shade	63	37
Absorbed (by fenestration) solar	63	37
Fluorescent lights, suspended, unvented	67	33
Fluorescent lights, recessed, vented to return air	59	41
Fluorescent lights, recessed, vented to return air and supply air	19	81
Incandescent lights	80	20
People		
Conduction, exterior walls	63	37
Conduction, exterior roofs	84	16
Infiltration and ventilation	0	100
Machinery and appliances (see Table 13)	20 to 80	80 to 20

Sources: Pederson et al. (1998), Hosni et al (1999).

Figure 12

Radiant/Convective Splits (from ASHRAE 2001 Fundamentals Handbook, Chapter 29)

Choosing Which Method to Use

No single load estimating method is best. Each method is appropriate for a specific application, depending on the point in the design process, time available to run the calculations, the sophistication of the user, and the desired level of accuracy in the results. The following recommendations are meant to guide the beginning designer as they become familiar with the various methods available.

- Preliminary design rough loads – rule of thumb check figures
- Schematic building loads – manual methods like storage load factor/ETD or simple ***block load*** software using transfer function or radiant time series
- Final loads for equipment selection – space, zone, and system loads using block load or system-based design software (HAP) using transfer function or radiant time series

With the release of the 2001 Fundamentals Handbook, ASHRAE began promoting the heat balance method for energy analysis and radiant time series method for pure load calculations.

External Space Loads

Earlier, when load components were first presented in Figure 1, external space loads were shown to include everything that directly caused the room temperature to vary, either above (heat gain), or below (heat loss) the set point. These loads can be further broken down into three subgroups:

- Weather-Related – made up of predominantly solar gain through glass, along with solar and transmission gains through walls and roof.
- Infiltration-Related – not directly weather-related, though outside wind speed affects the amount of airflow through the exterior envelope of the building. Infiltration (and exfiltration) occurs through any opening in the building envelope, whether it is planned like a window, door or HVAC unit sleeve, or unplanned gaps due to poor construction or building component deterioration.
- Adjacent-Space Related – made up of heat transfer across any interior partition with a temperature differential across the material assembly. Traditionally these have been thought of as walls, but horizontal assemblies that separate the floors in a building often have the required temperature differential to cause heat transfer.

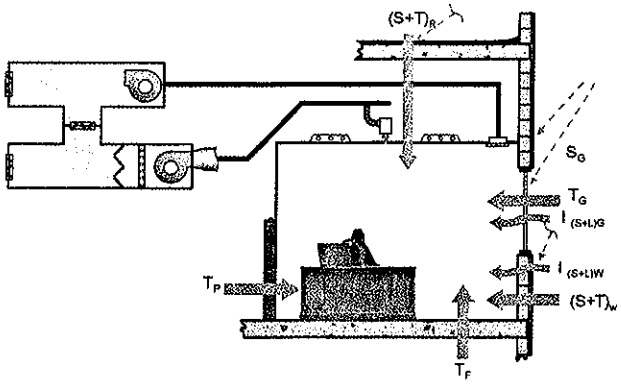


Figure 13

External Space Load Components

These heat gains and losses, which cause the space temperature to vary from set point, are the same no matter what kind of system is used, whether it is a window unit or a VAV terminal. Later in this TDP we will see that the system chosen will impact the load.

Design Conditions

As had been discussed, a ***temperature difference*** (Δt) is needed for heat to flow. So, some way to determine the peak or extreme outside air temperatures, along with the inside design temperatures, is needed to calculate the maximum heat flows through the building exterior envelope. The load estimating methods also need to calculate heat flow over time, so a means of determining a daily profile of hourly temperatures is needed.

Climatic Design Conditions

The designer is concerned about both cooling and heating; so two groups of climatic design data are needed. The most current ASHRAE Handbook of Fundamentals is the best source of outside weather data for design conditions across the globe. Tables are available for the United States, Canada, and other world locations.

The inside design temperatures are the comfort design conditions, generally 74 to 78 F design for cooling and 70 to 74 F design for heating.

Data is presented for dry-bulb dependent comfort cooling applications, wet-bulb dependent evaporation equipment (cooling towers, evaporative coolers, and fresh air ventilation systems) and dew point extreme conditions that often occur during months of moderate dry bulb temperatures.

Cooling

In the ASHRAE Climatic Design Data tables, there are three temperature variance choices, 0.4%, 1%, and 2%. These percent values indicate the number of hours in a year that the indicated temperatures may be exceeded. For instance, 0.4% db/mwb means that for 0.004 * 8760 hours/year = 35 hours per year the 0.4% cooling design dry bulb temperature for Chicago of 91° F may be exceeded. It is common for designers to use their best judgment when selecting which design conditions to use. However, beware of the creeping safety factor problem. Consult with your owner to see if being uncomfortable by only a couple degrees inside on those really hot days that do not occur every summer are worth the savings in both first cost and yearly cooling operating costs. Remember, going from the 0.4% column to the 1.0% column means only 53 additional hours per year, or only 7 to 10 occurrences a season, assuming approximately 6 hours of temperature overage on an above-design day. Therefore, a reduction of 3° F in the maximum outside temperature represents nearly a 20 percent drop in the Δt used to calculate the transmission heat transfer (88° F - 75° F, versus 91° F - 75° F).

TABLE 2–CORRECTIONS IN OUTDOOR DESIGN TEMPERATURES FOR TIME OF DAY
(For Cooling Load Estimates)

DAILY RANGE OF TEMPERATURE* (F)	DRY- OR WET-BULB	SUN TIME									
		AM			PM						
		8	10	12	2	3	4	6	8	10	12
10	Dry-Bulb	–9	–7	–5	–1	0	–1	–2	–5	–8	–9
	Wet-Bulb	–2	–2	–1	0	0	0	–1	–1	–2	–2
15	Dry-Bulb	–12	–9	–5	–1	0	–1	–2	–6	–10	–14
	Wet-Bulb	–3	–2	–1	0	0	0	–1	–1	–3	–4

TABLE 3–CORRECTIONS IN OUTDOOR DESIGN TEMPERATURES FOR TIME OF YEAR
(For Cooling Load Estimates)

YEARLY RANGE OF TEMPERATURE(F)*	DRY- OR WET-BULB	TIME OF YEAR								
		March	April	May	June	July	August	8	Oct.	Nov.
100	Dry-Bulb	–29	–19	–10	–3	0	0	–6	–16	–27
	Wet-Bulb	–14	–10	–5	–2	0	0	–3	–8	–14
95	Dry-Bulb	–29	–19	–10	–3	0	0	–6	–16	–27
	Wet-Bulb	–14	–10	–5	–2	0	0	–3	–8	–14

Figure 14

Outdoor Temperature Corrections (from Carrier SDM-1, Chapter 2)

Cooling is a time-dependant series of calculations. Many of the methods require a full day's time frame for the dynamics of the procedures. The practical reasons are that different thermal loads peak at different times. For example, the east peaks in the summer morning, while the west peaks in the summer afternoon (Figure 16 shows this graphically). The south side may peak in the fall because of the influence of the solar load created by the sun's lower position in the sky. However, when the system or coil load is determined, it may be in the summer mid-afternoon because of the effect of the high enthalpy (sensible and latent heat content) outdoor air. Figure 14 shows portions

of tables from SDM-1 that can be used to determine peak cooling temperature for various sunlit hours for a hot day in each month of the year. The tables are based on the ***daily range*** and the ***annual range*** (full tables are in the Appendix).

Figure 15 shows a graphical representation of a design day for Chicago in August.

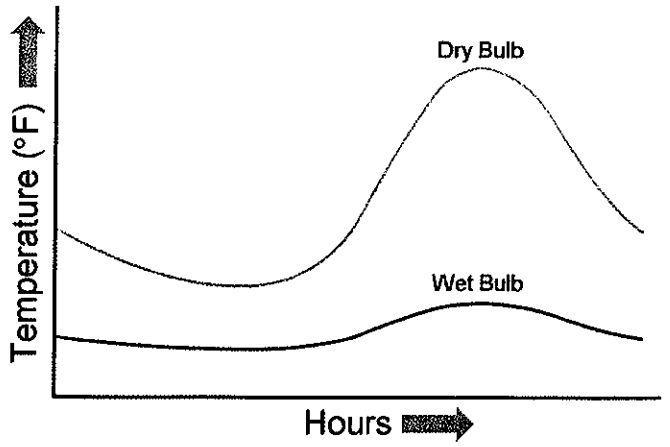


Figure 15

Design Temperature Profiles - August

Heating

Design heat loss is a simple calculation involving the design minimum outside air temperature. It is usually based on the ASHRAE 99.6 percent value or the judgment of the designer. For Chicago, the number is -6° F. When performing design heat loss calculations, there are no credits taken for any internal loads or solar heat gains. This is because the maximum heating losses usually occur in the morning hours, often before the building is occupied, meaning offsetting loads from lights, people, equipment, and even solar would not be present.

When selecting climatic design temperatures, the addition of a few degrees to the ASHRAE design data may be the first of many additive safety factors that creeps up on the designer.

Solar

To go along with the cooling design temperature profiles, a means of calculating peak effects from the sun is needed to figure out the solar loads through the glass. The solar load component represents the amount of radiant energy transmitted to the space through ordinary single pane glass. Hand calculation methods use tables of data, while computer software programs make this data ready to use for the site city. The solar data is in Btuh per ft^2 of glass area, comprised of both ***beam radiation*** (direct) and ***diffuse radiation*** (indirect, or scattered).

The compass points and number of exposures have an impact on the solar loads. The position of the sun not only varies with time of day, but significantly with time of year, which is determined by the latitude (angle of the sun) and longitude (sun time versus time of day) of the building site. For example, an east exposure has a morning peak in the summer. The south side has a peak around noon and it may be in the fall. The west glass peaks late afternoon in the summer, while the north glass receives only scattered diffuse sunrays in northern latitude locations like the United States. Occasionally, a cooling load study is done based on the building site orientation, especially if a large amount of glass is to be used.

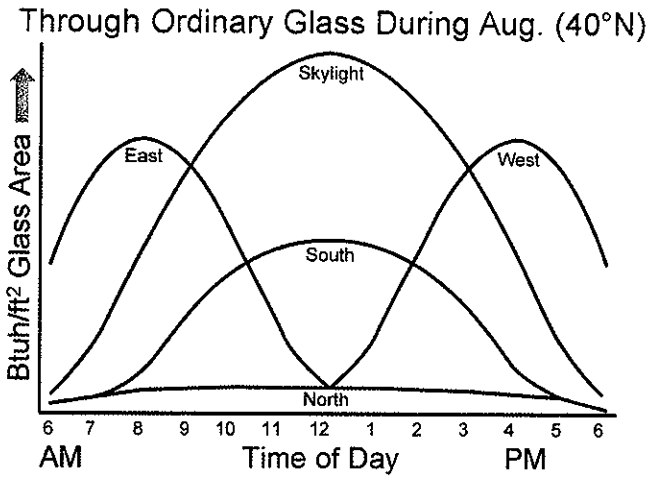


Figure 16

Solar Heat Gain

Since radiant energy is not a heat gain until it strikes some object and is absorbed, this does not represent the amount of heat to be removed by the air conditioning equipment. Glass types, special coatings, and internal and external shading all cut back on the amount of solar energy that comes into the room.

Coefficient Conversion

You can convert from one to another. SHGC = 0.87 * SC

The value that is eventually derived to modify the solar heat gain value is called either a shade coefficient (SC) or a ***solar heat gain coefficient*** (SHGC). The SHGC is the current rating method used for both the transfer function and radiant time series methods. These two coefficients are not the same. The ***shading coefficient*** should be used if the CLTD/CLF method is utilized.

Window and Glazing Types

The National Fenestration Rating Council (NFRC) is a good source for detailed information on windows and glazing types (simply called "glass"). Most manufacturers of windows and glass also offer technical information for computing solar heat gain when using their products.

Energy codes have all but done away with single pane 1/8-inch clear glass. Double-pane windows using two layers of glazing are most common, and on occasion in very cold climates triple-pane glazing assemblies are used. Glass comes in different thickness, 3/16 in., 1/4 in., and 1/8 in. Between the panes, there

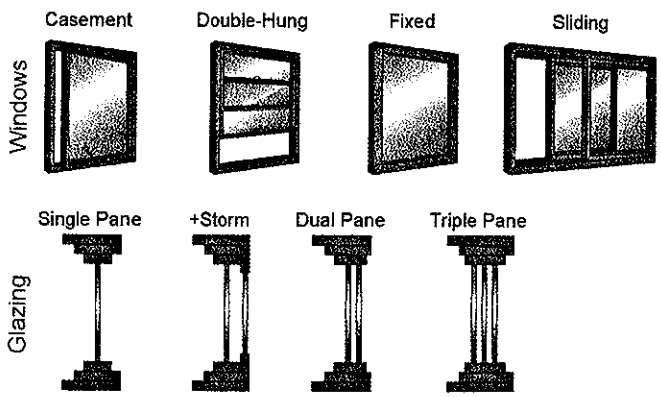


Figure 17

Window and Glazing Types

can be different gaps 1/4 in. and 1/2 in., filled with air, krypton, or argon. These refinements all reduce the SHGC and affect the overall U-factor.

The frame types, such as aluminum, wood, or vinyl, with or without thermal breaks, greatly affect the thermal performance and comfort for those seated near the windows.

Coatings

Further refinements in glazing performance come from inner and outer panes that have both tinting such as bronze, gray, or blue-green, and reflective coatings to reduce the transmission of solar energy with limited reduction of the visible light portion of the spectrum. Low-emission glass (low-e) is a clear glass that has a microscopically thin coating of metal oxide. This allows the sun's heat and light to pass trough the glass into the building. At the same time it blocks heat from leaving the room, reducing heat loss considerably. Buildings no longer need to excessively limit the glazing area to meet the energy conservation codes.

