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Heating, ventilation and air conditioning engineering and design

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HEATING, VENTILATION and AIR CONDITIONING ENGINEERING and DESIGN

by

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A Design Project

in

Partial Fulfillment

of the

Requirements for the Degree of

MASTER OF SCIENCE

in

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Abstract

The object of this Design Project was to learn and demonstrate the engineering and design of Heating, Ventilation and Air Conditioning systems. This was done in two parts. The first part was the design project. The design project consisted of designing the heating, ventilation and air-conditioning systems for a three story office building. The second part discusses the theoretical aspects of heating, ventilation and air-conditioning systems which are used in the design project.

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Introduction

"By the end of the 19^{th} century the concept of central heating was fairly well developed, and early in the twentieth century cooling for comfort got its start"[1]. Since this time there have been many developments to bring the industry to the level of design it has now. The greatest developments have been made in the detailed analytical methods needed for the design of large systems. The most recent major development has been in designing heating, ventilation and air conditioning systems through the use of computers. The Heating Ventilation and Air Conditioning field is very large in engineering and it has grown through the input of thousand of engineers, their names are too numerous to list.

This Design Project is broken down into two parts. The first part is the design of a heating ventilation and air-conditioning system for a three story office building. The second part discusses the theoretical aspect of heating ventilation and air conditioning engineering and design. .

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1 Faye C. Mc. Quiston and Jerald D. Parker. Heating Ventilation and Air Conditioning Analysis and Design, 2nd ed. New York: John Wiley and Sons, 1982

1. Design Proposal

1.1 Proposal

The purpose of the design project was to demonstrate the concepts and theory of heating, ventilation and air-conditioning design. This was done by designing the heating, ventilation and air-conditioning systems for a three story office building. The design used many different types of systems and combinations of systems.

1.2 The Building

The building used in this design was a three story office building approximately 160' by 100' or 48,000 square feet. The building was designed for tenant fit up. Tenant fit up means that the office space floor area is open and the tenant will install his own walls to create the tenant's desired office area.

The building's main entrance is two stories high and leads into the main corridor. This corridor is the same for the other two floors. The building's exterior wall area is approximately 40% glass and 60% brick. The building has no basement, it is slab on grade. The roof of the building is metal deck with a foam/roof membrane covering. Further details of the building (wall and roof sections) can be seen on the plans.

1.3 The Choice of Building Systems

The choice of systems for this building might not be the most optimal choice of cost and efficiency versus function. The systems were chosen to demonstrate the knowledge and understanding for the HVAC field.

The system in the building uses a water loop heat transfer system, using Water-to-Air Unitary heat pumps for the building perimeter and a Variable Air Volume system with central electric heat for the interior. The system uses a gas boiler and a fluid cooler to maintain the water temperature in the water loop. A more detailed discussion of these systems is given in the chapter "Discussion of Building Systems".

1.4 Depth of Analysis and Design

The heating and cooling load analysis was performed in detail. The psychrometric analysis and design portion of the project was performed in partial detail. What is meant by this is that the psychrometric analysis, duct static pressure calculations and water loop system static pressure calculations were only performed in detail for the second floor of this building. This was done because these calculations would be redundant work if preformed for the other two floors. The basis of this design project is to show the basic understanding of this field.

2. Building Loads

2.1 Method Used To Determine Loads

A computer program was used to determine the heating and cooling loads for the building. The computer program used was the Hourly Analysis Program by Carrier. This program used separate methods for determining the heating and cooling loads.

2.1.1 Heating Design Load Analysis - The " Design Point" analysis technique was used to determine the heating loads. A design point was used because usually the coldest part of the year is at night at a specific temperature. At this point the program analyzes transmission, ventilation and infiltration loads. This point selected is not associated with any particular month or hour.

2.1.2 Cooling Design Load Analysis - Cooling loads have to take into consideration of the loads caused by thermal storage and solar heat gain. Therefor, an hour by hour analysis was preformed to compute the cooling loads. From this analysis the maximum cooling load was determined. Often this maximum cooling load does not correspond to the maximum outdoor temperature.

2.1.3 Weather Data - Weather data for the cooling analysis was given by a pair of hourly wet bulb and dry bulb temperature profiles and one set of solar profiles per month. The design cooling load temperature data was derived using an empirical method and represents 1% cooling design conditions. For design heating load calculations, the 99% Winter design point was used. The program contains pre-stored weather data of this type for over 500 cities worldwide.

2.1.4 Schedule Data - Schedules were used to give hourly characteristics of internal heat gains. Such characteristics were lighting, occupancy, miscellaneous internal loads and equipment operations.

2.1.5 Define Spaces - Spaces in a building are areas which have load related elements like walls, glass and ceilings. Usually spaces are individual rooms or for vary large rooms there could be many spaces. For each space there is a space input sheet that must be filled out so its data can be entered into the computer. An example of this sheet is on the following two pages.

Figure 2-1, Space Input Sheet
Abridge from The Carrier Hourly Analysis Program

COMPLEX SPACE INPUT SHEET

SPACE NAME:

PAGE 1 OF 2

WALL INFORMATION (NUMBER OF WALL TYPES =)

	Weight (H, M, L or lb/sqft)	Ext Color (D, M, L)	U-Value (BTU/hr/sqft/F)
Wall Type 1			
Wall Type 2			
Wall Type 3			

NET WALL AREAS (SQFT)

Exposure	Wall Type 1	Wall Type 2	Wall Type 3
Northeast			
East			
Southeast			
South			
Southwest			
West			
Northwest			
North			

ROOF INFORMATION (NUMBER OF ROOF TYPES =)

	Weight (H, M, L or lb/sqft)	Ext Color (D, M, L)	U-Value (BTU/hr/sqft/F)	Area (sqft)
Roof Type 1				
Roof Type 2				
Roof Type 3				

GLASS INFORMATION (NUMBER OF GLASS TYPES =)

	U-Value (BTU/hr/sqft/F)	Glass Factor	Internal Shades?
Glass Type 1			
Glass Type 2			
Glass Type 3			

EXTERNAL SHADING INFORMATION

	Window Height (ft)	Window Width (ft)	Reveal Depth (in)	Overhang Height (in)	Overhang Extension (in)	Fin Separation (in)	Fin Extension (in)
Shade 1							
Shade 2							
Shade 3							

	Glass Type 1		Glass Type 2		Glass Type 3	
Exposure	Area (sqft)	Shade	Area (sqft)	Shade	Area (sqft)	Shade
Northeast						
East						
Southeast						
South						
Southwest						
West						
Northwest						
North						
Horizontal						

COMPLEX SPACE INPUT SHEET

SPACE NAME: PAGE 2 OF 2

INTERNAL LOADS

SPACE DATA	: Floor Area	= ______ sqft	Building Wt.	= ______
PEOPLE	: sqft/person	= ______	Total People	= ______
	Schedule No.	= ______	Activity Level	= ______
	Sensible Gain	= ______	Latent Gain	= ______
LIGHTING	: W/sqft	= ______	Total Watts	= ______
	Schedule No.	= ______	Wattage Mult.	= ______
	Fixture Type	= ______		
OTHER ELECTRIC	: W/sqft	= ______	Total Watts	= ______
	Schedule No.	= ______		
MISC SENSIBLE	: Load	= ______ BTU/hr	Schedule No.	= ______
MISC LATENT	: Load	= ______ BTU/hr	Schedule No.	= ______

PARTITIONS, INFILTRATION, GROUND

PARTITIONS (Next to Unconditioned Spaces)

	Area (sqft)	U-Value (BTU/hr/sqft/F)	Unconditioned Space Temp. Cooling (deg F or %)	Unconditioned Space Temp. Heating (deg F or %)
Walls				
Ceilings				
Floors				

INFILTRATION			GROUND ELEMENT		
Cooling :	CFM/sqft =	CFM	Area	:	sqft
Heating :	CFM/sqft =	CFM	Perimeter	:	ft
Typical :	CFM/sqft =	CFM	Depth	:	ft

KEYS:

People Activity Levels:

1. Seated at Rest (230S/120L)
2. Office Work (245S/205L)
3. Sedentary Work (280S/270L)
4. Medium Work (295S/455L)
5. Heavy Work (525S/925L)

Lighting Fixture Type:

1. Recessed, not vented
2. Vented
3. Free-hanging

2.1.6 Define Zones - Zones are a group of spaces that share the same general set of air system characteristics. Usually a zone is selected for group of spaces that will be under the same HVAC system. An example of a zone input sheet is on the following page.

2.1.7 Running the Program - The program can be run for a single hour calculation or multiple hour calculations. Usually multiple hour calculations are used to determine the peak cooling load of the system. A single point calculation is used to determine the peak heating load.

Figure 2-2, Zone Input Sheet
Abridge from The Carrier Hourly Analysis Program

ZONE INPUT SHEET

1. ZONE NAME AND TYPE

Zone Name : ____________
Job Name : ____________
Zone Type = 1 Normal Zone
2 Single-Space Zones

2. THERMOSTAT AND EQUIPMENT SCHEDULE

COOLING EQUIPMENT
Occupied cooling setpoint = ____________ F
Unoccupied cooling setpoint = ____________ F
Starting hour, occupied per. = ____________
No. hours in occupied period = ____________

HEATING EQUIPMENT
Heating thermostat setpoint = ____________ F

3. COOLING SYTEM PARAMETERS

SUPPLY AIR
Type of Input = 1 CFM/sqft
2 CFM
3 Temperature
Supply Air = ____________

VENTILATION AIR
Type of Input = 1 CFM/sqft
2 CFM
3 % supply air
4 CFM/person
Ventilation Air = ____________

SAFETY FACTOR
Clg Safety Factor = ____________ %

4. HEATING SYSTEM PARAMETERS

HEATING SOURCE
Type of heating sytem = 1 Warm Air
2 Hydronic
(If Warm Air:)
Supply Temperature = ____________ F
(If Hydronic:)
Water Temperature Drop = ____________ F

VENTILATION AIR
Type of Input = 1 CFM/sqft
2 CFM
3 % supply air
4 CFM/person
Ventilation Air = ____________

SAFETY FACTOR
Heating Safety Factor = ____________ %

5. OTHER SYSTEM PARAMETERS

SUPPLY FAN
Total static pressure = ____________ in wg
Total efficiency = ____________ %
Fan configuration = 1 (Draw-Thru) 2 (Blow-Thru)

EXHAUST AIR
Direct exhaust air flow rate = ____________ % of vent air

RETURN AIR
Is a return air plenum used ? Y N
(If Yes:)
% of lighting load to plenum = ____________ %
% of roof load to plenum = ____________ %
% of wall load to plenum = ____________ %

COIL DATA
Cooling coil bypass factor = ____________

6. SPACE SELECTION

Space Name	Qty	Space Name	Qty	Space Name	Qty
1		18		35	
2		19		36	
3		20		37	
4		21		38	
5		22		39	
6		23		40	
7		24		41	
8		25		42	
9		26		43	
10		27		44	
11		28		45	
12		29		46	
13		30		47	
14		31		48	
15		32		49	
16		33		50	
17		34			

2.2 Building Input

2.2.1 Weather Input - Weather data was taken from the area of Hartford Connecticut.

2.2.2 Schedule Data - There were two type of schedules, people and lighting & electric. The people schedule was as follows:

Week Day	100%	occup. 8:00 am to 5:00 pm
	20%	occup. 5:00 pm to 8:00 pm
	0%	occup. 8:00 pm to 8:00 am
Saturday	20%	occup. 8:00 am to 1:00 pm
	0%	occup. 1:00 pm to 8:00 am
Sunday	0%	occup. All Day

The lighting and electric schedule was as follows.

Week Day	100%	8:00 am to 8:00 pm
	0%	8:00 pm to 8:00 am
Saturday	100%	8:00 am to 1:00 pm
	0%	1:00 pm to 8:00 am
Sunday	0%	All Day

2.2.3 Defining Spaces - Since the floor areas were open, the spaces were divided into interior spaces and perimeter spaces. Perimeter spaces extended from the outside wall to approximately 12 feet to the inside portion of the building. This was done because it was assumed that these perimeter spaces were going to be made into private offices. The interior spaces were divided into

6 approximately equal areas. The names and locations of these spaces can be found in the plan section of this paper.

In reviewing the complex space input sheet, the first information that has to be given is the wall, roof and glass information data. The wall, roof and glass areas can be found in the appendix. The weight, exterior color, U-value for the walls were calculated to be 46 lb./sq.ft., dark and .0441 BTU/h/sq.ft./°F respectively. The weight, exterior color and the U-value for the roof were calculated to be 3.165 lb./sq.ft., dark and .075 BTU/h/sq.ft./°F respectively. The glass had a U-value of .53 BTU/h/sq.ft./F and a glass factor of .6. There were no internal shades or external shades

On the second page of the space input sheet the first section deals with internal loads. The space data varies with each space. The people data was given in the following form: The number of people were given in square feet per person and ASHRAE estimates that their are approximately 110 sq.ft./person for an office building. The schedule number was given of the occupancy schedule. The activity level was given to be 2, for office work.

The lighting and other electric were combined as 6.0 watts per square foot, which was an average given by ASHRAE. The schedule was associated with the lighting and electric schedule. There were no miscellaneous sensible or latent loads.

Partition data was entered according to the needs of each space. Infiltration data was entered as cfm/sq.ft.. The values inputted were .05 cfm/sq.ft. for perimeter spaces and 0.0 cfm/sq.ft. for interior spaces. Initially the air change method was used on this building but the values obtained from this method were too high for an internally dominated building. The ground element varied according to each space.

2.2.4 Defining Zones - The spaces were broken up into 6 different zones. Each floor had 2 zones, one zone was for the interior spaces the other was for the perimeter spaces.

Inputting the zones' data and referring back to the zone input sheet, the cooling system parameters, heating system parameters and other system data remained the same for all the zones. The cooling system parameters were given as a 75°F indoor dry bulb with a 50% relative humidity, a supply air temperature input type of 57°F, a ventilation air input of 20 cfm/person (a requirement given by Building Officials and Code Administrators, BOCA, code for an office building) and a factor of safety of 10%. The heating system parameters were given as an indoor dry-bulb temperature of 68°F, a heating source as warm air and a supply temperature 102°F, ventilation air was given as 20 cfm/person and a safety of factor 10%.

The other system data inputted was as follows: a return air plenum was used, 100% ventilation air was exhausted, the lighting load to the plenum was 10%, and the roof load to the plenum was 30%. The supply fan data was as follows: an exhaust static pressure of .5" of water, a draw through configuration, a coil by pass factor of .05, and a 24 hours of system operation.

The spaces for each zone depended on the specific zone. Space multipliers were 1. The input for each zone can be found in the appendix.

2.3 Determining Building Loads

Four sets of heating and cooling loads were run. The first set determined the maximum cooling and heating loads for the entire year. The second, third and fourth sets determined the cooling and heating loads for the months of January, March and October.

This was done to determine if a heat pump system was a viable system for this building.

2.4 Discussion of Load Calculations

2.4.1 Discussion of Output

An example of the output can be found in the appendix.

2.4.1.1 Maximum Heating and Cooling Loads - These loads were calculated in order that the HVAC system could be sized properly. This means the HVAC system has to meet the maximum load on the building. These loads are the extreme cases that the building's HVAC system would have to operate under. The results obtained through this analysis were checked against the average values obtained from ASHRAE and these results corresponded in an acceptable range. The results are to lengthy to be presented in this chapter.

2.4.1.2 Heating and Cooling Loads for January, March and October - Heating and cooling loads were calculated for these months to show that a heat pump system can be used efficiently in this building application. To determine if a heat pump system is viable there must be a need for cooling in the building in the Fall, Winter and Spring months so that the heat extracted to cool these areas can be transferred to other areas of the building. From studying the cooling loads for these months (found in the appendix) there is enough heat generated from the need for cooling to make a heat pump system viable. The largest generator of heat was found to be the interior portion of the building which is obvious from examining the building design. Examining the design load cooling summary sheets the largest contributor of the internal heat generated was due to the lighting and electric loads. The second largest contributor to the internal heat generated was the people.

Certain areas of the perimeter portion of the building also needed cooling in the winter months. These areas are on the southern exposer of the building. The largest heat gain in these areas was due to the solar loads.

3. Building Systems

3.1 General

The system used heat pumps to heat and cool the perimeter spaces of the building and used a self contained variable air volume system with central electric heat to cool and heat the interior spaces of the building. Return air for both systems flowed through a return air plenum located in the ceiling space. The system was design, through a water loop system, to transfer heat generated from the interior portion of the building to the perimeter portion for the Fall, Winter and Spring months.

3.2 Variable Air Volume System (VAV)

Each floor has a self contained VAV system that heats and cools the interior spaces of the building.

The system used a looped ductwork network to supply air to the VAV boxes. The VAV boxes control the volume of air that is distributed to the diffusers. Each VAV box is controlled by a thermostat that is located in the area that the VAV box serves. The diffusers are connected to the VAV box by flexible duct. This was done so that the diffuser can be relocated easily for tenant fitup. Air is returned to the system through return air louvers into the return air plenum located above the ceiling.

The self contained VAV unit discharges its heat generated during its cooling cycle into the water loop network.

3.3 Water Source Heat Pump System

Heat pumps are used to heat and cool the perimeter spaces of the building. The heat pumps are connected to the water loop system were it absorbs or rejects its heat. The heat pumps are connected to the diffusers by flexible duct. This was done in case the diffusers have to be relocated for tenant fitup. Air is returned through to the heat pumps through return louvers into the return air plenum located above the ceiling. Individual heat pumps are controlled by thermostats located in the respected area that the heat pump serves.

3.4 Ventilation System

There are two different types of ventilation systems, the first supplies ventilation air to the VAV systems and the second supplies air to the heat pump systems.

The first system supplies ventilation air to the return side of the self contained VAV units. Ventilation air is supplied to the VAV systems by a single duct that is connect to a ventilation fan mounted on the roof.

The second system supplies ventilation air to the heat pumps by supplying air into the plenum space in the area where the heat pump return is located. There are two separate ventilation fan systems. One system supplies one half of the building and the other system supplies the other half. The ventilation fans are mounted on the roof and a set of ducts channel the air to the heat pumps.

3.5 Exhaust System

There are two systems that exhaust air from the building. The first system and the largest system exhausts air from the return air plenum on each floor from a point near the return for the self contained VAV units. The air is exhausted through an exhaust duct by an exhaust fan mounted on the roof. The second exhaust system exhausts air from the bathrooms. Air is exhaust from the bathrooms because the BOCA code requires that the bathrooms be exhausted. The air is exhausted through an exhaust duct by an exhaust fan located on the roof.

3.6 Water Loop System, Boiler, Fluid Cooler, Air Separator, Compression Tank and Pumps

The water loop system is used to supply heat or remove heat that is needed for the operation of the heat pumps and to remove heat that is generated by the self contained VAV units. Connected to the system is a boiler and a fluid cooler. These components maintain the fluid temperature in the water loop so that the heat pumps and VAV units can function properly. An air Separator is used in the system to remove air and other gases from the water loop. A compression tank is used to absorb the expansion of water in the system. The pumps in the system drive the fluid through the piping system.

Each floors' loop is of a reverse return design. The loop is connected to a riser that leads to the penthouse, on the roof of the building. This is where the pumps, boiler and fluid cooler are located. The location of the penthouse is over the corridor area, in order that the noise generated by the equipment will not reach the office areas.

3.7 Controls and Control Sequence

The controls in this building will be of the electronic type and will be used to regulate the equipment in the building. Controls will not be selected because it is out of the scope of this project but the control sequences will be specified.

3.7.1 Cooling Season - During the cooling season both the heat pumps and the self contained VAV system will be rejecting heat into the water loop system. In this mode the fluid cooler is in operation and locks out the boiler from operating. If the fluid flow stops or the water loop temperature becomes too great, the heat pumps and VAV units will be shut down.

During daytime operation all heat pumps, VAV units, ventilation and exhaust systems will be operating. At night the exhaust and ventilation systems will shut down to conserves energy. In addition the heat pumps will be shut down and the VAV units will remain on to circulate air through the building. In this mode the VAV boxes will open to their maximum positions and the cooling system in the VAV units will shut down. In the morning the system will return back to its daytime operating mode.

3.7.2 Heating Season - During daytime operation the VAV System will be in the cooling mode and the heat pumps will be in the heating mode or cooling mode depending on the space they serve. The heat pumps thermostat will have a dead band area to keep the units from cycling from cooling to heating mode. During daytime operation the ventilation and exhaust systems will be operating. In this mode the boiler will be maintaining the water loop temperature. When the boiler is in operation it locks out the fluid cooler from operating. If the fluid flow stops or the water loop temperature falls below its minimum operating value the heat pumps and the VAV units will be shut down.

During night time operation the exhaust and ventilation systems will be shut down. The heat pumps and VAV units will be set to hold an indoor temperature of 55°F. In this mode the VAV boxes will be fully opened.

The system will begin morning warm up operation approximately 1 hour before the building is occupied. In this mode the ventilation and exhaust system will remained closed. This mode will begin with the heat pumps and the electric resistance heat located in the self Contained VAV units to begin there operation to heat the building. In this mode the VAV boxes will remain partially opened. After morning warm up the system will begin its normal daytime operating mode.

4. Psychrometrics

The psychrometric analysis will only be preformed on the second floor of the building. This was done because the other two floors' analyses would only be a variation of the second floor's analysis.

Two analyses were preformed for the VAV system. These were maximum cooling mode and morning warm-up mode. Three analyses were preformed for the heat pump systems. These were maximum cooling, morning warm-up and maximum heating mode.

4.1 Variable Air Volume System

4.1.1 Maximum Cooling - The following cooling load information is taken from the cooling load analysis preformed by the Carrier Hourly Analysis program.

Maximum Cooling Load occurs in July at 13:00 hours

Total Cooling Load 282,282 BTU/hr.
or 23.52 Tons
(232,322 sens, 49,960 lat)
Ventilation Load (1,400 cfm)
18,142 BTU/hr sens
34,890 BTU/hr lat

Supply Fan Load (6.7 Bhp) 17,059 BTU/hr sens

Coil Characteristics	
Coil ent. air Temp (DB/WB)	80.6/65.8°F
Coil leav. air Temp(DB/WB)	55.1/54.3°F
Cooling supply air Temp	57°F
Total cooling cfm	8,400 cfm
Coil bypass factor	.050

Below shows the system diagram. Energy entering the system is positive and energy leaving the system is negative.

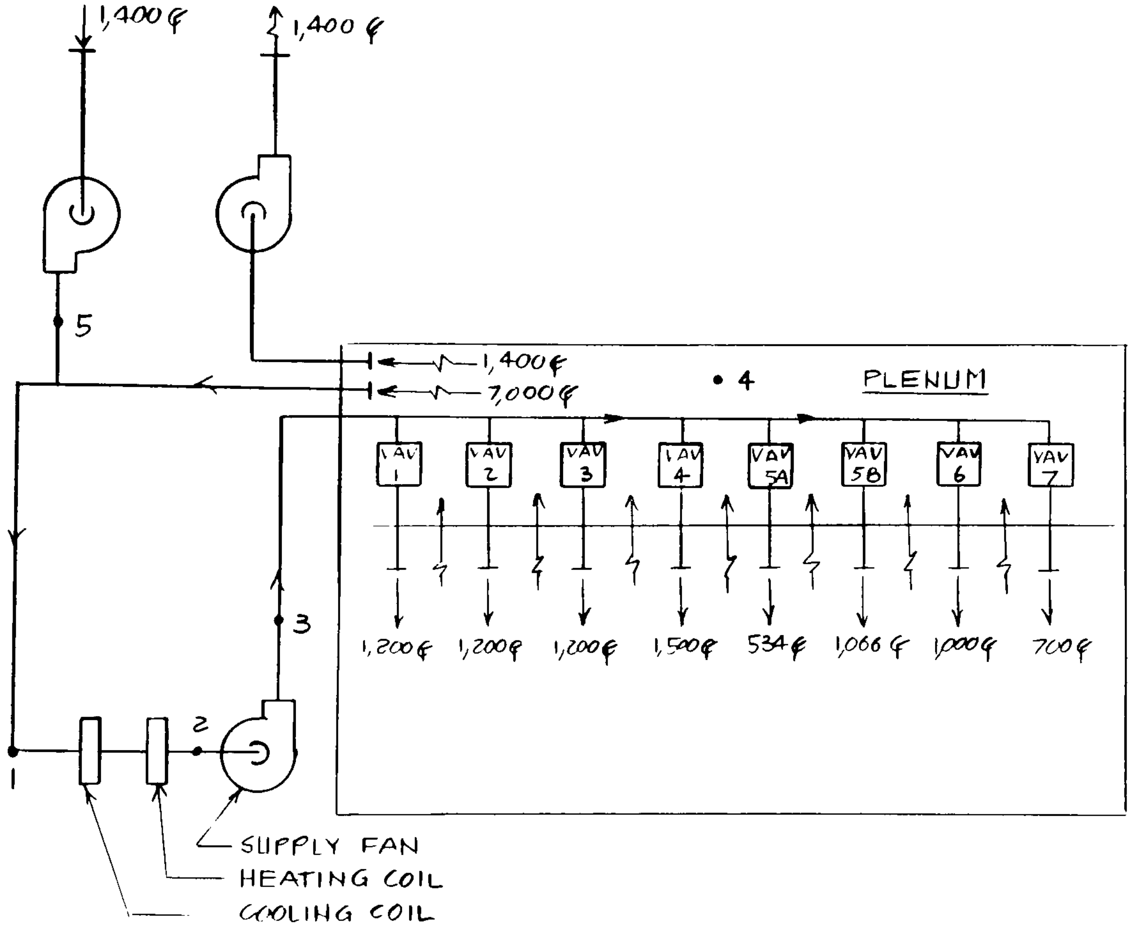


Figure 4-1, Floor 2, VAV System Diagram, Maximum Cooling

The property chart describes all the properties for each point in the system.

Table 4-1, <u>Property Chart - VAV Cooling</u>

Point	Temp (°F) DB	WB	V FT^3/lbma	h BTU/hr	Rel Hum %	W Hum Ratio
1	80.0	65.2	13.80	30.0	46	.010
2	55.0	54.0	13.14	22.6	94	.0086
3	57.5	55.0	13.20	23.05	85	.0086
4	78.0	63.0	13.73	28.6	45	.0090
5	87	71.5	14.10	36.5	50	.0138

The First Law Chart describes the processes of the system.

Table 4-2, <u>First Law Chart - VAV Cooling</u>

Points	Process	$\dot{q}$ (BTU/lbma/hr) $\dot{q}_s$	$\dot{q}_l$
1 to 2	cooling and dehumidifying	-232,322	-49,960
2 to 3	heating	17,059	0
3 to 4	heating and humidifying	197,121	15,070
4,5 to 1	adiabatic mixing of 2 gases	18,142	34,890

The following page diagrams this process on the psychrometric chart. The following pages beyond will show the calculations that were preformed to fill out the Property Chart and First Law Chart.

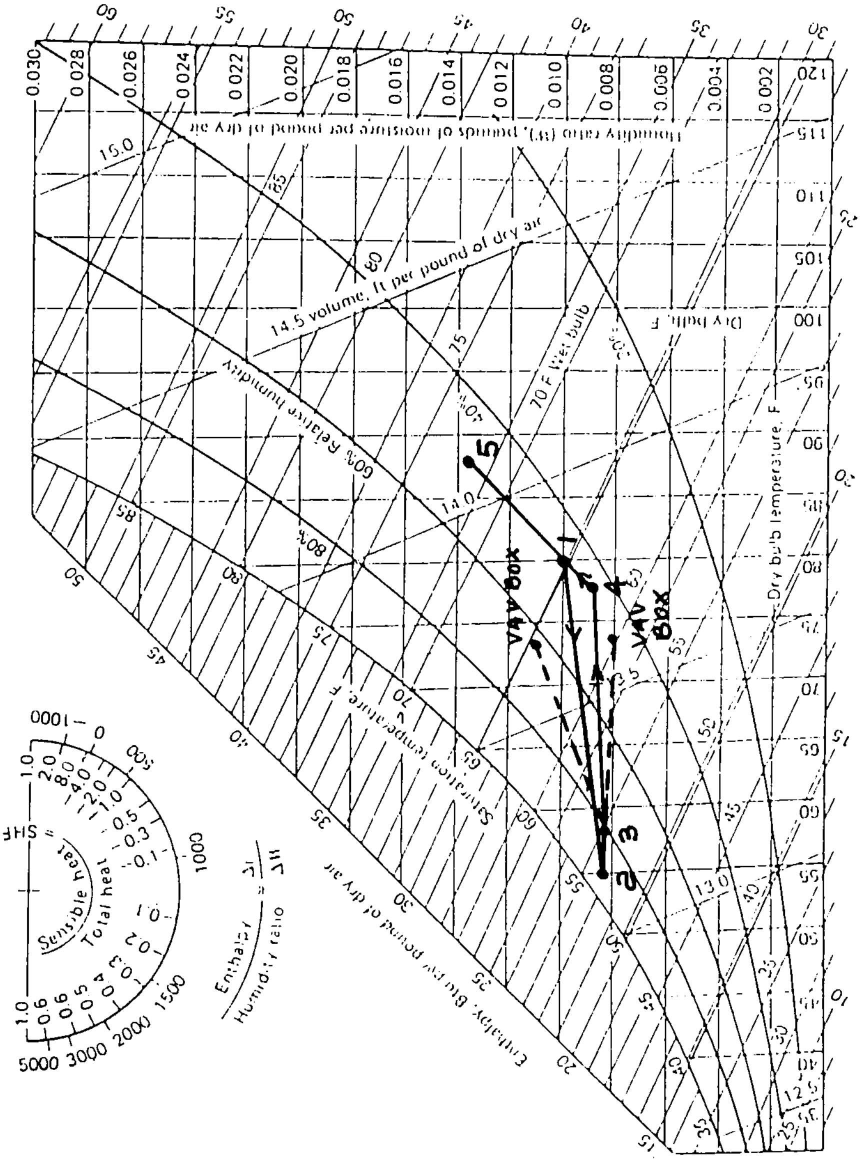


Figure 4-2, Floor 2, VAV Psychrometric Chart, Maximum Cooling

The process will begin with point 1.

Point 1 to Point 2

Cooling and dehumidifying of moist air.

$$\dot{q}_s = 232,322 \text{ BTU/hr}$$

$$\dot{q}_l = 49,960 \text{ BTU/hr}$$

$$\dot{m}_a = \frac{cfm}{v} * \frac{60 \text{ min}}{hr}$$

$$\dot{m}_a = \frac{8,400 \text{ ft}^3/\text{min}}{21. \text{ ft}^3/\text{lbma}} * \frac{60 \text{ min}}{hr} = 38,356 \text{ lbma/hr}$$

$$t_2 = 55 \text{ DB/ } 54 \text{ WB } ^\circ F$$

$$\dot{q}_s = \dot{m}_a * C_p * (t_1 - t_2)$$

$$t_1 = -\frac{\dot{q}_s}{\dot{m}_a * C_p} + t_2$$

$$t_2 = \frac{232,322 \text{ BTU/hr}}{(38,356 \text{ lbma/hr})*(.246 \text{ BTU/lbmaR})} + 55^\circ F$$

$$t_1 = 80^\circ F$$

$$\dot{q}_l = \dot{m}_a * (W_1 - W_2) * h_{fg}$$

$$W_1 = -\frac{\dot{q}_l}{\dot{m}_a * h_{fg}} + W_2$$

$$W_1 = \frac{49,960 \text{ BTU/hr}}{(38,356 \text{ lbma/hr})*(1062.14 \text{ BTU/lbmv})} + .0086$$

$$W_1 = .0098$$

Energy Balance Equation (steady state)

$$q_s + \dot{q}_l = \dot{m}_a h_1 \dot{-} m_a h_2 \doteq -282,282 \text{ BTU/hr}$$

Point 2 to Point 3

Heating of Moist Air

Energy Balance Equation (steady state)

$$\dot{q}_{fan} = \dot{m}_a h_3 - \dot{m}_a h_2 = \dot{m}_a*(h_3-h_2)$$

$$h_3 = \frac{\dot{q}_{fan}}{\dot{m}_a} + h_2$$

$$h_3 = \frac{17{,}059 \text{ BTU/hr}}{38{,}356 \text{ lbma/hr}} + 22.6 \text{ BTU/lbma}$$

$$h_3 = 23.05 \text{ BTU/lbma}$$

Mass Balance Equation

$$m_{a2} = m_{a3} = 38{,}356 \text{ lbma/hr}$$

Point 3 to Point 4

Heating and humidifying of moist air

$$\dot{q}_s = 197{,}121 \text{ BTU/hr}$$

$$\dot{q}_l = 15{,}070 \text{ BTU/hr}$$

Energy Balance Equation

$$q_s + q_l = m_a h_4 - m_a h_3 = m_a*(h_4-h_3)$$

$$h_4 = \frac{\dot{q}_s + \dot{q}_l}{\dot{m}_a} + h_3$$

$$h_4 = \frac{(197{,}121 + 15{,}070) \text{ BTU/hr}}{38{,}356 \text{ lbma/hr}} + 23.05 \text{ BTU/lbma}$$

$$h_4 = 28.6 \text{ BTU/lbma}$$

Mass Balance Equation

$$m_{a3} = m_{a4}$$

$$\dot{m}_{a3}*W_3 + \dot{m}_w = \dot{m}_{a4}*W_4$$

$$\dot{q}_1 = \dot{m}_w * h_w$$

$$\dot{m}_w = \frac{\dot{q}_1}{h_w} = \frac{15{,}070 \text{ BTU/hr}}{1050.85 \text{ BTU/lbma}} = 14.34 \text{ lbma/hr}$$

$$W_4 = W_3 + \dot{m}_w/\dot{m}_{a3} = .0086 + 14.34/38{,}356$$

$$W_4 = .0090$$

Points 4,5 to Point 1

Adiabatic mixing of two streams of moist air.

Energy Balance Equation (steady state)

$$m_{a4} * h_4 + m_{a5} * h_5 = m_{a1} * h_1$$

$$\dot{m}_{a4} = \frac{\text{cfm}}{v} * \frac{60 \text{ min}}{\text{hr}}$$

$$\dot{m}_{a4} = \frac{7{,}000 \text{ ft}^3/\text{min}}{22. \text{ ft}^3/\text{lbma}} * \frac{60 \text{ min}}{\text{hr}} = 30{,}590 \text{ lbma/hr}$$

Mass Balance Equation

$$m_{a4} + m_{a5} = m_{a1}$$

$$\dot{m}_{a5} = \dot{m}_{a1} - \dot{m}_{a4} = (38{,}356-30{,}590)\text{lbma/hr}$$

$$\dot{m}_{a5} = 7{,}766 \text{ lbma/hr}$$

$$\dot{m}_{a4} * W_4 + \dot{m}_{a5} * W_5 = \dot{m}_{a1} * W_1$$

$$W_5 = \frac{\dot{m}_{a1} * W_1 - \dot{m}_{a4} * W_4}{\dot{m}_{a5}}$$

$$W_5 = \frac{(38{,}356 \text{ lbma/hr})(.010) - (30{,}590 \text{ lbma/hr})(.009)}{7{,}766 \text{ lbma/hr}} = .0139$$

Back to the energy balance equation

$$h_5 = \frac{\dot{m}_{a1} * h_1 - \dot{m}_{a4} * h_4}{\dot{m}_{a5}}$$

$$h_5 = \frac{(38{,}356 \text{ lbma/hr})(30.2 \text{ BTU/lbma}) - (30{,}590 \text{ lbma/hr})(28.6 \text{ btu/lbma})}{7{,}766 \text{ lbma/hr}}$$

$$h_5 = 36.5 \text{ BTU/lbma}$$

<u>4.1.2 Morning Warm-up</u> - The morning warm-up cycle will raise the indoor air temperature from 55°F to 68°F. During this cycle the ventilation and exhaust systems are closed.

The question for this cycle was to determine how much supply air is required at a temperature of 102°F to raise the temperature in the room in approximately 1 hour. Below is the system diagram for this process and the next page shows this system on the psychrometric chart.

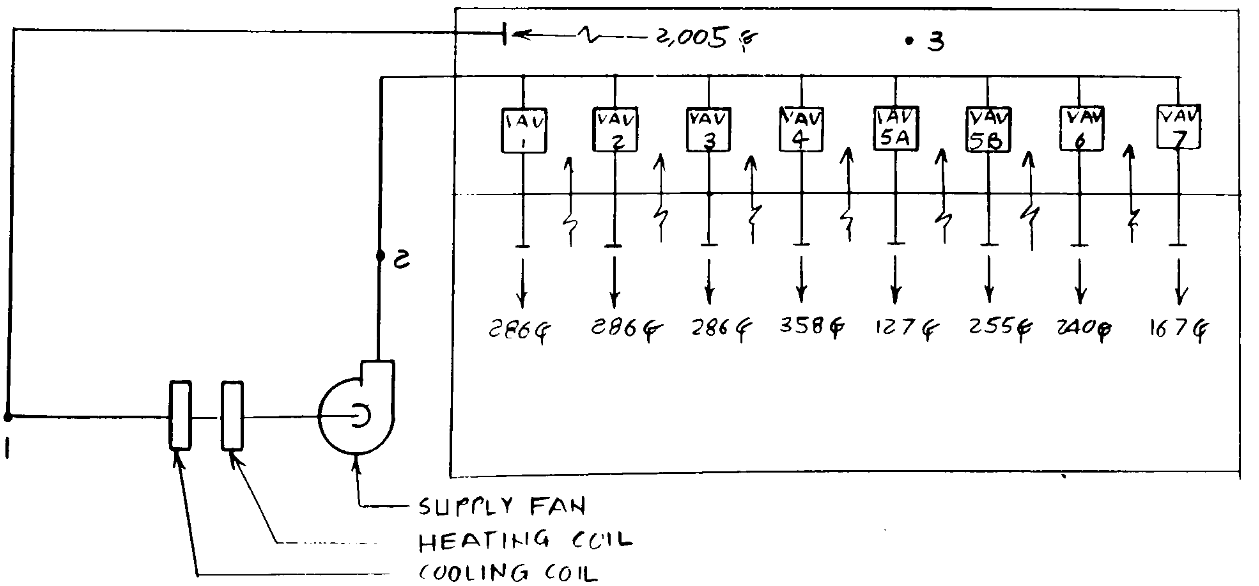


Figure 4-3, Floor 2, VAV System Diagram, Morning Warm-up

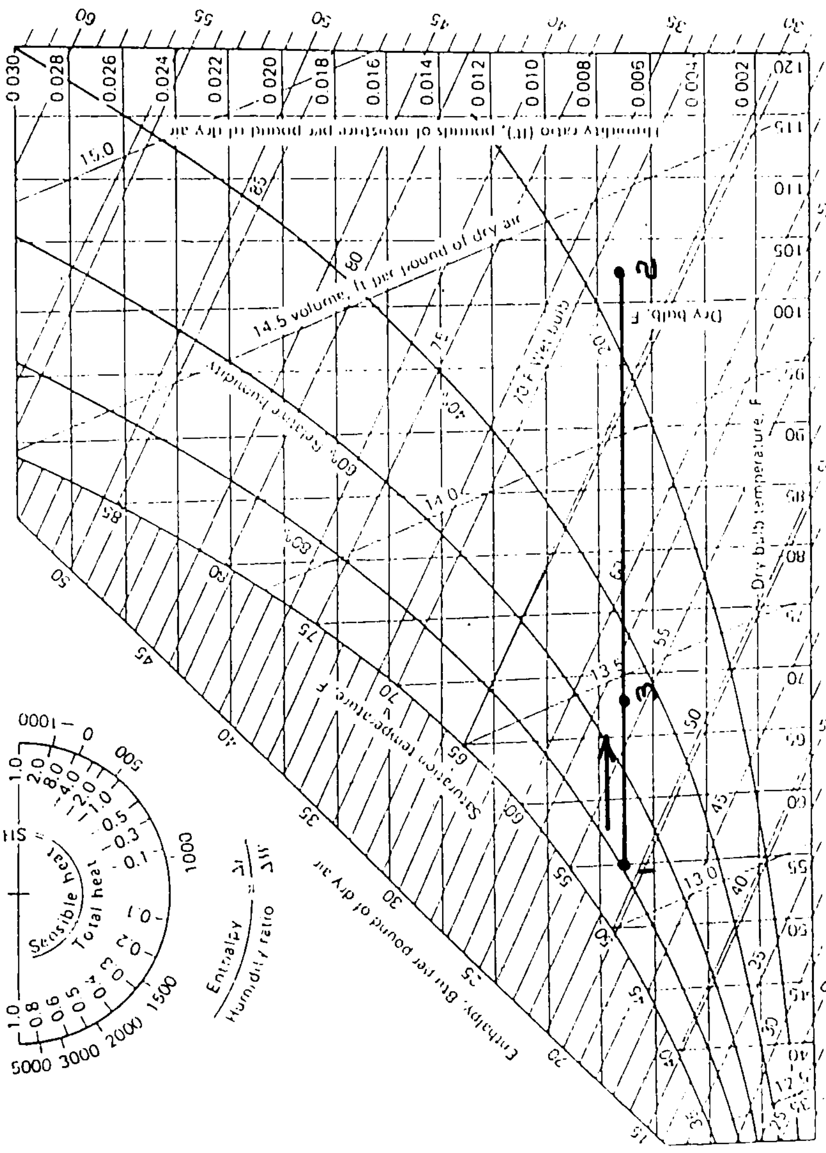


Figure 4-4, Floor 2, VAV Psychrometric Chart, Morning Warm-up

If the ventilation and exhaust systems where operational the resulting net ventilation loss would be 93,774 BTU/hr. This same load will be used to raise the temperature of the space so the heating portion of the system will not be oversized.

The volume of air that has to be raised in temperature is 117,000 ft^3. The temperature raise of this volume of air is from 55°F to 68°F or a öt of 13°F. The supply temperature of the air is 102°F.

The question was to determine the volume of air supplied at a temperature of 102°F and a öt of 47°F so a load of 94,000 BTU/hr is never exceeded on the coil.

<u>The Energy Balance Equation (steady state)</u>

$$94{,}000 \text{ BTU/hr} = \dot{m}_a * (h_1 - h_2)$$

$$\dot{m}_a = \frac{94{,}000 \text{ BTU/hr}}{(h_1 - h_2)} = \frac{94{,}000 \text{ BTU/hr}}{(32.4-21.2)\text{BTU/lbma}}$$

$$\dot{m}_a = 8{,}392.8 \text{ lbma/hr}$$

$$\text{cfm} = \frac{\dot{m}_a * v}{60} = \frac{(8{,}392.8 \text{ lbma/hr})*(15.33 \text{ ft}^3/\text{lbma})}{60 \text{ min/hr}}$$

$$\text{cfm} = 2{,}005 \text{ cfm}$$

All VAV boxes are closed to 24% of their capacity. The system operates until the temperature in the system has been raised to 68°F.

4.2 Heat Pump System

Even though the heat pumps are separate units, this is an analysis of the maximum heat pumps combined load.

4.2.1 Maximum Cooling Load - The following cooling load information is taken from the cooling load analysis preformed by the Carrier Hourly Analysis program.

Maximum Cooling Load occurs in August at 15:00 hours

Total Cooling Load 264,320 BTU/hr.
or 22.03 Tons
(229,536 sens, 34,784 lat)

Ventilation Load (847 cfm)
14,428 BTU/hr sens
19,316 BTU/hr lat

Supply Fan Load 18,476 BTU/hr sens

Coil Characteristics

Coil ent. air Temp (DB/WB)	78.4/64.4°F
Coil leav. air Temp(DB/WB)	55.1/54.5°F
Cooling supply air Temp	57°F
Total cooling cfm	9,140 cfm
Coil bypass factor	.050

Below shows the system diagram. Energy entering the system is positive and energy leaving the system is negative.

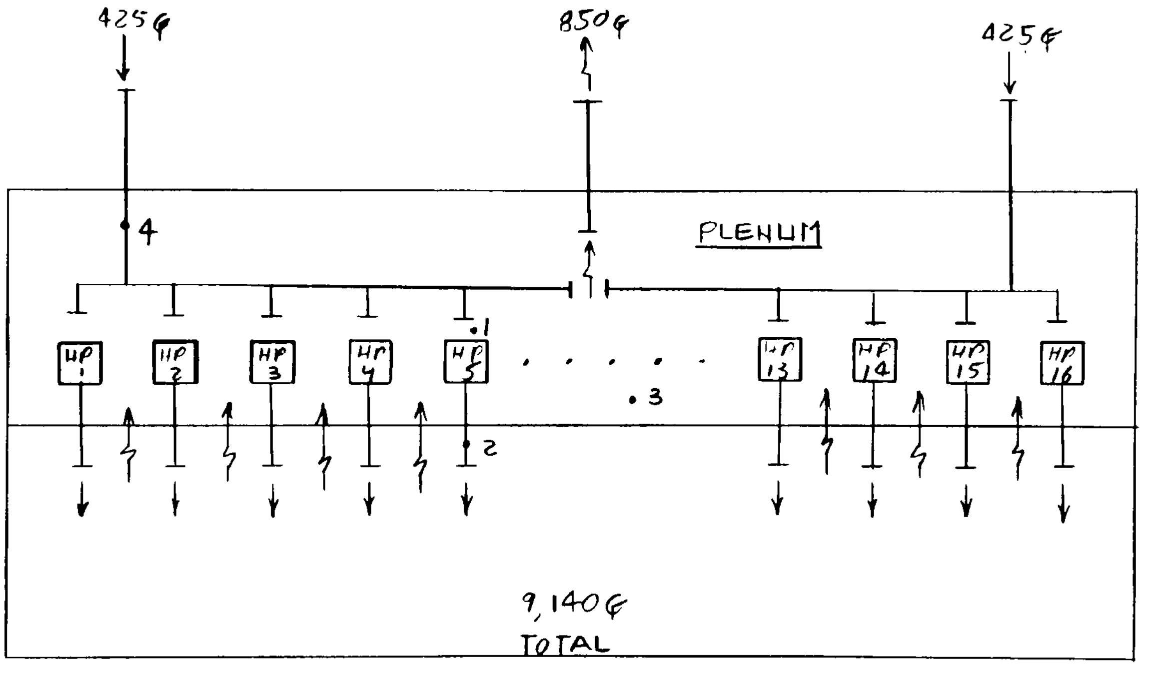


Figure 4-5, Floor 2, Heat Pump System Diagram, Maximum Cooling

The property chart describes all the properties for each point in the system.

Table 4-3, <u>Property Chart - Heat Pump Cooling</u>

Point	Temp (°F) DB	WB	V $FT^3/lbma$	h BTU/hr	Rel Hum %	W Hum Ratio
1	78.4	64.4	13.77	29.6	47	.0098
2	56.0	55.0	13.20	23.2	89	.0088
3	78.0	63.3	13.75	28.8	45	.0092
4	82.2	71.0	13.96	35.0	59	.0139

The First Law Chart describes the processes of the system.

Table 4-4, First Law Chart - Heat Pump Cooling

Points	Process	$\dot{q}$ (BTU/lbma/hr)	
		$\dot{q}_s$	$\dot{q}_l$
1 to 2	cooling and dehumidifying	-229,536	-34,784
2 to 3	heating and humidifying	215,109	15,468
3,4 to 1	adiabatic mixing of 2 gases	14,428	19,316

The following page diagrams this process on the psychrometric chart. The following pages beyond will show the calculations that were preformed to fill out the Property Chart and First Law Chart.

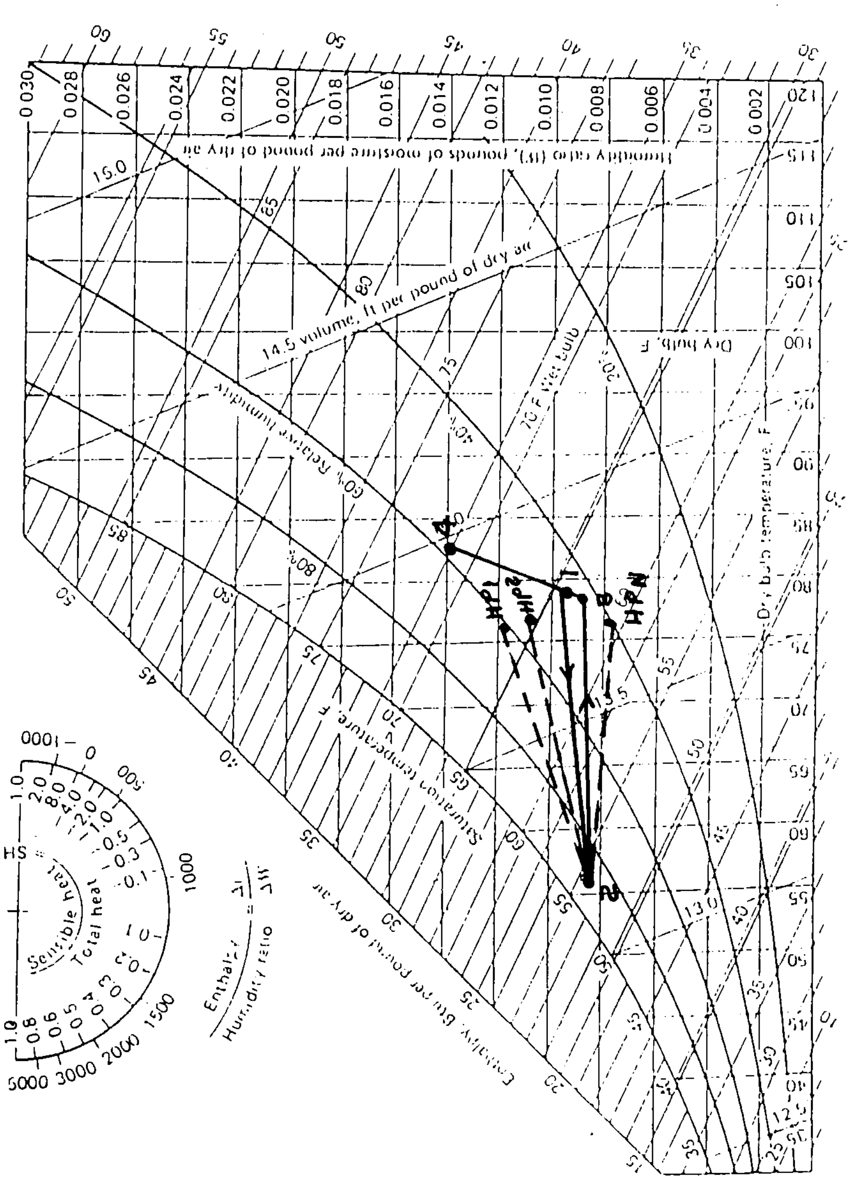


Figure 4-6, Floor 2, Heat Pump Psychrometric Chart Maximum Cooling

The process will begin with point 1.

Point 1 to Point 2

Cooling and dehumidifying of moist air.

$$\dot{q}_s = 229{,}536 \text{ BTU/hr}$$

$$\dot{q}_l = 34{,}784 \text{ BTU/hr}$$

$$\dot{m}_a = \frac{cfm}{v} * \frac{60 \text{ min}}{hr}$$

$$\dot{m}_a = \frac{9{,}140 \text{ ft}^3\text{/min}}{24. \text{ ft}^3\text{/lbma}} * \frac{60 \text{ min}}{hr} = 41{,}545 \text{ lbma/hr}$$

Energy Balance Equation (steady state)

$$\dot{q}_s + \dot{q}_l = \dot{m}_a h_1 - \dot{m}_a h_2 = -264{,}320 \text{ BTU/hr}$$

Point 2 to Point 3

Heating and humidifying of moist air

$$\dot{q}_s = 215{,}109 \text{ BTU/hr}$$

$$\dot{q}_l = 15{,}468 \text{ BTU/hr}$$

Energy Balance Equation (steady state)

$$\dot{q}_s + \dot{q}_l = \dot{m}_a h_3 - \dot{m}_a h_2 = \dot{m}_a*(h_3-h_2)$$

$$h_3 = \frac{\dot{q}_s + \dot{q}_l}{\dot{m}_a} + h_2$$

$$h_3 = \frac{(215{,}109 + 15{,}468) \text{ BTU/hr}}{41{,}545 \text{ lbma/hr}} + 23.2 \text{ BTU/lbma}$$

$$h_3 = 28.8 \text{ BTU/lbma}$$

Mass Balance Equation

$$\dot{m}_{a2} = \dot{m}_{a3}$$

$$\dot{m}_{a2}*W_2 + \dot{m}_w = \dot{m}_{a3}*W_3$$

$$\dot{q}_l = \dot{m}_w*h_w$$

$$\dot{m}_w = \frac{\dot{q}_1}{h_w} = \frac{15{,}468 \text{ BTU/hr}}{1050.85 \text{ BTU/lbma}} = 14.72 \text{ lbma/hr}$$

$$W_3 = W_2 + \dot{m}_w/\dot{m}_{a2} = .0088 + 14.72/41{,}545$$

$$W_3 = .0090$$

<u>Points 3, 4 to Point 1</u>

Adiabatic mixing of two streams of moist air.

<u>Energy Balance Equation (steady state)</u>

$$\dot{m}_{a3}*h_3 + \dot{m}_{a4}*h_4 = \dot{m}_{a1}*h_1$$

$$\dot{m}_{a3} = \frac{\text{cfm}}{v} * \frac{60 \text{ min}}{\text{hr}}$$

$$\dot{m}_{a3} = \frac{8{,}293 \text{ ft}^3\text{/min}}{25. \text{ ft}^3\text{/lbma}} * \frac{60 \text{ min}}{\text{hr}} = 36{,}188 \text{ lbma/hr}$$

<u>Mass Balance Equation</u>

$$\dot{m}_{a3} + \dot{m}_{a4} = \dot{m}_{a1}$$

$$\dot{m}_{a4} = \dot{m}_{a1} - \dot{m}_{a3} = (41{,}545\text{-}36{,}188)\text{lbma/hr}$$

$$\dot{m}_{a4} = 5{,}357 \text{ lbma/hr}$$

$$\dot{m}_{a3}*W_3 + \dot{m}_{a4}*W_4 = \dot{m}_{a1}*W_1$$

$$W_4 = \frac{\dot{m}_{a1}*W_1 - \dot{m}_{a3}*W_3}{\dot{m}_{a4}}$$

$$W_4 = \frac{(41{,}545 \text{ lbma/hr})(.0098) - (36{,}188 \text{ lbma/hr})(.0092)}{5{,}357 \text{ lbma/hr}} = .0139$$

Back to the energy balance equation

$$h_4 = \frac{\dot{m}_{a1} * h_1 - \dot{m}_{a3} * h_3}{\dot{m}_{a4}}$$

$$h_4 = \frac{(41{,}545 \text{ lbma/hr})(29.6 \text{ BTU/lbma}) - (36{,}188 \text{ lbma/hr})(28.8 \text{ btu/lbma})}{5{,}357 \text{ lbma/hr}}$$

$$h_4 = 35.0 \text{ BTU/lbma}$$

4.2.2 Morning Warm-up and Maximum Heating Load - First to be analyzed is the morning warm-up mode were the heat pumps warm the air from 55°F to 68°F. The second mode to be analyzed is the maximum winter operating condition.

Morning Warm-up - This cycle raises the indoor temperature from 55°F to 68°F with the ventilation and exhaust systems closed. Below is the system diagram schematic.

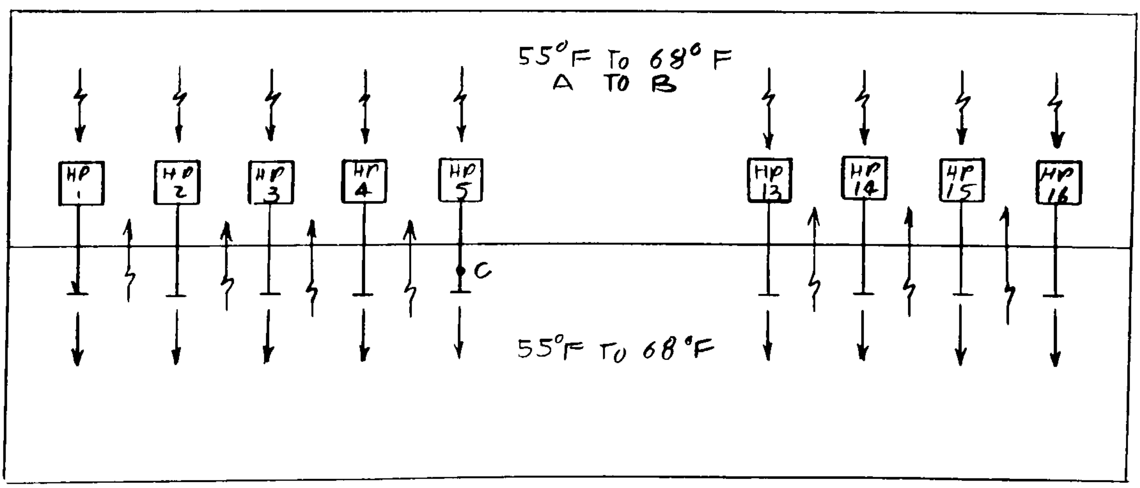


Figure 4-7, Floor 2, Heat Pump System Diagram, Morning Warm-up

Since the ventilation and exhaust systems are closed, the heat pumps will use their extra capacity to raise the temperature of the perimeter spaces.

The volume of air that has to raised in temperature is 55,872 ft^3.

The load seen on all the combined coils must not exceed 59,359 BTU/hr which is equal to the ventilation load if the ventilation system was in operation. This was done so the heat pumps would have enough capacity.

The total load on the heat pumps was 156,596 BTU/hr and at a flow rate of 2,649 cfm. The supply air temperature was 102°F.

The flow rate to raise the temperature:

$$cfm = \frac{59{,}395 \text{ BTU/hr}}{156{,}596 \text{ BTU/hr}} * 2{,}649 \text{ cfm} = 1{,}005 \text{ cfm}$$

The total number of BTU's required:

55,872 ft^3 from 55°F to 68°F

1005 cfm or 4,467 lbma/hr

55,872 ft_3 or 4,249 lbma

h_a = 21.2 BTU/lbma

h_b = 24.4 BTU/lbma

BTU's required =(4,249 lbma)*(24.4-21.2)BTU/lbma

BTU's required = 13, 597 btu

The heat pumps combined put out 59,395 BTU/hr.

The time needed for morning warm-up:

$$\text{time} = \frac{13{,}597 \text{ BTU}}{59{,}395 \text{ BTU/hr}} = .223 \text{ hr or } 14 \text{ mins.}$$

4.2.3 This process is shown on the psychrometric diagram located in the Maximum Heating section.

Maximum Heating - Even though the heat pumps are separate this is an analysis of the maximum heat pumps combined heating load.

From the Carrier Hourly Analysis program the following are values taken from the heating load analysis.

Maximum heating load occurs at an outside temperature of 3°F.

Total load	156,596 BTU/hr
Ventilation Load	59,395 BTU/hr
Supply cfm	2,649 cfm
Ventilation cfm	847 cfm
Supply Temp.	102°F
Room Temp.	68°F

Below is a system diagram of the system.

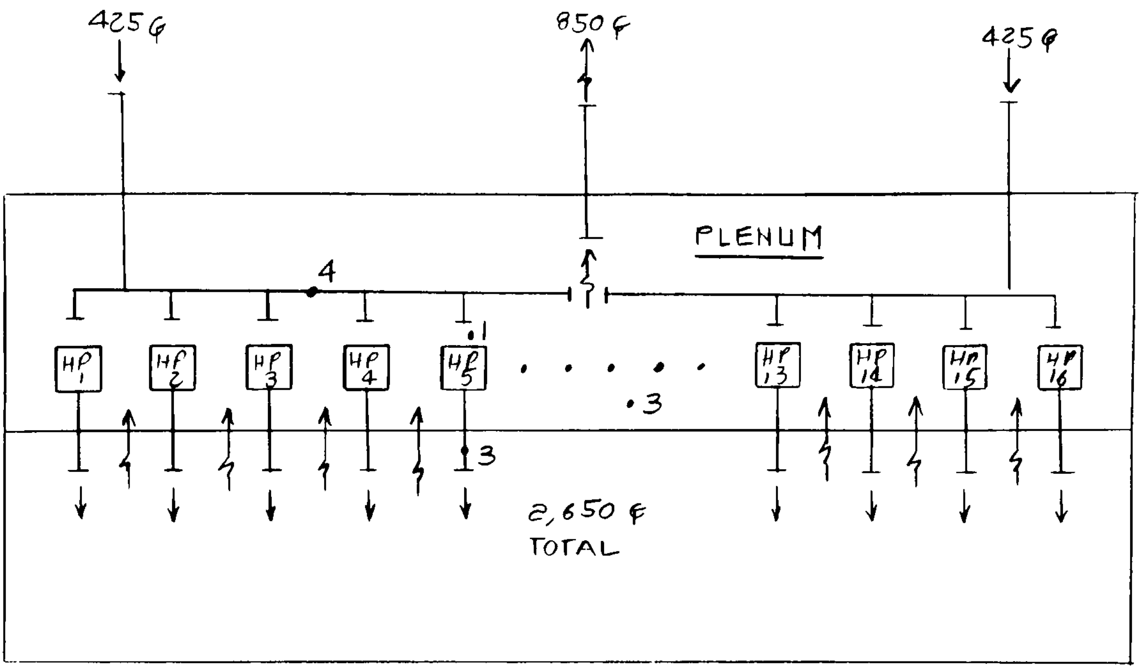


Figure 4-8, Floor 2, Heat Pump System Diagram, Maximum Heating

The property chart describes all the properties for each point in the system.

Table 4-5, <u>Property Chart - Heat Pump Heating</u>

Point	Temp (°F) DB	WB	V FT^3/lbma	h BTU/hr	Rel Hum %	W Hum Ratio
1	46.0	42.0	12.83	16.2	70	.0046
2	102.0	65.0	14.25	28.8	11	.0046
3A	68.0	57.0	13.46	24.4	50	.0074
3B	68.0	52.0	13.39	20.9	31	.0046
4	3.0	-	11.68	1.0	-	.0009

The First Law Chart describes the processes of the system.

Table 4-6, <u>First Law Chart - Heat Pump Heating</u>

Points	Process	$\dot{q}$ (BTU/lbma/hr) $\dot{q}_s$	$\dot{q}_l$
1 to 2	heating	156,596	-
2 to 3	cooling	-97,201	-
3,4 to 1	adiabatic mixing of 2 gases	-59,395	-

The following page diagrams this process on the psychrometric chart. The following pages beyond will show the calculations that were preformed to fill out the Property Chart and First Law Chart.

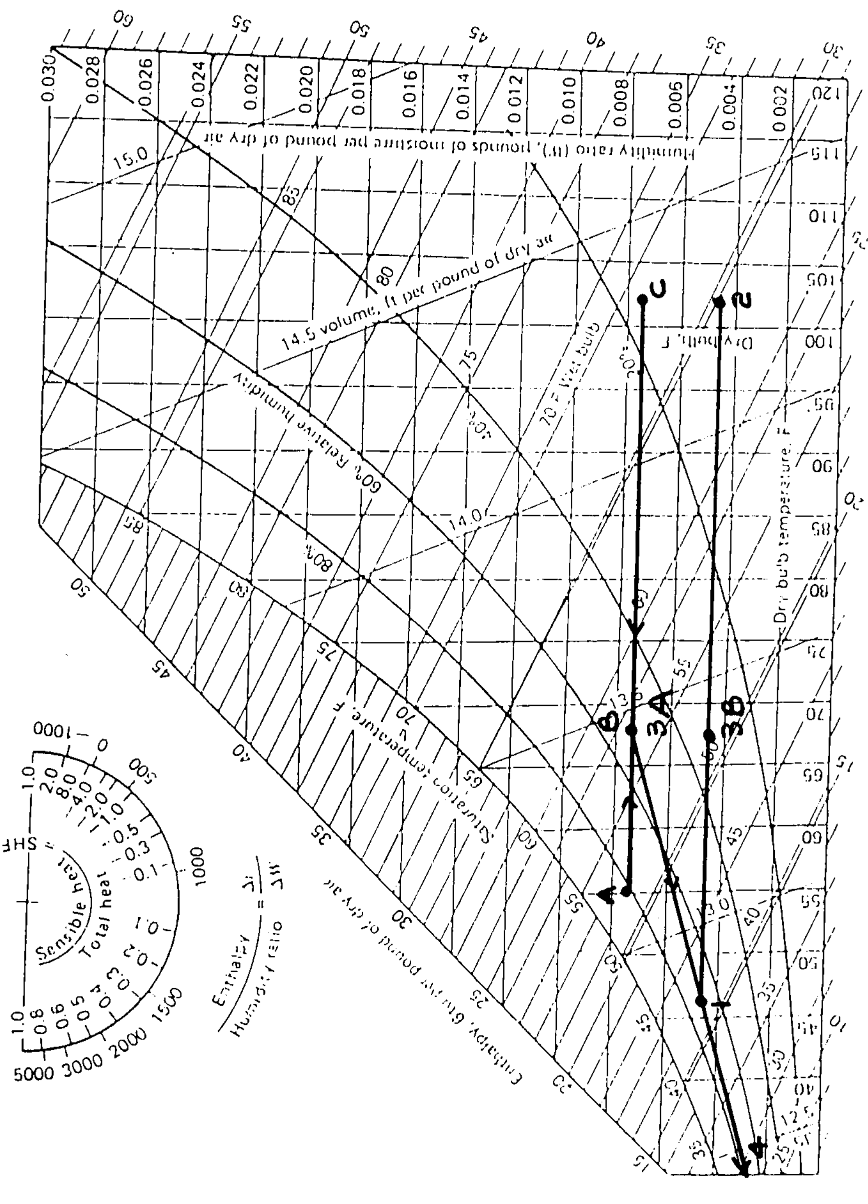


Figure 4-9, Floor 2, Heat Pump Psychrometric Chart, Maximum Heating

The process begins with points 3A and 4

Points 3A, 4 to Point 1

Adiabatic mixing of two streams of moist air.

Energy Balance Equation (steady state)

$$\dot{m}_{a3A}*h_{3A} + \dot{m}_{a4}*h_4 = \dot{m}_{a1}*h_1$$

Mass Balance Equation

$$\dot{m}_{a3A} + \dot{m}_{a4} = \dot{m}_{a1}$$

$$\dot{m}_{a3A} = \frac{1{,}800 \text{ cfm}}{26.\ ft^3/lbma} * \frac{60 \text{ min}}{hr} = 8{,}024 \text{ lbma/hr}$$

$$\dot{m}_{a4} = \frac{850 \text{ cfm}}{27.\ ft^3/lbma} * \frac{60 \text{ min}}{hr} = 4{,}366 \text{ lbma/hr}$$

$$\dot{m}_{a1} = (8{,}024 + 4{,}366)\text{lbma/hr} = 12{,}390 \text{ lbma/hr}$$

Back to Energy Balance Equation

$$h_1 = \frac{\dot{m}_{a3A}*h_{3A} + \dot{m}_{a4}*h_4}{\dot{m}_{a1}}$$

$$h_1 = \frac{(8{,}024 \text{ lbma/hr})*(24.4 \text{ BTU/lbma}) + (4{,}366 \text{ lbma/hr})*(1.0 \text{ BTU/lbma})}{12{,}390 \text{ lbma/hr}}$$

$$h_1 = 16.2 \text{ BTU/lbma}$$

$$v_1 = \frac{2650 \ ft^3/min}{12{,}390 \text{ lbma/hr}} * \frac{60 \text{ min}}{hr} = 12.83 \ ft^3/lbma$$

Point 1 to Point 3

Heating of Moist Air

Energy Balance Equation (steady state)

$$\dot{q} = m_a\dot{h_2} - m_a\dot{h_1} = m_a*(h_2-h_1)$$

$$h_2 = \frac{\dot{q}}{\dot{m}_a} + h_1$$

$$h_2 = \frac{156{,}596 \text{ BTU/hr}}{12{,}390 \text{ lbma/hr}} + 16.2 \text{ BTU/lbma}$$

$$h_2 = 28.8 \text{ BTU/lbma}$$

Mass Balance Equation

$$\dot{m}_{a1} = \dot{m}_{a2} = 12{,}390 \text{ lbma/hr}$$

Point 2 to Point 3B

cooling of moist air

Energy Balance Equation (steady state)

$$\dot{q} = \dot{m}_{a3B} * h_{3b} - \dot{m}_{a2} * h_2$$

$$h_{3B} = \frac{\dot{q}}{\dot{m}_a} + h_2$$

$$h_{3b} = \frac{-97{,}201 \text{ BTU/hr}}{12{,}390 \text{ lbma/hr}} + 28.8 \text{ BTU/hr}$$

$$h_{3b} = 20.9 \text{ BTU/hr}$$

Points 3A and 3B do not coincide, this is because as the system operates it loses moisture in the air through the exhaust system. The ventilation system does not replace this moisture because the ventilation air's moisture content is vary low. Points 3B, 1 and 2 will continue to move down on the psychrometric chart until the heating process is represented on a horizontal line. On this line all the points in the system will have the same humidity ratio as point 4 and the system will then be in equilibrium.

5. Components

In this chapter the sizing and selection of the equipment for the following systems will be discussed. The systems are: Variable Air Volume, Heat Pump, Ventilation, Exhaust and Piping systems.

5.1 Variable Air Volume System

In this section the sizing of the VAV boxes, Diffusers, Duct system and Self Contained VAV units will be discussed.

5.1.1 Variable Air Volume Boxes - In selecting the VAV boxes three considerations had to be addressed. These were cfm, configuration and noise. To keep noise to a minimum the air velocities in the boxes had to be kept to a reasonable level. The configurations and cfm's depend on the location the box serves. Below is a list of boxes selected and there corresponding names that refer to the drawings.

Table 5-1, VAV Boxes

Name	cfm
VB-1	2,000
VB-2	1,200
VB-3	800
VB-4	800
VB-5	1,200
VB-6	1,500

5.1.2 Diffusers - The diffusers that were selected were of the 2 way linear slot diffuser type. The same three considerations have to be met by the diffusers as was by the VAV boxes. These are cfm, configuration and noise. The cfm and configurations depend on the location of the diffusers. Below is a list of diffusers selected and their corresponding names that refer to the drawings.

Table 5-2, Diffusers

Name	# of Slots	length(ft)
A	4	2.5
B	4	4
C	4	5

5.1.3 Duct System - The duct system was design to provide the best possible situation where all the static pressures at the inlets of each VAV box were the same. This was done by designing a looped duct system which is shown on the drawings. The duct system was sized to provide a .1 inch static pressure drop per 100 ft. of duct. 45 degree branch tee's and round corners were used to limit friction in the system. Flexible duct was used between the VAV boxes and the diffusers so the location of the diffusers can be varied.

The method used to size the duct system was the equal friction method. Hand tabulation was used to determine the maximum static pressure in the system. Flow coefficients were taken from the ASHRAE Fundamentals book. The static pressure drop for the second floor's duct system was found to be 1.8 inches of water.

5.1.4 Self Contained Variable Air Volume Unit - The self contained VAV units must provide enough cooling and heating at the required cfm and must overcome the static pressure in the duct system.

The unit sized for the first floor must handle a cooling load of 334,906 BTU/hr or 27.91 tons. The water flow rate was determined to be 95 gpm. The electric coil sized for the first floor was 43,842 watts.

The unit sized for the second floor must handle a cooling load of 282,282 BTU/hr or 23.52 tons. The water flow rate was determined to be 90 gpm. The electric coil sized for the second floor was 27,485 watts.

5.1.5 The unit sized for the third floor must handle a cooling load of 310,259 BTU/hr or 25.85 tons. The water flow rate was determined to be 92 gpm. The electric coil sized for the third floor was 45,898 watts.

5.2 Heat Pump System

In this section the sizing of the heat pumps, diffusers and duct systems will be discussed.

5.2.1 Water Source Heat Pumps - Heat pumps were selected in order they provide enough cooling and heating for the system. The maximum heating and cooling loads were calculated using the Carrier Hourly Analysis Program. From sizing the heat pumps their corresponding heat of rejection, heat of absorption and water were noted so the piping system could be sized correctly.

The sizes and corresponding names used on the drawings are as follows:

Table 5-3, **Heat Pumps**

Name	Cooling load (BTUH)	Heating Load (BTUH)
HP1	12,000	15,100
HP2	15,000	17,000
HP3	19,000	24,000
HP4	23,000	27,800

5.2.2 Diffusers - Diffusers were selected in the same manner as the VAV systems' diffusers.

5.2.3 Duct System - Ducts were sized according to the equal friction method. In this method the ducts were sized at .1" static pressure drop per 100'. The final run of duct was flexible duct. This was done to ease the installation of the duct system and diffusers. Balancing dampers were used to balance the flow in the duct systems.

5.3 Ventilation System

The ventilation system was broken down into three smaller systems. Two of these systems supply ventilation air to the heat pump units and one system supplies ventilation air to the VAV units. A diagram of these systems and their respected air flows can be found on the following page.

5.3.1 Duct System - The duct system was sized according to the equal friction method. In this method ducts were sized at a .1" static pressure drop per 100' of duct. The static pressures were determined for the floor branches of the heat pump ventilation ducts. The other ducts' static pressures were assumed values. These values were assumed because calculating these values would be redundant work. The assumed values were .25" static pressures for all the ventilation systems.

5.3.2 Ventilation Fans - The ventilation fans have the following characteristics:

Table 5-4, Ventilation Fans

FAN	STATIC	CFM
VF-1	.25"	4,200
VF-2	.25"	1,800
VF-3	.25"	1,800

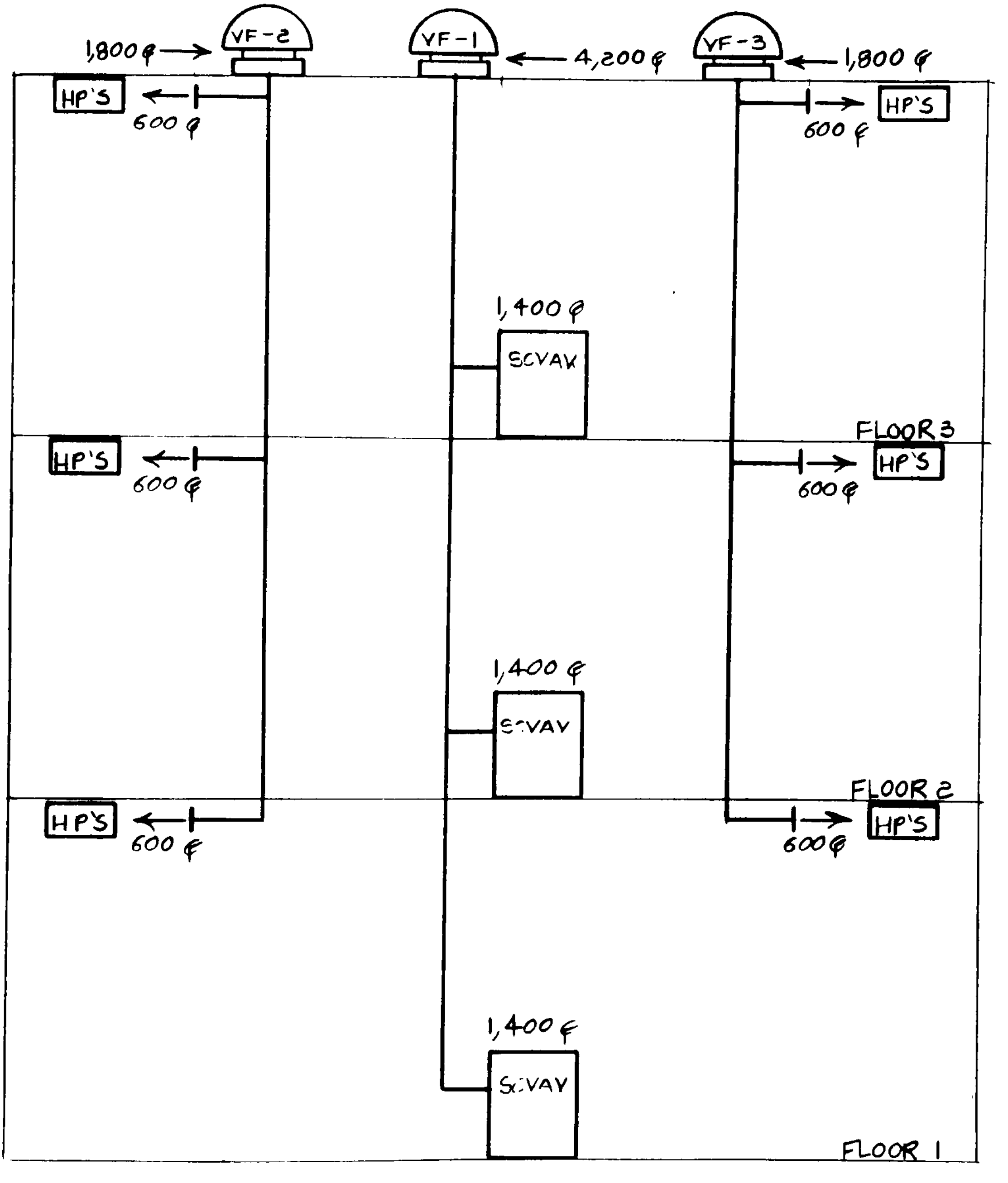


Figure 5-1, Ventilation Flow Diagram

5.4 Exhaust System

The exhaust system was broken down into two subsystems. The first subsystem exhausted air from the floors' plenum spaces the second exhausted air from the bathrooms. A diagram of these systems and their respected air flows can be seen on the following page.

5.4.1 Duct System - The duct system was sized according to the equal friction method. In this method ducts were sized at a .1" static pressure drop per 100' of duct. The ducts' static pressures were assumed values. These values were assumed because calculating these values would be redundant work. The assume values were .25" static pressures for all the exhaust systems.

5.4.2 Exhaust Fans - The exhaust fans have the following characteristics:

Table 5-5, Exhaust Fans

FAN	STATIC	CFM	HP	RPM
EF-1	.25"	5,700	1-1/2	750
EF-2	.25"	2,100	1/4	425

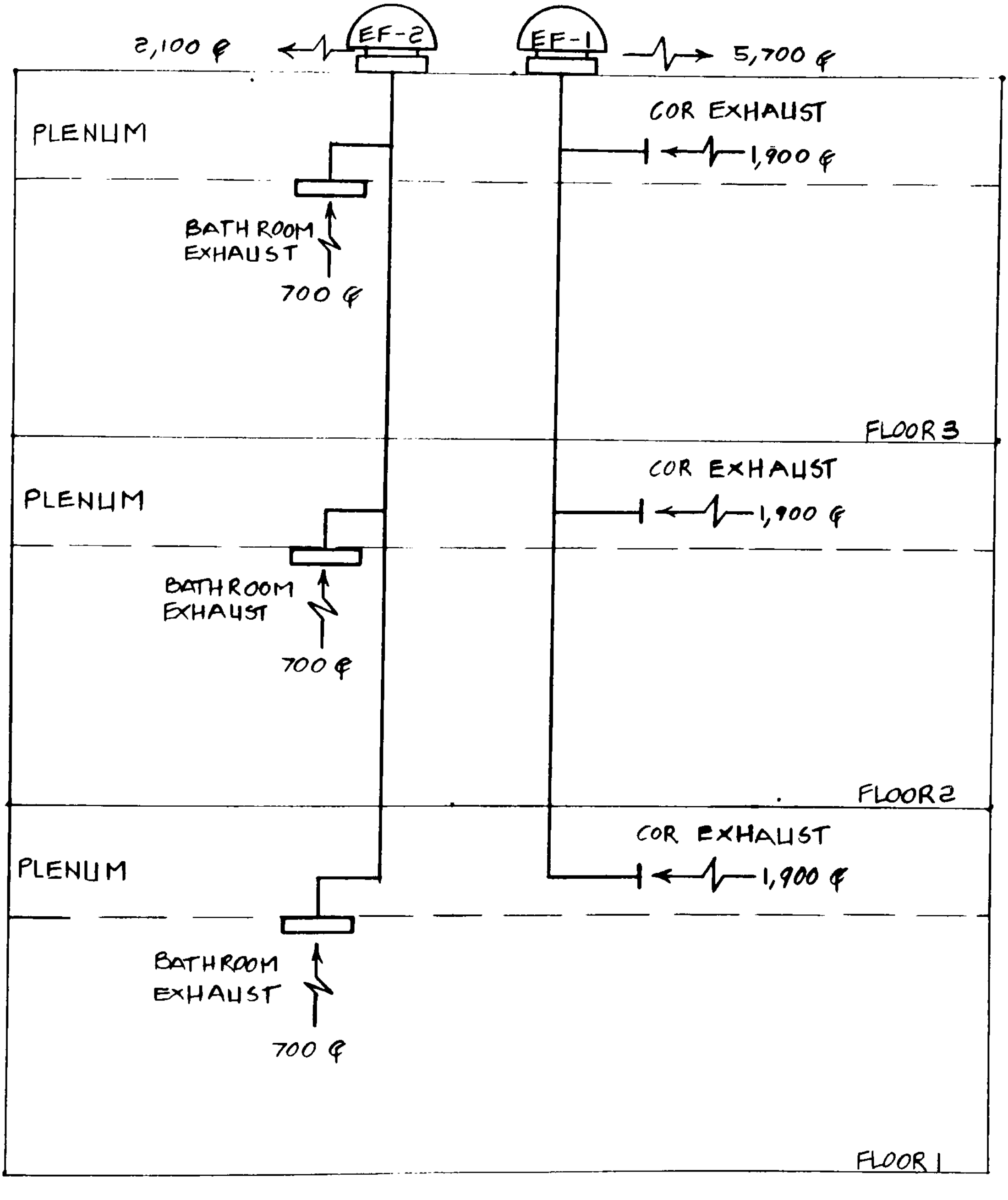


Figure 5-2, Exhaust Flow Diagram

5.5 Piping System

The piping system covers the floor loop piping, floor to floor piping, boiler, fluid cooler, air Separator, compression tank, and the pumps. The following page shows the schematic of the system. The fluid cooler is mounted on the roof and the boiler, air Separator, compression tank and pumps are located in the penthouse located on the roof. The system uses a glycol/water mixture for the heat transfer medium. This mixture is 40% glycol to reduce the freezing point of the fluid to a temperature of -14°F. This was done to protect the Fluid Cooler piping.

5.5.1 Floor Loop Piping System - Each Floors' water loop system was a reverse return design. This was done because this type of piping arrangement provides the most evenly distribution of water in the system. This is important because heat pumps can be damaged if they do not receive the proper amount of water.