Shading - Internal, External, and Adjacent Buildings

One way to cut down on the amount of solar beam energy is to reduce the effective glass area and/or reflect some radiation back to the outside. Internal shade type describes the type of drapes, shades or Venetian blinds used with the window. The presence of interior shades, and the characteristics of the shading device, affects conductive and convective heat flows through the window assembly, as well as the transmission of solar radiation into the building. Shade properties therefore influence the calculation of the overall window U-factor, the shading coefficient, and the transmission and solar loads for the window.

Internal shades, depending on color, reflect a portion of the solar radiation back out the window, reducing the SC. Light colors reflect solar more than dark colors. For example, the windows in the Chicago example building have a U of 0.650 and an SC of 0.658, without internal shades. With an open weave light-colored fabric, the U is improved to 0.544 because of the additional insulation, and the SC is lowered to 0.496. With a medium colored fabric, the U is unchanged but the SC is larger at 0.547. Because this is radiant heat, the way the solar is accounted for is affected. If the shades are drawn, the radiant solar energy strikes the shades and is absorbed because the shade material is lightweight the heat is released to room almost immediately. If the shades are not drawn, the solar comes into the room that has much more mass and the solar heat is absorbed and released later and at a slower rate, lowering the value of the peak load and shifting it to later in the day. External shading is commonly done with fins, overhangs, and reveals that are described in relation to the window. They reduce the area through which the sun directly shines. Hand calculation methods do not provide

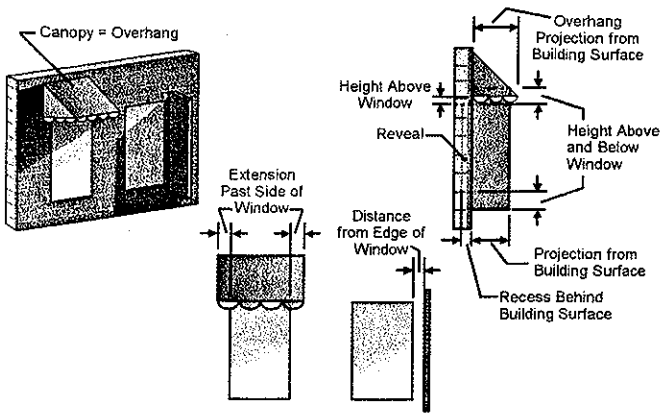


Figure 18

External Shading

accurate results for external shading. However, computer software can do the job with all the calculations needed to get hour-by-hour shading from exact solar angles, just as latitude and longitude are accounted for.

Shading (or even bounce light from reflective surfaces) from adjacent buildings is very difficult to calculate unless it shades surfaces all day. Then, just diffuse radiation can be assumed and a "north" exposure can be used to model this situation. Computer methods would need to reference the relationships of the adjacent building to walls and windows in the subject building in x-, y-, and z-coordinates. This would be very user-intensive and prohibitive from a practical standpoint. Therefore, designers will have to wait until these building interrelationships can be gained by CAD software communicating with load estimating software.

Transmission

If a temperature difference exists between the inside and outside surfaces, heat will flow by conduction to the cooler side. These are called transmission losses. There are varieties of surfaces that need to be accounted for in a cooling or heating load estimate to calculate all the conduction thermal loads. Remember, the equivalent temperature difference is used to account for the impact of solar loading on walls and roofs.

Wall and Door

Wall calculations can be performed manually using a U-factor for the wall materials assembly most closely matching the building design. There are tables of transmission coefficients (U-factor) in SDM-1 or older ASHRAE Fundamentals Handbooks that allows the user to select the most appropriate U-factor. The ASHRAE Fundamentals Handbook no longer has these tables since current methods like radiant time series use heat transfer coefficients layer-by-layer. The transfer function method uses the same method to calculate heat transfer as radiant time series.

However, quick comparisons of the overall U-factors will generally indicate relative resistance to heat flow of various assemblies. As discussed earlier, U-factors can be derived from tables of material R-values.

Door transmission gains or losses in commercial load estimates are sometimes ignored because they represent a very small fraction of the total load. If they are to be considered, a simple U-factor calculation is sufficient. This would not be the case if the doors were mostly glass, in which case they could be considered a separate window type.

Roof

The roof on a commercial building is usually horizontal and receives a lot of sunlight. Therefore, the effective temperatures are relatively high. If the roof is sloped, the area to be used in the transmission calculation is determined by measuring the area of the insulation layer at its location, either along the slope line or at the horizontal plane of the attic. The heat flow path starts with the ceiling material, then the air space above, all the way through the insulation and roof material assembly to the outside surface. Manual load calculations will require deriving the U-factor, or coefficients are internally computed for current software methods.

Windows and Skylights

Besides the solar load component, glass also has a transmission component. Similar to walls and roofs, the overall U-factor is used for this calculation.

On Grade and Below Grade

The temperature of the ground complicates calculation of the heat gain through the on grade floor or through basement walls below grade. The ground temperature near the surface is close to the outside air temperature, but further down the ground temperature approaches a rather constant value of from 45° F to 65° F, depending on the geographical area. There is very limited impact on the cooling load from the earth as most of the heat transfer is a heat loss. Heating loads, which will be discussed later, are much larger.

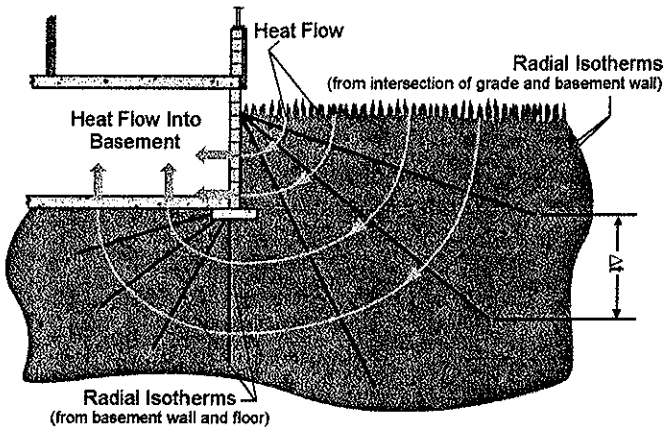


Figure 19

Transmission Below Grade

Infiltration

In addition to the other sources of heating or cooling loads, another tricky component is the effect of infiltration. These unplanned airflows must be heated or cooled to room conditions, often adversely affecting perimeter spaces. Experienced designers will say that estimating infiltration is like sticking a wet thumb in the air to estimate wind velocity.

Modern construction techniques result in tightly constructed buildings with minimal envelope infiltration. In low-rise buildings with fixed windows, and in those facilities operated under a slight positive pressure, infiltration can often be ignored. Most designers simply add a small safety factor to the load estimate to account for infiltration. The exceptions are high-rise buildings and those expected to be subjected to high wind loads.

Some old rules of thumb

that have been used for older buildings are 1 air change per hour in the summer and 2 air changes per hour in the winter.

Wind and Stack Effects

The pressure of the wind causes air to enter (infiltrate) the windward side of a building and leave (exfiltrate) the opposite (leeward) side. Stack effect, often called buoyancy, is caused by air density differences due to temperature and relative humidity differences between the indoor and outdoor air. Stack-effect induced airflow can be significant in tall buildings that are not sealed tightly at the floorline.

The relative importance of the wind and stack pressures in a building depends on building height, internal resistance to vertical airflow, location and flow resistance characteristics of exterior envelope openings, local terrain, and the immediate shielding of the building.

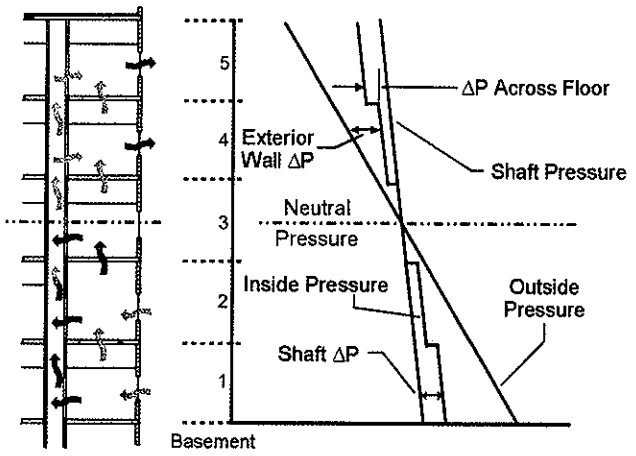


Figure 20

Stack Effect for an Idealized Building

Sensible and Latent Components

The infiltration air, when it is at a different temperature or moisture content, will impose additional space loads on the heating and cooling system.

Infiltration is treated as a space thermal load. The difference in air temperatures is a sensible load addition.

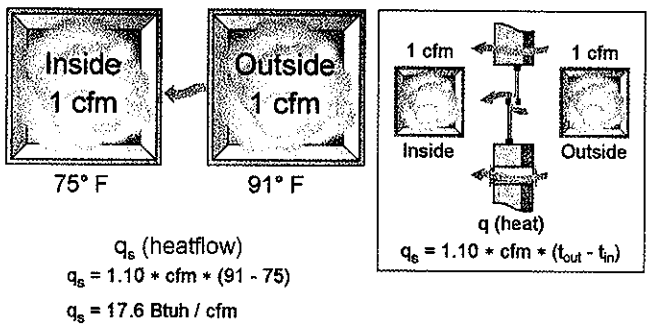


Figure 21

Infiltration, Sensible Component

The difference in moisture content is a latent load addition.

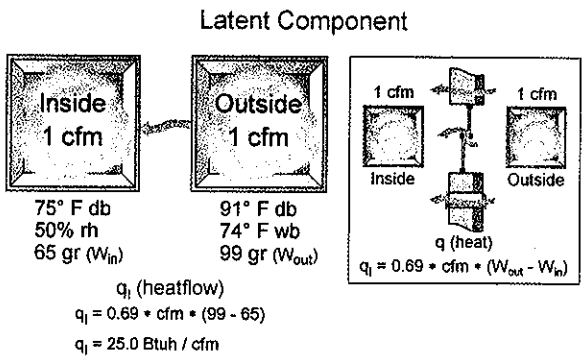


Figure 22

Infiltration, Latent Component

Partitions, Floors, and Ceilings

Some conditioned spaces in a building may be adjacent to unconditioned or partially conditioned spaces. When this occurs, heat is conducted by the temperature differences across the dividing material assemblies. It may be an interior wall (partition) next to a minimally heated warehouse. On the other hand, it may be a floor above an unconditioned parking garage or a ceiling that has a hot attic above it. All that is needed, of course, for a hand calculation is the U-factor, Δt, and area. Computer software may offer ways to change the temperature on the unconditioned side in some relationship with outside air to account for the dynamics over time.

Internal Space Loads

In non-residential buildings, the internal loads are a significant item in the load calculations. The designer must estimate the peak usage of lights, number of people, electric equipment, and plug load. The other item of judgment, besides how much, is when. The current dynamic methods used in computer software need the amount during the unoccupied hours as well to estimate the magnitude of pulldown loads. In some industrial applications items such as electric motors, exposed piping, tanks, and other process items need to be included in the calculation.

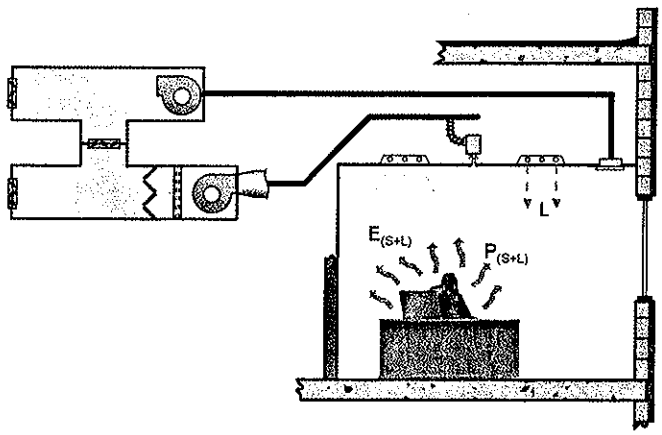


Figure 23

Internal Space Load Components

People

Heat is generated within the human body by oxidation – commonly called metabolic rate. This heat is carried to the surface of the body and dissipated by:

1. Conduction from the body to the objects the person touches and to the air surrounding them.

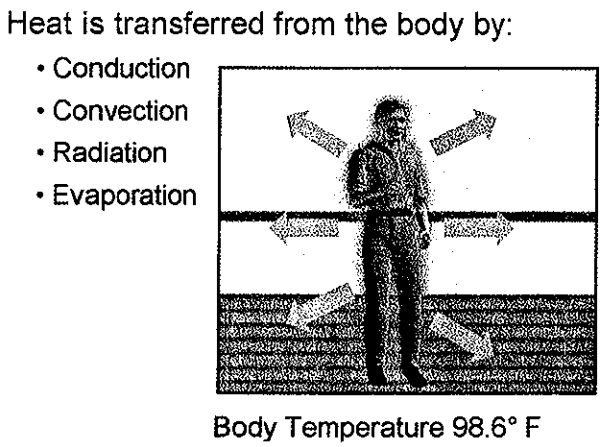


Figure 24

Heat Transfer from People

2. ***Convection*** from the body and respiratory tract into the air close to the person. Since warm air is lighter, or less dense, than cool air, the warm air floats upward. The warm air is replaced by cooler air, and the process continues.
3. Radiation from the body to the surrounding colder surfaces.
4. Evaporation of moisture from the body surface and in the respiratory tract to the surrounding air.

This chart shows representative values for latent and sensible heat gain from people. The values depend on room temperature and size (male, female or child). ASHRAE has tables and correction factors and computer software may have additional activity levels available for the designer.

A quick rule of thumb

Is 100 people represent 3 to 4 tons of total cooling load.

	Sensible Btuh	Latent Btuh	Total Btuh
Theatre	230	120	350
Office	245	205	450
Dancing	325	525	850
Bowling	525	925	1450

Figure 25

Heat Gain from People

Lighting

Lights generate sensible heat by the conversion of electrical power, or Watts (W), into light and heat. The energy is dissipated by radiation to the surrounding surfaces, by conduction into adjacent materials, and by convection to the surrounding air. Lights as a load component are typically referred to as lamp Watts per ft^2 of floor area. The radiant energy from lights is stored in the mass of the building and appears later as a load in the space. This is the same process that occurs with solar radiant energy, and current software methods factor this into the calculations.