The loop first supplied the self contained VAV unit and then supplied the heat pumps following a clockwise pattern around the building.

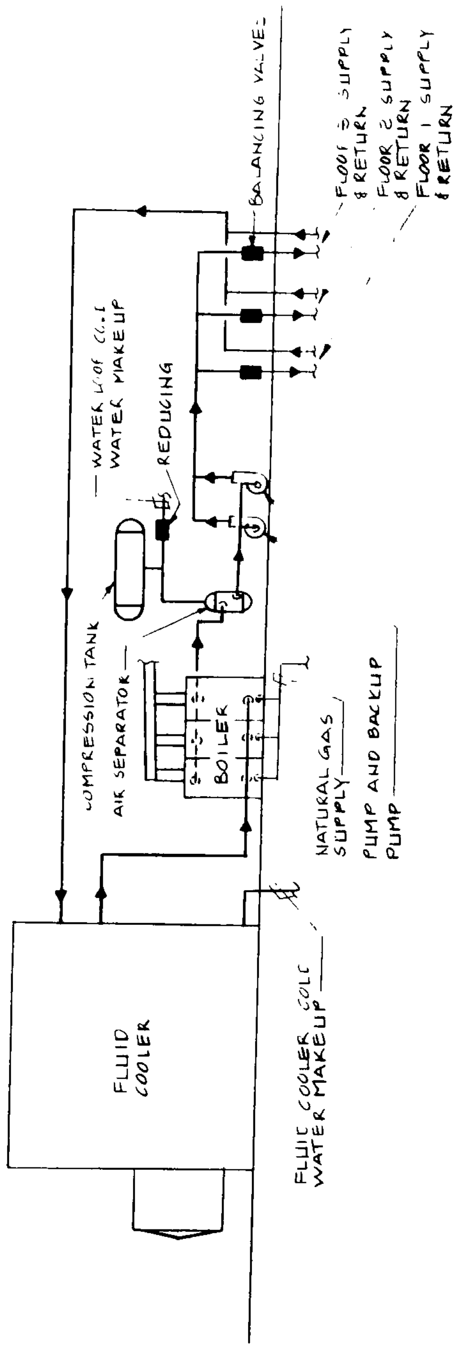


Figure 5-3, Piping Diagram

The pipe sizing and pressure calculations were calculated using the Carrier Water Piping design Program. This program sizes pipes by maintaining certain water velocities. "The pressure loss calculations are preformed using algorithms developed from charts for friction loss in pipes. These charts are from the Carrier Design Manual and are based on the density and viscosity of water at 60°F. "Correction factors are built into the program to allow the user to specify fluid temperatures other than 60°F and/or fluid type of solution ethylene glycol in water."1

The input and output of the second floor's water loop piping system can be found in the appendix. The output for the second floor piping system was a total system friction head of 31.76 ft. H_2O and a total system fluid volume of 376.17 gal.

The second floor's loop piping system was the only system sized using the computer. The other two floors' systems were not sized because this would be redundant work preformed.

1 Carrier E20-II Water & Refrigeratiot Piping Design Program, Carrier Corporation, 1986, pg. 4-1

5.5.2 Floor to Floor Piping System - A reverse return system was used to supply water to each floors' water loop system. Each floor had its own set of pipe risers that ran from the penthouse. The gpm's for each riser are as follows:

FLOOR 1	140.6 GPM
FLOOR 2	134.0 GPM
FLOOR 3	132.4 GPM

This piping system was not sized using the computer program because it would be redundant work.

5.5.3 Boiler - To determine the size of boiler needed net heat extracted by the system was first calculated. Then the net heat lost by the system was calculated. The total heat needed to be supplied by the boiler was 569,529 btu/hr. The temperature rise thorough the boiler was specified to be 20°F (65°F to 85°F). Flow rate through the boiler was 407 gpm.

The type of boiler chosen was a modular cast Iron gas-fired boiler. This type of boiler was chosen because it contains separate heating modules which improves the efficiency of the boiler. "This type of boiler improves efficiency over single burner larger boilers because actual heating loads are matched by firing more or fewer heating modules at long operating periods at full rated input and therefor at maximum efficiencies."2

5.5.4 Fluid Cooler - A closed circuit fluid cooler was chosen because it has the advantage of requiring only one set of pumps compared to a cooling tower and a heat exchanger system. The major problem with a fluid cooler was the possibility of freezing in the winter. To prevent this problem a glycol/water mixture was used. A mixture of 40% glycol to water was used to provide protection down to -14°F.

The cooler was seized to meet the following:

Heat Rejected by System	2,713,411 BTU/hr (from tabulated values)
Flow Rate	407 gpm
Entering Air Temp.	91 DB/74 WB °F
Leaving Water Temp.	85°F
Use 40% Glycol	

5.5.5 Air Separator - A centrifugal air seperator was selected. The Separator works on the action of centrifugal force rather than low velocity separation. The size of the unit was based on the fluid flow rate of 407 gpm.

2 Hydrotherm Catalog MRB9-1187 MultiTemp MR-B Series Engineering Manual, Hydrotherm Corp.

5.5.6 Compression Tank - A compression tank was needed in the system to confine the expansion of the fluid in the system when that fluid's temperature varied. The BOCA Mechanical code states the equations to be used to size the compression tank. Below is the Equation they specify:

$$V_T = \frac{(0.00041T - .0466)V_s}{(P_a/P_f) - (P_a/P_o)}$$

Where: V_T = Minimum tank volume (gal)
V_s = Volume of system (gal)
T = Average temperature of system (°F)
P_a = Atmospheric pressure (lb/in²)
P_f = Fill pressure (lb/in²)
P_o = Maximum operating pressure (lb/in²)

The Values for the system being sized were:

V_s = 1,700 gal
T = 80°F
P_a = 14.7 lb/in²
P_f = 50.0 lb/in²
P_o = 30.8 lb/in²

The equation yields the volume V_T = 76 gal.

The tank selected had a volume of 80 gallons.

5.5.7 Pumps - The fluid flow for the system was 407 gpm. Since only one section of the system had its static pressure calculated some assumptions of the rest of the systems were made. The assumptions that were made for the static pressures were as follows: The floor loop system had 34 feet of head, the floor to floor piping had 10 feet of head, the air Separator had 6 feet of head, the boiler had 4 feet of head, the fluid cooler had 13 feet of head and other piping in the system had 4 feet of head. The total head pressure was then assumed to be 71 feet of water.

Two pumps were chosen, one for regular use and the other for emergency backup. They were sized to deliver 407 gpm at a pressure of 71 feet of water.

6. Thermodynamics

Introduction

This chapter reviews the field of study of thermodynamics. The first section reviews the basic concepts of thermodynamics. The second section reviews the First Law of Thermodynamics. The third section reviews the Second Law of thermodynamics and the Vapor Compression Refrigeration Cycle.

6.1 Basic Concepts and Definitions

6.1.1 Thermodynamic System - a region in space or a quantity of matter bounded by a closed surface. This surface, also called the system boundaries, can be real or imaginary and either fixed or movable. There are two types of system, closed and open.

6.1.1.1 Closed System - is a system where no mass crosses the system boundary.

6.1.1.2 Open System - is a system where mass can enter or exit the system through the system boundary.

6.1.2 Surroundings - is anything outside the system that can be affected by the system.

6.1.3 Thermodynamic Properties

6.1.3.1 Mass (m) - the measure of the amount of matter that one has.

6.1.3.2 Volume (V) - the measure of the quantity of space.

6.1.3.3 Area (A) - the measure of the quantity of surface.

6.1.3.4 Temperature (T) - the measure of the degree of hotness or the degree of coolness.

6.1.3.5 Velocity (v) - the measure of the degree of speed

6.1.3.6 Force (F) -

6.1.3.7 Energy (E) - the capacity to do work. Energy has many forms: a) thermal, b) electrical, c) chemical, d) potential, e) kinetic, and f) nuclear

6.1.3.8 Pressure (p) - force per unit area

$p = {}^{F}/_{A}$

6.1.3.9 Internal Energy (U) - the energy posed by a system caused by the motion of the molecules and/or intermolecular forces.

6.1.3.10 Work (W) - the mechanism that transfers energy across the system boundary with differing pressures (or force of any kind)

6.1.3.11 Enthalpy (H) - is the sum of internal energy and the product of its pressure and volume.

$$H = U + pV$$

6.1.3.12 Entropy (S) - the measure of molecular disorder of a given system.

$$S_2 - S_1 = \int_1^2 \frac{dQ}{T}$$

REVERSIBLE

6.1.3.13 Heat (Q) - the mechanism that transfers energy across the boundaries of systems with differing temperatures. This transfer is always in the direction of lower temperature.

6.1.3.14 Specific heat at constant pressure (Cp) - is the rate of change of specific enthalpy of a substance with respect to the change in the temperature of the substance while maintaining a constant pressure:

$$Cp = (dh/dT)p$$

6.1.3.15 Specific heat at constant volume (Cv) - is the rate of change of specific internal energy of the substance with respect to a change in the temperature of the substance while maintaining a constant pressure.

$$Cv = (du/dT)v$$

6.1.3.16 Specific Enthalpy (per unit mass) (h)

6.1.3.17 Specific Entropy (per unit mass) (s)

6.1.3.18 Specific Volume (per unit mass) (v)

6.1.3.19 Specific Heat (per unit mass) (q)

6.1.3.20 Specific Internal Energy (per unit mass) (u)

6.1.3.21 Work (per unit mass) (w)

6.1.4 Flow Work (per unit mass) = pv

6.1.5 Property - A property of a system is any observable characteristic of the system. The state of the system is defined by listing its properties.

6.1.6 Process - A process is a change in state that can be defined as any change in the properties of a system. A process is described by specifying the initial and final equilibrium states and the path and interaction that take place across the system boundary during the process.

6.1.7 Cycle - A cycle is a process or a series of processes in which the initial and final states are the same.

6.2 First Law of Thermodynamics

Principle of the conservation of energy

$$\begin{pmatrix}\text{Energy supplied}\\ \text{to a system}\end{pmatrix} - \begin{pmatrix}\text{Energy removed}\\ \text{from a system}\end{pmatrix} = \begin{pmatrix}\text{increase}\\ \text{the system's}\\ \text{energy level}\end{pmatrix}$$

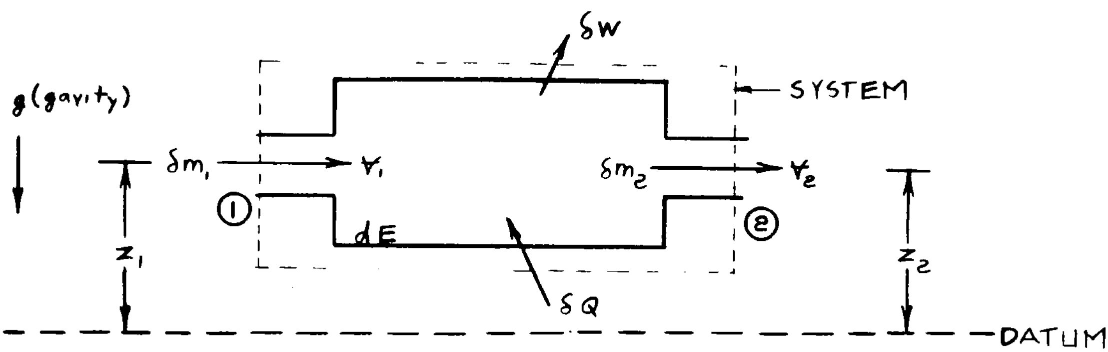


Figure 6-1, Thermodynamic System

δm_1 = mass entering the system

δm_2 = mass leaving the system

The First Law Equation is:

$$\delta m_1\ (u_1 + p_1 v_1 + ((v_1)^2)/2 + z_1 g) -$$
$$\delta m_2\ (u_2 + p_2 v_2 + ((v_2)^2)/2 + z_2 g) +$$
$$\delta Q - \delta W = dE$$
$$h = u + pv$$

Expressing the First Law Equation as a rate equation, we get:

$$\dot{m}_1(h_1 + (v_1)^2/_2 + z_1 g) - \dot{m}_2(h_2 + (v_2)^2/_2 + z_2 g)$$
$$+ \dot{Q} - \dot{W} = {}^{dE}/_{dt}$$

In most cases the fluid properties and flow rate crossing the boundaries remain constant which lets us integrate the First Law Equation. Doing this, we get the following:

$$\Sigma\ m_{in}(u + pv + {}^{v^2}/_2 + gz)_{in} -$$
$$\Sigma\ m_{out}(u + pv + {}^{v^2}/_2 + gz)_{out} + Q - W =$$
$$[m_f(u + {}^{v^2}/_2 + gz)_f - m_i(u + {}^{v^2}/_2 + gz)_i]_{SYSTEM}$$

For a steady state system, the First Law Equation reduces down to:

$$\Sigma\ m_{in}(u + pv + {}^{v^2}/_2 + gz)_{in} -$$
$$\Sigma\ m_{out}(u + pv + {}^{v^2}/_2 + gz)_{out} + Q - W = 0$$

For a closed system, the First Law reduces down to:

$$Q - W = [m(u_f - u_i)]_{SYSTEM}$$

6.3 Second Law of Thermodynamics

Specifies in what direction a process may proceed. There are two classical statements, Clausius and Kelvin.

6.3.1 Clausius statement - it is impossible to construct a device that executes a thermodynamic cycle so that the sole effect is to produce a transfer of heat energy from a body at a low temperature to a body at a high temperature.

6.3.2 Kelvin statement - It is impossible to construct a device that executes a thermodynamic cycle, exchanges heat energy with a single reservoir, and produces an equivalent amount of work.

6.3.3 In an open system, the Second Law of Thermodynamics can be written in terms of entropy as:

$$dS_{system} = (\delta Q/_T) + \delta m_i s_i - \delta m_e s_e + ds_{ir}$$

where: dS_{system} = total change within the system in time dt during the process.

$\delta m_i s_i$ = entropy increase caused by mass entering.

$\delta m_e s_e$ = entropy decrease caused by mass leaving.

$\delta Q/_T$ = entropy change caused by reversible heat transfer between system and surroundings

ds_{irr} = entropy created by irreversibilities

rearranging the above equation, we get:

$$\delta Q = T[(\delta m_e s_e - \delta m_i s_i) + ds_{SYSTEM} - ds_{irr}]$$

Integrating the Second Law Equation we get:

$$(s_f - s_i)_{SYSTEM} = \int_{rev} \delta Q / T + \Sigma(ms)_{in} - \Sigma(ms)_{out} + \Delta s_{produced}$$

6.3.4 Availability

Available energy - is energy in the form of shaft work or in a form completely convertible to shaft work by ideal processes. Energy which is part convertible and part nonconvertible into shaft work is said to be made up of an available part (the availability of energy) and an unavailable part (the unavailability of energy).

Irreversibility or available energy degraded - is the decrease in the available energy due to irreversibilities and is equal to the reversible work minus the actual work for the process.

Reversible work - refers to the maximum useful work obtained for a given change of state including heat supplied from other systems but excluding work done on the surroundings.

To determine the availability it is necessary to specify reference conditions. These conditions are:

a) at rest (zero velocity)
b) at a reference elevation
c) at the same pressure (p_0) and temperature (T_0) as the surroundings
d) at chemical equilibrium

For the steady-flow system, the availability formulation of the second law of thermodynamics is:

$$\dot{A}_Q + \dot{m}(a_f)_1 = \dot{m}(a_f)_2 + \dot{A}_W + \dot{I}$$

where: A_Q = rate of availability of heat transfer to the system
A_W = rate of availability of work transfer from the system
a_f = availability per unit flowing mass
I = irreversibility rate ($I>0$)

The availability of heat transfer rate Q that occurs at a temperature T is:

$$A_Q = (1 - {}^{To}/_T)Q$$

where T_o = temperature of the surroundings

Quite often A_Q is denoted by E_A and its converse unavailable, energy, by E_U. From this we get:

$$E_A = Q(1 - {}^{To}/_T)$$

and

$$E_U = Q - E_A = Q({}^{To}/_T)$$

Flow availability a_f is the maximum work that can be obtained by allowing a unit mass of the flowing fluid to come into equilibrium with the surroundings:

$$a_f = h + gz + v^2/2 - TS - (h_0 + gz_o + v_0^2/2 - T_0S)$$

6.3.5 Refrigeration and Heat Pump Cycles

The Clausius statement in the Second Law of Thermodynamics states that external work must be supplied to a system to transfer heat from a low temperature source to a high temperature source.

Refrigerator - to extract heat from a low temperature reservoir.

Heat pump - to supply heat to a high temperature reservoir.

6.3.6 Carnot Refrigerator or Heat pump

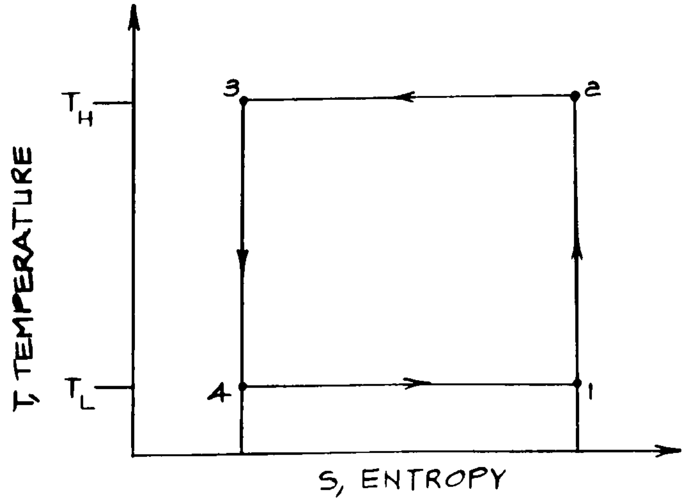


Figure 6-2, Carnot Cycle - Temperature Versus Entropy Diagram

$$(COP)_R = \frac{\text{heat extracted from low-temp reservoir}}{\text{work input for the cycle}}$$

$$(COP)_{R,CARNOT} = \frac{T_L(S_1 - S_4)}{(T_H - T_L)(S_1 - S_4)} = \frac{T_L}{T_H - T_L}$$

$$(COP)_{Hp} = \frac{\text{heat supplied to the high temp reservoir}}{\text{work input for the cycle}}$$

$$(COP)_{HP,CARNOT} = \frac{T_H(S_2 - S_3)}{(T_H - T_L)(S_2 - S_3)} = \frac{T_H}{T_H - T_L}$$

6.3.6.1 Vapor Compression Refrigeration Cycle (VCR)

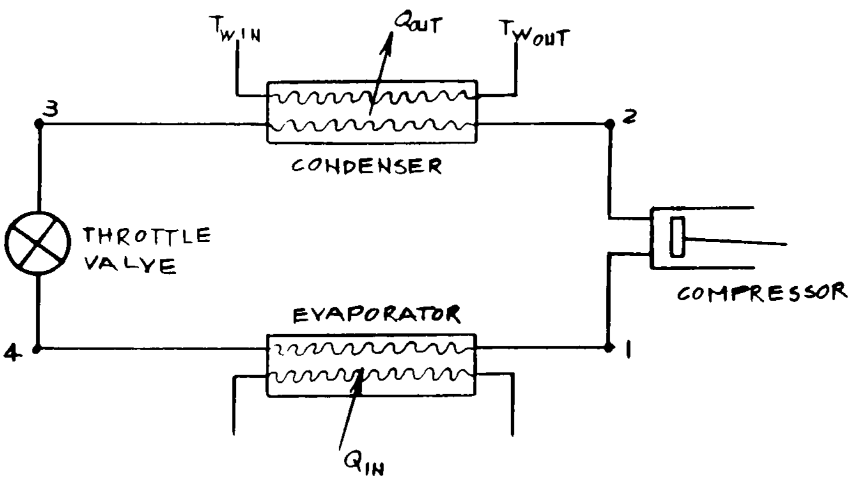


Figure 6-3, Vapor Compression Refrigeration System

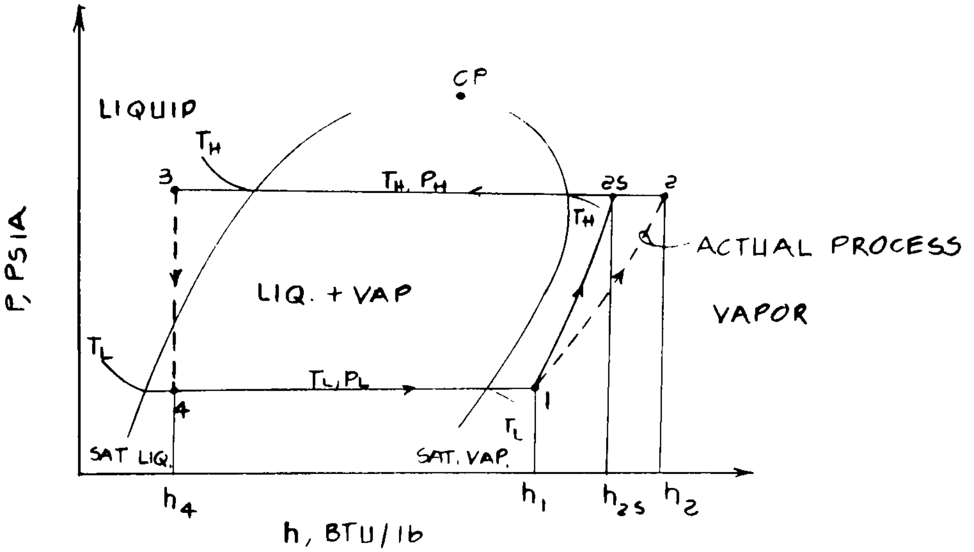


Figure 6-4, Vapor Compression Refrigeration Cycle, Pressure Versus Enthalpy Diagram

$h_4 - h_1$ = net refrigeration
$h_2 - h_1$ = actual work of compression
$h_3 - h_2$ = heat of rejection
$h_{2s} - h_1$ = isentropic work of compression

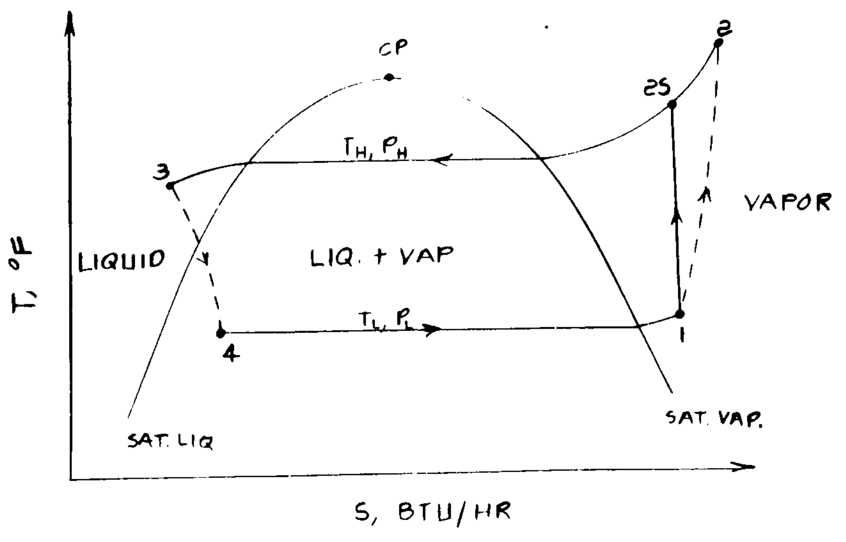


Figure 6-5, Vapor Compression Refrigeration Cycle, Temperature Versus Entropy Diagram

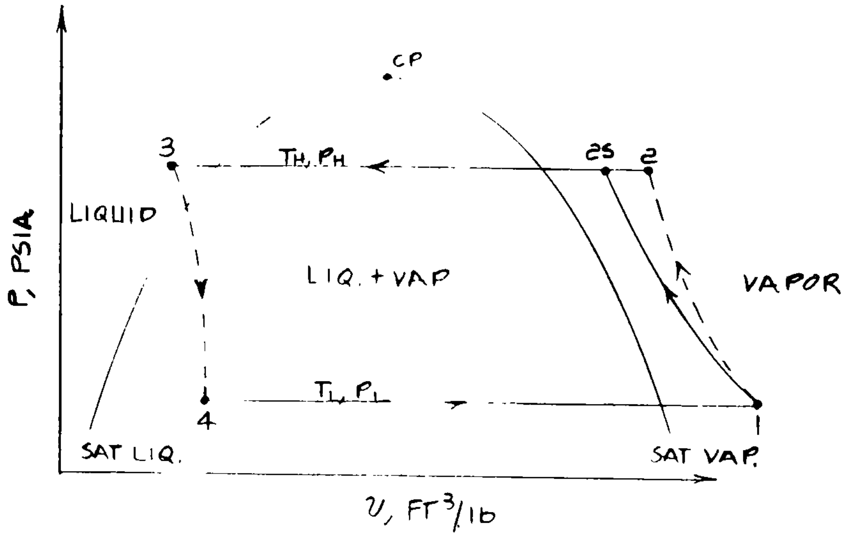


Figure 6-7, Vapor Compression Refrigeration Cycle, Pressure Versus Specific Volume

First Law Chart

Points	Process
1 - 2S	$S = S_1$ (isentropic)
1 - 2	real
2S - 3	$P = P_H$
2 - 3	real
3 - 4	real
4 - 1	$P = P_L$

Compressor

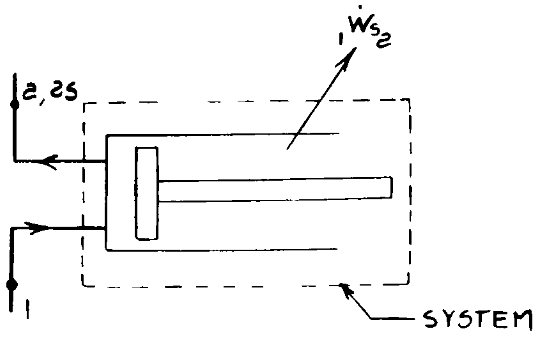


Figure 6-8, Compressor

a) Points	Process	2nd Law
1 - 2S	isentropic	$S=S_1$, $ds=0 \Rightarrow \delta q = 0$
1 -2	real	$ds = {}^{\delta q}/_T$

b) mass conservation $\dot{m}_1 = \dot{m}_{2S} = \dot{m}_2$

c) Energy Conservation (First Law)

assume: steady flow

$\Delta PE = \Delta KE = 0$

perfect insulation

Process 1 - 2S

$$\int_1^{2S} \delta q = \int_{h1}^{2S} \overset{h}{dh} + \int_1^{2S} \delta w_s \quad -{}_1w_{S_{2S}} = h_{2S} - h_1$$

Process 1 - 2

$$\int_1^{2} \delta q = \int_{h1}^{2} dh \overset{h}{+} \int_1^{2} \delta w_S \quad -{}_1w_{S_2} = h_2 - h_1$$

<u>Condenser</u>

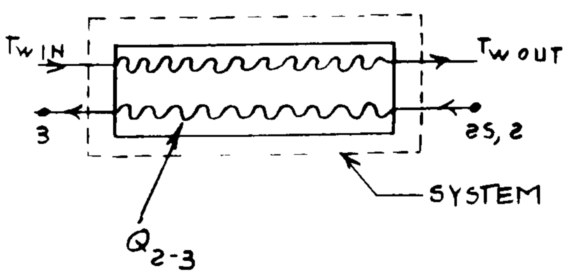


Figure 6-9, Condenser

a) Points	Process	2nd Law
2S - 3	$P = P_{2S}$	$\delta q_r = ds = dh + \delta w_5$ $\delta w_5 = -vdp = 0$
S - 3	real	$ds > {}^{\delta q}/_T$

b) Mass conservation $\dot{m}_2 = \dot{m}_3 = \dot{m}_{2S}$

c) Energy Conservation

$\delta w_5 = 0$, no work done
$\Delta PE = \Delta KE = 0$

Process 2S - 3

$$\int_{2S}^{3} \delta q = \int_{2S}^{3} dh + \int_{2S}^{3} \delta w_S$$

$$\int_{2S} \delta q_3 = h_3 - h_{2S}$$

Process 2 - 3

$$\int_{2}^{3} \delta q = \int_{2}^{3} dh + \int_{2}^{3} \delta w_S$$

$$\int_{2} \delta q_3 = h_3 - h_2$$

<u>Throttle</u>

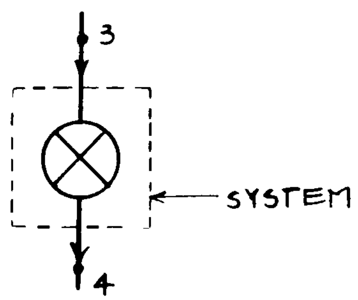


Figure 6-10, Throttle

a) Points	Process	2nd Law
3 - 4	real	$\delta q < Tds$

b) Mass conservation $\dot{m}_3 = \dot{m}_4$

c) Energy Conservation
$\delta w_S = 0$, no work
$Q = 0$, perfect insulation
$\Delta KE = \Delta PE = 0$

$$\dot{Q} = \int h\underline{p}V \cdot dA + \dot{W}s$$

$$0 = -h_3\underline{p}_3V_3A_3 = h_4\underline{p}_4V_4A_4$$

$$0 = -h_3\dot{m}_3 + h_4\dot{m}_4$$

$$h_3 = h_4$$

Evaporator

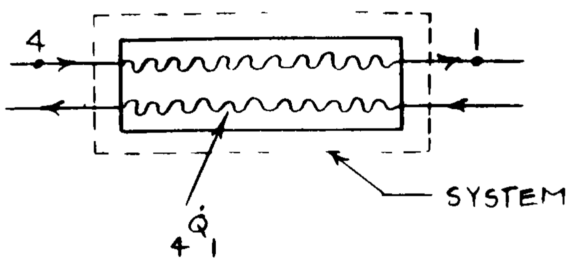


Figure 6-11, Evaporator

a)

Points	Process	2nd Law
3 - 4	$P = P_4$	$\delta q_r = Tds = dh + \delta w_s$
		$\delta w_s = -vdp = 0$

b) Mass conservation $\dot{m}_4 = \dot{m}_1$

c) Energy conservation
$\blacktriangle PE = \blacktriangle KE = 0$
steady state

$$\int_4^1 \delta q = \int_h^{h_1} dh + \int_4^1 \delta w$$

$$_4q_1 = h_1 - h_4$$

Efficiencies

Vapor Compression Refrigeration (COP_{VCR})

$$(COP)_{VCR} = \frac{(h_1 - h_4)}{(h_2 - h_1)}$$

Isentropic Compressor Efficiency $(\eta_{comp})_{isen}$

$$\eta_{comp\,isen} = \frac{h_{2s} - h_1}{h_2 - h_1}$$

Mechanical Compressor Efficiency $(n_{comp})_{mech}$

$$n_{\substack{comp\\mech}} = \frac{(\dot{m}g)_{ref}\ {}_1W_{s_2}}{(T\ X\ N)\ 2\pi}$$

where: T = Torque
N = RPMS
$_1W_{s_2}$ = work
m = mass flow rate
g = acceleration due to gravity

Overall Compressor Efficiency $(n_{overall})$

$$n_{overall} = n_{\substack{comp\\mech}}\ X\ n_{\substack{comp\\isen}} = \frac{(mg)\ {}_1W_{s_{2s}}}{(T\ X\ N)\ 2\pi}$$

6.3.6.2 Heat Pump

The heat pump system is the same as the Vapor Compression Refrigeration system.

$$(COP)_{Hp} = \frac{h_2 - h_3}{h_2 - h_1}$$

7. Fluid Flow

Introduction

This chapter reviews the basic concepts of fluid flow. This chapter also discusses head loss in pipes and static pressure losses in duct systems.

7.1 Eulerian Integral approach

To look at a volume in space and to write equations which govern the behavior of the fluids passing through it. An overall description of larger, finite size particles.

7.2 Basic Equations in Integral form for a Control Volume

7.2.1 Relation of system derivatives to the control volume formulation:

N = extensive property

$N = M, \bar{P}, \bar{H}, E, S$

$\underline{N}$ = intensive property

$\underline{N} = 1, \bar{V}, \bar{r} \times \bar{V}, e, s$

$$\frac{dN}{dt}_{system} = \frac{d}{dt}\int_{cv} \underline{N}\underline{P}dV + \int_{cs} \underline{N}\underline{P}\bar{V} \cdot d\bar{A}$$

7.2.2 Conservation of Mass

$N = M$

$\underline{N} = 1$

$$\frac{dM}{dt}_{system} = \frac{d}{dt}\int_{cv} \underline{P}dV + \int_{cs} \underline{P}\bar{V} \cdot d\bar{A}$$

$$\frac{dM}{dt}_{system} = 0$$

$$0 = \frac{d}{dt}\int_{cv} \rho dV + \int_{cs} \rho \bar{V} \cdot d\bar{A}$$

7.2.3 Momentum Equation

$$N = \bar{P}$$

$$\underline{N} = \bar{V}$$

$$\frac{d\bar{P}}{dt}_{system} = \frac{\rho}{dt}\int_{cv} \bar{v}\rho dV + \int_{cs} \bar{v}\rho\bar{v} \cdot d\bar{A}$$

$$\frac{d\bar{P}}{dt}_{system} = \bar{F}_{system} = \bar{F}_{on\ system} + \bar{F}_{control\ volume}$$

$$\bar{F} = \bar{F}_S + \bar{F}_B = \frac{d}{dt}\int_{cv} \bar{v}\rho dV + \int_{cs} \bar{v}\rho\bar{v} \cdot d\bar{A}$$

$$\bar{F}_B = \int \bar{B}\ dm = \int_{cv} \bar{B}\rho\ dv$$

$\bar{B}$ = body force per unit mass

$$\bar{F} = \int_A -pd\bar{A}$$

<u>7.3 Bernoulli Equation</u> (incompressible invisid flow) - along a stream line:

$$P/\rho + gz + v^2/2 = \text{constant (along a stream line)}$$

$$P_1/\rho + v_1^2/2 + gz_1 = P_2/\rho + v_2^2/2 + gz_2 \text{ (along a stream line)}$$

<u>7.4 Static, Stagnation, and Dynamic Pressures</u>

7.4.1 Static Pressure- would be the pressure measured by an instrument moving with the flow. It is the same pressure found in Bernoulli's Equation.

$$P/\rho + v^2/2 = \text{constant}$$

7.4.1.1 Wall Pressure Tap

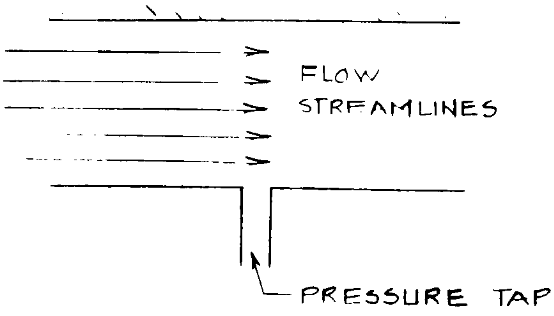


Figure 7-1, Wall Pressure Tap

7.4.1.2 Static Pressure Probe

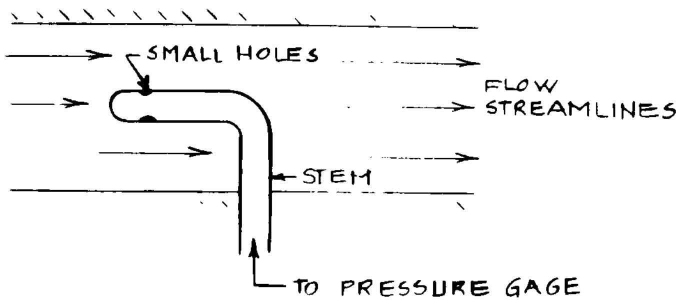


Figure 7-2, Static Pressure Probe

7.4.2 Stagnation Pressure - is that pressure obtained when a flowing fluid is decelerated to zero by a frictionless process.

P_0 = stagnation pressure

$$P_0/\underline{p} + v_0^2/2 = P/\underline{p} + v^2/2$$ Bernoulli's Equation

7.4.2.1 Stagnation Pressure Gauge

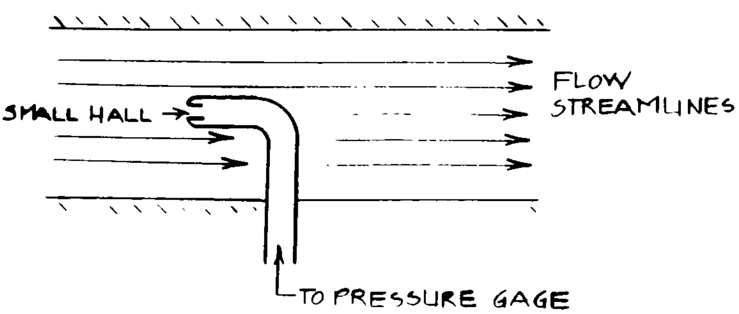


Figure 7-3, Stagnation Pressure Gauge

7.4.3 Dynamic Pressure - $\frac{1}{2}\rho v^2$

$\frac{1}{2}\rho v^2 = P_0 - P$

solving for the speed:

$$v^2 = \frac{2(P_0 - P)}{\rho}$$

7.5 Unsteady Bernoulli Equation

Corresponding equation for unsteady flow along a streamline.

$$P_1/\rho + v_1^2/2 + gz_1 = P_2/\rho + v_2^2/2 + gz_2 + \int_1^2 \frac{dv_s}{dt} ds$$

where: $(dV_s/dt)ds$ is the change in V_s along S

restrictions: 1) Incompressible flow

2) Frictionless flow

3) Flow along a streamline

7.6 Internal Incompressible Viscous Flow

An internal flow is any flow that is contained in pipes, ducts, nozzles, diffusers, sudden contractions and expansions, valves and fittings. This flow can be either laminar or turbulent.

$R_e = \rho \bar{V} D/\mu$ Reynold's Number

$R_e \approx 2300$ transition occurs from laminar to turbulent

7.6.1 Entrance Length and Fully Developed Flow

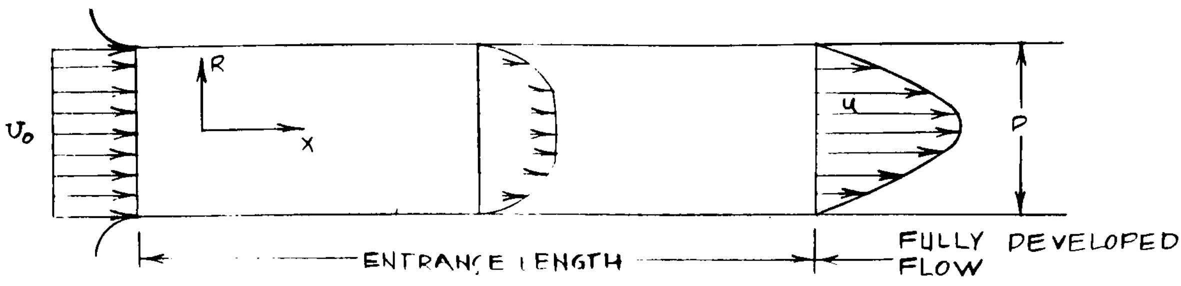


Figure 7-4, Entrance Length and Fully Developed Flow

$$\bar{V} = \underline{v}_\theta = (^1/_A) \int_{area} u dA$$

For laminar flow, the entrance length, L, is a function of the Reynold's number:

$$\frac{L}{D} \approx 0.06 \frac{\rho \bar{V} D}{M} \approx .06 Re D$$

7.7 Calculation of Head Loss in Piping Systems

Total head loss, h_{lt}, is regarded as the sum of major losses, h_l, due to frictional effects in fully-developed flow in constant-area tubes, and minor losses, h_{lm}, due to entrances, fittings, area changes, and so on. We consider major and minor losses separately.