Incandescent lamps convert approximately 10 percent of W into light. The remaining 90 percent of W turns into heat within the lamp. This heat then transfers into the space, 80 percent by radiation and 10 percent by conduction plus convection. This kind of light is used for task lighting, where its application reduces the need for higher overall lighting levels.

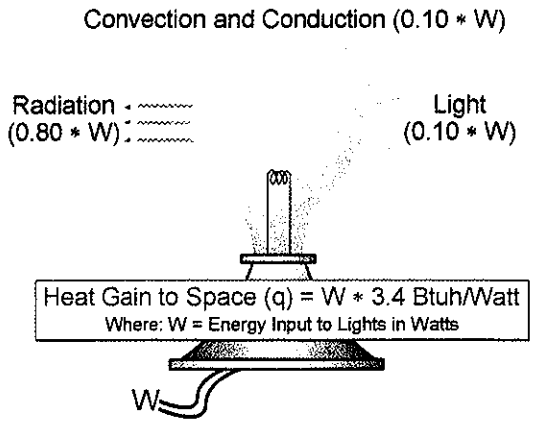


Figure 26

Lighting, Incandescent

Fluorescent light fixtures are more commonly used in commercial buildings for general lighting. Since about 23 percent of W is converted into light by fluorescent lamps, they are more efficient than incandescent lamps. Even though more efficient, a full 45 percent of W dissipates by conduction and convection to the space. Additionally, the final 10 percent of the input power turns into heat in the electronic ballast feeding the lamps.

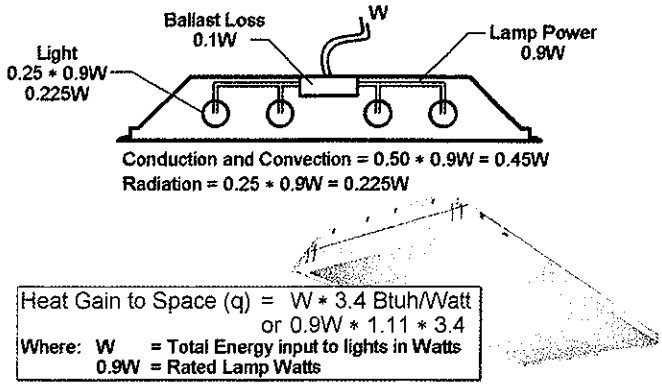


Figure 27

Lighting, Fluorescent

Electric Equipment (Plug Load)

Other devices such as computers, printers, copy machines, and appliances use electrical energy. Sometimes they are all lumped into a category of miscellaneous electrical equipment or plug load. Carrier SDM-1 or ASHRAE has tables of information for many kinds of appliances. For instance, a 12-cup coffee maker with two burners is 3750 Btuh sensible and 1910 Btuh latent.

Just taking the nameplate data and adding them up gives too large a plug load. In an article in the ASHRAE Journal (June 2000, *Heat Gain from Office Equipment*), it was reported that the average nameplate Watts for tested computers was 391, while the average heat dissipation was only 55 Watts. It was also found that heat gain was between 0.44 Watts per ft^2 to 1.05 Watts per ft^2, with an average of about 0.81 Watts per ft^2. The more unique the equipment or specialized the usage, i.e. medical laboratory, it is best to use published heat dissipation values for the actual equipment from the manufacturer (or installation and service manuals).

Electric Motors

Electric motors contribute sensible heat to a space by converting the electric power input (Watts or horsepower) to heat. Heat gain from electric motors depends on many factors, including motor horsepower, motor efficiency, motor hours used within a time period, how heavily the motor is loaded, and whether or not the motor and driven equipment is located inside or outside of the conditioned space or airstream.

Motor Nameplate or Rated Horsepower	Standard Efficiency	EPACT Min. Efficiency	Premium Efficiency	Heat Gain From Standard Eff. Motor in Location Btu/hr		
				A	B	C
1/20	35.0	-	-	364	127	236
1/12	35.0	-	-	606	212	394
1/8	35.0	-	-	909	318	591
1/6	35.0	-	-	1,212	424	788
1/4	54.0	-	-	1,178	636	542
1/3	56.0	-	-	1,515	848	667
1/2	60.0	-	-	2,121	1,273	848
3/4	72.0	-	-	2,651	1,909	742
1	75.0	82.5	-	3,393	2,545	848
1 1/2	77.0	84.0	-	4,958	3,818	1,140
2	79.0	84.0	-	6,443	5,090	1,353
3	81.0	86.5	89.5	9,426	7,635	1,791
5	82.0	87.5	89.5	15,518	12,725	2,793
7 1/2	84.0	88.5	91.7	22,723	19,088	3,636
10	85.0	89.5	91.7	29,941	25,450	4,491
15	86.0	91.0	93.0	44,390	38,175	6,215
20	87.0	91.0	93.0	58,506	50,900	7,606
25	88.0	91.7	93.6	72,301	63,625	8,676
30	89.0	92.4	93.6	85,787	76,350	9,437
40	89.0	93.0	94.5	114,382	101,800	12,582
50	89.0	93.0	94.5	142,978	127,250	15,728
60	89.0	93.6	95.0	171,573	152,700	18,873
75	90.0	94.1	95.4	212,083	190,875	21,208
100	90.0	94.1	95.4	282,778	254,500	28,278
125	90.0	94.5	95.4	353,472	318,125	35,347
150	91.0	95.0	95.8	419,505	381,750	37,755
200	91.0	95.0	96.2	559,341	509,000	50,341
250	91.0	95.4	96.2	699,176	636,250	62,926

NOTES: Ratings for open, drip-proof type, 4-pole, 1800 rpm motors
Totally-enclosed, fan-cooled (TEFC) motor ratings slightly higher
Consult motor manufacturer's ratings for specific data

Figure 28

Electric Motor Efficiencies

If the building is an industrial application, then the motors driving the process machinery also are a source of internal heat gain. There are sources beyond Figure 28 for data used in calculating motor heat loads, including manufacturers, ASHRAE 90.1 Energy Standard, and the Federal Energy Policy Act (EPACT). Minimum efficiency levels continue to climb because of the substantial use of electricity by motors.

Gas-Fired Equipment

Equipment within the space that is fired by natural gas is normally limited to cooking appliances and miscellaneous laboratory items like Bunsen burners. To limit the large amounts of both sensible and latent heat liberated by the natural gas combustion and the process itself, most of these devices are located under hoods directly exhausted to the outside. Even though, a certain percentage of the heat makes it into the space. Tables 51 and 52 in SDM-1 give values on an extensive list of gas-fired restaurant appliances and lab equipment. When high densities of equipment are present, like in medical laboratories and industrial pilot plants, data should be taken directly from the manufacturer's literature.

Piping, Tanks, and Evaporation

In most commercial applications, the level of insulation provided on both hot and cold piping and tanks keeps this ***internal load component*** at a negligible level. In industrial plants, such heat gains to the space are common, along with common items like furnaces and dryers. These contribute sensible heat to the space by convection and radiation from the outside surfaces, and frequently dryers contribute sensible and latent heat from the drying process.

Heat transmission tables (No. 54 through No. 58) can be found in SDM-1, and most manufacturers publish similar performance data to assist with either manual or software load estimating. Most estimating methods have an entry for miscellaneous loads where both sensible and latent load values may be entered. The user must sum up the sensible and latent gains for the piping lengths, equipment, or moisture transfer and then enter them into the manual form or computer software.

Bare and Insulated Piping

Both hot and cold piping exposed to the conditioned air in a room will have an effect on the load calculations. Hot water and steam piping heat transfer amounts depend on both the insulation type and thickness, and the temperature of the hot water or the pressure of the steam. Tables 54 and 55 in SDM-1 can be used in those cases where this may be a factor. Heat transfer coefficients for piping conveying cold fluids can be found in SDM-1, Table 56.

Closed and Open Tanks

Tanks that contain liquids at temperatures different than the room cooling or heating set points are a source of heat gain or loss. SDM-1 Table 57 may be used for sensible heat transfer from uninsulated closed tanks, and Table 58 gives similar data for free water surfaces for open tanks.

Steam Leaks, Absorption, and Evaporation

Steam leaking into the room has both a sensible and latent component. The sensible portion of the steam flow is calculated at

$$q_S = lb/hr * \Delta t * 0.45 Btu/lb$$

The temperature difference is between the room air dry bulb and steam temperature.

The latent portion is calculated at

$$q_L = lb/hr * 1050 Btu/lb$$

If moisture-absorbent materials (hygroscopic) are brought into the room, like large quantities of paper in a copy center, the moisture absorbed releases sensible heat (latent heat of vaporization at 1050 Btuh/lb of absorbed moisture) that needs to be included in the load calculation. If the hygroscopic material is removed from the space, an equivalent latent credit is included in the calculation.

Conversely, moisture evaporating at the room wet bulb (not heated or cooled from external sources), like a room full of plants or a reflecting pool, uses the room sensible heat for heat of evaporation. This form of latent gain should be deducted from the room sensible heat and added to the room latent heat. The total room heat gain does not change, but the effect on the sensible heat factor may be significant.

System Loads

The last group of load components to be considered when estimating a cooling load is those due to operating characteristics of the system serving the building or zone. These include loads associated with ventilation air, coil bypass, fan heat, duct heat transfer, duct airflow leakage, return plenum heat pickup, and in cases like underfloor air distribution, the supply plenum heat loss. Some of these items can only be accurately evaluated after the system is designed. Those computerized methods that perform system-based design loads are well suited to evaluate these items.

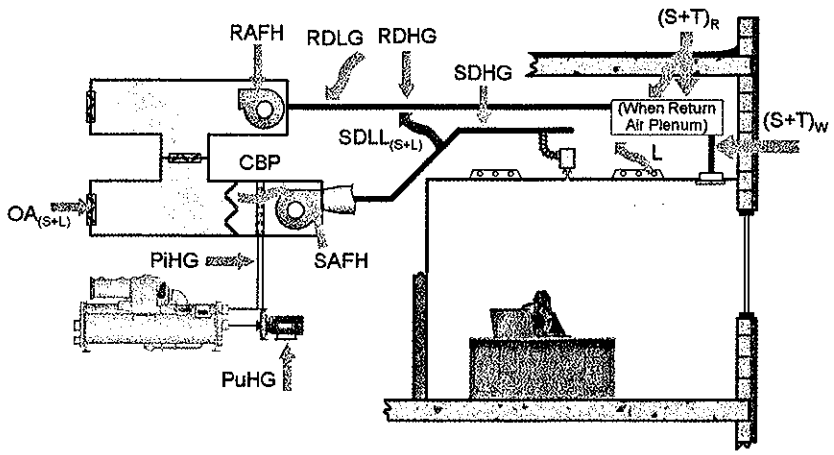


Figure 29

System Load Components

Ventilation Outdoor Air

Ventilation outdoor air normally creates a significant load for both heating and cooling estimates because of the large design Δt and/or moisture content of most locations. The first step is to determine the quantity of outdoor air needed. Air brought in from outside the building must be at least equal to the direct exhaust needs. Direct exhaust is that airflow which is directly exhausted from the building, not entering the main return air system. Toilet, laboratory, and kitchen hood exhausts are examples of direct exhaust.

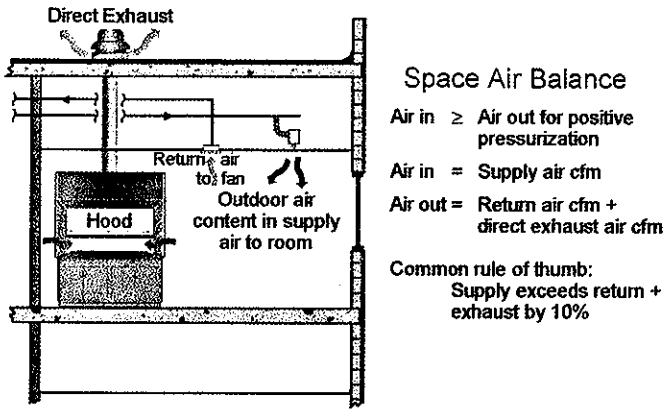


Figure 30

Ventilation, Space Air Balance

Codes and standards, notably the ASHRAE IAQ Standard 62, indicate the minimum amount of outdoor air to be used in different applications to provide an acceptable indoor air quality. Most of the time, these quantities are in excess of the direct exhaust requirements, especially in low-density occupancies like offices.

Maintain proper space pressure,

usually positive relative to the building exterior or adjacent circulation space, by supplying air in excess of the combined direct exhaust plus return air, regardless of the amount of outdoor air brought in at the main air handling unit.

Sensible and Latent Loads

Outdoor air brought in for ventilation has both sensible and latent load components. In simple calculations, the entering outdoor air is "assumed" the same as the outside conditions. Conditions like a hot rooftop alter the design climatic conditions and need to be accounted for in the load calculations. Since ventilation outdoor air is the same as infiltration air, the formulas shown in Figure 21 and Figure 22 can be used. In a few systems, like packaged terminal air conditioners (PTACs) and fan coils without dedicated makeup air systems, the ventilation air directly enters the space untreated by the cooling coil, so it is treated as a space load.

Duct Heat Transfer and Airflow Leakage

In transferring air from the system cooling coil to the conditioned spaces and back to the air-handling unit there are load components that must be considered. An accurate evaluation of these parameters can only be made when the air-conditioning system has been designed. Since load estimating is the first step in system design, a reasonable approximation of these parameters is necessary at this point.

Duct Heat Gain

Heat gains or losses in the supply and return ductwork will affect system performance and increase cooling and heating coil loads.