7.7.1 Major Head Losses (h_l)

$$h_l = (P_1 - P_2)/_{\rho} - g(z_2 - z_1)$$

for fully developed flow through a constant area pipe if $z_1 = z_2$ get $h_l = {}^{\Delta P}/_{\rho}$

7.7.1.1 Laminar Flow

$$\Delta P = 32\frac{L\mu\bar{V}}{D^2}$$

$$h_l = (64/R_e)(L\bar{V}^2/2D)$$

7.7.1.2 Turbulent Flow (experimental results)

$$h_l = \frac{fL\bar{V}^2}{2D}$$

f = friction factor
(determined experimentally)

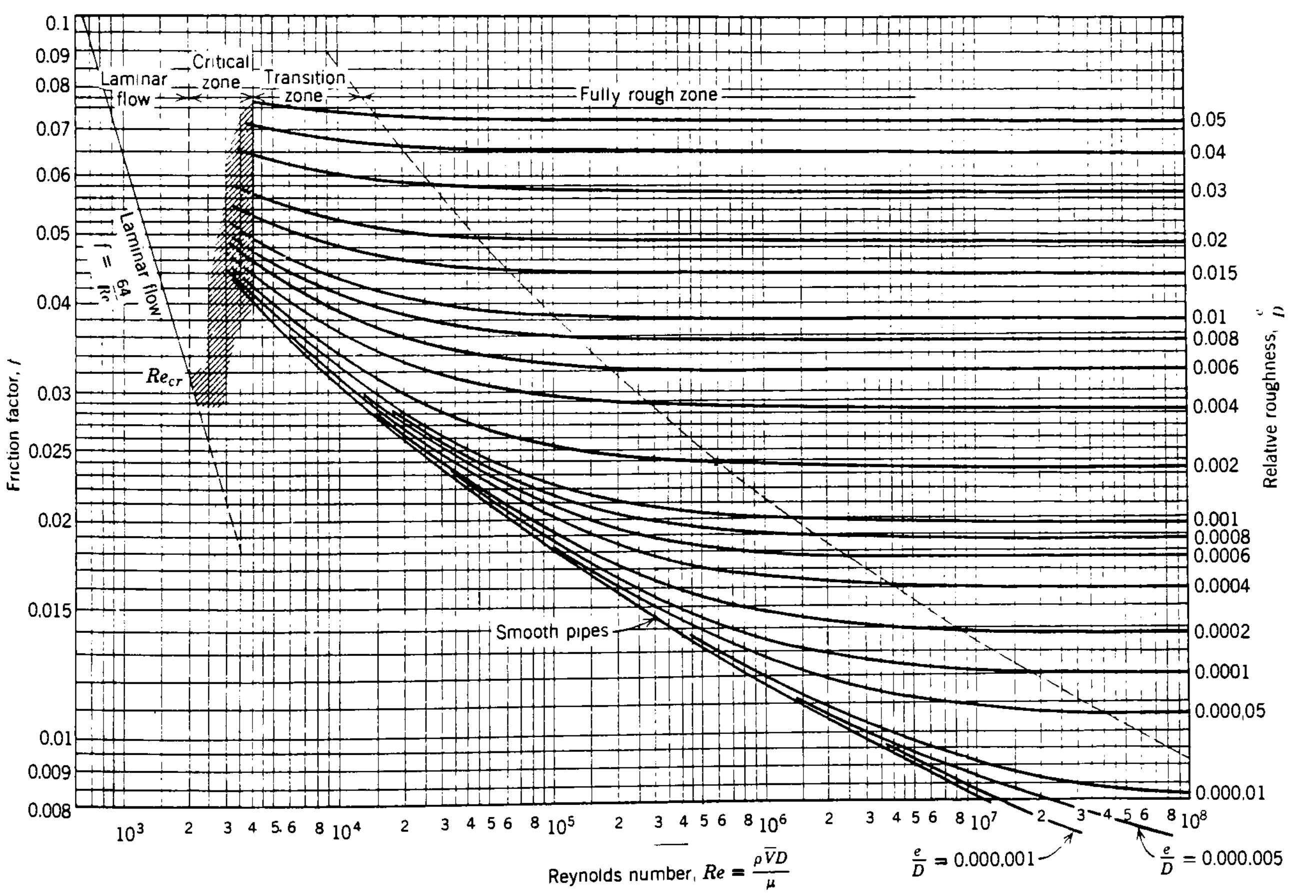


Figure 7-5, Friction Factor for Fully Developed Flow in Circular Pipes
Abridge from "Introduction to Fluid Mechanics", by Robert W. Fox and Alan T. McDonald, 3rd Ed. John Wiley and Sons. 1985

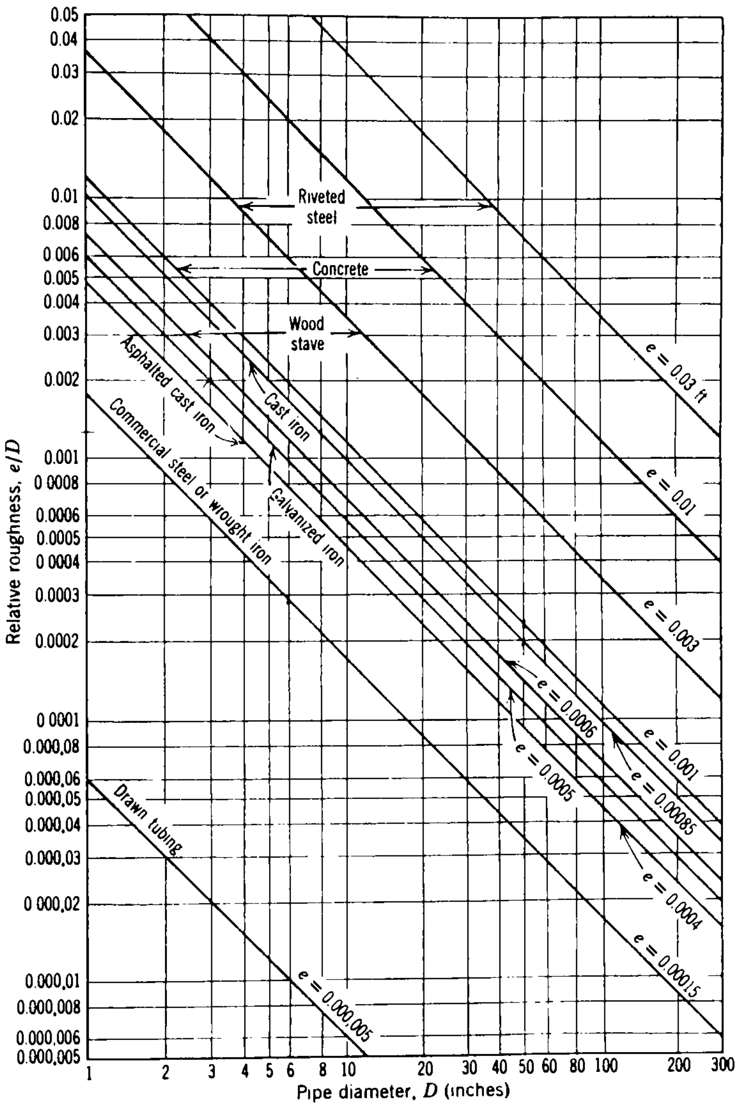


Figure 7-6, Relative Roughness Values for Pipes of Common Engineering Materials Abridge from "Introduction to Fluid Mechanics", by Robert W. Fox and Alan T. McDonald, 3rd Ed. John Wiley and Sons. 1985

The B lasuis correlation for turbulent flow in smooth pipes, valid for $R_e \leq 10^5$ is:

$$f = \frac{.3164}{R_e^{.25}}$$

The most widely used formula for friction factor is that due to Colebrook:

$$\frac{1}{f^{.5}} = -2.0\log\left(\frac{e/D}{3.7} + \frac{2.51}{R_e f^{.5}}\right)$$

estimated within one percent

$$f^{\circ} = .25\left[\log\left(\frac{e/D}{3.7} + \frac{5.74}{R_e^{.9}}\right)\right]^{-2}$$

7.7.2 Minor Losses (due to fittings, bends, and abrupt changes in area)

$$h_{lm} = K + \bar{V}^2/2$$

K = loss coefficient, determined experimentally

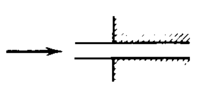
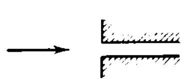
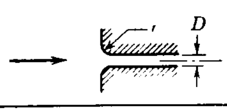
or

$$h_{lm} = f\,(L_e/D)\,(\bar{V}^2/2)$$

L_e = equivalent length of straight pipe, determined experimentally

7.7.2.1 Inlets and Exits

Table 7-1, Minor Loss Coefficients for Pipe Entrances
Abridge from "Introduction to Fluid Mechanics", by Robert W. Fox and Alan T. McDonald, 3rd Ed. John Wiley and Sons. 1985

Entrance Type		Minor Loss Coefficient, K^a
Reentrant		0.78
Square-edged		0.5
Rounded		r/D: 0.02, 0.06, ≥0.15; K: 0.28, 0.15, 0.04

[a] Based on $h_{l_m} = K(\bar{V}^2/2)$, where $\bar{V}$ is the mean velocity in the pipe.

7.7.2.2 Enlargements and Contractions

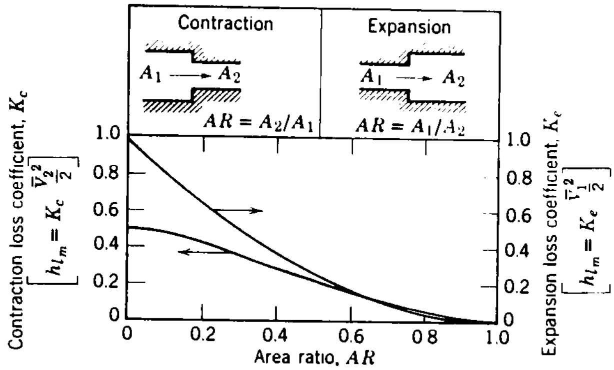
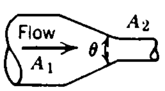


Figure 7-7, Loss Coefficients for Flow through Sudden Area Changes
Abridge from "Introduction to Fluid Mechanics", by Robert W. Fox and Alan T. McDonald, 3rd Ed. John Wiley and Sons. 1985

Table 7-2, Loss Coefficients for Gradual Contractions: Round and Rectangular Ducts
Abridge from "Introduction to Fluid Mechanics", by Robert W. Fox and Alan T. McDonald, 3rd Ed. John Wiley and Sons. 1985



	Loss Coefficient, K^a						
	Included Angle, θ, Degrees						
A_2/A_1	10	15–40	50–60	90	120	150	180
0.50	0.05	0.05	0.06	0.12	0.18	0.24	0.26
0.25	0.05	0.04	0.07	0.17	0.27	0.35	0.41
0.10	0.05	0.05	0.08	0.19	0.29	0.37	0.43

[a] Based on $h_{l_m} = K(\bar{V}_2^2/2)$

7.7.2.3 Diffusers

C_p is defined as the ratio of static pressure rise to inlet dynamic pressure.

$$C_p = \frac{P_2 - P_1}{\frac{1}{2}\rho\bar{V}_1^2}$$

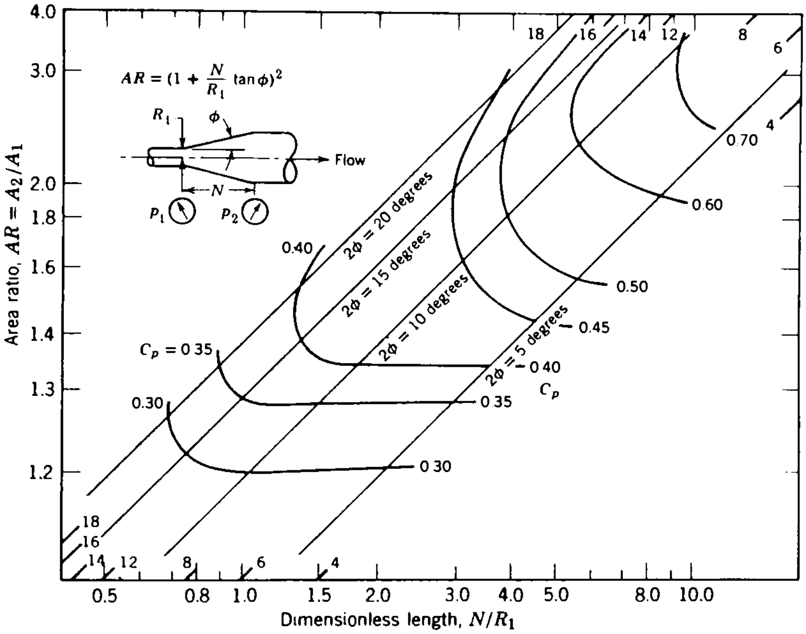


Figure 7-8. Pressure Recovery Data for Conical Diffusers with Fully Developed Turbulent Pipe Flow at Inlet
Abridge from "Introduction to Fluid Mechanics", by Robert W. Fox and Alan T. McDonald, 3rd Ed. John Wiley and Sons. 1985

$$h_{lm} = \frac{\bar{V}_1^2}{2}\left[\left(1 - \frac{1}{(AR)^2}\right) - C_p\right]$$

The Ideal Pressure Recovery Coefficient is:

$$C_{pi} = 1 - \left(1/(AR)^2\right),$$

for Ideal Flow where $h_{lm} = 0$

Then the head loss can be represented by:

$$h_{lm} = (C_{pi} - C_p)(\bar{V}_1^2/2)$$

7.7.2.4 Pipe Bends

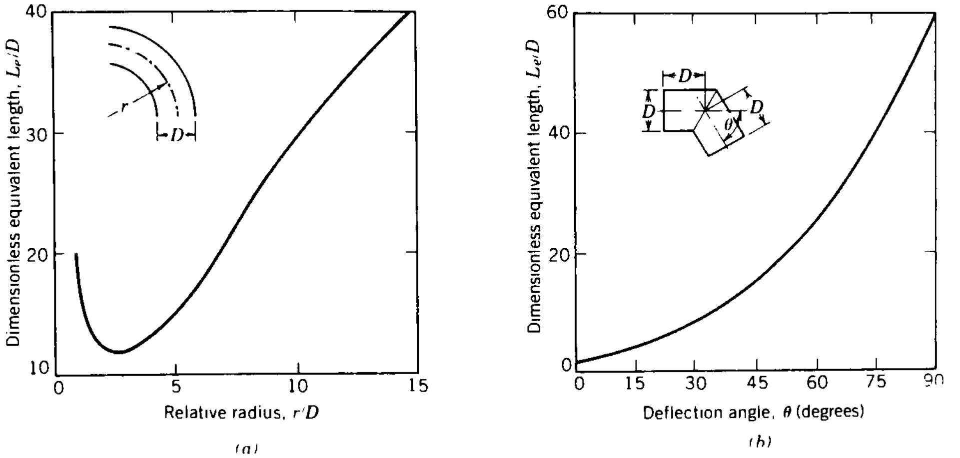


Figure 7-9, Representative Resistance values (L_e/D) or (a) 90° Pipe Bends and Flanged Elbows, and (b) Miter Bends
Abridge from "Introduction to Fluid Mechanics", by Robert W. Fox and Alan T. McDonald, 3rd Ed. John Wiley and Sons. 1985

7.7.2.5 Valves and Fittings (±10% accuracy)

Table 7-3, Representative Dimensionaless Equivalent Lengths (L_e/D) for Valves and Fittings
Abridge from "Introduction to Fluid Mechanics", by Robert W. Fox and Alan T. McDonald, 3rd Ed. John Wiley and Sons. 1985

Fitting Type	Equivalent Length,[a] L_e/D
Valves (fully open)	
Gate valve	8
Globe valve	340
Angle valve	150
Ball valve	3
Lift check valve: globe lift	600
: angle lift	55
Foot valve with strainer: poppet disk	420
: hinged disk	75
Standard elbow: 90	30
: 45	16
Return bend, close pattern	50
Standard tee: flow through run	20
: flow through branch	60

[a] Based on $h_{l_m} = f \frac{L_e}{D} \frac{\bar{V}^2}{2}$

*The ASHRAE fundamentals book has a greater list of these head loss coefficients.

7.8 Static Pressure Loss in Air Duct Systems

From Bernoulli's Equation:

$$\rho({V_1}^2/2) + (P_{z1} + P_1) + \alpha z_1 =$$

$$\rho({V_2}^2/2) + (P_{z2} + P_2) + \alpha z_2 + \Delta P_t$$

where: V = average duct velocity, fpm
ΔP_t = total pressure loss between station 1 and 2 in the system lbf/ft^2
α = $\rho g/gc$ specific weight, lbf/ft^2
P = pressure, lbf/ft^2, gauge pressure
P_z = represents atmosphere pressure at elevation z_1 and z_2

When the specific weight of the atmosphere is constant, the above equation reduces to:

$$\frac{\rho V_1^2}{2g_c} + P_1 = \frac{\rho V_2^2}{2g_c} + P_2 + \Delta P_t$$

7.8.1 Stack effect - caused by the differences in the properties of the gas in the stack as compared to the gas outside the stack (density).

$$\Delta P_t = \left(P_1 + \frac{\rho V_1^2}{2g_c}\right) - \left(P_2 + \frac{\rho V_2^2}{2g_c}\right) + \frac{g}{g_c}(\rho_a - \rho)(Z_2 - Z_1)$$

stack effect is given by:

$$P_{se} = (.192)(\rho_a - \rho)(Z_2 - Z_1)$$

where: P_{se} = stack effect, in. H_2O
ρ_a = density of atmosphere air lb/ft^3
ρ = density of air within the ducts lb/ft^3

12.9.2 Pressure loss (in. of H_2O) - total pressure in a duct system is caused by two components, static pressure and velocity pressure.

static pressure = P

Velocity pressure = $P_v = \rho V^2/2g_c$

Total pressure = $P_t = P + P_v$

7.8.2 Fluid resistance- cause of two types of losses, frictional losses and dynamic losses.

7.8.2.1 frictional losses - caused by friction of the molecules in the flow.

This loss can be calculated by the Darcy - Weisbach Equation:

$$\Delta P_{fr} = F_D \left(^{12L}/_{D}\right) P_v$$

where: ΔP_{fr} = friction losses in terms of total pressure (in. H_2O)
F_D = friction factor
L = length of duct, ft.
D = diameter of pipe, in.

For turbulent flow f_D can be determined by Colebrook's Equation:

$$\frac{1}{\sqrt{f_D}} = -2\log_{10}\left[\frac{12E}{3.7D} + \frac{2.51}{R_e\sqrt{f_D}}\right]$$

where: E = material absolute roughness factor, ft.

The above equation is a transcendental equation and must be solved using an iterative technique.

A friction chart for round ducts can be found in the ASHRAE fundamentals book, chapter 33, figure A-1.

Correction factors - used if there is a significant variation in temperature, barometric pressure (elevation), and humidity:

$$\Delta P_{f_{r,a}} = KK_m \Delta P_{f_{r,s}}$$

where: $\Delta P_{f_{r,a}}$ = friction loss in terms of total pressure at actual conditions, in. H_2O
$\Delta P_{f_{r,s}}$ = friction loss in terms of total pressure at standard conditions, in. H_2O

K = friction chart correction factor for density and/or viscosity
K_m = friction chart correction factor for duct roughness

K can be determined by two different methods:

method 1: $K = ((\rho_a/\rho_s)^{.9}(\mu_a/\mu_s)^{.1}$

a = actual conditions
s = standard conditions

method 2: $K = K_T K_E K_H$

K_T = temperature corrections
$K_T = [530/(T_a + 460)]^{.825}$

K_E = elevation corrections
$K_E = (B/29.921)^{.9}$
(B = barometric pressure, in Hg)
or
$K_E = [1 - (6.8754 \times 10^{-6})z]^{4.73}$
z = elevation, ft.

K_H = humidity correction
$K_H = [1 - .378(P_s/B)]^{.9}$
P_s = saturation pressure of water vapor at dewpoint temperature, in. Hg

K_m can be found in the ASHRAE fundamentals

book in chapter 33 on page 6.

Friction losses for noncircular ducts:

$$D_e = 4A/P$$

where : D_e = hydraulic diameter, in.
A = area, in^2
P = perimeter of cross-section, in.

Friction losses for rectangular ducts:

$$D_e = \frac{1.30(ab)^{.625}}{(a + b)^{.25}}$$

where: D_e = circular equivalent diameter
a,b = length of sides

D_e can also be found in ASHRAE fundamentals book, chapter 33, table A-2.

Friction losses for oval ducts: for aspect ratios from 2.0 to 4.1

$$D_e = 1.55(A^{.625}/P^{.25})$$

or

$$\Delta P_{fr} = .0245LK(P/4A)^{1.22}(V/1000)^{1.9}$$

where: ΔP_{fr} = oval duct friction loss, in. H_2O
K = correction factor
P = perimeter

Friction losses due to duct liners and fibrous glass ducts are usually given by manufacturers data.

7.8.2.2 Dynamic losses - are the result of flow disturbances caused by fittings that changes the airflow path's direction and/or area. These fittings include entries, exits, transitions and junctions.

Local Loss Coefficients

$$C_o = \frac{\Delta P_t}{P_{v,o}} \quad \text{or} \quad \Delta P_t = C_o P_{v,o}$$

where: C_o = local loss coefficient for section o
ΔP_t = fitting total pressure loss, in H_2O
$P_{v,o}$ = velocity pressure at section o, in H_2O

For converging and diverging flow junctions total pressure losses through the main section are calculated as:

$$\Delta P_t = C_{c,s} P_{v,c}$$

For total pressure losses through the branch section:

$$\Delta P_t = C_{c,b} P_{v,c}$$

To convert converging flow coefficients to upstream main and branch velocity pressures, the equations below are used.

$$C_s = \frac{C_{c,s}}{(1 - Q_b/Q_c)^2 (A_c/A_s)^2}$$

$$C_b = \frac{C_{c,b}}{(Q_b A_c/Q_c A_b)}$$

where: C_s = main local loss coefficient referenced to upstream velocity pressure
Q_b = branch airflow rate, cfm
Q_c = common airflow rate
A = respected? areas, in^2 or ft^2
C_b = branch local loss coefficient

Figure 7-10, Converging and Diverging Flows *loss coefficients can be found in ASHRAE fundamentals book chapter 33, tables B-1 through B-7

8. Heat Transfer

Introduction - This chapter focuses on the transmission of energy in the form of heat. Heat transfer occurs by three basic mechanisms or modes: conduction, convection, and radiation. This chapter discusses these three modes of heat transmission.

Conduction - the transmission of heat through a substance without motion.

Convection - the transmission of heat by the bulk movement of a fluid.

Radiation - the transmission of energy by electromagnetic radiation having a defined range of wavelengths.

8.1 Conduction

8.1.1 Steady State

Heat conduction can be expressed by:

$$\frac{q_x}{A} = -k\frac{dT}{dx}$$

where: q_x = heat flow in x direction
dT/dx = temperature gradient
k = thermal conductivity of the material
A = area

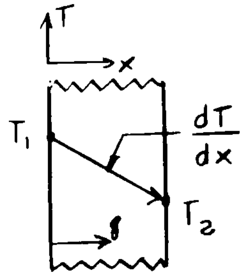


Figure 8-1, Conduction Through a Single Material

$\frac{q_x}{A}$ = is termed the heat flux and is sometimes written as q_x''

8.1.2 Unsteady state with internal heat generation

rate of energy conducted into control volume	+	rate of energy generated inside control volume	=	
rate of energy conducted out of control volume	+	rate of energy stored inside control volume		

8.1.2.1 A one dimensional system in rectangular coordinates

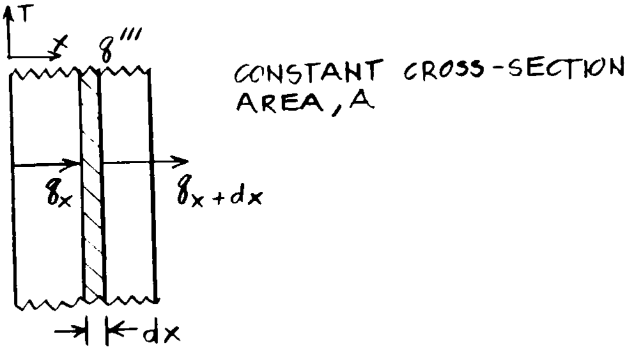


Figure 8-2, One Dimensional System in Rectangular Coordinates

For constant thermal conductivity the equation becomes:

$$\frac{d^2T}{dx^2} + \frac{q'''}{k} = \frac{1}{\alpha}\frac{dT}{dt}$$

where: $\alpha = \frac{k}{\rho c}$ thermal difusivity

c = specific heat

8.1.2.2 Cylindrical coordinates

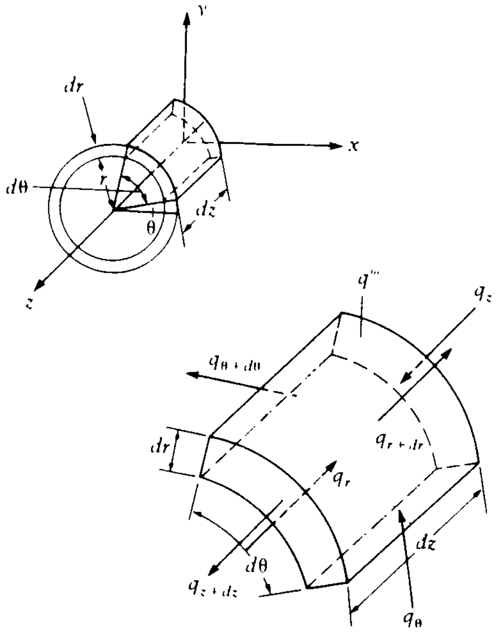


Figure 8-3, Volume Element in Cylindrical Coordinates

$$\frac{d^2T}{dr^2} + \frac{1}{r}\frac{dT}{dr} + \frac{1}{r^2}\frac{d^2T}{d\theta^2} + \frac{d^2T}{dz^2} + \frac{q'''}{h} = \frac{1}{\alpha}\frac{dT}{dt}$$

8.1.3 Steady state conduction in one dimension with no internal heat generation

$$q_x = -KA\frac{dT}{dx} = -KA\frac{dT}{dx}$$

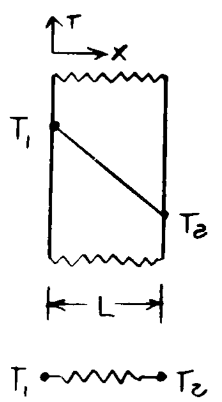


Figure 8-4, One Dimensional System

$$\frac{dT}{dx} = \frac{T_2 - T_1}{L}$$

$$q_x = \frac{-KA(T_2 - T_1)}{L} = \frac{T_1 - T_2}{(L/KA)}$$

8.1.3.1 Materials in Series

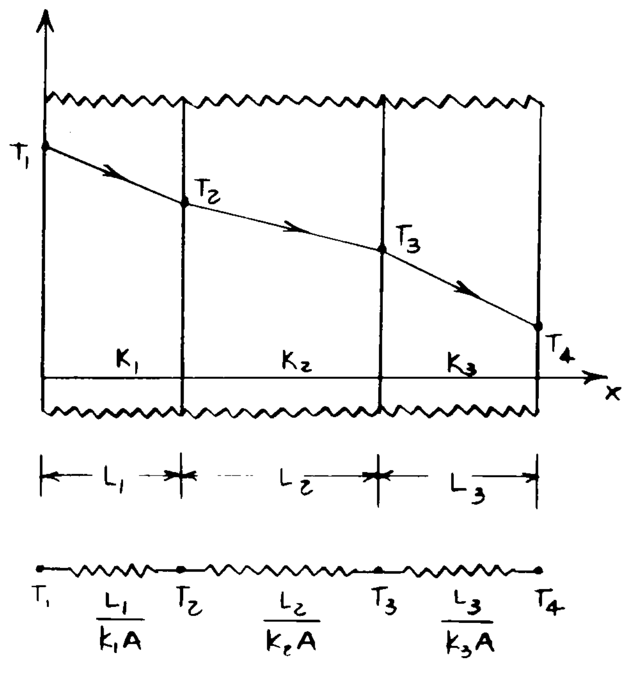


Figure 8-5, Materials in Series

$$q_x = \frac{T_1 - T_4}{\frac{L_1}{K_1A} + \frac{L_2}{K_2A} + \frac{L_3}{K_3A}} = \frac{\text{overall temp difference}}{\text{Sum of the Thermal conductivities}}$$

8.1.3.2 Materials in parallel

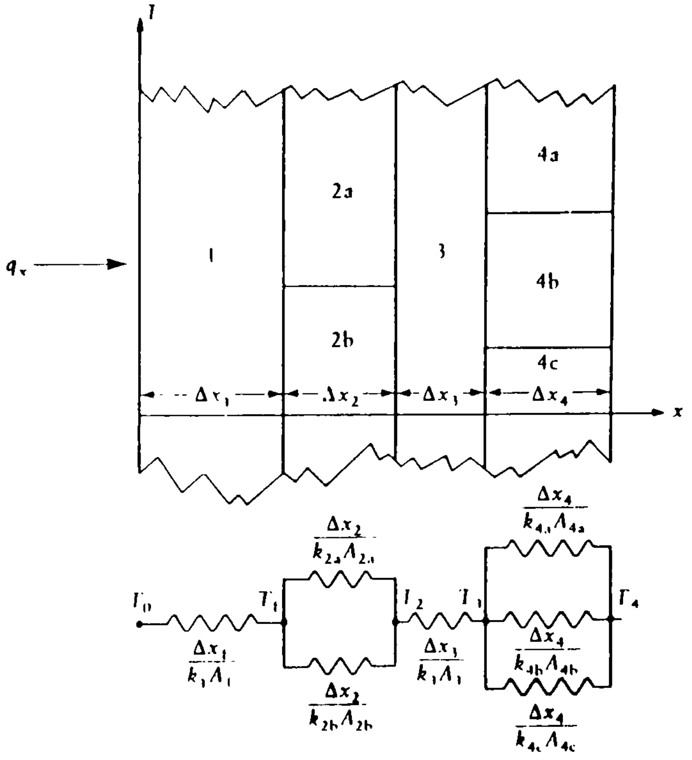


Figure 8-6, Composite Wall in Series/Parallel Arrangement
Abridge from "Engineering Heat Transfer", William S. Janna, 1st ed., PWS Publishers, 1986

$$q_x = \frac{T_0 - T_4}{\frac{\Delta x_1}{k_1 A_1} + \frac{\Delta x_2}{k_{2A}A_{2a} + k_{2b}A_{2B}} + \frac{\Delta x_3}{k_3 A_3} + \frac{\Delta x_4}{K_{4a}A_{4a} + K_{4c}A_{4c} + K_{4b} + A_{4b}}}$$

8.2 Convection

There are two kinds of convection, forced and free. In general, there are three types of flow situations: 1) closed - conduct, 2) external, and 3) unbounded flows.

$q''_c = h_c(T_w - T_\infty)$ (Newton's Law of Cooling)

where: h_c = convection coefficient
T_w = wall temperature
T_∞ = fluid temperature
q''_c = heat flux

Dimensional analysis is used to solve for h_c and the relevant physical properties and kinematic parameters of the flow situation. A more in-depth discussion is found in "Engineering Heat Transfer" by W.S. Janna in chapters 7, 8, 9, and 10.

8.3 Radiation

speed of light, $c_o = 9.836 \times 10^8\ {}^{ft}/_{sec}$

photon energy, $e = hv$

h = Planck's constant
$= 6.284 \times 10^{-37}$ BTU
v = frequency
$e = {}^{hc_o}/_{\phi}$
ϕ = wavelength

8.3.1 Emission and absorption at an opaque solid

absorptivity - amount an object absorbs

$$\alpha = {}^{q''a}/_{q''_i}$$

q''_i = incident energy flux

q''_a = absorbed energy flux

specular absorptivity

$\alpha_\phi = {}^{q''a\phi}/_{q''_{i\phi}}$ for a certain wavelength

values of $\alpha < 1$ are considered gray bodies

values of $\alpha = 1$ are considered black bodies

Rules for a black body:

1) A black body absorbs all incident radiation.

2) No surface can emit more energy than a black body at any given temperature and wavelength.

3) A black body emits radiant energy that depends on wavelength and temperature but is independent of direction

Emissivity - how much an object emits compared to a black body

$$\epsilon = (q''_e/q''_b)$$

Spectral emissivity - is wavelength dependent

$$\epsilon = (q''_{e\phi}/q''_{b\phi})$$

Wavelength and direction dependent

$$\epsilon_{\phi\theta\Phi} = \frac{q''_{e\phi\theta\Phi}}{q''_{b\phi\theta\Phi}} \quad \text{spherical coordinates}$$

$$\alpha_{\phi\theta\Phi} = \frac{q''_{a\phi\theta\Phi}}{q''_{i\phi\theta\Phi}} \quad \text{spherical coordinates}$$

diffuse object emits radiation equally in all directions

Transmissivity

$$T = q''_t/q''_i$$

Reflectivity

$$p = q''_r/q''_i$$

When light is incident on surface

$$\alpha + T + p = 1$$

8.3.2 Emittance, reflectivity, and transmissivity of real surfaces

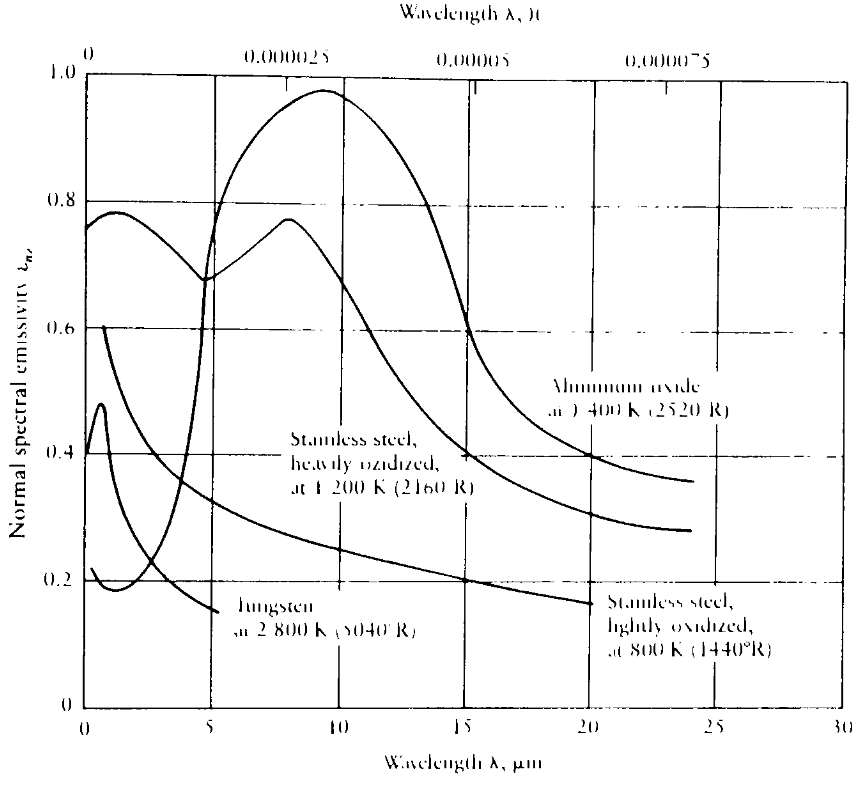


Figure 8-7, Variation of Normal Spectral Emissivity with Wavelength for Several Surfaces Abridge from "Engineering Heat Transfer", William S. Janna, 1st ed., PWS Publishers, 1986

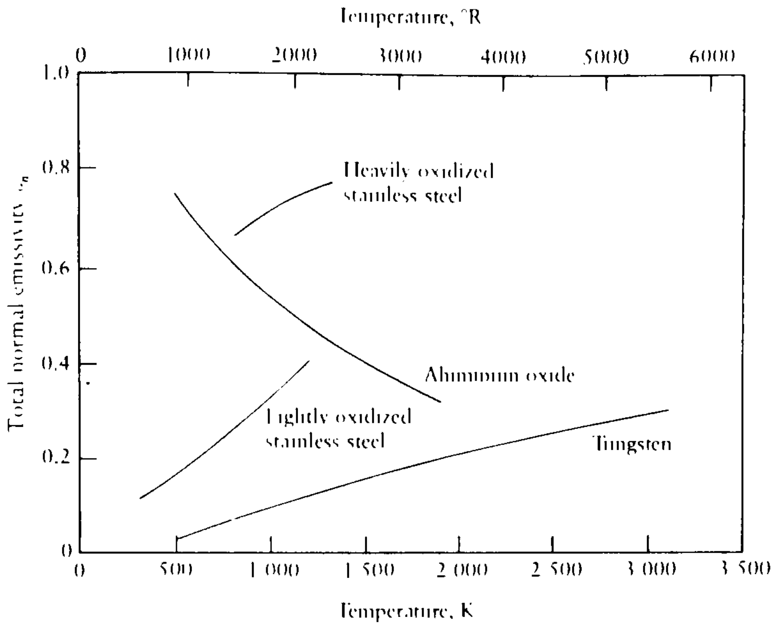


Figure 8-8, Variation of the Total Normal Emissivity ϵ_n With Temperature
Abridge from "Engineering Heat Transfer", William S. Janna, 1st ed., PWS Publishers, 1986

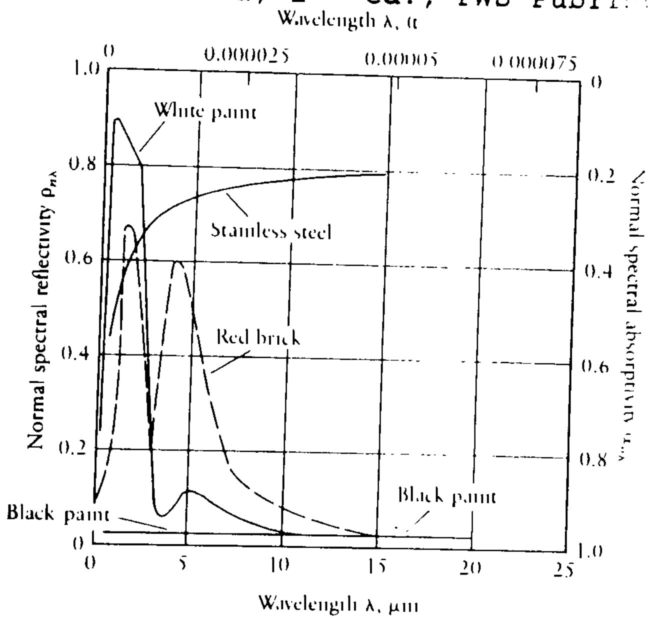


Figure 8-9, Variation of Normal Spectral Reflectivity $\rho_{\phi n}$ and Normal Spectral Absorptivity $\alpha_{\phi n}$ With Wavelength for Various opaque Surfaces
Abridge from "Engineering Heat Transfer", William S. Janna, 1st ed., PWS Publishers, 1986

Table 8-1, Low-temperature Emissivity and High-temperature Absorptivity of Various Surfaces Abridge from "Engineering Heat Transfer", William S. Janna, 1st ed., PWS Publishers, 1986

Surface material	Emissivity at atmospheric temperatures	Solar absorptivity
Highly polished "white" metals, gold, yellow brass	0.02–0.08	0.1–0.4
Clean, "dark" metals	0.1–0.35	0.3–0.6
Metallic-pigment paints	0.35–0.55	0.4–0.6
White, nonmetal surfaces	0.7–0.9	0.1–0.35
Dark-colored nonmetals	0.7–0.9	0.45–0.8
Black paint, asphalt, carbon, water	0.85–0.95	0.7–0.9

Table 8-2, Spectral Transmittance of Glass Abridge from "Engineering Heat Transfer", William S. Janna, 1st ed., PWS Publishers, 1986

	Transmittance, percent			
Wavelength, nm*	Clear, polished, $\frac{1}{4}$-in. plate	Gray tint, $\frac{1}{4}$-in. Solargray®	Bronze tint, $\frac{1}{4}$ in. Solarbronze®	Blue green, $\frac{1}{4}$ in. Solex®
Ultraviolet region				
300	1.1	0.0	0.0	0.0
320	0.3	0.0	0.0	0.0
340	29.4	2.1	0.3	0.7
360	74.8	33.3	22.8	33.3
380	79.3	44.2	34.1	44.3
Total solar	68.	34.	26.	35.
Visible region				
400	87.2	53.8	48.5	70.6
440	87.3	43.1	44.3	71.6
480	89.3	40.6	43.3	79.0
520	89.5	38.1	45.2	79.8
560	88.7	46.1	54.2	76.5
600	87.0	38.3	51.7	69.4
640	84.4	38.6	51.5	60.3
680	81.3	49.4	54.4	50.1
720	77.9	59.6	55.7	40.5
Total solar	88.	41.	51.	75.
Infrared region				
800	71.4	51.3	46.	27.1
900	65.8	43.6	39.0	18.8
1 000	63.4	40.5	36.3	15.8
1 100	62.9	38.9	35.2	15.3
1 200	63.8	38.2	35.8	16.3
1 400	69.9	44.4	43.7	21.1
1 600	73.8	54.3	55.3	32.6
1 800	75.9	56.3	57.8	40.0
2 000	77.1	58.5	58.9	40.3
Total solar	67.	46.	42.	22.

* One nanometer = 1×10^{-9} meters = 1×10^{-3} micrometers

9. Mass Transfer

Introduction - Mass Transfer is the transport of one component of a mixture, relative to the motion of the mixture and is the result of a concentration gradient. This mass transfer can occur by either diffusion or convection. This chapter discusses these two modes of mass transfer.

9.1 Molecular diffusion

The molecular diffusion of a gas into a second gas, a liquid, or a solid.

The diffusing component is denoted by B and the other component is denoted by A.

Molecular diffusion happens at ordinary temperatures and pressures and is caused by a density gradient.

In a gaseous medium containing gasses A and B, B diffuses through A in the direction to reduce the density gradient.

9.1.1 Fick's Law in general form

ρ_B/ρ is the fraction of component B to the binary mixture, mass ratio

C_B/C is the mole fraction equivalent

$$J_B = -\rho D_v \frac{d(\rho_B/\rho)}{dy}$$

$$J_B^* = -C D_v \frac{d(C_B/C)}{dy}$$

where: D_v = diffusion coefficient
J_B = diffusive mass flux
J_B^* = diffusive molar flux

or

$$J_{B*} = \underline{p}_B(v_B - v)$$
$$J_B^* = C_B(v_B - v)$$

where: $(v_B - v)$ = the velocity of component B relative to the velocity of the mixture

9.1.2 Fick's Law for Dilute Mixtures

The density of component B is small compared to the mixture. The diffuse mass flux can then be written as:

$$J_B = D_v(d\underline{p}_B/dy)$$

when $\underline{p}_B << \underline{p}$, $\blacktriangle\underline{p} < \blacktriangle\underline{p}_B$

This can be used without significant error for water vapor diffusing through air at atmosphere pressure and temperatures less than 80°F. This yields errors around 2%. For temperatures up to 140°F this will produce errors of about 10%.

9.1.3 Fick's Law for Mass Diffusion through Solids or Stagnant Fluids

$$\underline{p}_B << \underline{p} \text{ and } v = 0$$

$$J_B = \underline{p}_B(v_B - v) \approx \underline{p}_B v_B = \dot{m}_B''$$

$$\dot{m}_B'' = D_v(d\underline{p}b/dy)$$

9.1.4 Fick's Law for Ideal Gases with Negligible Temperature Gradient

$$P_B = \frac{\underline{C}_B\underline{R}_g\underline{T}}{M_b} \quad \text{approximating gas B as ideal}$$

when the gradient T is small

$$J_B = \frac{M_B D_v}{R_g T(dp_b/dy)}$$

or

$$J_B^* = \frac{\underline{D_v}}{R_g T(dp_b/dy)}$$

when B is dilute and ideal and T = const. $v_A = 0$

$$\dot{m}_B'' = \frac{\underline{M_B}\underline{D_v}}{R_g T(dp_b/dy)}$$

$$\dot{m}_B''^* = \frac{\underline{D_v}}{R_g T(dp_b/dy)}$$

9.1.5 The Diffusion Coefficient (D_v)

For a binary mixture, the diffusion coefficient, D_v, is a function of temperature, pressure and composition. Most of the time these coefficients are determined experimentally and are listed in table form. One of these table is shown in the ASHRAE Fundamentals Handbook in chapter 5, page 2, table 1.

9.1.6 The analogy between heat and mass diffusion

Flux = diffusivity x concentration gradient

Examination of the equation below:

$$\dot{m}''_B = D_v(^{d\underline{p}B}/_{dy})$$

which is Fick's law applied to mass diffusion of a dilute gas through solids or stagnant fluids is analogous to Fourier's Law:

$$q'' = -k(^{dT}/_{dx})$$

therefore, all of the heat transfer solutions for steady state and transient conduction are available for solving analogous steady state and transient mass transfer problems.

9.1.7 Molecular Diffusion in Liquids and Solids

Fick's Law:

$$\dot{m}_B'' = D_v(C_{B1} - C_{B2})/(Y_1 - Y_2)$$

where: C_B = molal concentration of a solute in a solvent, lbmol/ft^3

$Y_1 - Y_2$ = the difference that separates the two molal concentrations

The flow of liquid or gas through a porous or granular solid in the interstices and capillaries is termed structure sensitive diffusion . For this type of diffusion, the equation below is used:

$$\dot{m}_B'' = \bar{\mu}(\Delta P_B/\Delta y)$$

where:? $\bar{\mu}$ = permeability factor (experimentally found) (grains in)/(h ft^2 inHg)

Δy = material thickness

$\Delta P_B/\Delta y$ = water vapor pressure gradient, in hg/in

$\dot{m}_B''$ = mass flux grains/h ft^2

For more information see ASHRAE Fundamentals chapter 21, "Moisture in Building Construction".

9.2 Convection of Mass

Convection of mass involves the mass transfer mechanism of molecular diffusion and bulk fluid motion. The fluid motion in the region adjacent to a mass transfer surface is laminar or turbulent depending on geometry and flow conditions. Convective mass transfer is analogous to convective heat transfer.

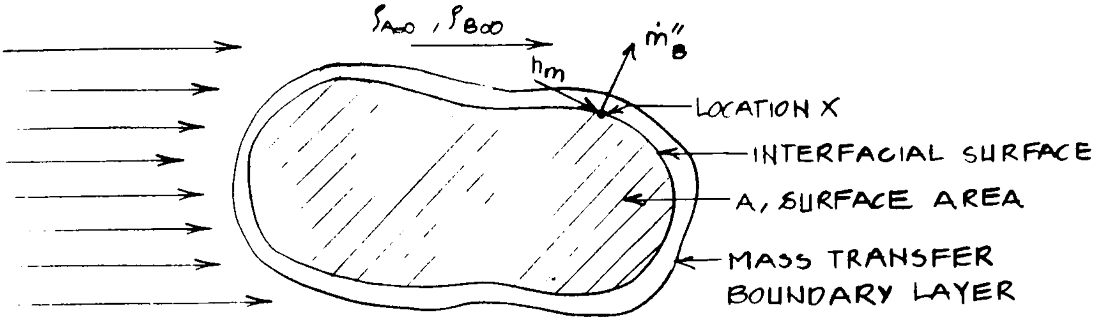


Figure 9-1, Mass Transfer for an External Flow

9.2.1 Mass Transfer Coefficients for External Flows

$$h_m = \dot{m}_B''/(\rho_{Bi} - \rho_{B\infty})$$

where: h_m = local external mass transfer coefficient, ft/hr
$\dot{m}_B''$ = mass flux of gas B from surface, lb_m/ft^2 h
ρ_{Bi} = concentration or density of gas B at the interface lb_m/ft^3
$\rho_{B\infty}$ = density of component B outside boundary layer, lbm/ft^3

If ρ_{Bi} and $\rho_{B\infty}$ are constant over the entire interface, then:

$$\dot{m}_B = \bar{h}_m A(\rho_{Bi} - \rho_{B\infty})$$

where:

$$\bar{h}_m = \frac{1}{A}\int_A h_m dA$$

9.2.2 Mass Transfer Coefficients for Internal Flows

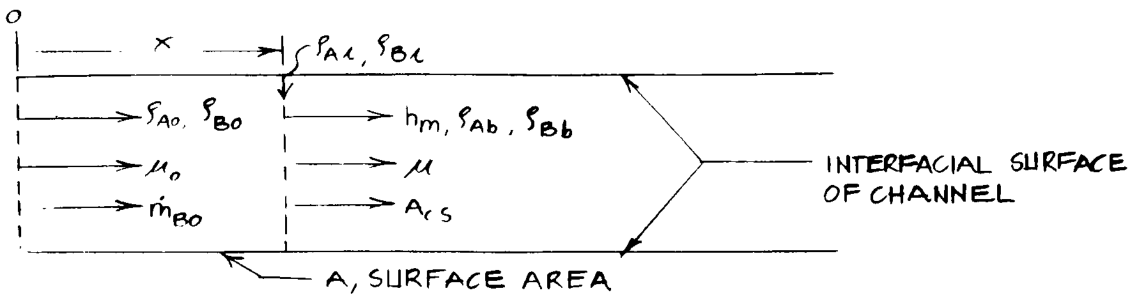


Figure 9-2, Mass Transfer for an Internal Flow

$$h_m = m_B''/(\rho_{Bi} - \rho_{Bb})$$

where: h_m = internal mass transfer coefficient, ft/h

$\dot{m}_B''$ = mass flux of gas B of the interface, lbm/ft^{2}h

ρ_{Bi} = density of B at surface lbm/ft^2

ρ_{Bb} = bulk density of gas B at location

$\bar{U}_b$ = average velocity of gas B at location x, ft/min

A_{cs} = cross-sectional area of channel at x, ft^2

U_B = velocity of component B in x direction, ft/min

ρ_B = density distribution of component B at x, lbm/ft^3

$$\bar{U}_B = \frac{1}{A_{cs}} \int_A U_B dA_{cs}$$

$$\rho_{Bb} = \frac{1}{U_b A_{cs}} \int_{A_{cs}} U_B \rho_B dA_{cs}$$

or

$$\rho_{Bb} = (m_{BO} + \int_A m_B'' \, dA) / (U_B A_{cs})$$

where: $\dot{m}_{BO}$ = mass flow rate of component B at x = 0 lbm/h

A = interfacial area of channel between x = 0 and x = x, ft^2

9.2.3 Eddy Diffusivity - Is the diffusion of mass in turbulent flow caused by random small mixing actions or eddy currents. Since the intensity of the turbulence is determined by the Reynold's number of the flow:

$$\dot{m}_B'' = \varepsilon_D(d\rho_B/dy)$$

where: ε_D = eddy diffusivity, ft^2/h

Because the data on eddy diffusivity are rare and difficult to obtain, the mass transfer coefficient is usually determined instead.

9.2.4 Application to Turbulent Flow

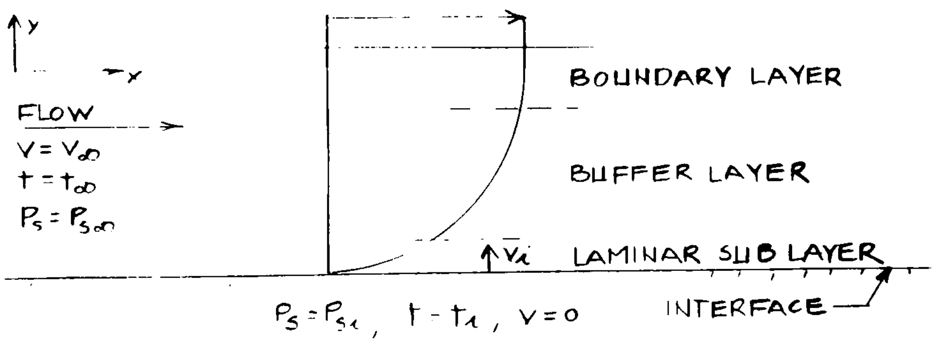


Figure 9-3, Mass Transfer for turbulent Flow Regions

The molecular mass diffusion is given by:

$$J_B = -D_v(d\rho_B/dy)_i$$

From Gibbs - Dalton Law the mass transfer from a wetted surface to the air stream is given by:

$$\dot{m}_B'' = -D_v(d\rho_B/dy)_i + \rho_{Bi}V_i$$

and

$$\rho_{Bi} = M_B P_{Bi}/R_g T_i \qquad \text{(Ideal gas)}$$

10. Psychrometrics

Introduction - Psychrometrics deals with determining thermodynamic properties of moist air and using these properties to analyze conditions and processes involving moist air.

The first section of this chapter discusses the properties that describe moist air. These properties are as follows: composition, molecular weight, and the gas constant. The second section discusses the fundamental parameters of moist air. These parameters are as follows: the Gibbs Dalton law for a mixture of perfect gases, humidity ratio, relative humidity, degree of saturation, dew point temperature, and enthalpy. The third section discusses the processes of moist air and the psychrometric chart. These processes are: adiabatic saturation, heating and cooling, cooling and dehumidifying, heating and humidifying, humidifying, and the adiabatic mixing of two streams of moist air.

10.1 Moist Air and the Standard Atmosphere

Approximation of the composition of dry air by volume fraction:

Gas	Volume Fraction	Molecular Weight
Nitrogen	.78084	28.016
Oxygen	.20948	32
Argon	.00934	39.944
Carbon Dioxide	.00031	44.010
Neon, Helium, Sulfur Dioxide, Hydrogen, and other minor gases	.00003	

10.1.1 The molecular weight of dry air is:

$$M_a = 28.965$$

and the gas constant Ra is:

$$Ra = \frac{\bar{R}}{M_a} = \frac{\bar{1}545.32}{28.965} = 53.352(\text{ft-lbf})/(\text{lbm - R})$$

where: R is $\bar{\text{t}}$he universal gas constant

$$\bar{R} = 1545.32^{(\text{ft - lbf})}/_{(\text{lbm mole} \bullet \text{R})}$$

10.1.2 the molecular weight of water is:

$$M_v = 18.105$$

and the gas constant for water vapor is:

$$R_v = \frac{1545.32}{18.015} = 85.78^{(\text{ft - lbf})}/_{(\text{lbm - R})}$$

10.1.3 the ASHRAE handbook's definition of the U.S. Standard Atmosphere:

a) acceleration due to gravity,

$$g = 32.174\ ^{\text{ft}}/_{\text{sec}^2}$$

b) temperature at sea level, 59°F

c) pressure at sea level, 29.921 in Hg

d) the atmosphere consists of dry air which behaves as a perfect gas

$$Pv = P/_{\underline{p}} = R_aT$$

where: $\underline{p}$ = Density
v = Specific Volume

10.2 Fundamental Parameters

Moist air, up to three atmospheres pressure, obeys the Ideal Gas Law with sufficient accuracy for engineering calculations.

10.2.1 The Gibbs Dalton Law for a mixture of perfect gases states that the mixture pressure is the sum of the partial pressure of its constituents.

$$P = P_1 + P_2 + P_3$$

for moist air:

$$P = P_{N_2} + P_{O_2} + P_{CO_2} + P_A + P_v$$

$$P_a = P_{N_2} + P_{O_2} + P_{CO_2} + P_A$$

so:

$$P = P_a + P_v$$

pressure of dry air +
pressure of water vapor

*Note: Each constituent in a mixture of perfect gases behaves as if the others were not present.

10.2.2 Humidity ratio, W (or "specific humidity") - the ratio of the mass of the water vapor m_v to the mass of the dry air m_a in the mixture.

$$W = {}^{m_v}/_{m_a}$$

10.2.3 Relative Humidity, ϕ - the ratio of the mole fraction of the water vapor, x_v in a mixture to the mole fraction x_s of the water vapor in a saturated mixture at the same temperature and pressure.

$$\phi = |X_v/X_s|_{t,p}$$

For a mixture of Ideal gases:

$$x_v = ({}^{P_v}/_{P}) \quad , \; x_s = ({}^{P_s}/_{P})$$

$$\phi = \frac{P_v}{P_s}$$

also:

$$\phi = |P_v/P_s|_{t,p}$$

10.2.4 Degree of Saturation, μ - the ratio of the humidity ratio W to the humidity ratio W_s saturated mixture at the same temperature and pressure.

$$\mu = |W/W_s|_{t,p}$$

10.2.5 Dew Point Temperature, t_d - is the temperature of saturated moist air at the same pressure and humidity ratio as the given mixture.

10.2.6 By using the perfect gas law we can derive a relationship between the relative humidity ϕ and the humidity ratio W:

$$m_v = P_v \frac{VM_v}{RT}$$

and:

$$m_a = P_a \frac{VM_a}{RT}$$

then:

$$W = \frac{M_v P_v}{M_a P_a}$$

For the air - water vapor mixture the above reduces to:

$$W = .6219(P_v/P_a)$$

also:

$$\phi = \frac{WP_a}{.6219 P_s}$$

10.2.7 Enthalpy (i) - the enthalpy of perfect gases is equal to the sum of the enthalpies of each constituent and is usually referenced to a unit mass of one constituent. For air - water vapor mixture, dry air is used as the reference because the amount of water vapor may vary during some processes.

$i = i_a + Wi_v$

If zero Fahrenheit or Celsius is selected as the reference state where the enthalpy of dry air is zero and the specific heats C_{pa} and C_{pv} are assumed constant, then:

$i_a = C_{pa}t$

$i_v = i_g + C_{pv}t$

where: i_g at 0°F of saturated water vapor is 1061.2 BTU/lbm and t is temperature °F

The total enthalpy then becomes:

$i = i_a + i_v$

$i = .240t + W(1061.2 + .444t)$ btu/lbma
where: C_{pa} and C_{pv} are .240 and .444 respectively.

<u>1.3 Adiabatic Saturation</u> - At a given pressure and temperature of an air-water vapor mixture one additional property is required to completely specify the state, except at saturation.

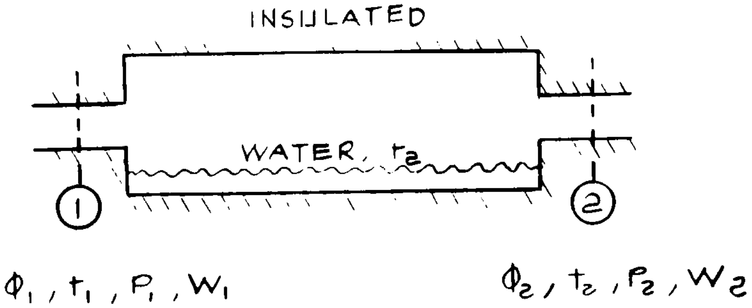


Figure 10-1, Adiabatic Saturation Device

The moist air leaving point 2 is saturated. The temperature t_2 is the adiabatic saturation temperature (or Thermodynamic Wet Bulb Temperature) t_{as}.

The energy balance equation yields:

$$i_{a1} + W_1 i_{v1} + (W_2 - W_1) i_w = W_2 i_{v2} + i_{a2}$$

or

$$W_1(i_{v1} - i_w) = C_{pa}(t_2 - t_1) + W_2(i_{v2} - i_w)$$

and

$$W_1(i_{v1} - i_w) = C_{pa}(t_2 - t_1) + W_2 i_{fg2}$$

where: i_w = enthalpy of liquid water
i_{fg} = enthalpy difference between saturated liquid and saturated vapor

10.3 The Psychrometric Chart

To ease engineering computations, a graphical representation of the properties of moist air was developed. This representation is known as the psychrometric chart and this chart is shown below.

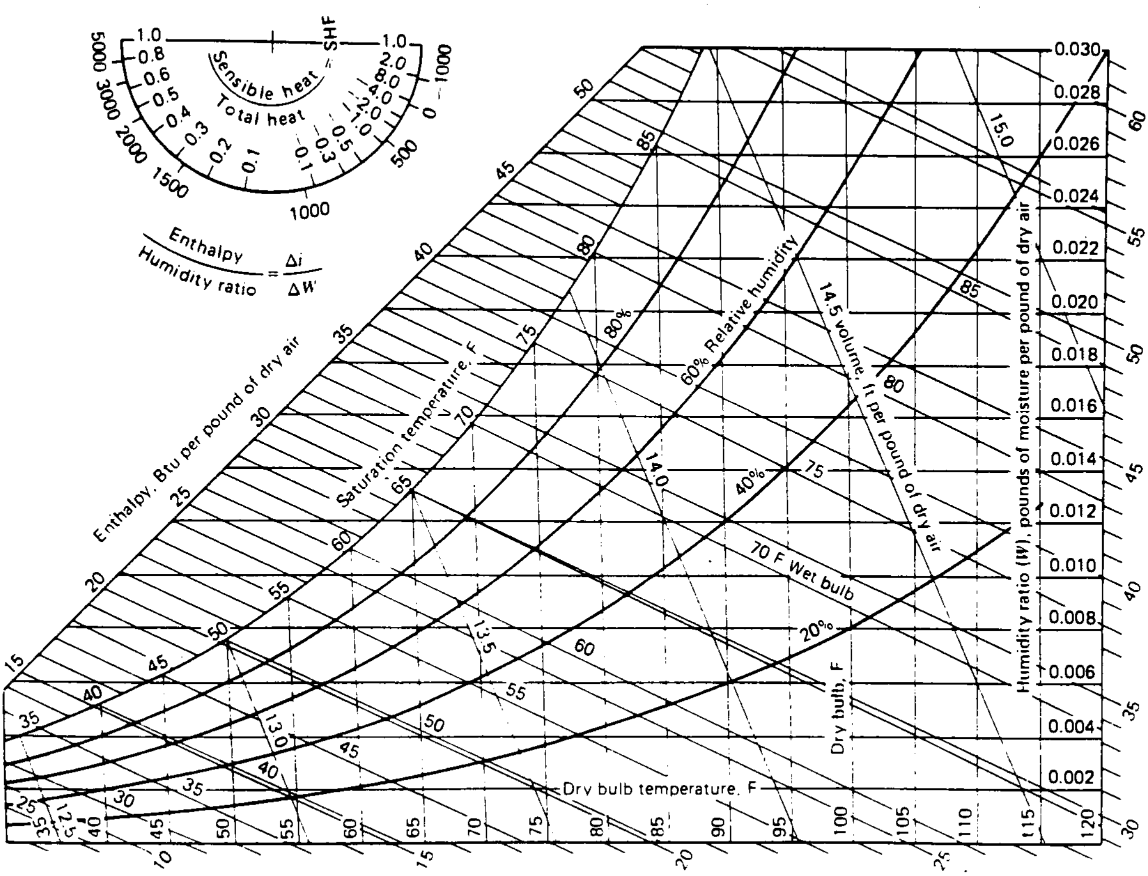


Figure 10-2, Psychrometric Chart
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

10.4 Classic Moist Air Processes

Two tools that will be used are the First Law of Thermodynamics (energy balance) and the law of Conservation of Mass.

10.4.1 Heating and Cooling of Moist Air - When air is heated or cooled without the loss or addition of moisture, the process yields a horizontal line on the psychrometric chart. This line represents a constant humidity ratio.

The steady state energy balance equation becomes:

$$\dot{m}_a i_1 + \dot{q} = \dot{m}_a i_2$$
$$i_1 = i_{a1} + W_1 i_{v1}$$
$$i_2 = i^{a2} + W_2 i_{v2}$$

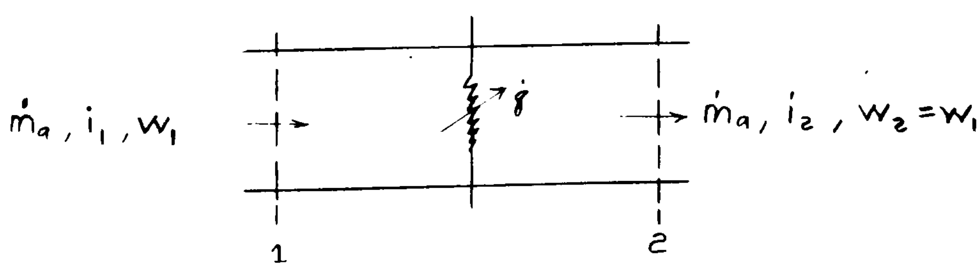


Figure 10-3, Heating or Cooling Device

$$\dot{q} = \dot{m}_a C_p(t_2 - t_1)$$

where: $C_p = C_{pa} + WC_{pv}$

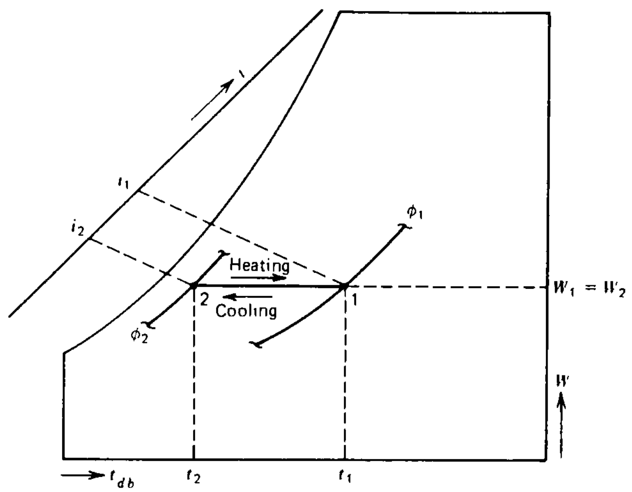


Figure 10-4, Sensible Heating and Cooling Process
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

10.4.2 Cooling and Dehumidifying of Moist Air - When moist air is cooled to a temperature below its dew point, some of the water vapor will condense and leave the air stream.

The steady state energy balance equation becomes:

$$\dot{m}_a i_1 = \dot{q} + \dot{m}_a i_2 + \dot{m}_w i_w$$

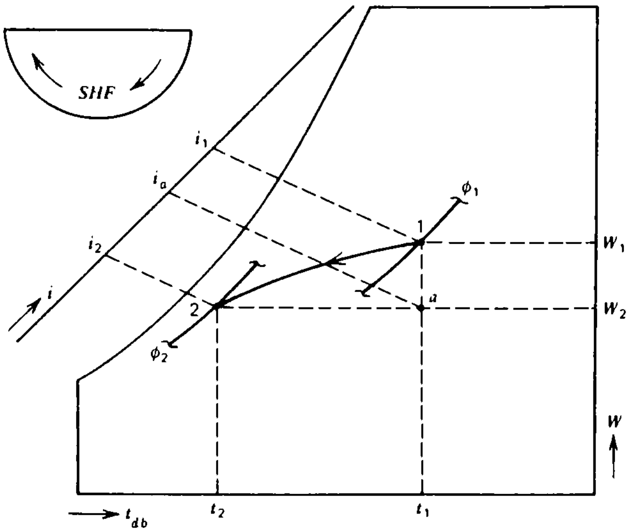


Figure 10-5, Cooling and Dehumidifying Process Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

For the steady state flow mass balance equation becomes:

$$\dot{m}_a W_1 = \dot{m}_w + \dot{m}_a W_2$$

Combining the energy balance equation and the mass balance equation yields:

$$\dot{q} = \dot{m}_a(i_1 - i_2) - \dot{m}_a(W_1 - W_2)i_w$$

$$\dot{q} = \dot{q}_s + \dot{q}_l$$

Sensible Heat Transfer

$$\dot{q}_s = \dot{m}_a c_p(t_1 - t_2) = \dot{m}_a(i_a - i_2)$$

Latent Heat Transfer

$$\dot{q}_l = \dot{m}_a(W_1 - W_2)i_{fg} = \dot{m}_a(i_1 - i_a)$$

The Sensible Heat Factor, SHF, is defined as:

$$SHF = \dot{q}_s/\dot{q}$$

The relationship between SHF and ΔW and Δt.

$$\dot{q}_s = \dot{m}_a C_p(t_2 - t_1)$$

$$\dot{q}_l = \dot{m}_a(W_2 - W_1)i_{fg}$$

$$SHF = \dot{q}_s/(\dot{q}_s+\dot{q}_l) = \frac{C_p(t_2 - t_1)}{C_p(t_2 - t_1) - (W_2 - W_1)i_{fg}}$$

or

$$\frac{(W_2 - W_1)}{(t_2 - t_1)} = \frac{1}{i_{fg}} (C_p) \frac{(SHF - 1)}{SHF}$$

10.4.3 Heating and Humidifying Moist Air

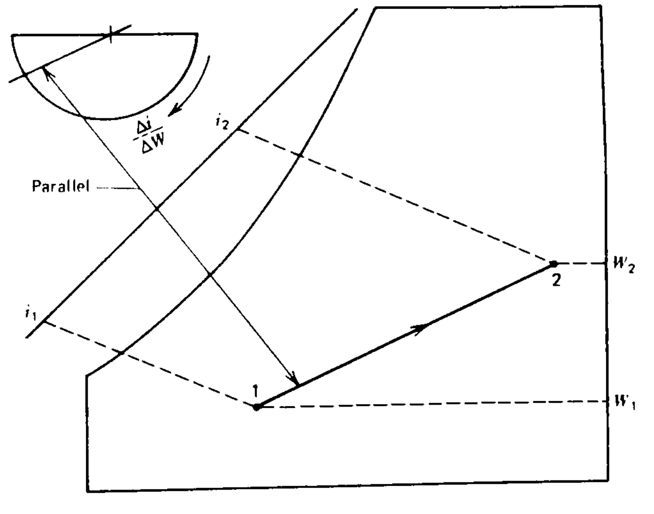


Figure 10-6, Heating and humidifying Process Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

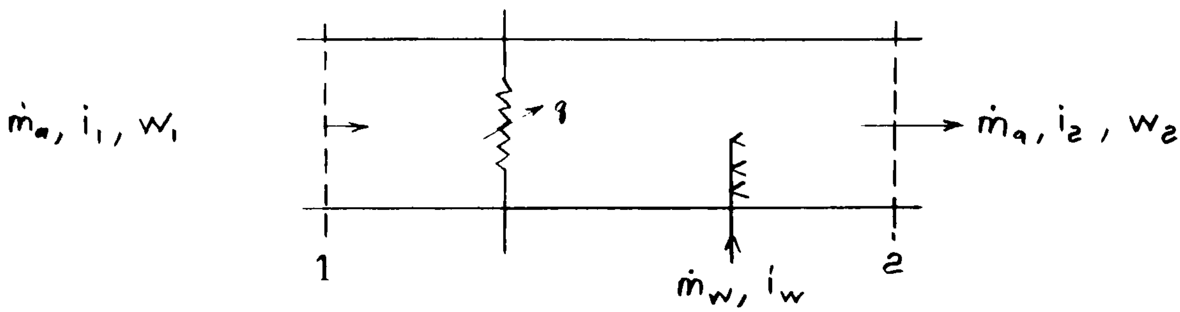


Figure 10-7, Heating and Humidifying Device

Steady State Energy Balance Equation

$$\dot{m}_a i_1 + \dot{q} + \dot{m}_w i_w = \dot{m}_a i_2$$

Mass Balance Equation on water yields:

$$\dot{m}_a W_1 + \dot{m}_w = \dot{m}_a W_2$$

Combining the above two equations yields:

$$\frac{i_2 - i_1}{W_2 - W_1} = \frac{\dot{q}}{\dot{m}_a(W_2 - W_1)} + i_w$$

or

$$\frac{\Delta i}{\Delta W} = \frac{\dot{q}}{\dot{m}_w} + i_w$$

10.5.4 Humidifying Moist Air - Moisture is frequently added to moist air without the addition of heat. From the Heating and Humidifying Moist Air section:

$$\frac{\Delta i}{\Delta W} = \frac{i_2 - i_1}{W_2 - W_1} = \frac{\dot{q}}{\dot{m}_w} + i_w$$

$$\dot{q}/\dot{m}_w = 0$$

The above equation becomes:

$$\Delta i/\Delta W = i_w$$

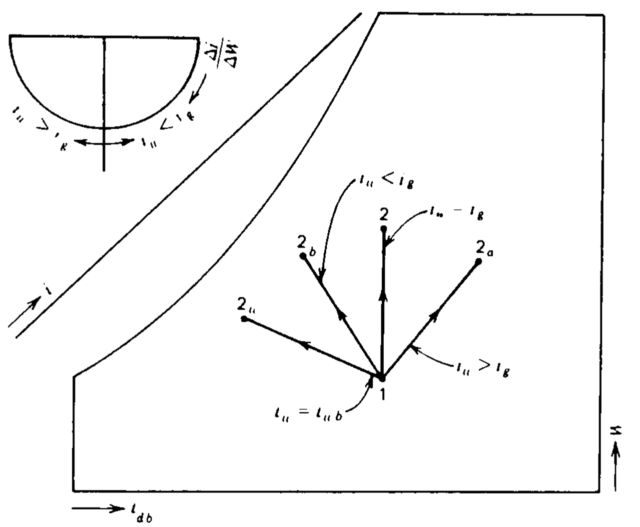


Figure 10-8, Humidification Process Without Heat Transfer;
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There are four different Humidifying processes.

10.4.4 Adiabatic Mixing of Two Streams of Moist Air (no heat gain or heat loss)

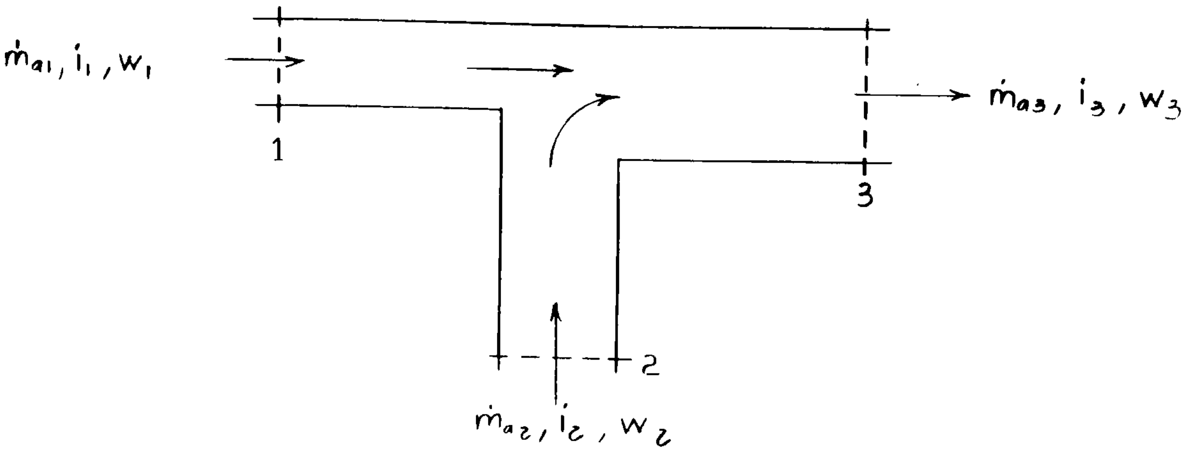

Figure 10-9, Adiabatic Mixing Device

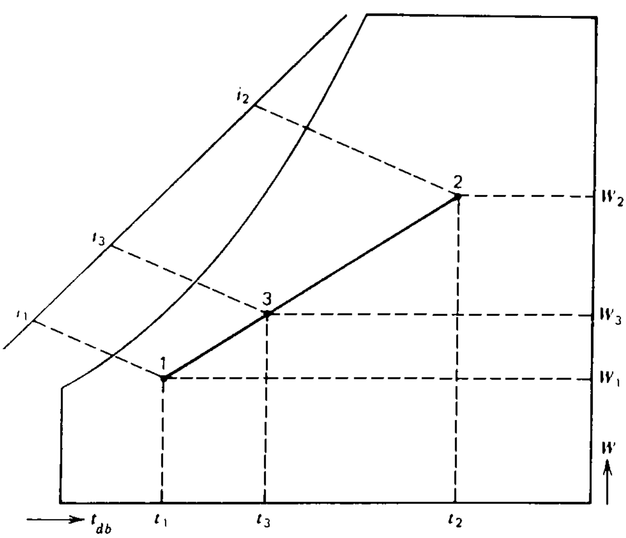

Figure 10-10, Adiabatic Mixing Process
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The steady state energy balance equation:

$$\dot{m}_{a1} i_1 + \dot{m}_{a2} i_2 = \dot{m}_{a3} i_3$$

The steady state mass balance equation for dry air:

$$\dot{m}_{a1} + \dot{m}_{a2} = \dot{m}_{a3}$$

The steady state mass balance equation for water vapor:

$$\dot{m}_{a1}W_1 + \dot{m}_{a2}W_2 = \dot{m}_{a3}W_3$$

Combining the above equations and eliminating m_{a3}, the relationships below are formed:

$$\frac{i_2 - i_3}{i_3 - i_1} = \frac{W_2 - W_3}{W_3 - W_1} = \frac{\dot{m}_{a1}}{\dot{m}_{a2}}$$

Point 3 must lie on a straight line between points 1 and 2. The length of the various line segments are proportional to the masses of dry air mixed.

$$\frac{\dot{m}_{a1}}{\dot{m}_{a2}} = \frac{\overline{3,2}}{\overline{1,3}} \qquad \frac{\dot{m}_{a1}}{\dot{m}_{a3}} = \frac{\overline{3,2}}{\overline{1,2}} \qquad \frac{\dot{m}_{a2}}{\dot{m}_{a3}} = \frac{\overline{1,3}}{\overline{1,2}}$$

10.4.5 Coil Bypass Factor - There is a point d which represents the apparatus dew point temperature of a cooling coil. The coil cannot cool all the air passing it to the coil temperature. This is an analogy to some of the air being brought to the coil temperature and the rest being bypassed around the coil. The fraction of air that is bypassed around the coils is b, known as the bypass factor. b is given by the following equation:

$$b = \frac{t_2 - t_d}{t_1 - t_d}$$

where: t_1 = coil entering temperature

t_2 = coil leaving temperature

t_d = apparatus dew point temperature

Then the heat removed by the cooling coil, $\dot{q}_{cs}$, is given by the following.

$$\dot{q}_{cs} = \dot{m}_{a1}c_p(t_1 - t_2)$$

or

$$\dot{q}_{cs} = \dot{m}_{a1}c_p(t_1 - t_d)(1-b)$$

10.5 The Goff and Grath Tables for Moist Air

Accurate thermodynamic properties of moist air were developed by Goff and Grath. These properties were calculated using statistical mechanics. To calculate properties at states other than saturation, the following equations are used.

$$v = v_a + \mu v_{as} + \overline{v}$$

$$i = i_a + \mu i_{as} + \overline{i}$$

$$s = s^a + \mu s_{as} + \overline{s}$$

where: $\mu = [W/W_s]_{t,p}$

$$\overline{v} = \frac{\mu(1 - \mu)A}{1 + 1.6078\mu W_s}$$

$$\overline{i} = \frac{\mu(1 - \mu)B}{1 + 1.6078\mu W_s}$$

$$\overline{s} = \frac{\mu(1 - \mu)C}{1 + 1.6078\mu W_s}$$

where A, B, C are given by the following:

Table 10-1, Constants A, B and C (standard atmospheric pressure)
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

For temperatures below 150°F v and i may be assumed to be zero.

Temperature		A		B		C	
F	C	ft^3/lbma	m^3/kga × 10^4	Btu/lbma	kJ/kga	Btu/(lbma-F) × 10^4	kJ/(kga-C) × 10^4
96	36	0.0018	1.124	0.0268	0.0623	0.4	1.675
112	45	0.0042	2.622	0.0650	0.151	0.9	3.768
128	54	0.0096	5.993	0.1439	0.335	2.0	8.374
144	63	0.0215	13.42	0.3149	0.733	4.2	17.58
160	71	0.0487	30.40	0.6969	1.62	9.1	38.10
176	80	0.1169	72.98	1.636	3.81	20.7	86.67
192	89	0.3363	209.9	4.608	10.7	56.7	237.4

Errors using the perfect gas relationship are given below:

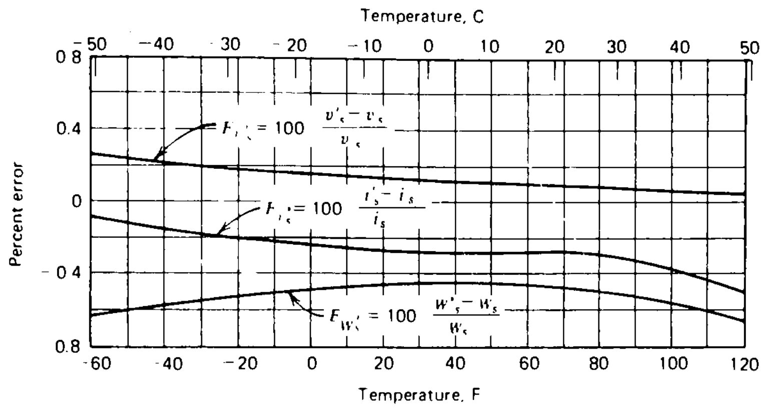


Figure 10-11, Error of Perfect Gas Relations in calculation of humidity ratio, enthalpy, and volume of standard air at standard atmospheric pressure
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

11. Comfort and Health

Introduction - Not everyone in a given space will be completely comfortable under one set of standard conditions. This is due to variation in a number of factors including clothing, age, sex, and activity level of each person.

The factors that influence comfort, in order of importance, are: 1) temperature, 2) radiation, 3) humidity, 4) air motion, and 5) the quality of the air with regard to odor, dust, and bacteria.

In this chapter the first section discusses the thermal interchanges between people and their environment. The second section discusses comfort conditions. The third section discusses indoor climate and health. The fourth discusses indoor air quality and the fifth describes air cleaning processes.

11.1 Thermal Interchanges Between People and Their Environments

The human body's temperature t_b varies with the balance between net heat produced by the body and that exchanged with the environment.

Heat generated in the body is mostly caused by metabolism, M. Also, the body can lose energy by doing work, W.

Heat can be transmitted into the body or away from the body by the following methods: Evaporation of body fluids (E), the exchange of radiation (R), and convection (C). The average body temperature during normal rest and exercise is 98.6 °F.

11.1.1 Heat Exchange Between the Body Surface and the Environment.

General Heat Balance Equation:

$$\Delta S = M - W - E + (R+C)$$

where: ΔS = time rate of heat storage; proportional to $\Delta T_b/\Delta\theta$, or the time rate of change in intrinsic body heat

M = rate of metabolism; proportional to oxygen consumption

W = mechanical work accomplished

E = rate of total evaporative heat loss, caused by the evaporation of body fluids

$R + C$ = dry heat exchange with environment; proportional to the difference between skin and environmental temperatures

R, C, and E are related to the body surface area. Most commonly used area is formulated by Dubois:

$$A_b = .202W^{.425}h^{.725}$$

where: A_b = Dubois surface area, m^2

W = weight, kg

h = height, m

For an average man,

$$h = 1.73m, W = 70kg, A_b = 1.8m^2$$

11.1.1.1 Metabolism (M) - in terms of O_2 consumption

$$M = (.23RQ + .77)(5.87)(\dot{V}_{O_2})(60/A_b),\ W/M^2$$

where: RQ = the respiration quotient, or ratio of $\dot{V}_{CO_2}$ exhaled to $\dot{V}_{O_2}$ inhaled; RQ may vary from .7 when using fat for metabolism at rest to 1.0 when using carbohydrates during heavy exercise.

$\dot{V}_{O_2}$ = oxygen consumption in liters per minute at standard condition (STPD?) of 0 ˚C, 101 KP_a

11.1.1.2 Work (W) - measured in kilopondmetres per minute (100kpm/min = 16.35 watts). A kilopondmetre is the energy required to raise a kilogram mass one meter in normal gravity.