With supply air from 50° F to 60° F flowing through the ductwork, and upwards of 90° F air surrounding the duct in an unconditioned space, a potential for heat gain to both the supply and return air exists. Typically, this heat source has a negligible affect on comfort air-conditioning systems because of the use of higher R-value insulation on the ducts has kept heat transfer low, somewhere under 2 percent. A conservative approach, therefore, would be to use 2 percent of the room sensible heat, assuming that for a specific hour, the sum of zone sensible loads is 120,000 Btuh. Therefore, with the duct heat gain at 2 percent of 120,000 Btuh, the supply duct heat gain would be 2,400 Btuh. With a lower Δt in the return ductwork, only 1 percent heat gain is normal. The first of these heat gains will be used to determine the change in temperature of the supply air between the time it leaves the air handler and the time it reaches the zone supply terminals/room diffusers. The second heat gain will be used to figure the rise in the return air before it reaches the air handler.

Duct Leakage Loss

Air leakage in supply ductwork increases the airflow requirement for the central supply fan, and in turn, increases the fan heat gain and power use. Air leakage in the return duct adds to the airflow, but since it usually compensates for the supply duct losses, it does not add to the fan cfm.

The greatest potential for lost capacity exists in the supply air duct and depends on the duct shape, the duct pressure, and the workmanship used in constructing the duct system. Of the above three parameters, the one that is the key toward approximating the leakage loss is duct pressure.

Lower pressure class ductwork (±0 to 2-in. static pressure) tends to be installed using rectangular ductwork with taped joint connections. Due to variations in workmanship, leakage tends to be an average of 5 percent of the supply air quantity.

Mid-level pressure class ductwork (±2 to +6-in. static pressure) is made from medium quality spiral duct with tight joints. Loose joints are not permissible due to the whistling noise that would result from a leak. Leakage in this case is around 2 to 3 percent of the supply air.

Higher pressure class ductwork (+6.0-in. and higher static pressure) is made from high quality spiral duct with positive leak-tight joints. Leakage tends to be 1 percent or less of the supply air quantity.

A general conservative approach would be to add 5 percent of the room sensible heat to cover the leakage loss in manual calculations.

Computer software may model this as an increase in supply air cfm, and the quantity that leaks from the supply duct is assumed to flow directly or indirectly into the return airstream.

Fan Horsepower and Motor Heat Gains

Draw-thru air handling units have the supply fan located downstream of the cooling coil, adding heat to the supply airstream due to the work done on the air to raise its pressure within the ductwork. This heat gain diminishes the capacity of the supply air to cool the room, requiring a greater airflow for the space load. A blow-thru supply fan arrangement increases the coil load without impacting the supply airflow to the space. A similar heat gain situation occurs if a return air fan is used in the air-handling system, but in this case it only affects the coil load.

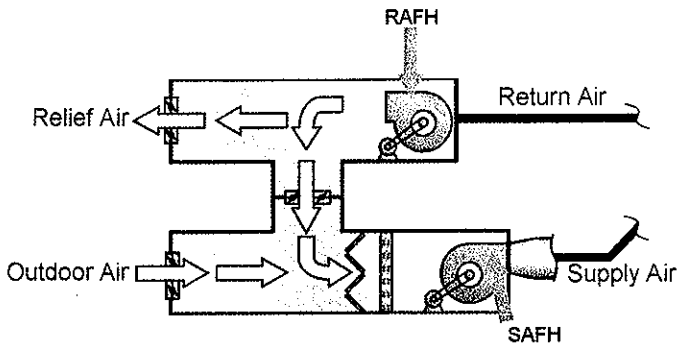
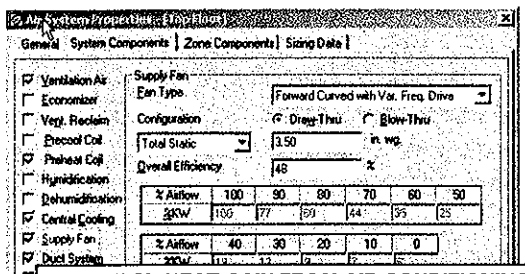


Figure 31

Fan Heat Gain Load Components

Should the motor driving the fan be located in the airstream, the electrical losses of the motor also show up as a heat gain. Figure 32 is a portion of Table 59 from SDM-1 (full table in Appendix), which allows an accurate estimate of this parameter. The table reveals that there are a number of considerations necessary to properly evaluate this parameter. A reasonable approach would be to add 5 percent of the room sensible heat to cover fan heat gain for draw-thru units. Blow-thru air-handling units have the fan located upstream of the cooling coil. Therefore, fan and motor heat gain would not be part of the system supply losses. Instead, they would be considered system return losses.



MANUAL
OR
SOFTWARE
METHOD

TABLE 59–HEAT GAIN FROM AIR CONDITIONING FAN HORSEPOWER, DRAW

FAN TOTAL PRESSURE† (In. of Water)	CENTRAL STATION SYSTEMS‡ Temp Diff Room to Supply Air					APPLIE Temp Room
	10 F	15 F	20 F	25 F	10F	10 F
	PERCENT OF ROOM SENSIBLE					
0.50	1.2	0.8	0.6	0.5	0.4	2.2
0.75	1.9	1.3	1.0	0.8	0.6	3.5
1.00	2.7	1.8	1.4	1.1	0.9	4.8

Figure 32

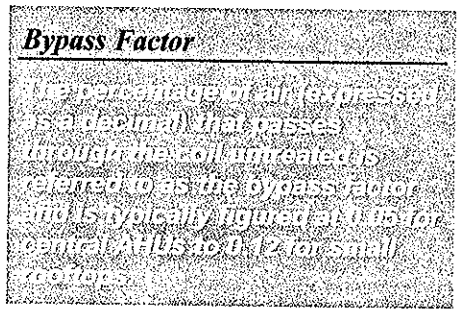
Fan Heat Gain

Computerized methods allow the designer to account for fan heat in a number of ways such as horsepower, kW, or static pressure. An estimate of total fan pressures is often the easiest way, as is shown in the figure.

Bypassed Outdoor Air

Since the cooling coil is not a perfect heat exchange device, some of the air entering the coil passes through the coil untreated. This represents a loss similar to a draw-thru fan heat gain that diminishes the supply air capacity to cool the room. Another way to look at it is a mixing of air-streams (conditioned and bypassed) that raises the coil leaving air temperatures above theoretically ideal conditions.

The coil bypass factor can be obtained from product literature for the specific equipment used in the load estimate. In most system design applications, however, the HVAC equipment has not yet been selected. In such a situation, specify a bypass factor that is generally representative of the type of equipment you intend to select (e.g., small, medium, or large rooftop units, central station air handlers of a particular type, etc.).



Plenums

Instead of running ductwork to convey return air from the spaces back to the air-handling unit, the building cavity between the suspended ceiling and floor or roof above can be used as a plenum. Now, with it becoming more common for building interior design to have a raised access flooring system, this underfloor plenum space, which is used for many building services, can also serve as a zone supply air plenum.

The distribution and accounting for some of the load components are affected as the plenums are at a different temperature than the room cooling and heating set points. The temperature of the return air in a plenum rises as heat is picked up, and likewise, the temperature of the supply air rises as cooling capacity is lost across the access floor and structural slab. Use of plenums influences the assignment of the heat gains, transferring portions of space loads over as coil loads.

Ceiling Plenums

When air is returned through the ceiling plenum, the cooling load caused by lights can be broken into two components. One component is that portion of the light heat, which flows directly into the space. The other component is the amount of heat absorbed as the return air passes over or through the fixture.

An accurate estimate of the two load components requires a detailed heat balance calculation and varies with the type of light fixture, type of ceiling, and the velocity of the return air over or through the fixture. However, for ordinary load estimating calculations, it is sufficient to calculate the total cooling load component for the light fixtures, and then distribute the load between the space and the return air as simple percentages of this total.

In a similar manner, a ceiling plenum under a roof can pick up even more heat from the roof load component. In this situation, the return air picks approximately 70 percent of the roof heat while 30 percent makes its way into the space. Again, 70 percent of the light heat reaches the space and the return air absorbs 30 percent of the light heat. The figure represent the composite effect of the various heat exchange processes that take place in this situation.

The return air heat transfer from the lights and roof are no longer space load components but are system load components directly affecting the cooling coil.

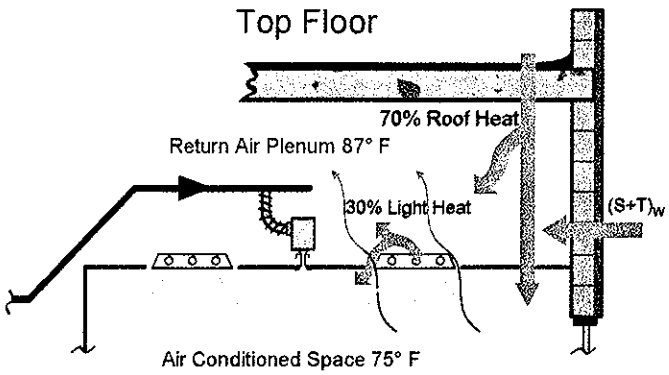


Figure 33

Ceiling Plenum, Top Floor

Underfloor Plenum

When an access flooring system is used as a supply plenum, the conditioned supply air is provided to the underfloor cavity and boosted by zone mixing fans before entering the conditioned area. The cool air in the plenum creates temperature differences across the plenum boundaries, both the perimeter walls and the floors (raised and structural). However, for the most part, the sensible load picked up comes from within the area to be conditioned, so nothing needs to be done to account properly for the heat transfer in the load calculation. The structural slab is usually not insulated, but the heat transferred to the floor below helps cool the material. A similar situation is occurring on the floor above, helping to cool the return air plenum below it, so at the main air handler the two load components cancel each other out.

Load Component Profile

Load component profiles are a visual representation of the dynamics of the major load components in a building or a zone over the course of a year. They are created by making some simplifying assumptions and plotting the results of peak cooling and heating load calculations on simple graph paper. The profiles plot the major thermal loads, such as transmission, internal loads and solar against outside dry bulb temperature. The loads are usually represented as Btuh ÷ 1000 or MBtuh.

The profiles let the designer visualize the relative size of each component and the relative change in the cooling and/or heating loads when changes are made to any individual component characteristic. This can be particularly helpful during schematic design when insulating values and areas of glass are being determined. Profiles are also an educational tool to begin the process of translating the fundamentals of load estimating to a system-based approach where the building overall loads can be related to the outside temperature and solar loading, much in the same way the HVAC system capacity responds to the changing outdoor air load conditions.

Peak Load Data for the Chicago Office Building:

Cooling (91° F/Jul 1500)

Transmission	58,755 Btuh
Lights	91,368 Btuh
People	14,557 Btuh
Electric	40,257 Btuh
Solar	69,444 Btuh

Heating (-6° F)

Transmission	252,768 Btuh

Figure 34

Example Building Load Data (Top Floor)

Building the Profile

The first step is to set up a grid of the outside dry bulb temperatures that cover the annual range for the subject building or zone on the horizontal axis. Sensible load values are plotted vertically in Btuh, both as positive heat gains (cooling), and negative loads or losses (heating).

The transmission peak loads are first plotted on the design heating and cooling temperature lines, using the data from the completed load estimate, in this case the top floor of the Chicago Office Building (Figure 34). Then a line is drawn between the design conditions to establish the slope of the load profile (Figure 35). The line crosses zero Btuh at the room set point. This uses the simplifying assumption of steady

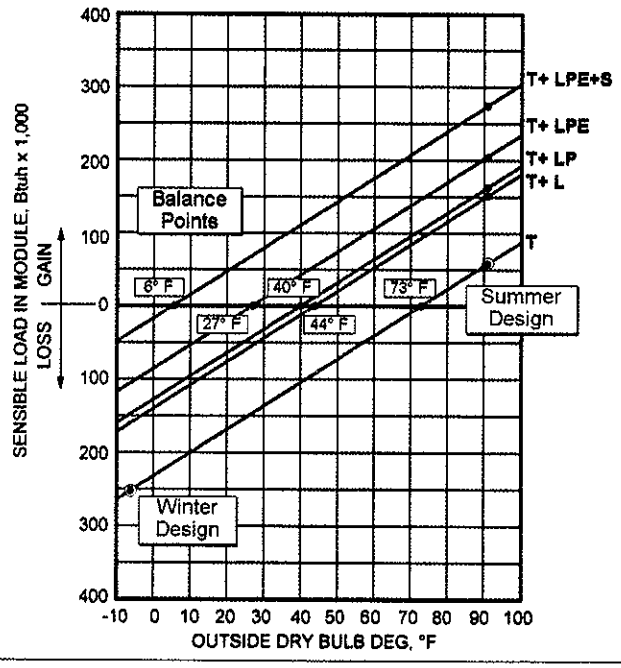


Figure 35

Completed Space Load Component Profiles

state heat flow. The slope of this line will change with different glass ratios, becoming steeper as the amount of glass increases. It can be seen that the net heating is less when it is cold outside and net cooling is greater when it warm outside.

Next, design cooling condition points should be established for lights, people, and miscellaneous electric (plug load). These internal loads are always cooling loads. Lines are drawn through each point parallel to the base profile line established by the transmission load. Without the addition of the transmission sloping line, these items would be straight horizontal lines as they are cooling loads at all temperatures. Adding the solar data completes the load profile (Figure 35). Here, a straight line based on the design cooling condition represents the solar data. In actuality the solar load varies significantly, both by time of day and time of year, requiring more complexity than a manual profile warrants.

Balance Points

Load profile balance points are those temperatures where the load lines cross over from cooling to heating. Looking at the completed profile, heat gain from lights will be offset by transmission losses when it is 44° F or lower outside. Adding people load causes the balance point to decrease to 40° F. When miscellaneous electrical loads are added, balance point drops even further to 27° F. As noted, the solar loads are highly variable, but it is easy to see how a building might require cooling even below freezing conditions outside.

The Chicago Office Building example shows the internal loads offsetting the transmission losses all the way down to 6° F. This extreme condition should be tempered with the infiltration load and the fact that not every light will be turned on or every person occupying the building. A more reasonable statement would be that our example building would most likely be in cooling down to the 30s, possibly even to below freezing. The eventual system selected for the building needs to accommodate this need for cooling during lower outside temperatures, either with an economizer cycle or through equipment selected to operate at the reduced temperatures.

Plotting on the Psychrometric Chart

A psychrometric plot of the system under design is necessary for a thorough understanding of the process and perhaps troubleshooting potential problems. If hand calculations are done, the process must be plotted manually on a psychrometric chart to assist in the design of the system. Some software load estimating programs can create these as part of the output.

State Points

When any two properties of an air-water mixture are known, the exact condition of the air can be located on the psychrometric chart, then all other properties can then be found from this one point. Such a point is referred to as a ***state point***. There are a number of state points that can be determined from running a cooling load estimate for an HVAC system. The simplest two are the room cooling set point and outside air design climatic condition. A third state point is the supply air temperature entering the space. This number is not needed for a basic manual load estimate, but is a common input for load estimating software. Efforts to model the HVAC system to refine the load require it as an input or it is a system data output when the designer uses other defining parameters such as airflows for either the air handler or the zone terminals/diffusers.

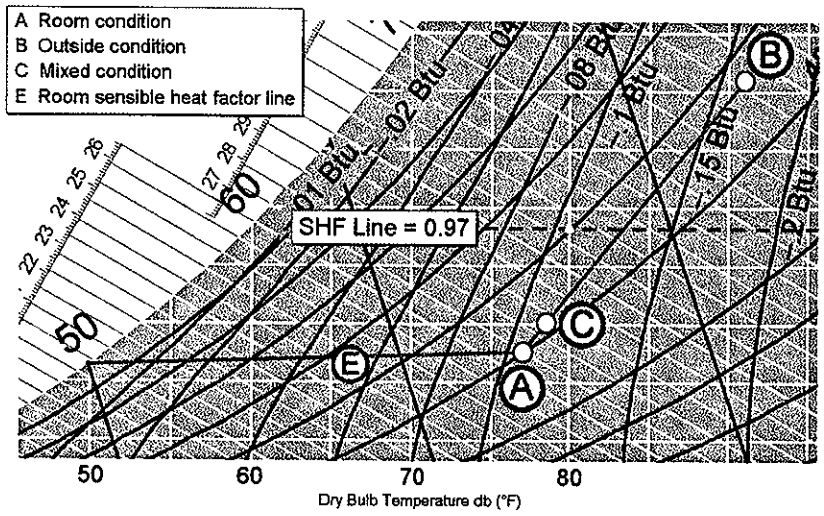


Figure 36

State Points and RSHF Line

Room Sensible Heat Factor (RSHF) Line

The room sensible heat factor is the ratio of the room sensible heat to the summation of room sensible and room latent heat. This ratio for a typical Chicago Office Building space is calculated as follows:

$$RSHF = \frac{RSH}{RSH + RLH} = \frac{6072\,Btuh}{6072\,Btuh + 160\,Btuh} = 0.97$$

The RSHF line is drawn through the room set point on Figure 36 as a line parallel to a sensible heat factor line of the same value (using the reference point on the chart). When adequate conditioned air is supplied to offset the room loads, the room set point will be satisfied, provided both the dry bulb and wet bulb temperatures of the supply air fall on the RSHF line. Colder air would need less airflow and warmer air more airflow.

System Lines

The psychrometric chart shown here has completed the circle by showing the effects of system loads and the performance of the cooling coil. The effect of fan heat (point 1 to 2), supply duct heat gain (point 2 to 3) are shown, along with point C, which is the effect of mixing outdoor air with the return air to arrive at the coil entering conditions. These advanced load-estimating concepts for system-based design will be covered in both the third and fourth TDPs in the Load Estimating series.

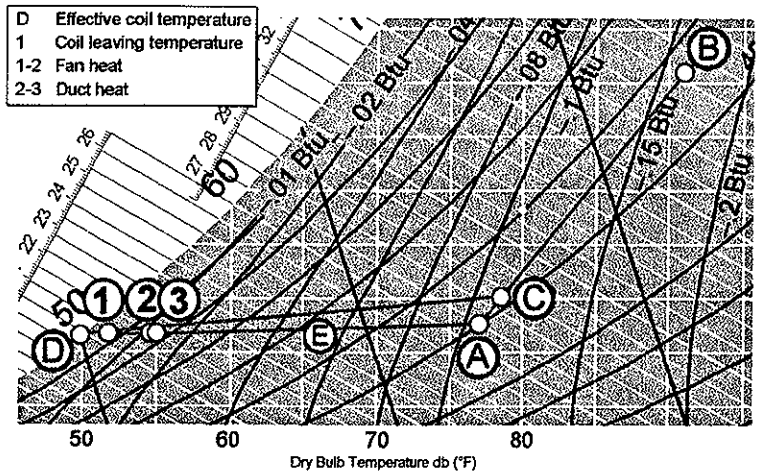


Figure 37

System Lines

Heating Load Estimate

Heating loads are a simplified version of cooling loads. Figure 38 shows that the envelope heating load losses are derived from steady state transmission and infiltration losses, while loads like ventilation outdoor air are most often system load components, which also apply to heating loads. The methods involved in calculating the loads have already been discussed in previous sections of this TDP. Determining the heat loss through basements and slab on grade floors becomes a little more complex than the simple $q = U * A * \Delta t$ transmission equation. The manual method, presented in detail in SDM-1, utilizes Form E-10A (found in Appendix). Most computer software load estimating methods automatically perform a heating load when calculating a cooling load.

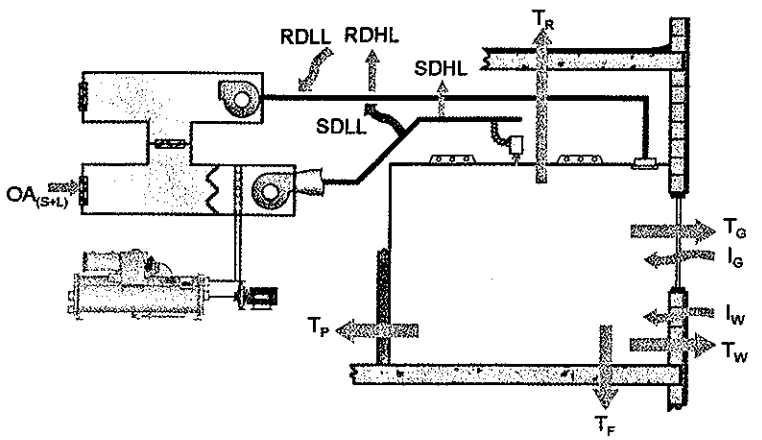


Figure 38

Heating Load Components

Design Conditions

To determine the temperature difference to be used across the exterior walls, the climatic design conditions need to be established first. From the ASHRAE Fundamentals Handbook, the heating 99.6 percent column value (reversal of 0.4 percent column used for cooling) for Chicago is -6° F. The room set point depends on the application. Comfort for an office building may be 70° F, while a warehouse may be only 45° F. Healthcare projects are applications that may require a wide range of set points in some areas, so you may need to design for 68° F cooling and 80° F heating.

Transmission

Transmission losses are normally the major heat transfer values in a space heating load estimate. The building walls, roof, glass, and foundation / basement comprise the load components. The temperature differences used are the differences between the room set point and the temperatures on the other side of the appropriate structural barrier (heating design climatic data, adjacent room temperature, or soil temperature). Heat loss through walls above grade is determined in the normal steady state transmission equation fashion.

Floor and Basement Heat Losses

For slab on grade and partially below-ground construction, the major heat loss takes place around the perimeter. The International Energy Conservation Code and ASHRAE 90.1 Energy Code require good levels of insulation along the full perimeter, extending down into the upper level of the soil, usually the first couple of feet, but no greater than the frost depth. The standard transmission equation is used to calculate the heat loss. While the floor loss is a minor percentage of the total heat loss, it is quite important for comfort, since cold floors mean an improperly heated building.

The heat loss from a full-depth basement is the result of a complex heat transfer process. The complexity is due to the variable ground temperature surrounding the basement. Below 8-ft, the ground is an excellent heat sink and can absorb large amounts of heat with little appreciable change in soil temperature. Above the 8-ft level, a temperature gradient exists due to the influence of the outside air temperature. The greatest variation exists at the surface and decreases down to the 8-ft level. In order to simplify the heat loss calculation, Figure 39 shows four possible calculations that could be involved.

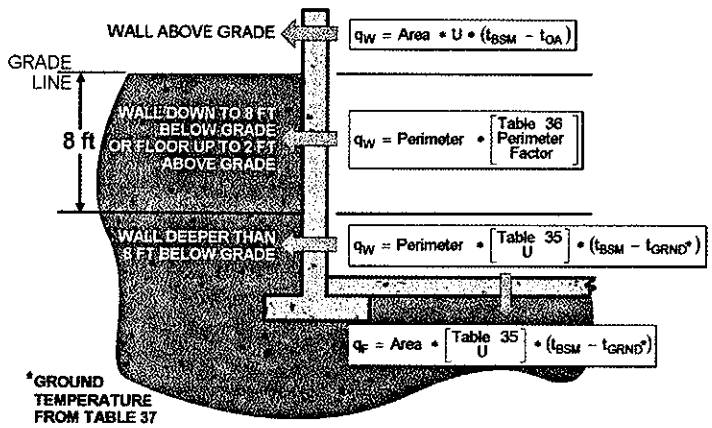


Figure 39

Basement and Floor Heat Losses

Basement Floor

The heat loss through a basement floor is normally small and relatively constant year round because the ground temperature under the floor varies little throughout the year.

SDM-1 Table 37 shows the ground temperature to vary only 25° F while the outside air temperature varies 50° F (from -30° F to +20° F). SDM-1 Table 35 also shows the transmission coefficient (U) for an uninsulated masonry floor of any thickness to be approximately $0.05\,\text{Btuh}\cdot\text{ft}^2\cdot°\text{F}$. Heat losses through the basement floor is therefore determined by the following equation.

$$q = U * A * (t_B - t_G)$$

Where:

U is $0.05\,\text{Btuh}\cdot\text{ft}^2\cdot°\text{F}$

A is the net floor area in ft^2

t_B is the basement temperature

t_G is the ground temperature

TABLE 37—GROUND TEMPERATURES

FOR ESTIMATING HEAT LOSS THROUGH BASEMENT LOSSES

Outdoor Design Temp (F)	–30	–20	–10	0	+10	+10
Ground Temp (F)	40	45	50	55	60	65

Figure 40

Ground Temperatures (from SDM-1, Chapter 5)

Walls 8-ft Below Grade

Like the basement floor, the heat loss through a masonry wall of any thickness may be approximated in the same manner as that done for a basement floor. The U-factor for masonry walls from Table 35 should be 0.08 instead of $0.05\,\text{Btuh}\cdot\text{ft}^2\cdot°\text{F}$.

TABLE 35—TRANSMISSION COEFFICIENT U–MASONRY FLOORS AND WALLS IN GROUND

(Use only in conjunction with *Table 35*)

Floor or Wall		Transmission Coefficient U (Btuh) (sq ft) (deg F)
* Basement Floor		0.05
Portion of Wall Exceeding 8 feet ground level	Below	0.08

*Some additional floor loss is included in perimeter factor, See *Table 36.*

Equations:

Heat loss through floor, Btuh = (area of floor, sq ft) * (U value) * (basement – ground temperature).

Heat loss through wall below 8 foot line, Btuh = (area of wall below 8 ft line, sq ft) * (U value) * (basement – ground temp).

NOTE: The factors in Tables 35 and 36 may be used for any thickness of uninsulated masonry wall or floor, but there must be a good contact (no air space which may connect to the outdoors) between the ground and the floor or wall. Where the ground is dry and sandy, or where there is cinder fill along wall or where the wall has a low heat transmission coefficient, the perimeter factor may be reduced slightly.

Figure 41

Transmission Coefficients (from SDM-1, Chapter 5)

Grade to 8-ft Below

In order to account for the variable temperature gradient, SDM-1 Table 36, provides a perimeter factor based on the distance of the floor (wall height) below grade. Heat loss is determined by the following equation:

$$q = L_p * Q * \Delta t$$

Where:

Q is the perimeter factor from Table 36

L_p is the lineal feet of basement wall (perimeter)

Δt is the difference between outside air and basement temperatures.

TABLE 36—PERIMETER FACTORS

FOR ESTIMATING HEAT LOSS THROUGH BASEMENT WALLS AND OUTSIDE STRIP OF BASEMENT FLOOR

(Use only in conjunction with *Table 35*)

Distance of Floor From Ground Level	Perimeter Factor (Q)
2 Feet above	0.90
At ground level	0.60
2 Feet below	0.75
4 Feet below	0.90
6 Feet below	1.05
8 Feet below	1.20

Equation:

Heat loss about perimeter, Btuh = (perimeter of wall, ft) * (perimeter factor) * (basement – outdoor temperature).

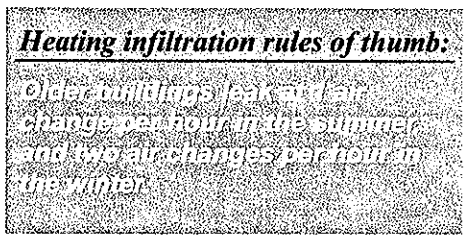
Figure 42

Perimeter Factors (from SDM-1, Chapter 5)

Infiltration

Infiltration air quantity in the winter can be more than in the summer because of the increased wind velocities, from an average of 7.5 mph in summer to 15 mph in winter. In addition, a much larger temperature difference exists between the outside and inside air temperatures.

Predicting infiltration rates before a building is constructed is not an easy task. Modern building construction techniques result in tightly constructed buildings with minimal infiltration. Infiltration can often be ignored for low-rise buildings with inoperable windows and those operated under a slight positive pressure. Most designers simply add a small safety factor to their heating load estimate to account for infiltration. The exceptions are high-rise buildings and those expected to be subjected to high wind loads. Unoccupied heating should be checked to see if including infiltration raises the total load because most buildings go to intermittent fan operation at night, allowing infiltration to occur.