11.1.1.3 Dry Heat Exchange (R+C), can be expressed in terms of the linear radiation heat coefficient h_r and an average convection heat transfer coefficient $\bar{h}_c$:

$$(R + C) = h_r(\bar{t}_{cl} - \bar{t}_r) + \bar{h}_c(\bar{t}_{cl} - t_a),\ W/M^2$$

where: $\bar{t}_{cl}$ = mean temperature of clothing surface

$\bar{t}_r$ = mean radiant temperature of environment

$\bar{t}_a$ = ambient air temperature

The above can be rearranged into:

$$(R + C) = h(\bar{t}_{cl} - \bar{t}_o),\ {}^W/_{M^2}$$

where: $t_o = (h_r\bar{t}_r + \bar{h}_c t_a)/(h_r + \bar{h}_c)$

$h = h_r + \bar{h}_c$

t_o is the average of the mean radiant and ambient temperatures weighted by their respective heat transfer coefficients. Heat loss from dry skin is given by:

$$(R + C) = h_{cle}(\bar{t}_{sk} - t_{cl}),\ {}^W/_{M^2}$$

or

$$(R + C) = h(\bar{t}_{sk} - t_o)F_{cle}$$

where: t_{sk} = skin temperature, average

h_{cle} = effective clothing conductance, ${}^W/_{M^2}$

I_a = insulation (resistance) of the ambient air = $1/h$, $m^2 \bullet {}^\circ C/W$

I_{cle} = insulation (resistance) of the clothing = $1/h_{cle}$, $m^2 \bullet {}^\circ C/W$

F_{cle} = effective thermal efficiency of clothing, dimensionless

$$F_{cle} = h_{cle}/(h + h_{cle}) = I_a/(I_a + I_{cl})$$

clothing insulation is often expressed in clo units (1 clo = .155 $m^2 \cdot {}^\circ C/W$)

11.1.1.4 Total Evaporative Heat Loss (E) (latent heat loss)

$$E = (60\Phi/A)(\Delta W/\Delta\theta),\ W^2/M^2$$

where: Φ = latent heat of vaporization at t_{sk} = 36°C

ΔW = change in body weight, kg

$\Delta\theta$ = change in time, min

The above coefficients can be found in the ASHRAE 1985 Fundamentals book, chapter 8.

11.2 Comfort Conditions

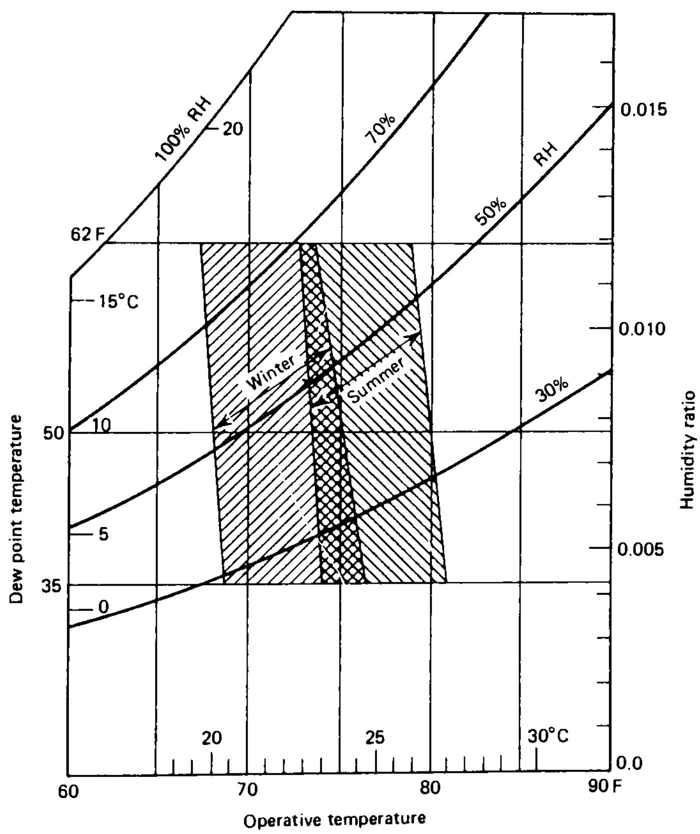


Figure 11-1, Acceptable Ranges of Operative Temperature and Humidity for Persons Clothed in Typical Summer and Winter clothing, at Light, Mainly Sedentary, Activity.
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

The comfort zones are appropriate for current seasonal clothing habits in the U.S.. Summer clothing is light slacks and short-sleeved shirts with a .5 clo value. Winter clothing is heavy slacks, a long-sleeved shirt, and a sweater or jacket with a .9 clo value. The temperatures are for sedentary and slight activity level for people. The winter zone is for airspeeds less than .15 m/s and the summer zone is for movements less than .25 m/s. The ASHRAE standard allows the summer comfort zone to extend above 26°C only if the average air movements is increased .275 m/s for each degree of Kelvin temperature increase to a maximum temperature of 28°C and air movements of .8 m/s.

11.3 Indoor Climate and Health

Many factors influence indoor environmental effects on humans. One major factor that influences the environment is the quality of air in it. There are many standards and codes which dictate the minimum quality of air in a space.

The quality of indoor air is a function of three major components: 1) the thermal climate, determined by wet bulb and dry bulb temperatures, relative humidity (rh), and air movement. 2) Atmospheric components, these included gasses and vapors normally present in the air, bioefluents, produced by biological process, volatile organic compounds (vocs), particle matter, inorganic gaseous combustion products, radon progeny and other radioactive substances, formaldehyde, pesticides, aerosols, viruses, bacteria and condensation nuclei. 3) Indoor air distribution or mixing pattern of ventilation air.

11.4 Control of Indoor Air Quality

This is done to improve indoor climate and health in an environment.

11.4.1 Ventilation - The requirement of outdoor air to ventilate indoor air. There are many standards and codes regarding the amount of outside air needed to ventilate a given space. These standards are usually depicted by ASHRAE. The codes are formulated by the town, state, and federal governments. Also coupled with ventilation is exhausting air from the space. Certain spaces require a minimum amount of air to be exhausted, like bathrooms, labs, hospitals, and certain work areas.

11.4.2 Washing - Air washing is used control temperature and humidity while also removing odorous vapors and particles. Air washing works in three ways. 1) through absorption of vapor from the air, 2) condensation of the vapor and 3) the direct removal of particles from the air.

11.4.3 Adsorption (not absorption) - is the adhesion of molecules to the surface of a solid, called the adsorbent. Types of adsorbents are: charcoal, zeolite, silica gel, mica, and others.

11.4.4 Odor Masking and Counteraction - is the technique of introducing a pleasant odor to cover an unpleasant odor. Caution should be taken because the masking odor might also be unpleasant.

11.5 The Cleaning of Air

Atmosphere dust is a complex mixture of smoke, mists, fumes, dry granular particles, fibers, living organisms such as mold spores, bacteria, and plant pollen that may cause diseases or allergic responses. Particles in the atmosphere range in size from less than 10^{-6}m up to the order of magnitude of the dimensions of leaves and insects. Different degrees of air cleanliness are required for various applications. There are three operating characteristics that distinguish various types of air cleaners. These are the efficiency, the air-flow resistance, and the dust-holding capacity.

11.5.1 Mechanisms of Particle Collection - Three broad categories of air cleaners:

A) fibrous media unit filter

B) renewable media filter

C) electronic air cleaners

12. Heat Transmission Through Building Structures

Introduction - This chapter discusses heat transmission through building structures by three modes: Conductance, Convection and Thermal Radiation.

12.1 Conductance

From the heat transfer section:

$$\dot{q} = -kA(dt/dx)$$

where: $\dot{q}$ = heat transfer rate, BTU/hr

k = thermal conductivity, BTU/(hr-ft-°F)

A = area normal to heat flow, ft^2

dt/dx = temperature gradient, °F/ft

$$\dot{q} = -kA(t_2 - t_1)/(x_2 - x_1)$$

or

$$\dot{q} = -(t_2 - t_1)/R'$$

where: $R' = \Delta x/kA$

and: $R = \Delta x/k$ (unit thermal resistance)

$c = 1/R$ (unit thermal conductance)

The next set of pages gives sets of thermal resistances and conductances for various materials These values can also be found in the ASHRAE Fundamentals book.

Table 12-1, Thermal Resistances for Some Steady state Conduction Problems
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

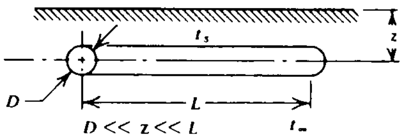
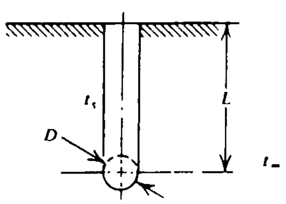
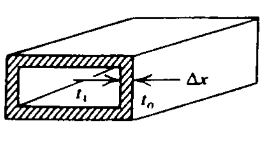
Number	System	Expressions for the Resistance R' $\dot{q} = \Delta t / R'$ (Btu/hr) or W
1.	Cylinder buried in a semi-infinite medium having a temperature at a great distance t_∞. The ground surface is assumed adiabatic. 	$R' = \dfrac{\left(\ln \dfrac{2L}{D}\right)\left(1 + \dfrac{\ln(L/2z)}{\ln(2L/D)}\right)}{2\pi k L}$
2.	A vertical cylinder placed in a semi-infinite medium having an adiabatic surface and temperature t_∞ at a great distance. 	$R' = \dfrac{\ln(4L/D)}{2\pi k L}$
3.	Conduction between inside and outside surfaces of a rectangular box having uniform inside and outside temperatures. Wall thickness Δx is less than any inside dimension. 	$R' = \dfrac{1}{k\left[\dfrac{A}{\Delta x} + 0.54\Sigma L + 1.2\Delta x\right]}$ ΣL = sum of all 12 inside lengths Δx = thickness of wall A = inside surface area

Table 12-2, Thermal Properties of Building and Insulating Materials at a Mean Temperature of 75°F (English Units)
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Material	Description	Density ρ ($\frac{lbm}{ft^3}$)	Thermal Conductivity k ($\frac{Btu\text{-}in.}{ft^2\text{-}hr\text{-}F}$)	Unit Conductance C ($\frac{Btu}{hr\text{-}ft^2\text{-}F}$)	Unit Resistance: Per Inch Thickness $1/k$ ($\frac{ft^2\text{-}hr\text{-}F}{Btu\text{-}in.}$)	Unit Resistance: For Thickness Listed $1/C$ ($\frac{hr\text{-}ft^2\text{-}F}{Btu}$)	Specific Heat ($\frac{Btu}{lbm\text{-}F}$)
BUILDING BOARD	Asbestos-cement board						
Boards, panels, subflooring, sheathing, woodbased panel products	¼ in. or 6 mm	120	—	16.5	—	0.06	0.24
	Gypsum or plasterboard						
	⅜ in. or 10 mm	50	—	3.10	—	0.32	0.26
	½ in. or 13 mm	50	—	2.22	—	0.45	—
	Plywood	34	0.80	—	1.25	—	0.29
	¼ in. or 6 mm	34	—	3.20	—	0.31	0.29
	⅜ in. or 10 mm	34	—	2.13	—	0.47	0.29
	½ in. or 13 mm	34	—	1.60	—	0.62	0.29
	¾ in. or 20 mm	34	—	1.07	—	0.93	0.29
	Insulating board, and sheathing, regular density						
	½ in. or 13 mm	18	—	0.76	—	1.32	0.31
	25/32 in. or 20 mm	18	—	0.49	—	2.06	0.31
	Hardboard, high density, standard tempered	63	1.00	—	1.00	—	0.32
	Particle board						
	Medium density	50	0.94	—	1.06	—	0.31
	Underlayment						
	⅝ in. or 16 mm	40	—	1.22	—	0.82	0.29
	Wood subfloor						
	¾ in. or 20 mm	—	—	1.06	—	0.94	0.33

Material	Description	Density ρ ($\frac{lbm}{ft^3}$)	Thermal Conductivity k ($\frac{Btu\text{-}in.}{ft^2\text{-}hr\text{-}F}$)	Unit Conductance C ($\frac{Btu}{hr\text{-}ft^2\text{-}F}$)	Unit Resistance: Per Inch Thickness $1/k$ ($\frac{ft^2\text{-}hr\text{-}F}{Btu\text{-}in.}$)	Unit Resistance: For Thickness Listed $1/C$ ($\frac{hr\text{-}ft^2\text{-}F}{Btu}$)	Specific Heat ($\frac{Btu}{lbm\text{-}F}$)
BUILDING PAPER	Vapor—permeable felt	—	—	16.7	—	0.06	—
	Vapor—seal, two layers of mopped 15 lb felt	—	—	8.35	—	0.12	—
FINISH FLOORING MATERIALS	Carpet and fibrous pad	—	—	0.48	—	2.08	0.34
	Carpet and rubber pad	—	—	0.81	—	1.23	0.33
	Tile—asphalt, linoleum, vinyl, or rubber	—	—	20.0	—	0.05	0.30
INSULATING MATERIALS	Mineral fiber—fibrous form processed from rock, slag, or glass						
Blanket and batt	Approximately 2–2¾ in. or 50–70 mm	0.3–2.0	—	0.143	—	7	0.17–0.23
	Approximately 3–3½ in. or 75–90 mm	0.3–2.0	—	0.091	—	11	0.17–0.23
	Approximately 5½–6½ in. or 135–165 mm	0.3–2.0	—	0.053	—	19	0.17–0.23
Board and slabs	Cellular glass	8.5	0.38	—	2.63	—	0.24
	Glass fiber, organic bonded	4–9	0.25	—	4.00	—	0.23

	Expanded polystyrene—molded beads	1.0	0.28	—	3.57	—	0.29
	Expanded polyurethane—R-11 expanded	1.5	0.16	—	6.25	—	0.38
	Mineral fiber with resin binder	15	0.29	—	3.45	—	0.17
LOOSE FILL	Mineral fiber—rock, slag, or glass						
	Approximately 3.75–5 in. or 75–125 mm	0.6–2.0	—		—	11	0.17
	Approximately 6.5–8.75 in or 165–222 mm	0.6–2.0	—		—	19	0.17
	Approximately 7.5–10 in. or 191–254 mm	—	—		—	22	0.17
	Approximately 7¼ in. or 185 mm	—	—		—	30	0.17
	Silica aerogel	7.6	0.17	—	5.88	—	—
	Vermiculite (expanded)	7–8	0.47	—	2.13	—	—
ROOF INSULATION	Preformed, for use above deck						
	Approximately ½ in or 13 mm	—	—	0.72	—	1.39	—
	Approximately 1 in. or 25 mm	—	—	0.36	—	2.78	—
	Approximately 2 in. or 50 mm	—	—	0.19	—	5.56	—
	Cellular glass	9	0.4	—	2.5	—	0.24
MASONRY MATERIALS Concretes	Lightweight aggregates including expanded shale, clay, or slate, expanded slags; cinders, pumice; vermiculite; also cellular concretes	200	5.2	—	0.19	—	—
		100	3.6	—	0.28	—	—
		80	2.5	—	0.40	—	—
		40	1.15	—	0.86	—	—
		20	0.70	—	1.43	—	—
	Sand and gravel or stone aggregate (not dried)	140	12.0	—	0.08	—	—

Material	Description	Density ρ	Thermal Conductivity k	Unit Conductance C	Unit Resistance: Per Inch Thickness $1/k$	Unit Resistance: For Thickness Listed $1/C$	Specific Heat
		$\frac{lbm}{ft^3}$	$\frac{Btu\text{-}in.}{ft^2\text{-}hr\text{-}F}$	$\frac{Btu}{hr\text{-}ft^2\text{-}F}$	$\frac{ft^2\text{-}hr\text{-}F}{Btu\text{-}in.}$	$\frac{hr\text{-}ft^2\text{-}F}{Btu}$	$\frac{Btu}{lbm\text{-}F}$
MASONRY UNITS	Brick, common	120	5.0	—	0.20	—	—
	Brick, face	130	9.0	—	0.11	—	—
	Concrete blocks, three-oval core—sand and gravel aggregate						
	4. in or 100 mm	—	—	1.4	—	0.71	—
	8 in. or 200 mm	—	—	0.9	—	1.11	—
	12 in. or 300 mm	—	—	0.78	—	1.28	—
	lightweight aggregate (expanded shale, clay slate or slag; pumice)						
	3 in. or 75 mm	—	—	0.79	—	1.27	—
	4 in. or 100 mm	—	—	0.67	—	1.50	—
	8 in. or 200 mm	—	—	0.50	—	2.00	—
	12 in. or 300 mm	—	—	0.44	—	2.27	—
PLASTERING MATERIALS	Cement plaster, sand, aggregate	116	5.0	—	0.20	—	—
	Gypsum plaster:						
	Lightweight aggregate						
	½ in. or 13 mm	45	—	3.12	—	0.32	—
	⅝ in. or 16 mm	45	—	2.67	—	0.39	—

	Lightweight aggregate on metal lath ¾ in. or 20 mm	—	—	2.13	—	0.47	—
ROOFING	Asbestos-cement shingles	120	—	4.76	—	0.21	—
	Asphalt roll roofing	70	—	6.50	—	0.15	—
	Asphalt shingles	70	—	2.27	—	0.44	—
	Built-up roofing ⅜ in. or 10 mm	70	—	3.00	—	0.33	0.35
	Slate, ½ in. or 13 mm	—	—	20.00	—	0.05	—
	Wood shingles—plain or plastic film faced	—	—	1.06	—	0.94	0.31
SIDING MATERIALS (on Flat Surface)	Shingles						
	Asbestos-cement	120	—	4.76	—	0.21	—
	Siding						
	Wood, drop, 1 in. or 25 mm	—	—	1.27	—	0.79	0.31
	Wood, plywood, ⅜ in. or 10 mm, lapped	—	—	1.59	—	0.59	0.29
	Aluminum or steel, over sheathing, hollowbacked	—	—	1.61	—	0.61	—
	Insulating board—backed nominal, ⅜ in or 10 mm	—	—	0.55	—	1.82	—
	Insulating board—backed nominal, ⅜ in. or 10 mm, foil-backed	—	—	0.34	—	2.96	—
	Architectural glass	—	—	10.00	—	0.10	—
WOODS	Maple, oak, and similar hardwoods	45	1.10	—	0.91	—	0.30
	Fir, pine, and similar softwoods	32	0.80	—	1.25	—	0.33
METALS	Aluminum (1100)	171	1536	—	0.00065	—	0.214
	Steel, mild	489	314	—	0.00318	—	0.120
	Steel, stainless	494	108	—	0.00926	—	0.109

*Abstracted by permission from *ASHRAE Handbook of Fundamentals*, 1977

For materials in series:

$$R'_T = ({}^{R_1}/_{A_1}) + ({}^{R_2}/_{A_2}) \;...$$

For materials in parallel:

$$R'_T = \frac{1}{\frac{A_1}{R_1} + \frac{A_2}{R_2} + \frac{A_3}{R_3} + \;...}$$

12.2 Convection

$$\dot{q} = hA(t - t_w)$$

where: $\dot{q}$ = heat transfer rate, BTU/hr

h = film coefficient, BTU/(hr-ft^2-°F)

t = bulk temperature of fluid, °F

t_w = wall temperature

$$\dot{q} = {}^{(t - t_w)}/_{R'}$$

where: R'= 1/hA (hr F)/BTU

r = 1/h =1/c (hr ft^2 F)/BTU

12.3 Thermal Radiation

$$\dot{q}_{12} = \frac{\sigma(T_1{}^4 - T_2{}^4)}{\frac{1 - \varepsilon_1}{A_1\varepsilon_1} + \frac{1}{A_1F_{12}} + \frac{1 - e_2}{A_2\varepsilon_2}}$$

where: σ = Boltzmann constant, .1713 x 10^{-8} BTU/(hr - ft^2 - R^4)

T = absolute temperature, R

ε = emittance

A = surface area, ft^2

F = configuration factor, function of geometry

The effective emittance E is given by:

$$\frac{1}{E} = \frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1$$

The following pages give convection coefficients and emittances for various material.

Table 12-3, Surface Unit Conductances and Unit Resistance for Air
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

		Surface Emittances											
		$\epsilon = 0.9$				$\epsilon = 0.2$				$\epsilon = 0.05$			
		h		*R*		*h*		*R*		*h*		*R*	
Position of Surface	**Direction of Heat Flow**	$\frac{Btu}{hr\text{-}ft^2\text{-}F}$	$\frac{W}{m^2\text{-}C}$	$\frac{hr\text{-}ft^2\text{-}F}{Btu}$	$\frac{m^2\text{-}C}{W}$	$\frac{Btu}{hr\text{-}ft^2\text{-}F}$	$\frac{W}{m^2\text{-}C}$	$\frac{hr\text{-}ft^2\text{-}F}{Btu}$	$\frac{m^2\text{-}C}{W}$	$\frac{Btu}{hr\text{-}ft^2\text{-}F}$	$\frac{W}{m^2\text{-}C}$	$\frac{hr\text{-}ft^2\text{-}F}{Btu}$	$\frac{m^2\text{-}C}{W}$
STILL AIR													
Horizontal	Upward	1.63	9.26	0.61	0.11	0.91	5.2	1.10	0.194	0.76	4.3	1.32	0.232
Sloping—45 degrees	Upward	1.60	0.09	0.62	0.11	0.88	5.0	1.14	0.200	0.73	4.1	1.37	0.241
Vertical	Horizontal	1.46	8.29	0.68	0.12	0.74	4.2	1.35	0.238	0.59	3.4	1.70	0.298
Sloping—45 degrees	Downward	1.32	7.50	0.76	0.13	0.60	3.4	1.67	0.294	0.45	2.6	2.22	0.391
Horizontal	Downward	1.08	6.13	0.92	0.16	0.37	2.1	2.70	0.476	0.22	1.3	4.55	0.800
MOVING AIR (Any Position)													
wind is 15 mph or 6.7 m/s (for winter)	Any	6.0	34.0	0.17	0.029								
Wind is $7\frac{1}{2}$ mph or 3.4 m/s (for summer)	Any	4.0	22.7	0.25	0.044								

*Adapted by permission from *ASHRAE Handbook of Fundamentals*, 1977
*Conductances are for surfaces of the stated emittance facing virtual blackbody surroundings at the same temperature as the ambient air Values are based on a surface-air temperature difference of 10 deg F and for surface temperature of 70 F

Table 12-4, Reflectance and Emittance of Various Surfaces and Effective Emittance of Air Spaces Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

			Effective Emittance E of Air Space	
Surface	Reflectance in Percent	Average Emittance ϵ	With One Surface Having Emittance ϵ and Other 0.90	With Both Surfaces of Emittance ϵ
Aluminum foil, bright	92–97	0.05	0.05	0.03
Aluminum sheet	80–95	0.12	0.12	0.06
Aluminum coated paper, polished	75-84	0.20	0.20	0.11
Steel, galvanized, bright	70–80	0.25	0.24	0.15
Aluminum paint	30–70	0.50	0.47	0.35
Building materials—wood, paper, glass, masonry, nonmetallic paints	5–15	0.90	0.82	0.82

*Adapted by permission from *ASHRAE Handbook of Fundamentals*, 1977

Table 12-5, Unit Thermal Resistance of a Plane 3/4 inch Air Space
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Position of Air Space	Direction of Heat Flow	Mean Air Temperature F	Mean Air Temperature C	Temperature Difference F	Temperature Difference C	$E = 0.05$ $\frac{ft^2\text{-hr-F}}{Btu}$	$E = 0.05$ $\frac{m^2\text{-C}}{W}$	$E = 0.2$ $\frac{ft^2\text{-hr-F}}{Btu}$	$E = 0.2$ $\frac{m^2\text{-C}}{W}$	$E = 0.82$ $\frac{ft^2\text{-hr-F}}{Btu}$	$E = 0.82$ $\frac{m^2\text{-C}}{W}$
Horizontal	Up	90	32	10	6	2.22	0.391	1.61	0.284	0.75	0.132
		50	10	30	17	1.66	0.292	1.35	0.238	0.77	0.136
		50	10	10	6	2.21	0.389	1.70	0.299	0.87	0.153
		0	−18	20	11	1.79	0.315	1.52	0.268	0.93	0.164
		0	−18	10	6	2.16	0.380	1.78	0.313	1.02	0.180
45° Slope	Up	90	32	10	6	2.78	0.490	1.88	0.331	0.81	0.143
		50	10	30	17	1.92	0.338	1.52	0.268	0.82	0.144
		50	10	10	6	2.75	0.484	2.00	0.352	0.94	0.166
		0	−18	20	11	2.07	0.364	1.72	0.303	1.00	0.176
		0	−18	10	6	2.62	0.461	2.08	0.366	1.12	0.197
Vertical	Horizontal	90	32	10	6	3.24	0.571	2.08	0.366	0.84	0.148
		50	10	30	17	2.77	0.488	2.01	0.354	0.94	0.166
		50	10	10	6	3.46	0.609	2.35	0.414	1.01	0.178
		0	−18	20	11	3.02	0.532	2.32	0.408	1.18	0.210
		0	−18	10	6	3.59	0.632	2.64	0.465	1.26	0.222
45° Slope	Down	90	32	10	6	3.27	0.576	2.10	0.370	0.84	0.148
		50	10	30	17	3.23	0.569	2.24	0.394	0.99	0.174
		50	10	10	6	3.57	0.629	2.40	0.423	1.02	0.180
		0	−18	20	11	3.57	0.629	2.63	0.463	1.26	0.222
		0	−18	10	6	3.91	0.689	2.81	0.495	1.30	0.229

[a] Adapted by permission from *ASHRAE Handbook of Fundamentals*, 1977

[b] Effective emittance of the space E is given by Eq 14-11) Credit for an air space resistance value cannot be taken more than once and only for the boundary conditions established. Resistances of horizontal spaces with heat flow downward are substantially independent of temperature difference.

Table 12-6, Unit Thermal Resistance of a Plane 3.5 inch Air Space
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Position of Air Space	Direction of Heat Flow	Mean Air Temperature F	C	Temperature Difference F	C	$E = 0.05$ $\frac{ft^2\text{-hr-F}}{Btu}$	$\frac{m^2\text{-C}}{W}$	$E = 0.2$ $\frac{ft^2\text{-hr-F}}{Btu}$	$\frac{m^2\text{-C}}{W}$	$E = 0.82$ $\frac{ft^2\text{-hr-F}}{Btu}$	$\frac{m^2\text{-C}}{W}$
Horizontal	Up	90	32	10	6	2.66	0.468	1.83	0.322	0.80	0.141
		50	10	30	17	2.01	0.354	1.58	0.278	0.84	0.148
		50	10	10	6	2.66	0.468	1.95	0.343	0.93	0.164
		0	−18	20	11	2.18	0.384	1.79	0.315	1.03	0.181
		0	−18	10	6	2.62	0.461	2.07	0.365	1.12	0.197
45° Slope	Up	90	32	10	6	2.96	0.521	1.97	0.347	0.82	0.144
		50	10	30	17	2.17	0.382	1.67	0.294	0.86	0.151
		50	10	10	6	2.95	0.520	2.10	0.370	0.96	0.169
		0	−18	20	11	2.35	0.414	1.90	0.335	1.06	0.187
		0	−18	10	6	2.87	0.505	2.23	0.393	1.16	0.204
Vertical	Horizontal	90	32	10	6	3.40	0.598	2.15	0.479	0.85	0.150
		50	10	30	17	2.55	0.449	1.89	0.333	0.91	0.160
		50	10	10	6	3.40	0.598	2.32	0.409	1.01	0.178
		0	−18	20	11	2.78	0.490	2.17	0.382	1.14	0.201
		0	−18	10	6	3.33	0.586	2.50	0.440	1.23	0.217
45° Slope	Down	90	32	10	6	4.33	0.763	2.49	0.438	0.90	0.158
		50	10	30	17	3.30	0.581	2.28	0.402	1.00	0.176
		50	10	10	6	4.36	0.768	2.73	0.481	1.08	0.190
		0	−18	20	11	3.63	0.639	2.66	0.468	1.27	0.224
		0	−18	10	6	4.32	0.761	3.02	0.532	1.34	0.236

•Adapted by permission from *ASHRAE Handbook of Fundamentals*, 1977.
ᵃEffective emittance of the space E is given by Eq. (4-11) Credit for an air space resistance value cannot be taken more than once and only for the boundary conditions established. Resistances of horizontal spaces with heat flow downward are substantially independent of temperature difference

Table 12-7, Unit Thermal Resistance of Plane Horizontal Air Spaces With Heat Flow Downward, Temperature Difference 10 Deg F
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Air Space				E = 0.05		E = 0.2		E = 0.82	
Thickness		Mean Temperature		$\frac{ft^2\text{-hr-F}}{Btu}$	$\frac{m^2\text{-C}}{W}$	$\frac{ft^2\text{-hr-F}}{Btu}$	$\frac{m^2\text{-C}}{W}$	$\frac{ft^2\text{-hr-F}}{Btu}$	$\frac{m^2\text{-C}}{W}$
Inch	mm	F	C						
¾	20	90	32	3.29	0.579	2.10	0.370	0.85	0.150
		50	10	3.59	0.632	2.41	0.424	1.02	0.180
		0	−18	4.02	0.708	2.87	0.505	1.31	0.231
1½	38	90	32	5.35	0.942	2.79	0.491	0.94	0.166
		50	10	5.90	1.04	3.27	0.576	1.15	0.203
		0	−18	6.66	1.17	4.00	0.707	1.51	0.266
3½	89	90	32	8.19	1.44	3.41	0.601	1.00	0.176
		50	10	9.27	1.63	4.09	0.720	1.24	0.218
		0	−18	10.32	1.82	5.08	0.895	1.64	0.289

*Adapted by permission from *ASHRAE Handbook of Fundamentals*, 1977

12.4 Overall Heat Transfer Coefficient (U)

$$U = \frac{1}{R'A} = \frac{1}{R}, \quad BTU/(hr - ft^2 - F)$$

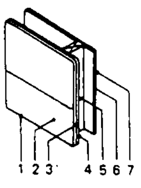
The heat transfer rate is given by:

$$\dot{q} = UA\Delta t$$

Tabulated heat transfer coefficients are given in the following pages and can also be found in the ASHRAE Fundamentals book.

Table 12-8, Coefficients of Transmission U of Frame Walls, BTU/(hr ft^2 F)
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Replace Air Space with 3.5-in. R-11 Blanket Insulation (New Item 4)



	1		2	
	Resistance (R)			
Construction	**Between Framing**	**At Framing**	**Between Framing**	**At Framing**
1. Outside surface (15 mph wind)	0.17	0.17	0.17	0.17
2. Siding, wood, 0.5 in. × 8 in. lapped (average)	0.81	0.81	0.81	0.81
3. Sheathing, 0.5-in. asphalt impregnated	1.32	1.32	1.32	1.32
4. Nonreflective air space, 3.5 in. (50 F mean; 10 deg F temperature difference)	1.01	—	11.00	—
5. Nominal 2-in. × 4-in. wood stud	—	4.38	—	4.38
6. Gypsum wallboard, 0.5 in.	0.45	0.45	0.45	0.45
7. Inside surface (still air)	0.68	0.68	0.68	0.68
Total Thermal Resistance (R)	$R_i = 4.44$	$R_s = 7.81$	$R_i = 14.43$	$R_s = 7.81$

Construction No. 1: $U_i = 1/4.44 = 0.225$; $U_s = 1/7.81 = 0.128$. With 20% framing (typical of 2-in. × 4-in. studs @ 16-in. o.c.), $U_{av} = 0.8\,(0.225) + 0.2\,(0.128) = 0.206$

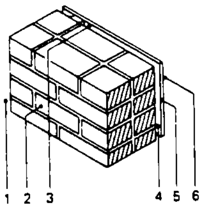
Construction No. 2: $U_i = 1/14.43 = 0.069$; $U_s = 0.128$. With framing unchanged, $U_{av} = 0.8(0.069) + 0.2(0.128) = 0.081$

[a] Adapted by permission from *ASHRAE Handbook of Fundamentals*, 1977

[b] U factor may be converted to W/(m^2-C) by multiplying by 5.68

Table 12-9, Coefficients of Transmission U of Solid Masonry Walls, BTU/(hr ft^2 F)
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Replace Furring Strips and Air Space with 1-in. Extruded Polystyrene (New Item 4)



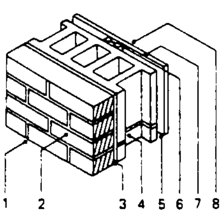
	1 Resistance (R)		2
Construction	Between Furring	At Furring	
1. Outside surface (15 mph wind)	0.17	0.17	0.17
2. Common brick, 8 in.	1.60	1.60	1.60
3. Nominal 1-in. × 3-in. vertical furring	—	0.94	—
4. Nonreflective air space, 0.75 in. (50 F mean; 10 deg F temperature difference)	1.01	—	5.00
5. Gypsum wallboard, 0.5 in.	0.45	0.45	0.45
6. Inside surface (still air)	0.68	0.68	0.68
Total Thermal Resistance (R)	R_i = 3.91	R_f = 3.84	R_i = 7.90 = R_f

Construction No. 1: U_i = 1/3.91 = 0.256; U_f = 1/3.84 = 0.260. With 20% framing (typical of 1-in × 3-in vertical furring on masonry @ 16-in. o.c.), U_{av} = 0.8 (0.256) + 0.2 (0.260) = 0.257
Construction No 2: U_i = U_f = U_{av} = 1/7.90 = 0.127

*U factor may be converted to W/(m²-C) by multiplying by 5.68
*Adapted by permission from *ASHRAE Handbook of Fundamentals*, 1977

Table 12-10, Coefficients of Transmission U of Masonry Cavity Walls, BTU/(hr ft^2 F)
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Replace Cinder Aggregate Block with 6-in. Light-weight Aggregate Block with Cores Filled (New Item 4)



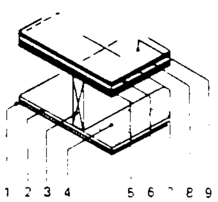
	1		2	
	Resistance (R)			
Construction	Between Furring	At Furring	Between Furring	At Furring
1. Outside surface (15 mph wind)	0.17	0.17	0.17	0.17
2. Face brick, 4 in.	0.44	0.44	0.44	0.44
3. Cement mortar, 0.5 in.	0.10	0.10	0.10	0.10
4. Concrete block, cinder aggregate, 8 in.	1.72	1.72	2.99	2.99
5. Reflective air space, 0.75 in (50 F mean; 30 deg F temperature difference)	2.77	—	2.77	—
6. Nominal 1-in. × 3-in. vertical furring	—	0.94	—	0.94
7. Gypsum wallboard, 0.5 in., foil backed	0.45	0.45	0.45	0.45
8. Inside surface (still air)	0.68	0.68	0.68	0.68
Total Thermal Resistance (R)	R_i = 6.33	R_f = 4.50	R_i = 7.60	R_f = 5.77

Construction No. 1: U_i = 1/6.33 = 0.158; U_f = 1/4.50 = 0.222. With 20% framing (typical of 1-in. × 3-in. vertical furring on masonry @ 16-in. o.c.), U_{av} = 0.8 (0.158) + 0.2 (0.222) = 0.171
Construction No. 2: U_i = 1/7.60 = 0.132, U_f = 1/5.77 = 0.173 With framing unchanged, U_{av} = 0.8(0.132) + 0.2(0.173) = 1.40

*U factor may be converted to W/(m²-C) by multiplying by 5.68
*Adapted by permission from *ASHRAE Handbook of Fundamentals*, 1977

Table 12-11, Coefficients of Transmission U of Ceilings and Floors, BTU/(hr ft^2 F)
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Assume Unheated Attic Space above Heated Room with Heat Flow Up—Remove Tile, Felt, Plywood, Subfloor and Air Space—Replace with R-19 Blanket Insulation (New Item 4)



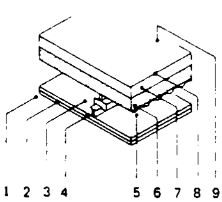
Heated Room Below Unheated Space Construction (Heat Flow Up)	1 Resistance (R) Between Floor Joists	At Floor Joist	2 Between Floor Joists	At Floor Joists
1. Bottom surface (still air)	0.61	0.61	0.61	0.61
2. Metal lath and lightweight aggregate, plaster, 0.75 in.	0.47	0.47	0.47	0.47
3. Nominal 2-in. × 8-in. floor joist	—	9.06	—	9.06
4. Nonreflective airspace, 7.25-in	0.93[a]	—	19.00	—
5. Wood subfloor, 0.75 in.	0.94	0.94	—	—
6. Plywood, 0.625 in.	0.78	0.78	—	—
7. Felt building membrane	0.06	0.06	—	—
8. Resilient tile	0.05	0.05	—	—
9. Top surface (still air)	0.61	0.61	0.61	0.61
Total Thermal Resistance (R)	R_i = 4.45	R_s = 12.58	R_i = 20.69	R_s = 10.75

Construction No. 1 U_i = 1/4.45 = 0.225; U_s = 1/12.58 = 0.079 With 10% framing (typical of 2-in. joists 16-in o.c.), U_{av} = 0.9 (0.225) + 0.1 (0.079) = 0.210
Construction No. 2 U_i = 1/20.69 = 0.048; U_s = 1/10.75 = 0.093. With framing unchanged, U_{av} = 0.9 (0.048) + 0.1 (0.093) = 0.053

[a]Use largest air space (3.5 in.) value shown in Table 4-5
[b]U factor may be converted to W/(m²-C) by multiplying by 5.68
[c]Adapted by permission from *ASHRAE Handbook of Fundamentals*, 1977

Table 12-12, Coefficients of Transmission U of Flat Built Up Roofs, BTU/(hr ft^2 F)
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Add Rigid Roof Deck Insulation, C = 0.24 (R = 1/C) (New Item 7)



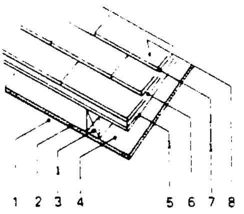
Construction (Heat Flow Up)	1 Resistance (R)	2
1. Inside surface (still air)	0.61	0.61
2. Metal lath and lightweight aggregate plaster, 0.75 in.	0.47	0.47
3. Nonreflective air space, greater than 3.5 in. (50 F mean; 10 deg F temperature difference)	0.93[a]	0.93[a]
4. Metal ceiling suspension system with metal hanger rods	0[b]	0[b]
5. Corrugated metal deck	0	0
6. Concrete slab, lightweight aggregate, 2 in.	2.22	2.22
7. Rigid roof deck insulation (none)	—	4.17
8. Built-up roofing, 0.375 in.	0.33	0.33
9. Outside surface (15 mph wind)	0.17	0.17
Total Thermal Resistance (R)	4.73	8.90

Construction No. 1: U_{av} = 1/4.73 = 0.211
Construction No. 2: U_{av} = 1/8.90 = 0.112

[a]Use largest air space (3.5 in.) value shown in Table 4-5
[b]Area of hanger rods is negligible in relation to ceiling area
U factor may be converted to W/(m²-C) by multiplying by 5.68.
[c]Adapted by permission (from *ASHRAE Handbook of Fundamentals* 1977

Table 12-13, Coefficients of Transmission U of Pitched Roofs, BTU/(hr ft^2 F)
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Find U_o for same Construction 2 with Heat Flow Down (Summer Conditions)



Construction 1 (Heat Flow Up) (Reflective Air Space)	1 Resistance (R) Between Rafters	1 At Rafters	2 Between Rafters	2 At Rafters
1. Inside surface (still air)	0.62	0.62	0.76	0.76
2. Gypsum wallboard 0.5 in., foil backed	0.45	0.45	0.45	0.45
3. Nominal 2-in. × 4-in. ceiling rafter	—	4.38	—	4.38
4. 45 deg slope reflective air space, 3.5 in. (50 F mean, 30 deg F temperature difference)	2.17	—	4.33	—
5. Plywood sheathing, 0.625 in.	0.78	0.78	0.78	0.78
6. Felt building membrane	0.06	0.06	0.06	0.06
7. Asphalt shingle roofing	0.44	0.44	0.44	0.44
8. Outside surface (15 mph wind)	0.17	0.17	0.25	0.25
Total Thermal Resistance (R)	$R = 4.69$	$R = 6.90$	$R = 7.07$	$R = 7.12$

Construction No. 1: $U_i = 1/4.69 = 0.213$; $U_s = 1/6.90 = 0.145$. With 10% framing (typical of 2-in rafters @ 16-in. o.c.), $U_o = 0.9(0.213) + 0.1(0.145) = 0.206$

Construction No. 2: $U_i = 1/7.07 = 0.141$; $U_s = 1/7.12 = 0.140$. With framing unchanged, $U_o = 0.9(0.141) + 0.1(0.140) = 0.141$

[a]Heat flow upward: Roof pitch is 45 degrees

[b]U factor may be converted to W/(m^2-C) by multiplying 5.68

[c]Adapted by permission from *ASHRAE Handbook of Fundamentals* 1977

Table 12-14, Overall Coefficients of Heat Transmission (U-Factor) of Windows and Skylights, BTU/(hr ft^2 F) Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

	Exterior Vertical Panels				Exterior Horizontal Panels (Skylights)	
	Summer[m]		Winter[l]			
Description	No Indoor Shade	Indoor Shade[n]	No Indoor Shade	Indoor Shade[n]	Summer[j]	Winter[i]
Flat Glass[c]						
Single Glass	1.04	0.81	1.10	0.83	0.83	1.23
Insulating Glass, Double[d]						
3/16 in. air space[e]	0.65	0.58	0.62	0.52	0.57	0.70
1/4 in. air space[e]	0.61	0.55	0.58	0.48	0.54	0.65
1/2 in. air space[f]	0.56	0.52	0.49	0.42	0.49	0.58
1/2 in. air space, low emittance coating						
e = 0.20	0.38	0.37	0.32	0.30	0.36	0.4[illegible]
e = 0.40	0.45	0.44	0.38	0.35	0.42	0.52
e = 0.60	0.51	0.48	0.43	0.38	0.46	0.56
Insulating Glass, Triple[d]						
1/4 in. air space[e]	0.44	0.40	0.39	0.31		
1/2 in. air space[f]	0.39	0.36	0.31	0.26		
Storm Windows						
1 in. to 4 in. air spaces[e]	0.50	0.48	0.50	0.42		
Plastic bubbles[k]						
Single Walled					0.80	1.15
Double Walled					0.46	0.70

[a] Reprinted by permission from *ASHRAE Handbook of Fundamentals* 1977
[b] See Table 4-6a for adjustments for various windows and sliding patio doors
[c] Emittance of uncoated glass surface = 0.84
[d] Double and triple refer to number of lights of glass
[e] 0.125-in. glass
[f] 0.25-in. glass
[g] Coating on either glass surface facing air space; all other glass surfaces uncoated
[h] Window design 0.25-in. glass, 0.125-in. glass, 0.25-in. glass
[i] For heat flow up
[j] For heat flow down
[k] Based on area of opening, not total surface area
[l] 15 mph outdoor air velocity, 0 F outdoor air, 70 F inside air temp natural convection
[m] 7.5 mph outdoor air velocity, 89 F outdoor air, 75 F inside air natural convection, solar radiation 248.3 Btu/(hr ft^2)
[n] Values apply to tightly closed venetian and vertical blinds, draperies, and roller shades

Table 12-15, Adjustment Factors for Coefficients U of Table 12-14, BTU/(hr ft^2 F)
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Description	Single Glass	Double or Triple Glass	Storm Windows
Windows			
All glass	1.0	1.0	1.0
Wood sash—80% glass	0.9	0.95	0.90
Metal sash—80% glass	1.0	1.20	1.20
Sliding glass doors			
Metal frame	1.0	1.10	
Wood frame	0.95	1.0	

*Reprinted by permission from *ASHRAE Handbook of Fundamentals*, 1977

Table 12-16, Coefficients of Transmission U of Slab Doors, BTU/(hr ft^2 F)
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

	Solid Wood No Storm Door		Storm Door[a] Wood		Storm Door[a] Metal		Summer No Storm Door	
Nominal Thickness	Btu/hr-ft²-F	W/m²-C	Btu/hr-ft²-F	W/m²-C	Btu/hr-ft²-F	W/m²-C	Btu/hr-ft²-F	W/m²-C
1 in. or 25 mm	0.64	3.63	0.30	1.70	0.39	2.21	0.61	3.46
1¼ in. or 30 mm	0.55	3.12	0.28	1.59	0.34	1.93	0.53	3.01
1½ in. or 38 mm	0.49	2.78	0.27	1.53	0.33	1.87	0.47	2.67
2 in. or 50 mm	0.43	2.44	0.24	1.36	0.29	1.65	0.42	2.38
	Steel Door Heavy Type							
1¾ in. or 44 mm								
Mineral fiber core	0.59	3.35					0.58	3.29
Solid urethane foam core	0 40	2.27					0.39	2.21
Solid polystyrene core with thermal break	0.47	2.67					0 46	2.61

[a]Approximately 50 percent glass for wood doors; values for metal doors are indepenoent of glass percentage
*Adapted by permission from *ASHRAE Handbook of Fundamentals*, 1977

Table 12-17, Heat Loss Through Basement Floors (for Floors More Than 3 ft Below Grade)
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Depth of Floor Below Grade		Narrowest Width of House							
		20 ft (6.1 m)		24 ft (7.3 m)		28 ft (8.5 m)		32 ft (9.8 m)	
ft	m	Btu/(hr-ft^2-F)	W/(m^2-C)	Btu/(hr-ft^2-F)	W/(m^2-C)	Btu/(hr-ft^2-F)	W/(m^2-C)	Btu/(hr-ft^2-F)	W/(m^2-C)
4	1.22	0.035	0.198	0.032	0.182	0.027	0.153	0.024	0.136
5	1.52	0.032	0.182	0.029	0.165	0.026	0.148	0.023	0.131
6	1.83	0.030	0.170	0.027	0.153	0.025	0.142	0.022	0.125
7	2.13	0.029	0.165	0.026	0.148	0.023	0.131	0.021	[illegible]

*Adapted by permission from *ASHRAE GRP 158 Cooling and Heating Load Calculation Manual* 1979
*For a depth below grade of 3 ft or less, treat as a slab on grade

Table 12-18, Heat Loss Rate for Below-Grade Walls With Insulation on Inside Surface
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Depth Wall Extends Below Grade*		Insulation over Full Surface						
		Resistance						
		R-4		*R*-8		*R*-13		(hr-ft^2-F)/Btu
			R-0.7		*R*-1.4		*R*-2.3	(m^2-C)/*W*
ft	m	Btu/(hr-ft^2-F)	W/(m^2-C)	Btu/(hr-ft^2-F)	W/(m^2-C)	Btu/(hr-ft^2-F)	W/(m^2-C)	
4	1.52	0.110	0.625	0.075	0.426	0.057	0.324	
5	1.83	0.102	0.579	0.071	0.403	0.054	0.307	
6	2.13	0.095	0.539	0.067	0.380	0.052	0.295	
7	1.22	0.089	0.505	0.064	0.363	0.050	0.284	
		Wall Insulated to a Depth of 2 ft Below Grade						
4	1.52	0.136	0.772	0.102	0.579	0.090	0.511	
5	1.83	0.128	0.727	0.100	0.568	0.091	0.517	
6	2.13	0.120	0.681	0.097	0.550	0.089	0.505	
7	1.22	0.112	0.636	0.093	0.528	0.086	0.488	

*Adapted by permission from *ASHRAE GRP 158 Cooling and Heating Load Calculation Manual* 1979
*For a depth below grade of 3 ft or less, treat as a slab on grade

Table 12-19, Heat Loss of Concrete Floors Less Than 3 ft Below Grade
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Heat Loss per Unit Length of Exposed Edge

Outdoor Design Temperature		Edge Insulation*						
		$R = 5$		$R = 2.5$		None		(hr-ft²-F)/Btu
		$R = 0.88$		$R = 0.44$				(m²-C)/W
F	C	Btu/(hr-ft)	W/m	Btu/(hr-ft)	W/m	Btu/(hr-ft)	W/m	
−20 to −30	−29 to −34	50	48	60	58	75	72	
−10 to −20	−23 to −29	45	43	55	53	65	62	
0 to −10	−18 to −23	40	38	50	48	60	58	
+10 to 0	−12 to −18	35	34	45	43	55	53	
+20 to +10	−7 to −12	30	29	40	38	50	48	

*Adapted by permission from *ASHRAE GPP 158 Heating and Cooling Load Calculation Manual* 1979
*Insulation is assumed to extend 2 ft (0.61 m) either horizontally under slab or vertically along foundation wall.

Table 12-20, Floor Heat Loss for Concrete Slabs With Embedded Warm Air Perimeter Heating Ducts (Per Unit Length of Heated Edge)
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Outdoor Design Temperature		Edge Insulation					
		$R = 2.5$ (hr-ft²-F)/Btu $R = 0.44$ (m²-C)/W Vertical Extending Down 18 in. or 0.46 m Below Floor Surface		$R = 2.5$ (hr-ft²-F)/Btu $R = 0.44$ (m²-C)/W L-Type Extending at Least 12 in. or 0.3 m Deep and 12 in. or 0.3 m Under		$R = 5$ (hr-ft²-F)/Btu $R = 0.88$ (m²-C)/W L-Type Extending at Least 12 in. or 0.3 m Deep and 12 in. or 0.3 m Under	
F	C	Btu/(hr-ft)	W/m	Btu/(hr-ft)	W/m	Btu/(hr-ft)	W/m
−20	−29	105	101	100	96	85	82
−10	−23	95	91	90	86	75	72
0	−18	85	82	80	77	65	62
10	−12	75	72	70	67	55	53
20	−7	62	60	57	55	45	43

*Adapted by permission from *ASHRAE GRP-158 Cooling and Heating Load Calculation Manual*, 1979
*Includes loss downward through inner area of slab.

Table 12-21, Transmission Coefficients U For Horizontal Bare Steel Pipes and Flat Surfaces, $BTU/(hr\ ft^2\ F)$
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Pipe Size Inches	Temperature Difference F Between Pipe Surface and Surrounding Air, Air at 80 F[c]									
	50	100	150	200	250	300	350	400	450	500
1/2	2.12	2.48	2.80	3.10	3.42	3.74	4.07	4.47	4.86	5.28
3/4	2.08	2.43	2.74	3.04	3.35	3.67	4.00	4.40	4.79	5.21
1	2.04	2.38	2.69	2.99	3.30	3.61	3.94	4.33	4.72	5.14
1 1/4	2.00	2.34	2.64	2.93	3.24	3.55	3.88	4.27	4.66	5.07
1 1/2	1.98	2.31	2.61	2.90	3.20	3.52	3.84	4.23	4.62	5.03
2	1.95	2.27	2.56	2.85	3.15	3.46	3.78	4.17	4.56	4.97
2 1/2	1.92	2.23	2.52	2.81	3.11	3.42	3.74	4.12	4.51	4.92
3	1.89	2.20	2.49	2.77	3.07	3.37	3.69	4.08	4.46	4.87
3 1/2	1.87	2.18	2.46	2.74	3.04	3.34	3.66	4.05	4.43	4.84
4	1.85	2.16	2.44	2.72	3.01	3.32	3.64	4.02	4.40	4.81
4 1/2	1.84	2.14	2.42	2.70	2.99	3.30	3.61	4.00	4.38	4.79
5	1.83	2.13	2.40	2.68	2.97	3.28	3.59	3.97	4.35	4.76
Vertical surface	1.84	2.14	2.42	2.70	3.00	3.30	3.62	4.00	4.38	4.79
Horizontal surface Facing upward	2.03	2.37	2.67	2.97	3.28	3.59	3.92	4.31	4.70	5.12
Horizontal surface Facing downward	1.61	1.86	2.11	2.36	2.64	2.93	3.23	3.60	3.97	4.37

[a]Values are for flat surfaces greater than 4 ft^2 or 0.4 m^2
[b]U in $W/(m^2\text{-}C)$ equals $Btu/(hr\text{-}ft^2\text{-}F)$ times 5.678
[c]The temperature difference in C equals F divided by 1.8. An air temperature of 27 C corresponds to 80 F.
[d]Adapted by permission from *ASHRAE Handbook of Fundamentals*, 1977

13. Solar Radiation

Introduction - The sun is the source of most of the energy used by humans. It drives the winds and ocean currents, it furnishes the energy required for plants to grow, it created our oil and coal resources, and it furnishes warmth to us both directly and indirectly.

Solar radiation has a great effect on heat gain and heat loss in buildings. Position of the sun and the clarity of the atmosphere have a great impact on the gain and loss of heat in a building.

In this chapter, the prediction of the sun will be given for any time and place on the earth. Also, the total amount of radiation striking the earth at any time and location will be discussed.

13.1 The motion of the Earth About the Sun

The motion of the earth is an elliptical orbit about the sun. One rotation of the earth takes approximately 365¼ days. The plane the earth creates as it orbits around the sun is called the elliptic plane or the orbital plane.

The mean distance from the center of the sun to the center of the earth is approximately 92.9×10^6 miles. The perihelion distance (the closest position to the sun) is 98.3 percent of the mean distance and occurs January 4. The aphelion distance (the farthest position from the sun) is 101.7 percent of the mean distance and occurs on July 5. Because of this, the earth receives about 7 percent more total radiation in January than in July.

It takes the earth 24 hours to make one rotation about its axis. Its axis is tilted 23.5 degrees with respect to the its orbital plane around the sun.

The diagram below shows the motion of the earth around the sun:

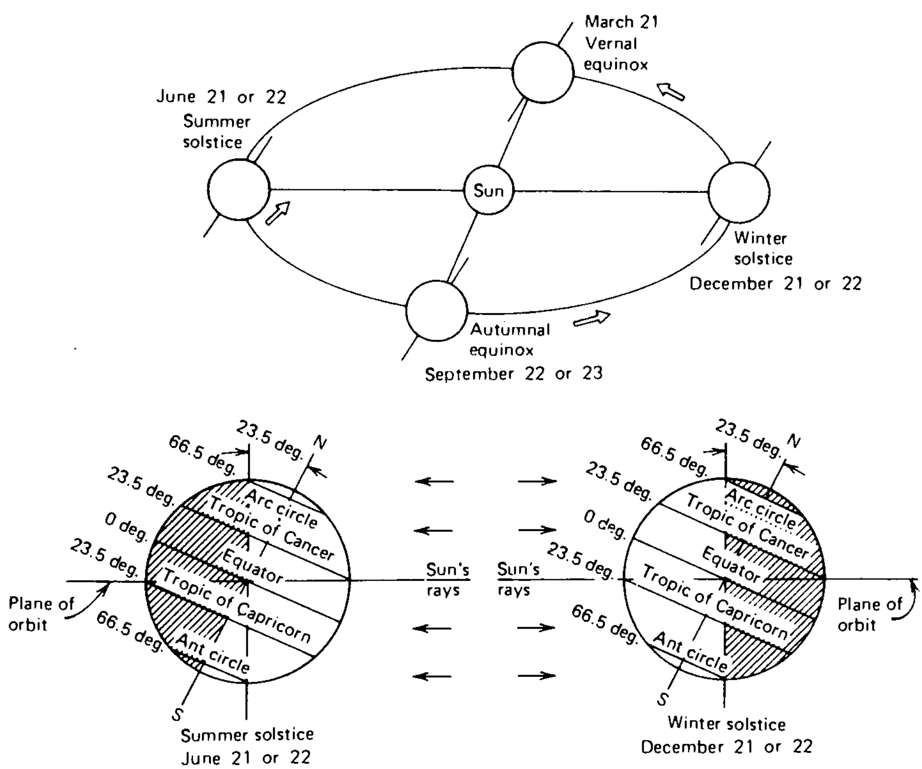


Figure 13-1, The Effect of the Earth's Tilt and Rotation About the Sun
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13.2 Time

The rotation of the earth is 24 hours about its own axis. The earth is divided into longitudinal lines passing through the poles into 360° of circular arcs. 15° of arc corresponds to 1/24 of a day or one hour of time.

The time of the location on the 0° longitudinal line is called Universal Time or Greenwich Civil Time (GCT) and is located going through Greenwich, England. The Local Civil Time (LCT) is determined by the longitude of the observer. The difference between GCT and LCT is 4 minutes for every 1° of longitude.

Clocks are usually set for the same reading throughout a zone covering approximately 15° of longitude. For the United States there are 4 standard time zones.

Eastern Standard Time,	EST	75°
Central Standard Time,	CST	90°
Mountain Standard Time,	MST	105°
Pacific Standard Time,	PST	120°

Since days are not precisely 24 hours long in length, solar time has slightly variable days because of nonsymetry of the earth's orbit, irregularities of the earth's speeds, and other factors. Time measured by the apparent daily-motion of the sun is called solar time (LST - Local Solar Time). Local Solar Time can be evaluated by the following.

LST = LCT + (equation of time)

Table 13-1, The Equation of Time
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

	Day			
Month	**1**	**8**	**15**	**22**
January	−3:16	−6:26	−9:12	−11:27
February	−13:34	−14:14	−14:15	−13:41
March	−12:36	−11:04	− 9:14	− 7:12
April	− 4:11	2:07	− 0:15	1:19
May	2:50	3:31	3:44	3:30
June	2:25	1:15	0:09	− 1:40
July	− 3:33	− 4:48	− 5:45	− 6:19
August	− 6:17	5:40	− 4:35	− 3:04
September	− 0:15	2:03	4:29	6:58
October	10:02	12:11	13:59	15:20
November	16:20	16:16	15:29	14:02
December	11:14	8:26	5:13	1:47

*Reprinted from "The American Ephemeris and Nautical Almanac," U.S. Naval Observatory, Washington, D.C.

13.3 Solar Angles

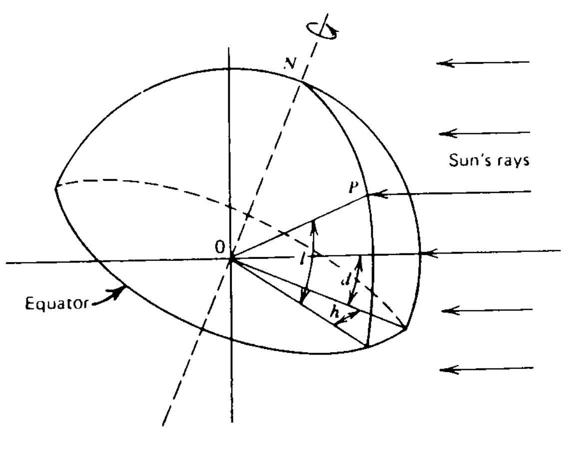

Figure 13-2, Latitude, Hour Angle, and Sun's Declination
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

P is the location on the earth which is of interest. l is the latitude which is the angle between OP and the projection of OP on the equatorial plane. h is the hour angle and is the angle between the projection of OP on the equatorial plane and the projection on that plane of a line passing from the center of the earth to the center of the sun. d is the declination angle and is the angle between the line connecting the centers of the sun and earth and the projection of that line on the equatorial plane.

For heating, ventilation and air conditioning computations it is convenient to define the sun's position in terms of the solar altitude, ß, and the solar azimuth, ø, which depend on l, h, and d.

The solar altitude, ß, is the angle between the sun's ray and the projection of that ray on a horizontal surface. It is given by:

$$\sin\beta = \cos l \cosh + \sin l \sin d$$

The sun's zenith angle, δ, is the angle between the sun's rays and a perpendicular to the horizontal plane.

$$\delta = 90^\circ - \beta$$

The solar azimuth, ø, is the angle in the horizontal plane between south and the projection of the sun's rays on that plane. It is given by:

cosØ = (sinß sinl - sind)/ (cosß cosl)

These angles are seen on the diagram below:

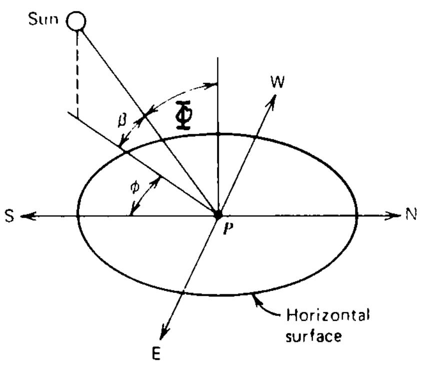

Figure 13-3, The Solar Altitude, Zenith Angle, and Azimuth Angle
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For the vertical surfaces the angle measured in the horizontal plane between the projection of the sun's rays on that plane and a normal to the vertical surface is called the wall solar azimuth, Φ. This angle can be seen below:

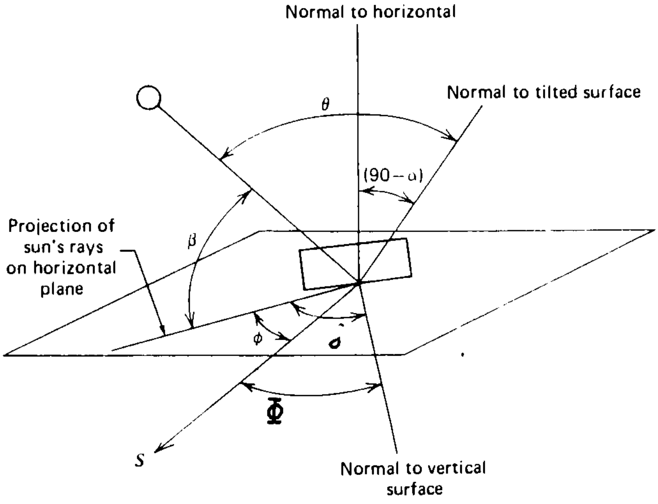

Figure 13-4, Wall Solar Azimuth, Wall Azimuth, Angle of Tilt for an Arbitrary Tilt
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

The wall solar azimuth, Φ, is given by:

$$\Phi = \phi \pm \delta$$

The angle of incidence, θ, is the angle between the sun's rays and the normal to the surface. The tilt angle, α, is the angle between the surface and the normal to the horizontal surface.

The angle of incidence, θ is given by:

$$\cos\theta = \cos\beta \cos\Phi \cos\alpha + \sin\beta \sin\alpha$$

Then for the vertical surface:

$$\cos\theta = \cos\beta \cos\Phi$$

For a horizontal surface:

$$\cos\theta = \sin\beta$$

External shading of windows in an effective way in reducing solar loads and may produce reductions of up to 80%.

For the inset window shown below:

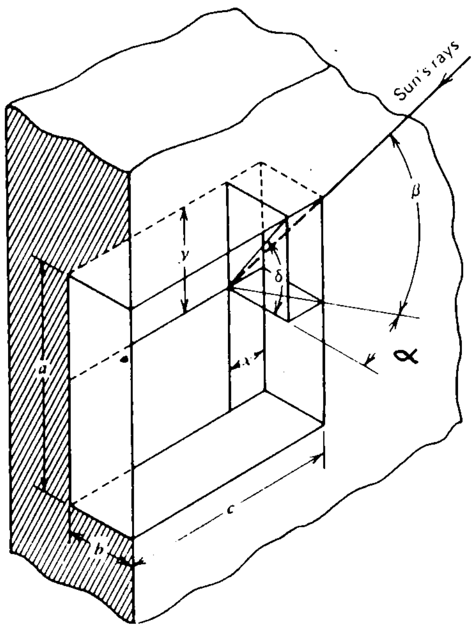

Figure 13-5, Shading of Window Set Back From Plane of a Building
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Dimensions x and y are given by:

$$x = b \tan\alpha$$

$$y = b \tan\delta$$

where:

$$\tan\delta = \tan\beta / \cos\alpha$$

13.4 Solar Irradiation

The Mean Solar Constant G_{SC} is the rate of irradiation on a surface normal to the sun's rays beyond the earth's atmosphere and at the mean of the earth-sun distance.

$$G_{sc} = 428 \text{ btu/(hr}\cdot\text{ft}^2)$$

Even though the atmosphere absorbs some radiation and G_{sc} is not the precise value for a point on the earth it can be used for most HVAC calculations.

The sun acts like a black body at a temperature of 10,800°F.

The total irradiation, G_t is given by:

$$G_t = G_{ND} + G_d + G_R$$

where: G_{ND} = Normal direct irradiation
G_d = Diffuse irradiation
G_R = Reflected irradiation

The depletion of the sun's rays by the earth's atmosphere depends on the composition of the atmosphere (cloudiness, dust, pollutants, atmospheric pressure, and humidity). The diagram below shows a spectral distribution of solar irradiation:

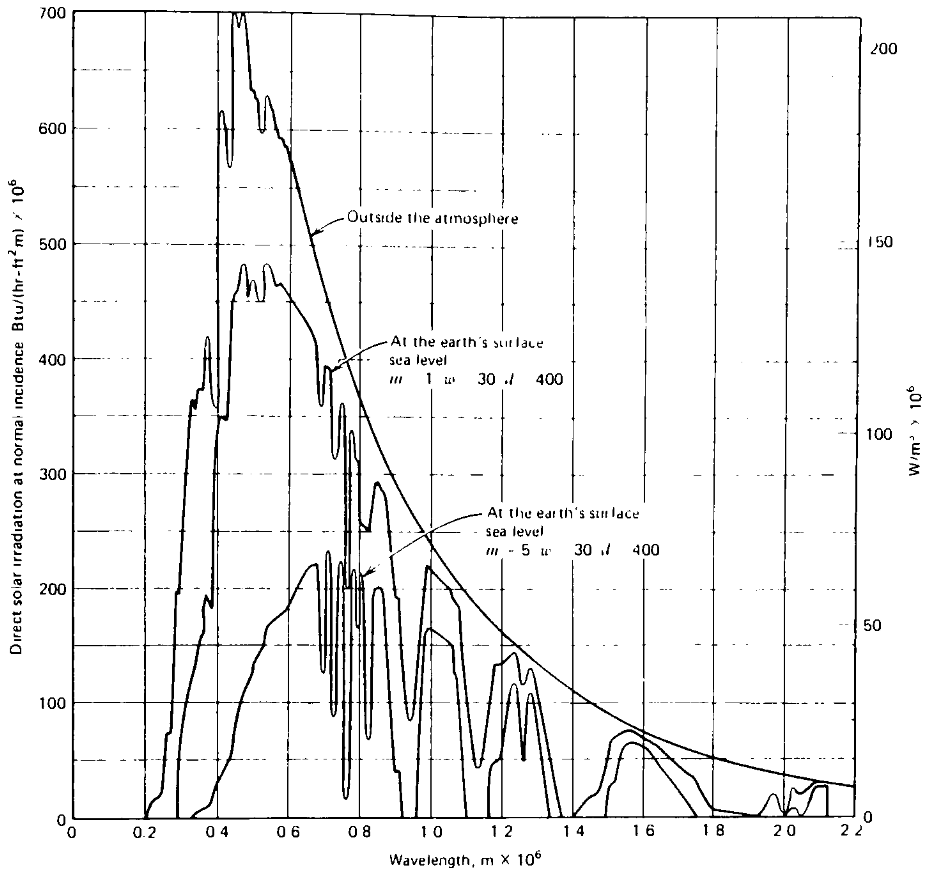

Figure 13-6, Spectral Distribution of Direct Solar Irradiation at Normal Incidence During Clear Days
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

where: m = The ratio of the mass of air at atmosphere in actual sun to earth path to the mass that would exist if the sun were directly overhead at sea level.
w = specified value for water vapor
d = specified value for dust

The value of solar irradiation on a clear day:

$$G_t = G_{ND}$$

where:

$$G_{ND} = \frac{A}{\exp(B/\sin \beta)}$$

where: G_{ND} = Normal direction irradiation, btu/(hr·ft²)

A = Apparent solar irradiation at air mass equal to zero, btu/(hr·ft²)

B = Atmospheric extinction coefficient

β = Solar altitude

The values of A and B are given below. A more detailed set of values can be found in the ASHRAE Fundamentals Book.

Table 13-2, Extraterrestrial Solar Radiation and Related Data for Twenty-First Day of Each Month, Base Year 1964
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

	G_o Btu/(hr-ft²	Equation of Time, min.	Declination, deg	*A* Btu/(hr-ft²)	*B* (Dimensionless Ratios)	C
Jan	442.7	−11.2	−20.0	390	0.142	0.058
Feb	439.1	−13.9	−10.8	385	0.144	0.060
Mar	432.5	− 7.5	0.0	376	0.156	0.071
Apr	425.3	+ 1.1	+11.6	360	0.180	0.097
May	418.9	+ 3.3	+20.0	350	0.196	0.121
June	415.5	− 1.4	+23.45	345	0.205	0.134
July	415.9	− 6.2	+20.6	344	0.207	0.136
Aug	420.0	− 2.4	+12.3	351	0.201	0.122
Sep	426.5	+ 7.5	0.0	365	0.177	0.092
Oct	433.6	+15.4	−10.5	378	0.160	0.073
Nov	440.2	+13.8	−19.8	387	0.149	0.063
Dec	443.6	+ 1.6	−23.45	391	0.142	0.057

*Reprinted by permission from *ASHRAE Handbook of Fundamentals*, 1977.
To convert Btu/(hr-ft²) to W/m², multiply by 3.1525

The angle of incidence, θ, is used to relate the normal direct irradiation G_{ND} to the direct irradiation G_D of other surface orientations.

$$G_N = G_{ND}\cos\theta$$

Diffuse radiation, $G_{d\theta}$, is the radiation which is in a diffuse form and strikes on non-horizontal surface.

$$G_{d\theta} = CG_{ND}F_{WS}$$

where: C = the ratio of diffuse to direct normal irradiation

F_{WS} = the configuration factor or angle factor between the wall and sky.

$F_{WS} = (1 + \cos\varepsilon)/2$

ε is the tilt angle of the surface from horizontal

$\varepsilon = 90^\circ - \alpha$

The reflected solar energy from the ground, G_r, is given by:

$$G_r = G_{tH}\rho_g F_{Wg}$$

where: G_r = Rate at which energy is reflected onto a wall, (btu/hrft^2)

G_{tH} = Rate at which energy is strikes the horizontal surface in front of the wall

F_{Wg} = Configuration factor

$F_{Wg} = (1 - \cos\varepsilon)/2$

Logic steps to determine total irradiation on a tilted objected:

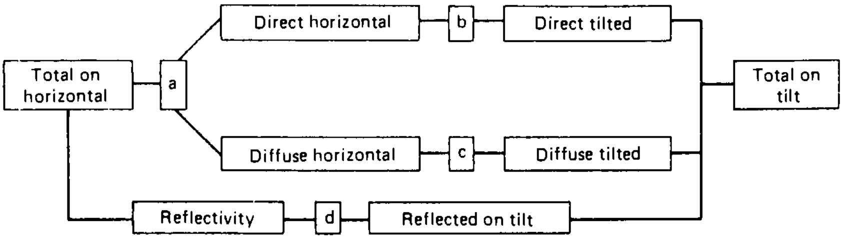

Figure 13-7, Conversion of Horizontal Insolation to Insolation on a Tilted Surface
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14. Space Heat Load

Introduction - In a building there are two kinds of heat losses; 1) The heat transmitted through walls, ceilings, floors, glass or other surfaces. and 2) The heat required to warm outdoor air entering the space.

The actual heat loss problem is transient due to changing outdoor temperatures and conditions. But there are occasions where outdoor conditions remain constant for a period of two to four days, like cold spells, snow storms, and periods of cloudy weather. Therefor, for design purposes the heat loss is usually estimated for steady state heat transfer for some reasonable design temperature.

General procedure for calculating design heat losses in a structure:

1) Select outdoor design conditions
2) Select indoor design conditions
3) Estimate the temperature in any adjacent unheated space
4) Select the transmission coefficients and compute the heat loss due to the structure
5) Compute the heat loss due to infiltration
6) Compute the heat loss due to ventilation
7) Sum the losses

This chapter discusses all the steps for calculating the design heat losses in a structure.

14.1 Outdoor Design Conditions

The ASHRAE Fundamentals Book has a recorded listing of the outdoor weather conditions for various parts of the United States and the World.

For general buildings, the design temperatures should generally be the 97½ percent value as specified by ASHRAE Standard 90-A Energy Conservation in New Building Design.

Note: Wind conditions which are abnormally high can generate peak heat load conditions above outdoor design temperatures.

14.2 Indoor Design Conditions

The Comfort and Health Chapter specifies the indoor design conditions needed for different space uses.

It should be noted that indoor design temperatures should be kept as low as possible so heating equipment will not be oversized. ASHRAE 90-A specifies 72°F. Over sizing of equipment causes reduced efficiencies.

14.3 Calculation of Heat Losses

14.3.1 Structural Heat Loss

$$\dot{q} = UA(t_i - t_o)$$

This calculation is preformed for each window, wall, roof, floor, door and etc. The work sheet below gives an orderly tabulation to the process.

HEAT LOSS WORKSHEET

1 Room					Total House														
2 Exposed Wall, feet																			
3 Room Dimensions, feet																			
4 Ceiling Height, feet	Direction Room Faces																		
Type Exposure		Constr	Coeff	Δt	Area or Length	Btuh Sens	Btuh Lat	Area or Length	Btuh Sens	Btuh Lat	Area or Length	Btuh Sens	Btuh Lat	Area or Length	Btuh Sens	Btuh Lat	Area or Length	Btuh Sens	Btuh Lat
5 Gross Exposed Walls and Partitions	a																		
	b																		
	c																		
	d																		
6 Windows and Doors	a																		
	b																		
	c																		
7 Net Exposed Walls and Partitions	a																		
	b																		
	c																		
	d																		
8 Ceilings	a																		
	b																		
9 Floors	a																		
	b																		
10 Infiltration Windows and Doors	North																		
	West																		
	South																		
	East																		
11 Ventilation	Resid																		
12 Sub Total Btuh Loss																			
13 Duct Loss, Btuh																			
14 Total Loss, Btuh																			
15 Heat Sources, Btuh																			
16 Air Quantity, cfm																			
17																			
18																			

Figure 14-1, Heat Loss Work Sheet
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14.3.2 Infiltration Heat Loss

Two types of heat losses are present: 1) Sensible and 2) Latent.

Sensible heat loss is due to the cold air being raised to indoor temperatures. It is given by:

$$\dot{q}_s = \dot{m}_o C_p (t_i - t_o)$$

where: $\dot{m}_o$ = Mass flow rate
C_p = Specific heat capacity of the moist air

or based on volume flow rate:

$$\dot{q}_s = \dot{Q}C_p(t_i - t_o)/v_o$$

where: $\dot{Q}$ = Volume flow rate
v_o = Specific volume

The latent heat requirement is due to humidifying the dry cold air that enters the structure. It is given by:

$$\dot{q}_l = \dot{m}_o(W_i - W_o)i_{fg}$$

where: $(W_i - W_o)$ = difference in design humidity ratios
i_{fg} = Latent heat of vaporization at indoor conditions

in terms of volume flow rate:

$$\dot{q}_l = \dot{Q}(W_i - W_o)i_{fg}/v_o$$

14.3.3 Determining Air Infiltration Rates

There are two methods used: 1) Air Change Method and 2) Crack Method.

1) The Air Change Method is based upon the number of air changes per hour based on how many windows and doors a room has. This method produces satisfactory results. The table below gives reasonable precision for residential and light commercial applications:

Table 14-1, Air Changes Taking Place Under Average Conditions in Residences. Exclusive of Air Provided for Ventilation
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Kind of Room or Building	**Number of Air Changes Taking Place per Hour**
Rooms with no windows or exterior doors	$\frac{1}{2}$
Rooms with windows or exterior doors on one side	1
Rooms with windows or exterior doors on two sides	$1\frac{1}{2}$
Rooms with windows or exterior doors on three sides	2
Entrance halls	2

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[a]For rooms with weather-stripped windows or with a storm sash, use $\frac{2}{3}$ these values.

A total allowance of one half the sum of the individual rooms should be taken because air that enters the building on the windward side is assumed to leave the building on the leeward side. The heating load due to infiltration should generally be assigned to those rooms on the windward side of the building where the air enters.

2) The Crack Method is generally considered the most accurate. It estimates the infiltration based on the characteristics of the windows and doors and the pressure difference between the inside and outside.

The volume flow rate for the crack method can be estimated by:

$\dot{Q} = AC\Delta P^n$

where: A = Cross-section area of crack
C = Flow coefficient, which depends on the type of crack and the nature of the flow in the crack
ΔP = Pressure difference, $P_o - P_i$
n = Exponent that depends on the nature of the flow in the crack, $.4 < n < 1.0$

The pressure difference in the crack is given by:

$\Delta P = \Delta P_w + \Delta P_s + \Delta P_p$

where: ΔP_w = Pressure difference due to wind
ΔP_s = Pressure difference due to the stack effect
ΔP_p = Pressure difference due to building pressurization

All pressure differences are positive when they cause air to flow into the building.

ΔP_w - Pressure difference due to wind

$$\Delta P_w = \frac{\rho(\bar{V}^2_w - \bar{V}^2_f)}{2g_c}$$

where: ρ = Air density
$\bar{V}_w$ = Wind speed
$\bar{V}_f$ = Final velocity of the wind at the building boundary and assumed to be zero, $V_f = 0$

Pressure Coefficient, C_p is used because $\bar{V}_f$

is not really equal to zero.

$$C_p = \Delta P_w / \Delta P_{wt}$$

where: ΔP_{wt} = the pressure difference when $V_f = 0$

Finally:

$$\frac{\Delta P_w}{C_p} = \frac{p}{2g_c} \bar{V}^2_w$$

The Table below gives some approximate values for a rectangular building with normal and quartering wind.

Table 14-2, Pressure Coefficients for a Rectangular Building
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Building Wall	Normal Wind	Quartering Wind
Windward	0.95	0.70
Sides	-0.40	-
Leeward	-0.15	-0.50

ΔP_s - Pressure difference due to stack effect. Stack effect is caused by different densities in air. In winter the air outside the building is more dense causing infiltration on the bottom of the building. In the summer the process is reversed.

There is some point in the building where there is a neutral point where there is no infiltration or exfiltration. Theoretically this point is at the center of the building but due to building openings it might vary from this position.

ΔP_{st} (theoretical) is given by:

$$\Delta P_{st} = \frac{P_o\ h\ g}{R_a\ g_c} (1/T_o - 1/T_i)$$

where: P_o = Outside pressure
h = vertical distance from neutral point
T_o = Outside temperature, R
T_i = Inside temperature, R
R_a = Gas constant for air

Due to the resistance of the stack effect due to floors, a draft coefficient is used, C_d, to relate theoretical values to actual values.

$$C_d = \Delta P_s / \Delta P_{st}$$

This gives the stack pressure difference, ΔP_s to be:

$$\Delta P_s = \frac{C_d\ P_o\ h\ g}{R_a\ g_c} (1/T_o - 1/T_i)$$

C_d ranges from 1.0 for buildings with no doors in the stair wells to .65 for most modern office buildings. Below is a graph of pressure difference due to stack effect.

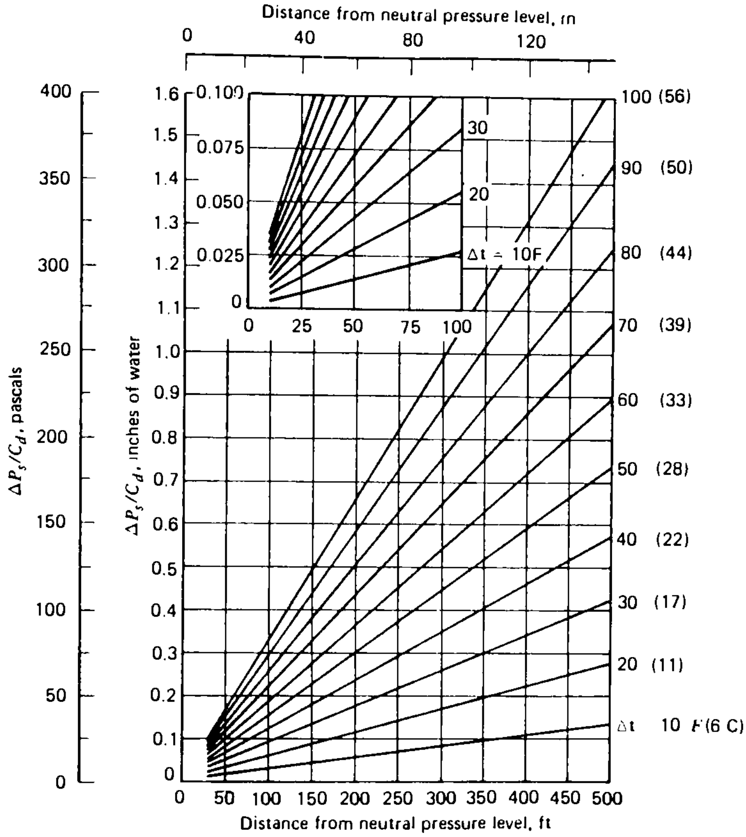

Figure 14-2, Pressure Difference Due to Stack Effect
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

$\blacktriangle P_p$ - Pressure difference due to building pressurization. This pressure is difficult to determine because it depends on the pressure differences due to wind and stack effect. $\blacktriangle P_p$ will vary from 0 to 1/3 of the total pressurization except this value will be negative.

The following pages give air leakage rates for windows, doors, and walls.