No Credits

The heating load is estimated for the winter design temperatures, which usually occur at night. Consequently, no credit is taken for the heat given off by internal sources (people, lights, etc.). Some buildings, such as telephone exchanges, which have a continuous reliable source of heat, could use their heat source as a credit. Typically, however, a conservative approach to sizing heating equipment is taken, i.e., sizing for the worst-case condition. Due to the large difference between outside and inside air temperatures, space comfort could be adversely affected in worst-case conditions. Worse yet, frozen pipes, tanks, etc. could lead to very costly damage if the heating capacity was not available.

Warm-Up Safety Factor

If night setback temperature control is used to conserve energy, then some warm-up safety factor is needed as equipment reserve capacity to be available at the start of the occupied cycle to bring the space up to set point in a reasonable amount of time. A warm-up safety factor of 25 percent is often used in colder climates.

Summary

Load estimating is very important to HVAC system design because it not only calculates the many heat transfer load components that make up cooling and heating loads, it can model the dynamics of the building, along with the unique response of a specific HVAC system.

Heat transfer theory is relatively straightforward, utilizing simple formulas and computer software that models the complex dynamics of building heat storage and time-dependant heat balance equations.

Like many other aspects of HVAC design, simple check figures and rules of thumb allow the load estimating process to begin early in the design cycle when rough approximations are needed by the design team.

Load estimate output data is useful to the designer in many ways. Load profiles give the designer a visual representation of the load estimate relationship to outside air temperature and permit analysis of key load component impacts on building performance. Plotting state points and the RSHF line on the psychrometric chart begin the process of representing the interaction between the building load estimate and HVAC system response to the load.

Always make sure the building data is accurate, that the load estimate output is consistent with the building/HVAC system design, and do not forget to use the simplest method of estimating that fits the current need. Towards the middle of the design process, once most of the building component and system equipment details have been firmed up, use a software load estimating process that will be easy to update and keep current as the designs are finalized.

Work Session 1

1. Why is it important to understand the sources of the various building load components and how to combine them? _____

   _____

2. What three categories can all load components be classified into?

   _____

   _____

   _____

3. Explain what infiltration is. _____

   _____

4. Explain why it is difficult to accurately estimate this load component _____

   _____

5. Under what conditions would ventilation air be considered a room load? _____

   _____

6. Name the four supply air losses that exist between the coil of a central air handler and the room supply air terminal.

   _____ _____

   _____ _____

7. Would a blow through fan arrangement in a central air handler cause you to calculate the supply air fan heat (SAFH) as a loss between the coil and supply air terminal? (Yes/No)? Why? _____

   _____

8. Why must exhaust air be considered in a typical air conditioning system?

   _____

   _____

9. Define Grand Total Heat (GTH). _____

   _____

10. Define refrigeration load. _____

_____

11. What is the meaning behind the ASHRAE 0.4%, 1%, and 2% cooling db/mwb temperature columns? _____

_____

12. Of what use is the maximum design summer wet bulb temperature? _____

_____

13. In what project locations might one expect to use the maximum wb/mdb temperature columns? _____

14. Why is the peak of the horizontal glass curve so much greater than that of the south glass?

_____

_____

15. Why is the solar gain through the west glass nearly zero before noon? Why isn't it zero?

_____

_____

16. Explain why the actual cooling load caused by the sun's rays penetrating a window is smaller than and occurs at a peak time later than the instantaneous solar heat. _____

_____

_____

_____

17. Can the simple steady state equation ($q = U * A * \Delta t$) be used to determine the wall and roof transmission loads? _____

_____

18. Explain what Equivalent Temperature Difference (ETD) represents.

_____

_____

_____

19. Determine the sensible and latent load created by 100 people dancing (moderately) in a room whose temperature is maintained at 75° F. _____

_____

_____

20. Explain why stored energy from lights is generally neglected in calculating the light load component for a building. _____

_____

21. Explain why an accurate estimate of supply duct losses is not possible at the load-estimating phase of an air conditioning system design. _____

_____

_____

22. What is a typical percent of RSH used to estimate supply duct heat gain? _____

_____

23. Indicate where you would estimate the fan heat losses for the following:

	Supply Duct	Return Duct
A. Draw through fan	_____	_____
B. Blow through fan	_____	_____

24. Explain what bypassed outdoor air load is. _____

_____

25. When the ceiling plenum is located under a roof, _____ % of the roof load is absorbed by the return air, and _____ % is transmitted to the space. _____ % of the light heat is transmitted to the space, while _____ % is picked up by the return air.

26. Name the load components that make up a building heating load.

_____

_____

_____

27. What load components could be used as a heating load credit? Explain why. _____

_____

_____

_____

Appendix

Glossary

Cooling Check Figures, Commercial System Quick Reference

Approximate Design Air Distribution cfm (Cooling), Commercial Systems Quick Reference

Air Conditioning Load Estimate Form E-20A, Catalog No. 797-006

ASHRAE Standard 90.1 References for HVAC System Designers

Table 2 – Corrections in Outdoor Design Temperature for Time of Day, SDM-1

Table 3 – Corrections in Outdoor Design Conditions for Time of Year, SDM-1

Table 59 –Heat Gain Form Air Conditioning Fan Horsepower, Draw-Thru System, SDM-1

Heating Load Estimate Form E10A, Cat. No. 797-005

References

Work Session Answers

Load Components Terms List

	External Load Components
I_{LG}	infiltration, latent, glass
I_{SG}	infiltration, sensible, glass
I_{LW}	infiltration, latent, wall
I_{SW}	infiltration, sensible, wall
S_G	solar, glass
S_R	solar, roof
S_W	solar, wall
T_C	transmission, ceiling
T_F	transmission, floor
T_G	transmission, glass
T_P	transmission, partition
T_R	transmission, roof
T_W	transmission, wall
	Internal Load Components
E_L	equipment, latent
E_S	equipment, sensible
L	lighting
P_L	people, latent
P_S	people, sensible

	System Load Components
CBP	coil bypass
L	lighting
OA_L	outdoor air, latent
OA_S	outdoor air, sensible
PiHG	piping heat gain
PuHG	pump heat gain
RAFH	return air fan heat
RDHG	return duct heat gain
$RDLG_L$	return duct leakage gain, latent
$RDLG_S$	return duct leakage gain, sensible
SAFH	supply air fan heat
SDHG	supply duct heat gain
$SDLL_L$	supply duct leakage loss, latent
$SDLL_S$	supply duct leakage loss, sensible
S_R	solar, roof
S_W	solar, wall
T_R	transmission, roof
T_W	transmission, wall

Glossary

Word	**Definition**
airflow	movement of air, usually within boundaries such as ducts: commonly expressed as cfm.
applied equipment	field-assembled and commissioned equipment that has a range of selectable performance capacities that can be used either singly or in combinations to create an HVAC system or subsystem.
block load	maximum building or HVAC system heating or cooling load that occurs within the hours of a day, or days of a year; not necessarily as large as the sum of the individual zone loads within the block load area.
Btuh (Btu per hour)	the basic unit for measuring the rate of heat transfer within an HVAC system, British thermal units per hour
bypass factor	percentage of airflow that passes through a coil or filter untreated by the heat transfer surface or media.
cfm	cubic feet per minute. The unit of measure of the volume rate of air flow, as in a heating system.
check figure	a general sizing guideline used to evaluate preliminary equipment selections, capacities and fluid flows (i.e. cfm, gpm)
conduction	heat transfer by which heat is moved from molecule to molecule of a substance by a chain collision of those molecules
convection	heat transfer within a fluid by the movement of heated molecules from one place to another
energy simulation	software-based modeling of equipment or system performance for obtaining energy usage over time.
equivalent temperature difference (ETD)	alternative dry bulb temperature difference that creates the same transmission heat transfer as the actual dry bulb temperature difference, plus the transmission associated with the solar load component.
external load component	load component originating outside the space, zone, or building, but not associated with the HVAC equipment or distribution systems; normally external to the building, like solar and transmission loads.
heat	a form of energy that can be transferred by conduction, convection, or radiation; only transferred from a warmer substance to a colder substance.
heat extraction rate	rate at which heat is removed from the conditioned space.
indoor air quality (IAQ)	controlling the types and levels of indoor contaminants; involves material selections and moisture control within buildings, adequate ventilation with outdoor air, proper filtration, and the design, operation, and maintenance of comfort air conditioning systems for occupant benefit
internal load component	load component originating within the space, zone, or building, such as lighting, people and equipment.
life cycle cost	cost of equipment over its entire life, including operating and maintenance cost.
load calculation	mathematical calculation of the heat transfer associated with any of the various load components.
load component	heat transfer associated with a building or system element
load estimate	approximation of the heating and cooling requirements of a space, zone, system or plant; made up of individual load calculations for the associated load components.

packaged equipment	factory-assembled and tested equipment that has published performance capacities that can be used either singly or in combinations to create an HVAC system or subsystem.
pulldown load	cooling load at the initiation of a system on cycle; often made up of residual or stored load left over at the initiation of the off cycle.
R-value (thermal resistance)	quantity determined by the temperature difference, at steady state, between two defined surfaces of a material or construction that induces a unit heat flow rate through a unit area. $R = DT/q$.
room	an area within a building bounded on all sides with walls, a ceiling/roof and floor; rooms are considered spaces.
rule of thumb	a useful design guideline having wide application but not intended to be strictly accurate or reliable in every situation.
set point	point at which the desired value of the controlled variable is set.
safety factor	additional or reserve capacity or capability built into a system or component to account for unexpected situations or guard against failure or loss of set point.
shading coefficient (SC)	ratio of absorbed and transmitted solar heat relative to fenestration fitted with shading devices to that occurring with unshaded single strength glass.
solar temperature	alternative outside air temperature which, in the absence of all radiation exchanges, would give the same rate of heat entry into the surface as would exist with the actual combination of incident solar radiation, radiant energy exchange with the sky and other outdoor surroundings, and convective heat exchange with the outdoor air; used to determine equivalent temperature difference for sunlit building assemblies.
solar heat gain coefficient (SHGC)	fraction of incident irradiance that enters through the glazing and becomes heat gain; includes both the directly transmitted portion and the absorbed and re-emitted portion.
space	An area within a building not necessarily bounded on all sides by walls or a ceiling, such as a lobby or hall; rooms are considered spaces.
state point	A point or location on a psychrometric chart, pressure-enthalpy diagram, etc. that indicates the physical conditions of a mixture or process.
system load component	load component originating outside the conditioned space, zone, or building, and associated with the HVAC equipment or distribution systems, such as fan heat gain and coil bypass.
temperature difference (Δt)	difference between the temperatures of two substances, surfaces, or environments involving transfer of heat.
ton, ton of refrigeration	value used for cooling equipment capacity; equivalent to 12,000 Btuh.
transmission	transport of substances, energy, or indicated values from one place to another with or without impedances.
U-factor (thermal transmittance)	heat transmission in unit time through unit area of a material or construction and the boundary air films, induced by unit temperature difference between the environment on each side. *Note* - This heat transmission rate has been called the overall coefficient of heat transfer.
zone	a space, room or rooms, or building that is treated as a single comfort control area by an HVAC system.
zone load	peak cooling or heating load for the zone, occurring at a particular time of day and day of year; most likely, zones within an HVAC system peak at different times and dates.

Cooling Check Figures

Building Type	Cooling (ft^2/ton)	Lights/Equip. (Watts/ft^2)[1]	People (ft^2/person)	Ventilation[2]
Apartments				
Rental Suites	450	1.0	600	0.35 air changes per hour, but not
Hallways	550	2.0	-	less than 15 cfm/person
Auditoriums/Assembly	250 (20 seats/ton)	2.0	15	15 cfm/person
Bakeries[3]	225	6.0[1]	80	20 cfm/person
Banks	230	4.0[1]	30	15 cfm/person
Barber Shops	250	3.5[1]	45	15 cfm/person
Bars/Taverns/Pubs	100	1.0	20	30 cfm/person
Bathrooms (public)				
with Blow Driers[2]	200	8.0[1]	50	50 cfm/water closet
with Towel Dry	400	2.0	50	50 cfm/water closet
Beauty Shops	180	5.0[1]	45	25 cfm/person
Bowling Alleys	250 (1.5 tons/alley)	2.5	40	25 cfm/person
Clothing Shops (retail)	280	2.0	50	0.30 cfm/ft^2
Computer Rooms	80	6.0[1]	-	15 cfm/person
Department Stores				
Main Floors	300	2.0	35	0.30 cfm/ft^2
Upper Floors	380	2.0	50	0.20 cfm/ft^2
Doctor's Office				
Waiting Rooms	330	2.0	30	15 cfm/person
Examination Rooms	380	2.0	50	15 cfm/person
Drug Stores	250	3.0[1]	50	15 cfm/person
Educational				
Classrooms	280	2.0	20	15 cfm/person
Hallways	350	2.0	-	0.10 cfm/ft^2
Factories				
Light Manufacturing	240	3.0[1]	100	15 cfm/person
Medium Manufacturing	150	10.0[1]	200	15 cfm/person
Heavy Manufacturing	80	45.0[1]	300	15 cfm/person
Florists	250	3.0[1]	125	15 cfm/person
Gambling Casinos[4]	100	15.0[1]	25	30 cfm/person
Gift Shops	300	2.0	50	15 cfm/person
Grocery Stores	350	2.0	125	15 cfm/person
Hospital/Nursing				
Patient Rooms	300	1.5	100	25 cfm/person
Hallways	400	2.0	-	0.05 cfm/ft^2
Exam/Procedure Rooms	300	3.0[1]	50	15 cfm/person
Waiting Rooms	330	2.0[1]	30	15 cfm/person
Hotel, Motel, Dorm				
Guest Rooms	500	1.0	150	30/room
Hallways	560	2.0	-	0.05 cfm/ft^2
Lobby/Public	220	2.0	30	15 cfm/person
Offices				
One-Story	350	3.0[1]	150	20 cfm/person
Mid-Rise	400	3.0[1]	150	20 cfm/person
High-Rise	450	3.0[1]	150	20 cfm/person
Reception	350	2.0	50	15 cfm/person
Post Offices	280	2.0	30	15 cfm/person
Residences				
15% Glass[5]	700[6]	1.5[1]	400	0.35 air changes
25% Glass[5]	550[6]	1.5[1]	400	not<15 cfm/person
Restaurants				
Kitchen	180	12.0[1]	100	15 cfm/person
Fast Food Service, Dining	150	3.25[1]	30	20 cfm/person
Formal Dining	250	3.25[1]	30	20 cfm/person
Shoes Stores	300	2.0	50	15 cfm/person
Smoking Lounges	100	1.0	20	60 cfm/person

NOTES: There is no substitute for an accurate load analysis of a building, but these figures will help you to get a ballpark idea of the occupancy, lighting, ventilation and cooling requirements of a building in a timely manner. All figures assume the outdoor air ventilation rates shown.

[1] Where indicated, the lighting load includes other equipment such as driers, stoves, computers, etc.

[2] Ventilation rates are based on ASHRAE Standard 62-1999.

[3] Bakery figures assume that half of the floor space is dedicated to the kitchen and half to seating/serving/display area.

[4] The figures for the casino include some are allocated for a modest restaurant and for public bathrooms.

[5] In residences, the percent shown is the ratio of glass square footage to total outside wall square footage, expressed as a percent. A typical standard is 15%. A custom-built standard is 25%.

[6] Do not count unexposed, below-grade areas. They have no cooling load.

Approximate Design Air Distribution cfm (Cooling)

Building Type	Zone Primary Air cfm/ft² (by exposure)		
	East/South/West	North	Core
Apartments			
Rental suites	1.0	0.6	--
Hallways	--	--	0.7
Auditoriums and places of assembly	1.6		
Bakeries	1.8		
Banks	1.75		
Barber shops	1.6		
Bars/Taverns	4.0		
Bathrooms (public)[1]			
w/blow driers (2)	--	--	2.0
w/towel try	--	--	1.0
Beauty shops	2.2		
Bowling alleys	1.6		
Clothing shops (retail)	1.6	1.4	1.2
Computer rooms	7.5		
Department and discount stores			
Main floor	1.6	1.45	1.2
Upper floors	1.3	1.1	0.9
Doctor's offices			
Waiting rooms	1.2		
Examination rooms	1.0		
Drug stores	1.6		
Educational facilities			
Classrooms	1.6	1.3	1.2
Hallways	--	--	1.2
Factories			
Light manufacturing	1.7		
Medium manufacturing	2.7		
Heavy manufacturing	5.0		
Florists	1.6		
Gambling casinos	4.0		
Gift shops	1.3		
Grocery stores	1.3		1.1
Hospitals/nursing homes			
Patient rooms	1.4	1.2	--
Hallways	--	--	1.0
Exam/procedure rooms	--	--	1.3
Waiting rooms	1.5	1.3	1.0
Hotels/motels/dormitories			
Guest rooms	1.0	0.7	--
Hallways	--	--	0.7
Lobby/public areas	2.0	1.8	1.7
Jewelry stores	1.6		
Kitchens	2.2		
Libraries and museums	1.7	1.3	1.2
Offices			
One story	1.5	1.2	1.0
Mid-rise	1.3	1.0	0.8
High-rise	1.2	1.0	0.8
Reception	--	--	1.15
Post offices	1.4		
Residences[2]			
15% glass	0.6		
25% glass	0.75		
Shoe stores	1.3		
Smoking lounges	4.0		

Air quantities vary according to internal load assumptions, local code, designer preference, and air system selection criteria.

[1] Be sure to observe general building code for minimum exhaust. Bathroom figures shown above assume that the bathrooms are maintained at building design temperature.

[2] Percent shown is percent glass area of total outside wall area. 15% is typical; 25% is generous (as used for custom-built homes).

Air Conditioning Load Estimate

Carrier® Turn to the Experts.

Sheet ____ of ____ Date: _____
Prepared by: _____
Name of job: _____
Location: _____ Latitude: _____

Office: _____
System Name/Type: _____ Job No. _____
Approved: _____ Date: _____

Space used for: _____

Size: ____ x ____ = ____ ft² x ____ ft = ____ ft³

Item	Area or Quantity	Sun Gain or Temp. Diff.	Factor	Btuh Millions	Btuh 1,000's	Btuh 100's
SOLAR GAIN — GLASS						
Glass ( )	ft² x		x			
Glass ( )	ft² x		x			
Glass ( )	ft² x		x			
Glass ( )	ft² x		x			
Skylight	ft² x		x			
SOLAR & TRANSMISSION GAIN — WALLS & ROOF						
Wall ( )	ft² x		x			
Wall ( )	ft² x		x			
Wall ( )	ft² x		x			
Wall ( )	ft² x		x			
Roof–sun	ft² x		x			
Roof–shaded	ft² x		x			
TRANSMISSION GAIN — EXCEPT WALLS & ROOF						
All Glass	ft² x		x			
Partition	ft² x		x			
Ceiling	ft² x		x			
Floor	ft² x		x			
Infiltration	cfm x		x 1.10			
INTERNAL HEAT						
People			x			
Power	hp or kw x					
Lights	Watts x		3.4			
Appliances	Watts x		3.4			
Additional Heat Gains			x			
			subtotal			
Storage	ft² x		x (-)			
			subtotal			
Safety Factor			%			
Room Sensible Heat (RSH) ■						
Supply Duct Heat Gain	%+ Supply Duct Leak Loss		%+ Fan hp	%		
Outdoor Air	cfm x	° F x	BF x 1.10			
Effective Room Sensible Heat (ERSH) ■						
LATENT HEAT						
Infiltration	cfm x		gr/lb x 0.68			
People	x					
Steam	lb/hr x 1050					
Appliances, etc.						
Additional Heat Gains						
Vapor Trans.	ft² x 1/100 x		gr/lb x			
			subtotal			
Safety Factor			%			
Room Latent Heat (RLH)						
Supply Duct Leakage Loss			%			
Outdoor Air	cfm x	gr/lb x	BF x 0.68			
Effective Room Latent Heat (ERLH)						
Effective Room Total Heat (ERTH) ■						
OUTDOOR AIR HEAT						
Sensible	cfm x	° F x (1-	BF) x 1.10			
Latent	cfm x	gr/lb x (1-	BF) x 0.68			
Return Duct Heat Gain	%RSH + Return Duct Leakage Gain		% RSH			
Return Air or Blow-thru Fan			hp x 2545			
Grand Total Heat (GTH) or Dehumidifier Load) ■						
Pump	% GTH + Piping Heat Gain		% GTH			
REFRIGERATION LOAD ■						

Estimate for: Local Time Sun Time | Peak Load: Local Time Sun Time

Equipment Operation Hours/Day

Location	Conditions db	wb	%rh	Dp	gr/lb
Outdoor (OA)					
Room (RM)					
Difference		xxx	xxx	xxx	

OUTDOOR AIR

Ventilation: ____ people x ____ cfm/person = ____
____ ft² x ____ cfm/ ft² = ____
cfm ventilation ■ ____

Infiltration:
Swinging, revolving doors ____ people x ____ cfm/person = ____
Open doors ____ doors x ____ cfm/door = ____
Exhaust fan ____
Crack ____ feet x ____ cfm/ft = ____
cfm infiltration ■ ____
cfm outdoor air thru apparatus ■ ____ cfm_{OA}

APPARATUS DEWPOINT & DEHUMIDIFIED AIR QUANTITY

ESHF: Effective sensible heat factor = ERSH / ERTH = ____

ADP: Indicated ADP = ____ ° F Selected ADP = ____ ° F

Dehum. Rise: (1 - ____ BF) x (T_{RM} ____ ° F – T_{ADP} ____ ° F) = ____ ° F

Dehum. cfm: ERSH / (1.10 x ° $F_{Dehum.\ Rise}$) = ____ cfm_{DA}

SUPPLY AIR QUANTITY

Outlet Temp. Diff.: RSH / (1.10 x cfm_{DA}) = ____ ° F (RM – Outlet Air)*
*If this ΔT is too high, determine supply cfm for desired difference by supply air quantity formula.

Supply cfm: RSH / (1.10 x ° $F_{Desired\ Difference}$) = ____ cfm_{SA}

Bypass cfm: ____ cfm_{SA} - ____ cfm_{DA} = ____ cfm_{BA}

RESULTING ENT & LVG CONDITIONS AT APPARATUS

T_{RM} ____ ° F + (cfm_{OA} / cfm**) x (T_{OA}) ° F - T_{RM} ____ ° F = T_{EDB} ____ ° F

T_{ADP} ____ ° F + ____ BF x (T_{EDB}) - T_{ADP} ____ ° F = T_{LDB} ____ ° F

From Psych. Chart: T_{EWB} ____ ° F, T_{LWB} ____

**When bypassing a mixture of outdoor and return air, use supply cfm. When bypassing return air only, use dehumidified cfm.

NOTES

Form No. E-20A Cat. No. 797-006

ASHRAE Standard 90.1-2001
References for HVAC System Designers

This standard, *Energy Standard for Buildings Except Low-Rise Residential Buildings,* addresses many aspects relating to design of the system that should be considered during the selection process. Before listing those items, a quick review of what the standard is trying to address and accomplish is in order.

- Since it's beginning in 1975, this standard has had the purpose of providing minimum requirements for the energy-efficient design of buildings.
- The provisions of the standard apply to all building energy use related components and systems, with Chapter 6 addressing Heating, Ventilating and Air-Conditioning.
- Three compliance paths are available.

Item	Reference #	Notes
Simplified Approach Option for HVAC Systems (Option 1)	6.1.3	Works for smaller systems. Many pre-qualifiers
Mandatory Provisions (used with both of the next two items)	6.2	Deals with equipment efficiency, controls, construction, insulation, and balancing.
Prescriptive Path	6.3	Option 2 additive to Mandatory Provisions. Deals with economizers, simultaneous heating and cooling limitations, air system design and control, hydronic system design and control, heat rejection equipment, energy recovery, exhaust hoods, radiant heating system, and hot gas bypass limitations.
Energy Cost Budget Method	11.	Option 3 additive to Mandatory Provisions. Requires simulation of HVAC system design and *design energy cost* cannot exceed *energy cost budget.*
Mechanical Equipment Efficiency	6.2.1	Selected equipment must meet tabulated values.
Load Calculations	6.2.2	Heating and cooling load estimates must be run before equipment selection occurs.
Thermostatic Controls	6.2.3.1	Each zone shall be individually controlled.
Dead Band	6.2.3.1.2	$\geq$ 5° F required, unless special application.

Zone Isolation	6.2.3.2.5	HVAC systems serving zones operating non-simultaneously shall be divided into isolation areas.
Ventilation Control for High-Occupancy Areas	6.2.3.9	Controls shall automatically reduce outdoor air when spaces are partially occupied.
System Balancing	6.2.5.3	Combined with required VFDs, throttling losses in ductwork and piping are minimized.
Economizers	6.3.1	Required on most systems, per Table 6.3.1
Integrated Economize Controls	6.3.1.3	Economizers must be integrated with mechanical cooling system, with limited exceptions
Simultaneous Heating and Cooling Limitations	6.3.2	Prevents reheating, recooling or mixing of heated and cooled airstreams, with limited exceptions that permit VAV terminals.
Dehumidification	6.3.2.3	Does not allow humidistatic controls that violate 6.3.2, unless strict exceptions apply.
Humidification	6.3.2.4	If hydronic cooling is used and humidification set point exceeds 35° F, then a hydronic economizer is required.
Fan Power Limitation	6.3.3.1	Ratio of fan system power to supply fan airflow controlled by Table 6.3.3.1
VAV Fan Control	6.3.3.2	Guidelines for proper design (i.e. VFDs on motors $\geq$ 30hp)
Hydronic System Design and Control	6.3.4	Guidelines for proper design (i.e. VFDs on motors $\geq$ 50 hp and 100ft head)
Heat Rejection Equipment	6.3.5.2	Fan speed control on motors $\geq$ 7½ hp.
Exhaust Air Energy Recovery	6.3.6.1	Required on individual systems $\geq$ 5000 cfm and have minimum outdoor air supply $\geq$ 70% design airflow.
Exhaust Hoods	6.3.7	Special requirements for kitchen hoods and fume hood systems

TABLE 2—CORRECTIONS IN OUTDOOR DESIGN TEMPERATURES FOR TIME OF DAY

(For Cooling Load Estimates)

DAILY RANGE OF TEMPERATURE* (F)	DRY- OR WET-BULB	SUN TIME									
		AM			PM						
		8	10	12	2	3	4	6	8	10	12
10	Dry-Bulb	− 9	− 7	−5	−1	0	−1	−2	− 5	− 8	− 9
	Wet-Bulb	− 2	− 2	−1	0	0	0	−1	− 1	− 2	− 2
15	Dry-Bulb	−12	− 9	−5	−1	0	−1	−2	− 6	−10	−14
	Wet-Bulb	− 3	− 2	−1	0	0	0	−1	− 1	− 3	− 4
20	Dry-Bulb	−14	−10	−5	−1	0	−1	−3	− 7	−11	−16
	Wet-Bulb	− 4	− 3	−1	0	0	0	−1	− 2	− 3	− 4
25	Dry-Bulb	−16	−10	−5	−1	0	−1	−3	− 8	−13	−18
	Wet-Bulb	− 4	− 3	−1	0	0	0	−1	− 2	− 3	− 5
30	Dry-Bulb	−18	−12	−6	−1	0	−1	−4	−10	−15	−21
	Wet-Bulb	− 5	− 3	−1	0	0	0	−1	− 3	− 4	− 6
35	Dry-Bulb	−21	−14	−7	−1	0	−1	−6	−12	−18	−24
	Wet-Bulb	− 6	− 4	−2	0	0	0	−1	− 3	− 5	− 7
40	Dry-Bulb	−24	−16	−8	−1	0	−1	−7	−14	−21	−28
	Wet-Bulb	− 7	− 4	−2	0	0	0	−2	− 4	− 6	− 9
45	Dry-Bulb	−26	−17	−8	−2	0	−2	−8	−16	−24	−31
	Wet-Bulb	− 7	− 5	−2	0	0	−1	−2	− 4	− 8	−10

*The daily range of dry-bulb temperature is the difference between the highest and lowest dry-bulb temperature during a 24-hour period on a typical design day. (See *Table 1* for the value of daily range for a particular city).

Equation: Outdoor design temperature at any time = Outdoor design temperature from *Table 1* + Correction from above table.

TABLE 3—CORRECTIONS IN OUTDOOR DESIGN CONDITIONS FOR TIME OF YEAR

(For Cooling Load Estimates)

YEARLY RANGE OF TEMPERATURE(F)*	DRY- OR WET-BULB	TIME OF YEAR								
		March	April	May	June	July	August	Sept.	Oct.	Nov.
120	Dry-Bulb	−39	−22	−11	−4	0	0	−9	−24	−44
	Wet-Bulb	−23	−12	− 5	−2	0	0	−4	−13	−27
115	Dry-Bulb	−33	−22	−11	−4	0	0	−8	−20	−36
	Wet-Bulb	−18	−11	− 5	−2	0	0	−4	−10	−21
110	Dry-Bulb	−30	−20	−11	−4	0	0	−6	−17	−31
	Wet-Bulb	−15	−10	− 5	−2	0	0	−3	− 8	−16
105	Dry-Bulb	−30	−20	−11	−4	0	0	−6	−17	−29
	Wet-Bulb	−15	−10	− 5	−2	0	0	−3	− 8	−14
100	Dry-Bulb	−29	−19	−10	−3	0	0	−6	−16	−27
	Wet-Bulb	−14	−10	− 5	−2	0	0	−3	− 8	−14
95	Dry-Bulb	−29	−19	−10	−3	0	0	−6	−16	−27
	Wet-Bulb	−14	−10	− 5	−2	0	0	−3	− 8	−14
90	Dry-Bulb	−29	−19	−10	−3	0	0	−6	−16	−26
	Wet-Bulb	−14	−10	− 5	−2	0	0	−3	− 8	−14
85	Dry-Bulb	−29	−19	− 9	−3	0	0	−5	−16	−25
	Wet-Bulb	−14	−10	− 5	−2	0	0	−3	− 8	−14
80	Dry-Bulb	−24	−16	− 8	−3	0	0	−4	−12	−20
	Wet-Bulb	−13	− 9	− 4	−2	0	0	−2	− 6	−11
75	Dry-Bulb	−14	− 9	− 4	−1	0	0	−3	− 7	−15
	Wet-Bulb	− 7	− 5	− 2	0	0	0	−2	− 4	− 8
70	Dry-Bulb	−13	− 9	− 4	−1	0	0	−2	− 7	−14
	Wet-Bulb	− 6	− 4	− 2	0	0	0	−1	− 4	− 6
65	Dry-Bulb	−11	− 8	− 4	−1	0	0	−2	− 6	−12
	Wet-Bulb	− 6	− 4	− 2	0	0	0	−1	− 3	− 6
60	Dry-Bulb	− 9	− 7	− 3	−1	0	0	−2	− 5	−10
	Wet-Bulb	− 4	− 3	− 2	0	0	0	−1	− 3	− 5
55	Dry-Bulb	− 6	− 5	− 3	−1	0	0	−2	− 4	− 8
	Wet-Bulb	− 3	− 3	− 2	0	0	0	−1	− 2	− 4
50	Dry-Bulb	− 5	− 4	− 3	−1	0	0	−2	− 4	− 7
	Wet-Bulb	− 3	− 2	− 1	0	0	0	−1	− 2	− 3

*Yearly range of temperature is the difference between the summer and winter normal design dry-bulb temperatures (*Table 1*).

Equation: Outdoor design temperature = Outdoor design temperature from *Table 1* + Corrections from above table.

TABLE 59—HEAT GAIN FROM AIR CONDITIONING FAN HORSEPOWER, DRAW-THRU SYSTEM‡‡

	FAN TOTAL PRESSURE† (In. of Water)	CENTRAL STATION SYSTEMS‡ Temp Diff Room to Supply Air					APPLIED OR UNITARY SYSTEM** Temp Diff Room to Supply Air				
		10 F	15 F	20 F	25 F	30 F	10 F	15 F	20 F	25 F	30 F
		PERCENT OF ROOM SENSIBLE HEAT*									
Fan Motor Not in Conditioned Space or Air Stream	0.50	1.2	0.8	0.6	0.5	0.4	2.2	1.5	1.1	0.9	0.7
	0.75	1.9	1.3	1.0	0.8	0.6	3.5	2.4	1.8	1.4	1.2
	1.00	2.7	1.8	1.4	1.1	0.9	4.8	3.2	2.4	1.9	1.6
	1.25	3.9	2.6	1.9	1.6	1.3	6.5	4.3	3.2	2.6	2.2
	1.50	4.6	3.1	2.3	1.9	1.6	7.8	5.2	3.9	3.1	2.6
	1.75	5.4	3.6	2.7	2.2	1.8	9.1	6.1	4.6	3.6	3.0
	2.00	6.2	4.1	3.1	2.5	2.1	10.4	6.9	5.2	4.2	3.5
	3.00	10.4	6.9	5.2	4.2	3.5	16.7	11.2	8.4	6.7	5.6
	4.00	15.3	10.2	7.7	6.1	5.1					
	5.00	19.2	12.8	9.6	7.7	6.4					
	6.00	24.4	16.3	12.2	9.9	8.2					
	8.00	38.0	25.4	19.0	15.2	12.7					
Fan Motor†† in Conditioned Space or Air Stream	0.50	1.6	1.1	0.8	0.6	0.5	2.7	1.8	1.4	1.1	0.9
	0.75	2.6	1.8	1.3	1.1	0.9	4.2	2.8	2.1	1.7	1.4
	1.00	3.6	2.4	1.8	1.5	1.2	5.8	3.8	2.9	2.3	1.9
	1.25	5.0	3.4	2.5	2.0	1.7	7.6	5.1	3.8	3.1	2.6
	1.50	6.0	4.0	3.0	2.4	2.0	9.2	6.1	4.6	3.7	3.1
	1.75	7.0	4.7	3.5	2.8	2.4	10.7	7.2	5.4	4.3	3.6
	2.00	8.0	5.4	4.0	3.2	2.7	12.2	8.2	6.1	4.9	4.1
	3.00	13.2	8.8	6.6	5.3	4.4	19.5	13.1	9.8	7.8	6.5
	4.00	19.0	12.7	9.5	7.6	6.4					
	5.00	23.8	15.9	11.9	9.5	8.0					
	6.00	30.0	20.0	15.0	12.0	10.0					
	8.00	45.5	30.3	22.8	18.2	15.2					

*Excludes from heat gain, typical values for bearing losses, etc. which are dissipated in apparatus room.

†Fan Total Pressure equals fan static pressure plus velocity pressure at fan discharge. Below 1200 fpm the fan total pressure is approximately equal to the fan static. Above 1200 fpm the total pressure should be figured.

‡70% fan efficiency assumed.

**50% fan efficiency assumed.

††80% motor and drive efficiency assumed.

‡‡For draw-thru systems, this heat is an addition to the supply air heat gain and is added to the room sensible heat. For blow-thru systems this fan heat is added to the grand total heat; use the RSH times the percent listed and add to the GTH.

Heating Load Estimate

Carrier
Turn to the Experts.

Sheet ____ of ____ Date: _____
Prepared by: _____
Name of job: _____
Location: _____ Latitude: _____

Office: _____
System Name/Type: _____ Job No. _____
Approved: _____ Date: _____

HEATING CONDITIONS

Room _____db _____wb _____% rh _____gr/lb
Outside _____db _____wb _____% rh _____gr/lb
Diff. _____° F Diff. _____gr/lb

TEMPERATURE OF AIR ENTERING UNIT

____ % outdoor air x _____° F = _____° F
____ % recirculated air x _____° F* = _____° F
Total (average entering air temp.) = _____° F

*Room temperature plus temperature correction for heat of unit

TRANSMISSION

Surface	ft²	Trans. Coeff.	Btuh per °F Diff.	Total Btuh per °F Diff.	Temp. Diff.	Btuh

References

Carrier Corp.

Commercial Systems Quick Reference (CSQR), Catalog No. 795-205.

System Design Manual – Load Estimating, SDM-1, Catalog No. 510-304.

User Guide – Load Estimating Using Storage Load Factors and Equivalent Temperatures, Catalog No. 791-090.

ASHRAE

ASHRAE Handbook – 2001 Fundamentals:

Chapter 27, Climatic Design Information

Chapter 29, Nonresidential Cooling and Heating Load Calculation Procedures

ASHRAE Standard 62 – Ventilation for Acceptable Indoor Air Quality.

ASHRAE Standard 90.1 – Energy Efficient Design of New Buildings (Except Low Rise Residential Buildings).

Work Session Answers

1. To perform a load estimate for the selection of the heating and cooling equipment that will condition the various building spaces.
2. External, internal, and system
3. The passage of outdoor air into the conditioned space due to positive outside wind pressure.
4. Due to the tremendous variability of building geometry, building construction quality and wind patterns.
5. When introduced untreated directly into the space and/or when bypassed through or around an air handling unit coil.
6. Supply Duct Heat Gain, Supply Duct Leakage Loss, Supply Air Fan Heat, and Bypassed Outdoor Air.
7. No. The load would be calculated on the return side of the coil since this is physically where the fan would be located.
8. Since ventilation air is continually being drawn into the system, a similar quantity of return air must be exhausted (or relieved) in order to maintain balanced airflow through the system. With no exhaust, air pressure would build in the system, preventing the influx of any new ventilation air.
9. The total load seen by the coil consists of room loads, supply duct losses and outdoor air loads.
10. In a chilled water system, GTH plus pumping and piping losses between the coil and the chiller. In a DX system refrigeration load equals GTH.

11. They mean that the dry bulb and mean coincident wet bulb design temperature values shown are exceeded on average 0.4% (35 hours), 1% (88 hours), or 2% (175 hours) of the 8760 hours in a year.

12. For selecting wet bulb performance related equipment such as cooling towers.

13. In warm maritime areas that are close to the ocean.

14. In August the sun is relatively high in the sky. Thus, a great deal of solar energy striking the south glass is reflected off the glass. The horizontal glass on the other hand has the sun's rays striking it more directly with little reflection taking place. Most of the energy penetrates the glass.

15. The west is in shade up to noon. Even though in the shade the glass is still subject to diffuse radiation.

16. The sun's rays penetrating the glass are in the form of radiant energy must first strike a surface to be converted into heat. Not all of this heat shows up immediately in the space because some of it flows into the mass of the building by conduction. Over time, this stored heat makes its way into the space. Thus, the peak load caused by the sun's rays occurs later in the day and has a lower magnitude then the instantaneous heat gain.

17. Even though the storage effect and solar loading of the external wall or roof structure makes the flow of heat time-related and thus not a steady-state heat flow situation, the use of ETD allows the load allocation to be completed using the steady state equation.

18. ETD is a fictitious temperature difference that allows one to stop the flow of heat (like a frame in a camera) at one point in time and use the steady state heat transfer equation to calculate the load. It is the equivalent of the complex storage related heat transfer process that is actually taking place through the wall or roof.

19. 100 people * 325 Btuh/person = 32,500 Btuh sensible
    100 people * 525 Btuh/person = 52,500 Btuh latent

20. Carpeting is widely used. The primary target for radiant energy from the lights is the floor. Carpet insulates the floor from radiant light rays. The radiant energy strikes the carpet and is converted into heat. The heat is then dissipated almost immediately into the space.

    Unless the air conditioning equipment is run for an hour or two before lights are turned on, the mass of the building will already be saturated with stored heat from the night before. As the light heat is absorbed in the mass, the equivalent amount of energy is almost simultaneously released to the space.

21. At the load estimating phase of the project, the designer has not designed the ductwork or piping systems and in many instances has not as yet decided on the type of system to be used. Thus estimates must be made.

22. 2%-3% for mid-level pressure class ductwork (±2-in. to +6-in. static pressure). Lower pressure class ductwork would likely have 5% leakage.

23. A. Supply Duct

    B. Return Duct

24. This represents a loss that air leaving the coil must overcome before entering the room. Some air passing through the cooling coil untreated due to the coil bypass factor phenomenon causes it.

25. 70%, 30%, 70%, 30%

26. Transmission, infiltration, and ventilation

27. Only load sources that are reliable and can be counted on as being there. Solar and lights are typically not used for this reason. Generally, the peak heating load occurs in the early morning hours when the lights are not normally on and the sun has not yet risen.

Prerequisites:

An understanding of air conditioning equipment and systems along with psychrometric properties of air-water mixtures. This knowledge can be gained from TDP-103, Concepts of Air Conditioning, and TDP-201, Psychrometrics, Level 1: Introduction.

Learning Objectives:

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

- Understand the importance of load estimating and its role in HVAC design
- Understand the heat transfer theory behind conduction, convection, radiation and thermal storage
- Choose the appropriate load estimating method for each project, from manual check figures to computerized heat balances
- Categorize load components as to external, internal, and system groups, and calculate each
- Evaluate building loads using a load profile
- Plot state points and the room sensible heat factor line on the psychrometric chart

Supplemental Material:

Carrier Commercial Systems Quick Reference, ENG-05A, Catalog No. 795-205

Carrier Block Load, version 3.05. HVAC load estimating program for commercial buildings

Carrier Hourly Analysis Program (HAP), version 4.20a. HVAC system-based design and energy analysis program for commercial buildings

Air Conditioning Load Estimate Form No. E-20A, Catalog No. 797-006

Heating Load Estimate Form No. E-10A, Catalog No. 797-005

System Design Manual, Part 1, Load Estimating, Catalog No. 510-304

Load Estimating, Level 3, Application, Form No. TDP-302, Catalog No. 796-035

Load Estimating, Level 4, Theory, Form No. TDP-304, Catalog No. 796-036 (in development)

Instructor Information

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



Turn to the Experts.

Carrier Corporation
Technical Training
800 644-5544
www.training.carrier.com

Form No. TDP-301A
Supersedes T200-04A

Cat. No. 796-034
Supersedes 791-004