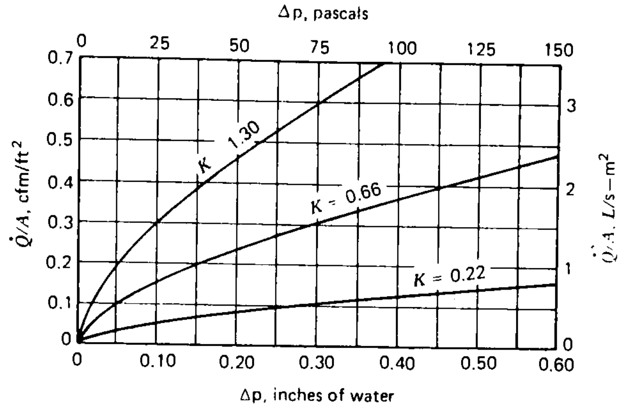

Figure 14-3, Curtain Wall Infiltration for One Room or One Floor
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

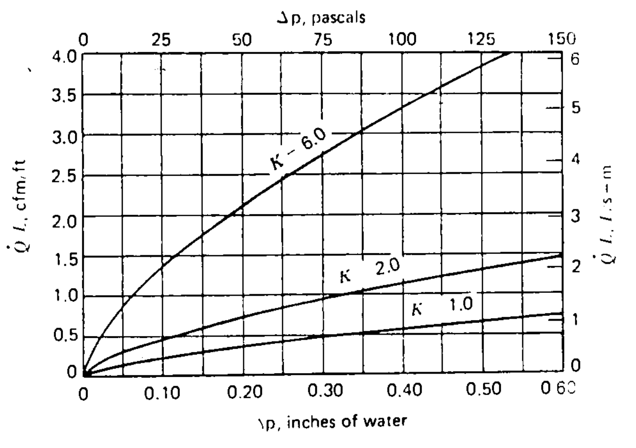

Figure 14-4, Window and Residential Type Door Infiltration Characteristics
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

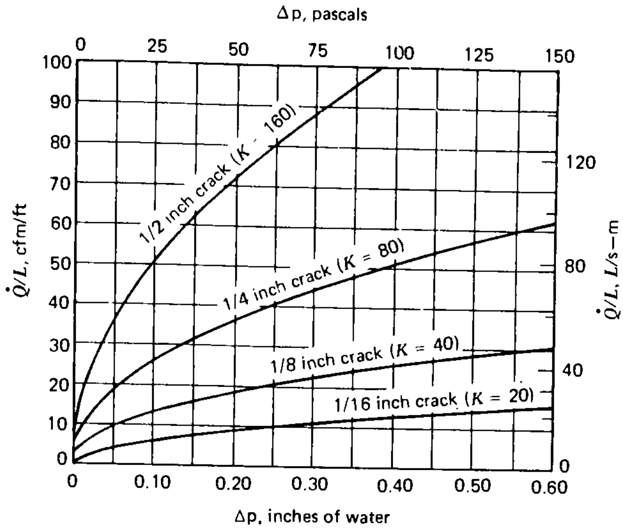

Figure 14-5, Infiltration Through Closed Swinging Door Cracks
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Table 14-3, Window Classification
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

	Wood Double-Hung (Locked)	**Other Types**
Tight Fitting Window $K = 1.0$	Weatherstripped Average Gap (1/64 in crack)	Wood Casement and Awning Windows; Weatherstripped Metal Casement Windows; Weatherstripped
Average Fitting Window $K = 2.0$	Non-Weatherstripped Average Gap (1/64 in. crack)	All Types of Vertical and Horizontal Sliding Windows; Weatherstripped. Note: if average gap (1/64 in. crack) this could be tight fitting window
	or Weatherstripped Large Gap (3/32 in. crack)	Metal Casement Windows; Non-Weatherstripped Note: if large gap (3/32 in. crack) this could be a loose fitting window
Loose Fitting Window $K = 6.0$	Non-Weatherstripped Large Gap (3/32 in. crack)	Vertical and Horizontal Sliding Windows; Non-Weatherstripped

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Table 14-4, Residential-Type Door Classification
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Tight Fitting Door $K = 1.0$	Very small perimeter gap and perfect fit weatherstripping—often characteristic of new doors
Average Fitting Door $K = 2.0$	Small perimeter gap having stop trim fitting properly around door and Weatherstripped
Loose Fitting Door $K = 6.0$	Large perimeter gap having poor fitting stop trim and weatherstripped or Small perimeter gap with no weatherstripping

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Table 14-5, Curtain Wall Classification
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Leakage Coefficient	**Description**	**Curtain Wall Construction**
$K = 0.22$	Tight Fitting Wall	Constructed under close supervision of workmanship on wall joints. When joints seals appear inadequate they must be re-done
$K = 0.66$	Average Fitting Wall	Conventional construction procedures are used
$K = 1.30$	Loose Fitting Wall	Poor construction quality control or an older building having separated wall joints

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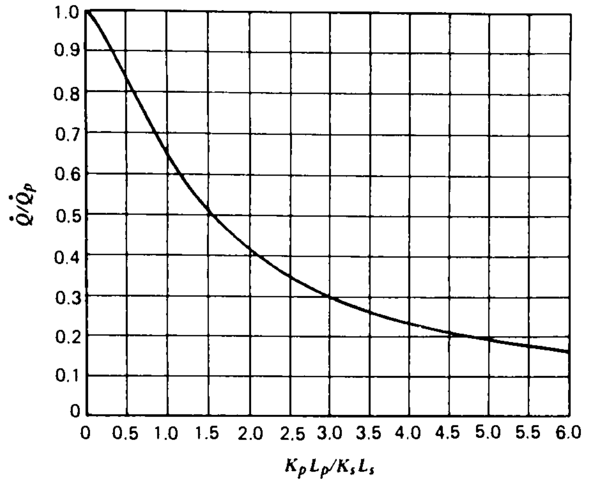

Figure 14-6, Infiltration for Storm-Prime Combination Windows
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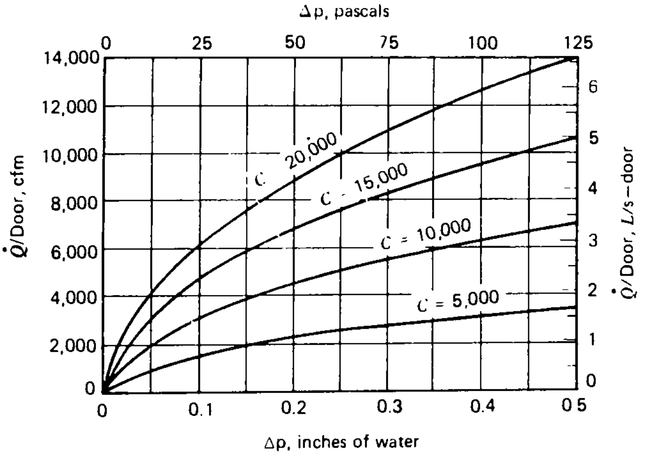

Figure 14-7, Swinging Door Characteristics With Traffic
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

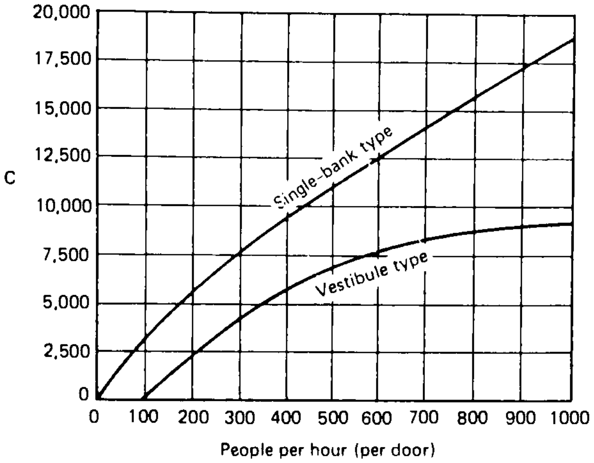

Figure 14-8, Flow Coefficient Dependence on Traffic Rate
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

There are two categories for calculating building infiltrations: 1) Low rise buildings - less than five stories, stack effect and wall leakages can be neglected and 2) High rise buildings - more than five stories, stack effect may be dominant.

For low rise buildings the crack length should be based on the windward but never less than one half the total crack length of the building.

14.4 Heat Losses From Air Ducts

Heat loss may be estimated by the following:

$$\dot{q} = UA_s \Delta t_m$$

where: U = Overall heat transfer coefficient
A_s = Surface area of the duct
Δt_m = Mean temperature difference

For duct insulation ASHRAE Standard 90-A should be followed where:

$$R = \Delta t/15 \text{ (hr ft}^2\text{ F)/btu}$$

14.5 Auxiliary Heat Sources

The heat sources supplied by people, lights, motors, and machines should always be estimated but actual allowances for these heat sources requires careful considerations.

14.6 Intermittently Heated Structures

When a structure is not heated on a continuous basis, the heating equipment capacity may have to be enlarged to assure that the temperature can be raised to a comfortable level within a reasonable amount of time.

14.7 Estimating Fuel Requirements

Digital computers are often used today to calculate fuel requirements, but where computers are uneconomical there are two hand calculation methods: 1) Degree Day procedure and 2) Bin method.

14.7.1 The Degree Day Procedure

The number of Degree Days, DD, is given by:

$$DD = (t - t_a)N/24$$

where: DD = Number of degree days
N = The number of hours which the average temperature t_a is computed
t = 65°F

For various locations the number of degree days are given in the ASHRAE Fundamentals book

Fuel requirements are given by:

$$F = \frac{24\ DD\ \dot{q}}{n(t_i - t_o)H}$$

where: F = The quantity of fuel required for the period desired
DD = The degree days for the period desired
$\dot{q}$ = The total heat loss based on design conditions, t_i and t_o btu/hr
n = an efficiency factor, .65 for gas and 1 for electric
H = The heating of fuel, btu/unit volume or mass
C_D = Correction factor given in the figure below

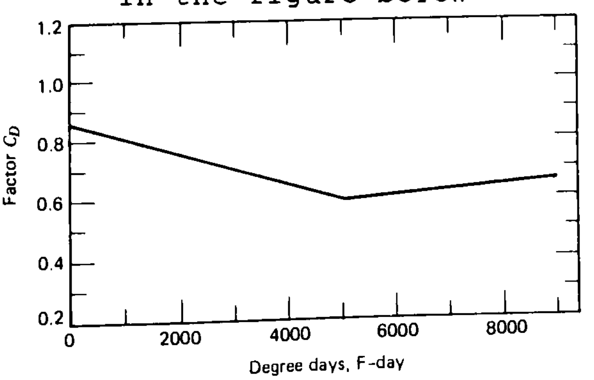

Figure 14-9, Correction Factor C_D
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14.7.2 Bin Method

This method is a hand calculation procedure where energy requirements are determined at many outdoor temperature conditions. The bins are usually 5°F in size and the day is divided into 3 - 8 hour shifts. The bin method requires hourly weather data and equipment characteristics.

15. The Cooling Load

Introduction - Unlike heating calculations, cooling load calculations must use transient analysis to satisfactory solve for the cooling loads. This is because of the strong transient effects caused by the hourly variation in solar radiation. Also, the heat gain by a building is not directly transmitted to the interior, a majority of this heat is stored in the structure.

In determining seasonal energy requirements for either heating or cooling both the solar inputs and transient effects due to storage must be considered for accurate results. These thermal simulations are usually preformed using a computer with some type of assumed or historical weather data. The computations the computer makes are often preformed on an hourly basis or a fraction of an hourly basis.

The first section in this chapter discusses the heat gain, cooling load and heat extraction rate. The second section discusses outdoor and indoor design conditions. The third section discusses the cooling load temperature difference method for determining the cooling load. The fifth section discusses the cooling load caused by internal sources. The sixth and seventh sections discusses the cooling load caused by infiltration and the summation of the total heating gain and cooling load. The eighth section discusses the determining of the cooling load using the transfer function method. The last section discusses the heat extraction rate and room temperature.

15.1 Heat Gain, Cooling Load, and Heat Extraction Rate

6.1.1 Heat Gain - The rate at which energy is transferred to or generated within a space. It has two components, sensible and latent heat. Heat gains usually occur in the following forms:

1. Solar radiation through openings.
2. Heat conduction through boundaries with convection and radiation from the inner surface into the space.
3. Sensible heat convection and radiation from internal objects.
4. Ventilation and infiltration air.
5. Latent heat gains generated within the space.

15.1.1 Cooling load - The rate which energy must be removed from the space to maintain the temperature and humidity at the design conditions.

The cooling load will differ from the heat gain at any instant in time. This is because much of the heat gain is absorbed by the building structure. Below shows this effect:

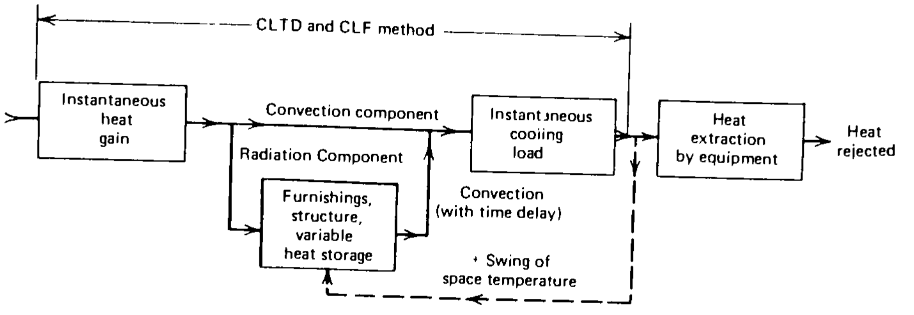

Figure 15-1, Schematic Relation of Heat Gain to Cooling Load
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

15.1.2 Heat Extraction Rate - The rate at which energy is removed from the space by the air conditioning equipment.

The figure below shows the relation between heat gain, cooling load and th mass of the structure.

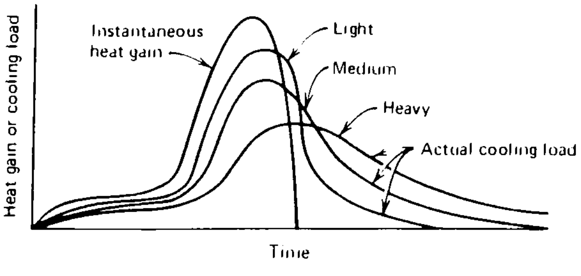

Figure 15-2, Actual Cooling Load and Solar Heat Gain for Light, Medium, and Heavy Construction
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15.1.3 Computational Viewpoint

The heat gain for the interior space at a given time is:

$$\dot{q}_{\theta} = \dot{q}_{i,\theta} + \dot{q}_{r,\theta} + \dot{q}_{s,\theta} + \dot{q}_{l,c}$$

where: $\dot{q}_{i,\theta}$ = Convective heat transfer from inside surfaces of boundaries at time θ.

$\dot{q}_{r,\theta}$ = Radiation heat transfer between the inside surfaces of boundaries and other interior surfaces at time θ.

$\dot{q}_{s,\theta}$ = Rate of solar energy entering the through the windows at time θ.

$\dot{q}_{l,\theta}$ = Rate of heat generation by lights, people, and other internal sources at time θ.

The sensible cooling load for the space at a given time may be expressed as:

$$\dot{q}_{c,\theta} = \dot{q}_{i,\theta} + \dot{q}_{I,\theta} + \dot{q}_{v,\theta} + \dot{q}_{sc,\theta} + \dot{q}_{lc,\theta}$$

where: $\dot{q}_{I,\theta}$ = Rate of heat gain due to infiltration at time θ.

$\dot{q}_{v,\theta}$ = Rate of heat gain due to outdoor ventilation air at time θ.

$\dot{q}_{sc,\theta}$ = Rate of energy convected from the inner window and other interior surfaces to the room air at time θ.

$\dot{q}_{l,\theta}$ = Rate of heat generated by lights, people, and other internal sources which is convected into the room air at time θ.

The equations describing the conduction heat transfer in the boundary walls, floors and ceilings must be solved simultaneously because they are coupled through the inside surface temperature. Also, the convective components of the solar and internal heat generation are related to the room air temperature.

Consider the heat conduction problem with heat transfer through a wall or roof section with variable solar radiation and variable outdoor temperature. The heat conduction equation becomes nonlinear:

$$\frac{di}{d\theta} = \frac{K}{pc}\frac{d^2t}{dx^2} \qquad \text{(Homogeneous slab)}$$

where: t = Local temperature at a point in the slab, °F
θ = Time, hour
K/pc = Thermal diffusivity of the slab, ft^2/hr
x = length, ft

Boundary conditions:
$x = 0$ (outside surface)

$$-K_w(dt/dx)_{x=0} = h_o(t_o(\theta) - t_{wo}) + \dot{q}_r(\theta)$$

where: q_r = Net solar radiation heat transfer

$x = L$ (inside surface)

$$-K_w(dt/dx)_{x=L} = h_i(t_{wi} - t_i) + q_r(\theta)$$

The non-linear, time dependent boundary condition at the outside air surface is the primary obstacle in obtaining a solution to the differential equation.

To eliminate this problem the concept of sol-air temperature is introduced. The sol-air temperature, t_e, is the fictitious temperature that in the absence of all radiation exchanges gives the same rate of heat transfer to the exterior surface as actually occurs by solar radiation and convection.

The heat transfer to the outer surface in terms of sol-air temperature is:

$$\dot{q}_o = h_o A(t_e - t_{wo})$$

In terms of the actual outdoor temperature, t_o, the heat transfer rate is given by:

$$\dot{q}_o = h_o A(t_o - t_{wo}) + \alpha A G_t - \varepsilon \blacktriangle R A$$

where: h_o = Coefficient of heat transfer btu/(hr ft^2 F)

ε = Emittance of the surface

$\blacktriangle R$ = Difference between the long wavelength radiation incident on the surface from the sky and the radiation emitted from a black at outdoor air temperature, btu/(hr ft^2)

α = Absorpstance of the wall surface

G_t = Total incidence solar radiation upon the surface, btu/(hr ft^2)

A = Surface area, ft^2

Combining the above two equations the following relationship is formed:

$$t_e = t_o + \alpha G_t / h_o - \varepsilon \blacktriangle R / h_o$$

t_e varies harmonically

$$0 < \varepsilon \blacktriangle R/ho < 7^\circ F \text{ (horizontal surface)}$$

$$.15 < \alpha / h_o < .30 \text{ (hr ft}^2\ ^\circ\text{F)/btu}$$

The outside boundary is then greatly simplified to:

$$-K_w(\underline{d}t/\underline{d}x)_{x=0} = h_o(t_e(\theta) - t_{wo})$$

With the assumption that t_e is a harmonic function, Fourier series solutions to the differential equation are possible using a digital computer. It is very difficult to solve the Fourier series problem on a computer. To reduce and simplify the time required for computations, transformed methods have been applied to this problem. A digital computer is still needed but calculations are very rapid.

A hand calculation method has been developed from the transfer function method which produces results within 5 percent of the transfer function method. This method is called the Cooling Load Temperature Difference Method (CLTD)

15.2 Outdoor and Indoor Design Conditions

The same method used in the Heat Loss section may also be applied here.

15.3 The CLTD Method

The CLTD method makes use of a temperature differences in the case of walls and roofs and cooling load factors (CLF) in the case of solar gains through windows and internal heat sources. The CLTD and CLF vary with time and are a function environmental conditions and building parameters.

These factors have been derived for a fixed set of surfaces and environmental conditions; Therefor, correction factors must be applied. In general, calculations proceed as follows:

For roofs and walls:

$$\dot{q}_\theta = UA(CLTD)_\theta$$

where: U = Overall heat transfer coefficient, btu/(hr ft^2 °F)
A = area, ft^2
CLDT = Temperature difference which gives the cooling load at time θ, °F

The CLTD accounts for the thermal response (lag) in the heat transfer through the wall or roof, as well as the response (lag) due to radiation of part of the energy from the interior surface of the wall to objects within the space.

For solar gain through gass:

$$\dot{q}_\theta = A(SC)SHGF(CLF)_\theta$$

where: A = area, ft^2
SC = Shading coefficient (internal shades)
SHGF= Solar heat gain factor, btu/(hr ft^2)
CLF = Cooling load factor for time θ

The SHGF is the maximum for a particular month, orientation, and latitude. The CLF accounts for the variation of the SHGF with time, the massiveness of the structure and the internal shade. Again the CLF accounts for the thermal response (lag) of the radiant part of the solar input.

For internal heat sources:

$$\dot{q}_\theta = q_i(CLF)_\theta$$

where: q_i = Instantaneous heat gain from lights, people and equipment, btu/hr
CLF_θ = Cooling load factor for time θ

The time of day when the peak cooling load will occur must be estimated. In fact two different types of peaks need to be determined. First the peak for each room and secondly the peak for the total building which is made up of many different spaces.

The following sections will show various cooling load components.

15.4 Cooling Load- External Sources

15.4.1 Walls and Roofs - The following tables, Table 15-1 and Table 15-2 give CLTD values in degrees F which were computed for the following conditions.

A. Dark surface for solar radiation absorption.

B. Inside temperature of 78°F.

C. Out door maximum temperature of 95°F and an outdoor daily range of 21°F.

D. Solar radiation for 40 degrees North latitude on July 21.

E. Outside convective film coefficient of 3.0 btu/(hr ft^2 °F).

F. Inside convective film coefficient of 1.46

btu/(hr ft^2 °F)

G. No forced ventilation or air ducts in the ceiling space.

* To convert CLTD values from degrees F to C multiply by $^5/_9$.

CLTD can be adjusted using the following relations:

$$CLTD_{cor} = (CLTD + LM)K + (78 - t_i) + (t_{om} - 85)$$

where: LM = A correction factor for latitude and month from table 15-3, °F

K = Color adjustment factor
$.5 \text{ (light)} \le K \le 1 \text{ (dark)}$

t_i = Room design temperature, °F

t_{om}= Outdoor mean temperature
$t_{om} = t_o - DR/2$, °F

Table 15-1, Cooling Load Temperature Differences for Calculating Cooling Load From Flat Roofs

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Roof No.	Description of Construction	Weight lb/ft²	U Btu/(hr-ft²-F)	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
	Without Suspended Ceiling																										
1	Steel sheet with 1-in (or 2-in) insulation	7 (8)	0.213 (0.124)	1	2	3	3	5	1	6	19	34	49	61	71	78	79	77	70	59	45	30	18	12	8	5	3
2	1-in wood with 1-in insulation	8	0.170	6	3	0	1	3	3	2	4	14	27	39	52	62	70	74	74	70	62	51	38	28	20	14	9
3	4-in l.w. concrete	18	0.213	9	5	2	0	2	3	3	1	9	20	32	44	55	64	70	73	71	66	57	45	34	25	18	13
4	2-in h.w. concrete with 1-in (or 2-in) insulation	29	0.206 (0.122)	12	8	5	3	0	1	1	3	11	20	30	41	51	59	65	66	66	62	54	45	36	29	22	17
5	1-in wood with 2-in insulation	19	0.109	3	0	3	4	5	7	6	3	5	16	27	39	49	57	63	64	62	57	48	37	26	18	11	7
6	6-in l.w. concrete	24	0.158	22	17	13	9	6	3	1	1	3	7	15	23	33	43	51	58	62	64	62	57	50	42	35	28
7	2.5-in wood with 1-in insulation	13	0.130	29	24	20	16	13	10	7	6	6	9	13	20	27	34	42	48	53	55	56	54	49	44	39	34
8	8-in l.w. concrete	31	0.126	35	30	26	22	18	14	11	9	7	7	9	13	19	25	33	39	46	50	53	54	53	49	45	40
9	4-in h.w. concrete with 1-in (or 2-in) insulation	52 (52)	0.200 (0.120)	25	22	18	15	12	9	8	8	10	14	20	26	33	40	46	50	53	53	52	48	43	38	34	30
10	2.5-in wood with 2-in insulation	13	0.093	30	26	23	19	16	13	10	9	8	9	13	17	23	29	36	41	46	49	51	50	47	43	39	35
11	Roof terrace system	75	0.106	34	31	28	25	22	19	16	14	13	13	15	18	22	26	31	36	40	44	45	46	45	43	40	37
12	6-in h.w. concrete with 1-in (or 2-in) insulation	(75) 75	0.192 (0.117)	31	28	25	22	20	17	15	14	14	16	18	22	26	31	36	40	43	45	45	44	42	40	37	34
13	4-in wood with 1-in (or 2-in) insulation	17 (18)	0.106 (0.078)	38	36	33	30	28	25	22	20	18	17	16	17	18	21	24	28	32	36	39	41	43	43	42	40
	With Suspended Ceiling																										
1	Steel sheet with 1-in (or 2-in) insulation	9 (10)	0.134 (0.092)	2	0	2	−3	4	4	1	9	23	37	50	62	71	77	78	74	67	56	42	28	18	12	8	5
2	1-in wood with 1-in insulation	10	0.115	20	15	11	8	5	3	2	3	7	13	21	30	40	48	55	60	62	61	58	51	44	37	30	25
3	4-in l.w. concrete	20	0.134	19	14	10	7	4	2	0	0	4	10	19	29	39	48	56	62	65	64	61	54	46	38	30	24
4	2-in h.w. concrete with 1-in insulation	30	0.131	28	25	23	20	17	15	13	13	14	16	20	25	30	35	39	43	46	47	46	44	41	38	35	32
5	1-in wood with 2-in insulation	10	0.083	25	20	16	13	10	7	5	5	7	12	18	25	33	41	48	53	57	57	56	52	46	40	34	29
6	6-in l.w. concrete	26	0.109	32	28	23	19	16	13	10	8	7	8	11	16	22	29	36	42	48	52	54	54	51	47	42	37
7	2.5-in wood with 1-in insulation	15	0.096	34	31	29	26	23	21	18	16	15	15	16	18	21	25	30	34	38	41	43	44	44	42	40	37
8	8-in l.w. concrete	33	0.093	39	36	33	29	26	23	20	18	15	14	14	15	17	20	25	29	34	38	42	45	46	45	44	42
9	4-in h.w. concrete with 1-in (or 2-in) insulation	53 (54)	0.128 (0.090)	30	29	27	26	24	22	21	20	20	21	22	24	27	29	32	34	36	38	38	38	37	36	34	31
10	2.5-in wood with 2-in insulation	15	0.072	35	33	30	28	26	24	22	20	18	18	18	20	22	25	28	32	35	38	40	41	41	40	39	37
11	Roof terrace system	77	0.082	30	29	28	27	26	25	24	23	22	22	22	23	23	25	26	28	29	31	32	33	33	33	33	32
12	6-in h.w. concrete with 1-in (or 2-in) insulation	77 (77)	0.125 (0.088)	29	28	27	26	25	24	23	22	21	21	22	23	25	26	28	30	32	33	34	34	34	33	32	31
13	4-in wood with 1-in (or 2-in) insulation	19 (20)	0.083 (0.064)	35	34	33	32	31	29	27	26	24	23	22	21	22	22	24	25	27	30	32	34	35	36	37	36

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*CLTD may be converted to degrees C by multiplying by 5/9

Table 15-2, Cooling Load Temperature Differences for Calculating Cooling Load From Sunlit Walls
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

North Latitude Wall Facing	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
												Solar Time, hr												
Group A Walls																								
N	14	14	14	13	13	13	12	12	11	11	10	10	10	10	10	10	11	11	12	12	13	13	14	14
E	24	24	23	23	22	21	20	19	19	18	19	19	20	21	22	23	24	24	25	25	25	25	25	25
S	20	20	19	19	18	18	17	16	16	15	14	14	14	14	14	15	16	17	18	19	19	20	20	20
W	27	27	26	26	25	24	24	23	22	21	20	19	19	18	18	18	18	19	20	22	23	25	26	26
Group B Walls																								
N	15	14	14	13	12	11	11	10	9	9	9	8	9	9	9	10	11	12	13	14	14	15	15	15
E	23	22	21	20	18	17	16	15	15	15	17	19	21	22	24	25	26	26	27	27	26	26	25	24
S	21	20	19	18	17	15	14	13	12	11	11	11	11	12	14	15	17	19	20	21	22	22	22	21
W	29	28	27	26	24	23	21	19	18	17	16	15	14	14	14	15	17	19	22	25	27	29	29	30
Group C Walls																								
N	15	14	13	12	11	10	9	8	8	7	7	8	8	9	10	12	13	14	15	16	17	17	17	16
E	22	21	19	17	15	14	12	12	14	16	19	22	25	27	29	29	30	30	30	29	28	27	26	24
S	21	19	18	16	15	13	12	10	9	9	9	10	11	14	17	20	22	24	25	26	25	25	24	22
W	31	29	27	25	22	20	18	16	14	13	12	12	12	13	14	16	20	24	29	32	35	35	35	33
Group D Walls																								
N	15	13	12	10	9	7	6	6	6	6	6	7	8	10	12	13	15	17	18	19	19	19	18	16
E	19	17	15	13	11	9	8	9	12	17	22	27	30	32	33	33	32	32	31	30	28	26	24	22
S	19	17	15	13	11	9	8	7	6	6	7	9	12	16	20	24	27	29	29	29	27	26	24	22
W	31	27	24	21	18	15	13	11	10	9	9	9	10	11	14	18	24	30	36	40	41	40	38	34
Group E Walls																								
N	12	10	8	7	5	4	3	4	5	6	7	9	11	13	15	17	19	20	21	23	20	18	16	14
E	14	12	10	8	6	5	6	11	18	26	33	36	38	37	36	34	33	32	30	28	25	22	20	17
S	15	12	10	8	7	5	4	3	4	5	9	13	19	24	29	32	34	33	31	29	26	23	20	17
W	25	21	17	14	11	9	7	6	6	6	7	9	11	14	20	27	36	43	49	49	45	40	34	29
Group F Walls																								
N	8	6	5	3	2	1	2	4	6	7	9	11	14	17	19	21	22	23	24	23	20	16	13	11
E	10	7	6	4	3	2	6	17	28	38	44	45	43	39	36	34	32	30	27	24	21	17	15	12
S	10	8	6	4	3	2	1	1	3	7	13	20	27	34	38	39	38	35	31	26	22	18	15	12
W	17	13	10	7	5	4	3	3	4	6	8	11	14	20	28	39	49	57	60	54	43	34	27	21
Group G Walls																								
N	3	2	1	0	1	2	7	8	9	12	15	18	21	23	24	24	25	26	22	15	11	9	7	5
E	4	2	1	0	-1	11	31	47	54	55	50	40	33	31	30	29	27	24	19	15	12	10	8	6
S	4	2	1	0	−1	0	1	5	12	22	31	39	45	46	43	37	31	25	20	15	12	10	8	5
W	6	5	3	2	1	1	2	5	8	11	15	19	27	41	56	67	72	67	48	29	20	15	11	8

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*CLTD may be converted to degrees C by multiplying by 5/9

Table 15-3, CLTD Correction For Latitude and Month Applied to Walls and Roofs, North Latitudes
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Lat.	Month	N	NE NNW	NE NW	ENE WNW	E W	ESE WSW	SE SW	SSE SSW	S	HOR
24	Dec	-5	7	9	-10	-7	-3	3	9	13	13
	Jan/Nov	4	-6	-8	-9	-6	-3	3	9	13	-11
	Feb/Oct	-4	-5	-6	-6	-3	-1	3	7	10	-7
	Mar/Sept	-3	-4	-3	-3	-1	1	1	2	4	-3
	Apr/Aug	-2	-1	0	1	-1	-2	-1	-2	-3	0
	May/Jul	1	2	2	0	0	-3	-3	-5	-6	1
	Jun	3	3	3	1	0	-3	-4	-6	-6	1
32	Dec	-5	-7	10	-11	-8	-5	2	9	12	-17
	Jan/Nov	-5	-7	-9	-11	8	-4	2	9	12	-15
	Feb/Oct	-4	-6	-7	-8	-4	-2	4	8	11	-10
	Mar/Sep	3	-4	-4	-4	-2	-1	3	5	7	-5
	Apr/Aug	-2	-2	-1	2	0	-1	0	1	1	-1
	May/Jul	1	1	1	0	0	-1	-1	-3	-3	1
	Jun	1	2	2	1	0	-2	-2	-4	-4	2
40	Dec	-6	-8	10	-13	-10	-7	0	7	10	-21
	Jan/Nov	5	-7	10	-12	-9	-6	1	8	11	-19
	Feb/Oct	-5	7	8	-9	-6	-3	3	8	12	-14
	Mar/Sep	4	5	5	-6	-3	1	4	7	10	-8
	Apr/Aug	2	3	2	-2	0	0	2	3	4	-3
	May/Jul	0	0	0	0	0	0	0	0	0	0
	Jun	1	1	1	0	1	0	0	-1	-1	2
48	Dec	-6	-8	-11	-14	-13	-10	-3	2	6	-25
	Jan/Nov	-6	-8	-11	13	-11	-8	-1	5	8	-24
	Feb/Oct	-5	-7	-10	-11	-8	-5	1	8	11	-18
	Mar/Sep	-4	-6	-6	-7	-4	-1	4	8	11	-11
	Apr/Aug	-3	3	-3	3	-1	0	4	6	7	-5
	May/Jul	0	-1	0	0	1	1	3	3	4	0
	Jun	1	1	2	1	2	1	2	2	3	2

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CLTD correction may be converted to degrees C by multiplying by 5/9

Table 15-4, Roof Construction Code
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Roof No.	Description	Code Number of Layers (see Table 7-6)
1	Steel sheet with 1-in. insulation	A0,E2,E3,B5,A3,E0
2	1-in. wood with 1-in. insulation	A0,E2,E3,B5,B7,E0
3	4-in. l.w. concrete	A0,E2,E3,C14,E0
4	2-in. h.w. concrete with 1-in. insulation	A0,E2,E3,B5,C12,E0
5	1-in. wood with 2-in. insulation	A0,E2,E3,B6,B7,E0
6	6-in. l.w. concrete	A0,E2,E3,C15,E0
7	2.5-in. wood with 1-in. insulation	A0,E2,E3,B5,B8,E0
8	8-in. l.w. concrete	A0,E2,E3,C16,E0
9	4-in. h.w. concrete with 1-in. insulation	A0,E2,E3,B5,C5,E0
10	2.5-in. wood with 2-in. insulation	A0,E2,E3,B6,B8,E0
11	Roof terrace system	A0,C12,B1,B6,E2,E3,C5,E0
12	6-in. h.w. concrete with 1-in. insulation	A0,E2,E3,B5,C13,E0
13	4-in. wood with 1-in. insulation	A0,E2,E3,B5,B9,E0

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Table 15-5,Wall Construction Group Description
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Group No.	Description of Construction	Weight (lb/ft^2)	U-value Btu/(hr·ft^2·F)	Code Numbers of Layers (see Table 7-4)
4-in. Face Brick + *(Brick)*				
D	4-in Common Brick	90	0.415	A0,A2,C4,E1,E0
C	1-in. Insulation or Air space + 4-in. Common Brick	90	0.174–0.301	A0,A2,C4,B1/B2,E1,E0
B	2-in. Insulation + 4-in. Common Brick	88	0.111	A0,A2,B3,C4,E1,E0
A	Insulation or Air space + 8-in. Common Brick	130	0.154–0.243	A0,A2,C9,B1/B2,E1,E0
4-in. Face Brick + *(H.W. Concrete)*				
B	2-in. Insulation + 4-in concrete	97	0.116	A0,A2,B3,C5,E1,E0
A	Air Space or Insulation + 8-in or more Concrete	143 190	0.110–0.112	A0,A2,B1,C10/11,E1,E0
4-in. Face Brick + *(L.W. or H.W. Concrete Block)*				
E	4-in. Block	62	0.319	A0,A2,C2,E1,E0
D	Air Space or Insulation + 4-in. Block	62	0.153–0.246	A0,A2,C2,B1/B2,E1,E0
C	Air Space or 1-in Insulation + 6-in or 8-in Block	73 89	0.221–0.275	A0,A2,B1,C7/C8,E1,E0
B	2-in Insulation + 8-in Block	89	0.096–0.107	A0,A2,B3,C7/C8,E1,E0
H.W. Concrete Wall + *(Finish)*				
E	4-in. Concrete	63	0.585	A0,A1,C5,E1,E0
D	4-in. Concrete + 1-in. or 2-in. Insulation	63	0.119–0.200	A0,A1,C5,B2/B3,E1,E0
C	2-in. Insulation + 4-in Concrete	63	0.119	A0,A1,B6,C5,E1,E0
C	8-in. Concrete	109	0.490	A0,A1,C10,E1,E0
B	8-in. Concrete + 1-in. or 2-in. Insulation	110	0.115–0.187	A0,A1,C10,B5/B6,E1,E0
A	2-in. Insulation + 8-in Concrete	110	0.115	A0,A1,B3,C10,E1,E0
L.W. and H.W. Concrete Block + *(Finish)*				
F	4-in. Block + Air Space/Insulation	29	0.161–0.263	A0,A1,C2,B1/B2,E1,E0
E	2-in Insulation + 4-in Block	29 37	0.105–0.114	A0,A1,B3,C2/C3,E1,E0
E	8-in. Block	47–51	0.294–0.402	A0,A1,C7/C8,E1,E0
D	8-in. Block + Air Space/Insulation	41–57	0.149–0.173	A0,A1,C7/C8,B1/B2,E1,E0
Metal Curtain Wall				
G	With/without Air Space + 1-in./2-in./3-in. Insulation	5–6	0.091–0.230	A0,A3,B5/B6/B12,A3,E0
Frame Wall				
G	1-in. to 3-in. Insulation	16	0.081–0.178	A0,A1,B1,B2/B3/B4,E1,E0

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Table 15-6, Thermal Properties and Code Numbers of Layers Used in Calculations of Coefficients for Roof and Wall
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Description	Code Number	Thickness and Thermal Properties				
		x	k	ρ	c	R
Outside surface resistance	A0					0.333
1-in. Stucco (asbestos cement or wood siding plaster, etc.)	A1	0.0833	0.4	116	0.20	0.208
4-in. face brick (dense concrete)	A2	0.333	0.75	130	0.22	0.444
Steel siding (aluminum or other lightweight cladding)	A3	0.005	26.0	480	0.10	0.0002
Finish	A6	0.0417	0.24	78	0.26	0.174
Air space resistance	B1					0.91
1-in. insulation	B2	0.083	0.025	2.0	0.2	3.32
2-in. insulation	B3	0.167	0.025	2.0	0.2	6.68
3-in. insulation	B4	0.25	0.025	2.0	0.2	10.03
1-in. insulation	B5	0.0833	0.025	5.7	0.2	3 33
2-in. insulation	B6	0.167	0.025	5.7	0.2	6.68
1-in. wood	B7	0.0833	0.07	37.0	0.6	1.19
2.5-in. wood	B8	0.2083	0.07	37.0	0.6	2.98
4-in. wood	B9	0.333	0.07	37.0	0.6	4.76
2-in. wood	B10	0.167	0.07	37.0	0.6	2.39
3-in. wood	B11	0.25	0.07	37.0	0.6	3.58
3-in. insulation	B12	0.25	0.025	5.7	0.2	10.0
4-in. clay tile	C1	0.333	0.33	70.0	0.2	1.01
4-in. l.w. concrete block	C2	0.333	0.22	38.0	0.2	1.51
4-in. h.w. concrete block	C3	0.333	0.47	61.0	0.2	0.71
4-in. common brick	C4	0.333	0.42	120	0.2	0.79
4-in. h.w. concrete	C5	0.333	1.0	140	0.2	0.333
8-in. clay tile	C6	0.667	0.33	70	0.2	2.02
8-in. l.w. concrete block	C7	0.667	0.33	38.0	0.2	2.02
8-in. h.w. concrete block	C8	0.667	0.6	61.0	0.2	1.11
8-in. common brick	C9	0.667	0.42	120	0.2	1.59
8-in. h.w. concrete	C10	0.667	1.0	140	0.2	0.667
12-in. h.w. concrete	C11	1.0	1.0	140	0.2	1.00
2-in. h.w. concrete	C12	0.167	1.0	140	0.2	0.167
6-in. h.w. concrete	C13	0.5	1.0	140	0.2	0.50
4-in. l.w. concrete	C14	0.333	0.1	40	0.2	3.33
6-in. l.w. concrete	C15	0.5	0.1	40	0.2	5.0
8-in. l.w. concrete	C16	0.667	0.1	40	0.2	6.67
Inside surface resistance	E0					0.685
0.75-in. plaster; 0.75-in. gypsum or other similar finishing layer	E1	0.0625	0.42	100	0.2	0.149
0.5-in. slag or stone	E2	0.0417	0.83	55	0.40	0.050
0.375-in. felt membrane	E3	0.0313	0.11	70	0.40	0.285
Ceiling air space	E4					1.0
Acoustic tile	E5	0.0625	0.035	30	0.20	1.786

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*Units: x = ft; c = Btu/(lb-F); k = Btu/(hr-ft-F); R = (hr-ft^2-F)/Btu; ρ = lb/ft^3

15.4.2 Fenestration - Heat admission or loss through fenestration areas are affected by many factors of which the following are the most significant

a. Solar radiation intensity and incident angle.

b. Difference between outdoor and indoor air temperatures.

c. Velocity and direction of flow across the exterior and interior surfaces.

d. Low temperature radiation exchange between the surfaces of the glass and the surroundings.

e. Exterior and interior shading.

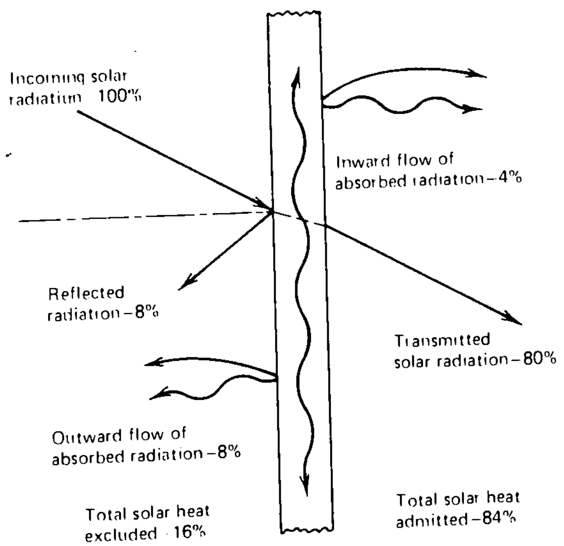

Figure 15-3, Distribution of Solar Radiation Falling on Clear Plate Glass
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

For fenestration the heat gain can be represented as:

Total Heat Gain = Solar Heat Gain + Conduction Heat Gain

The conduction heat gain for glass is calculated in the same way as for walls and roofs. Table 15-7 gives CLTD values for glass, 78°F indoor - 95°F max outdoor temperature and a 21°F daily range.

Table 15-7, Cooling Load Temperature Difference for Conduction Though Glass and Conduction Through Doors Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Solar Time, hr	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
CLTD, F	1	0	−1	−2	−2	−2	−2	0	2	4	7	9	12	13	14	14	13	12	10	8	6	4	3	2

Corrections: The values in the table were calculated for an inside temperature of 78 F and an outdoor maximum temperature of 95 F with an outdoor daily range of 21 F. The table remains approximately correct for other outdoor maximums (93 − 102 F) and other outdoor daily ranges (16 − 34 F), provided the outdoor daily average temperature remains approximately 85 F. If the room air temperature is different from 78 F, and/or the outdoor daily average temperature is different from 85 F, correct as shown in Eq. (7-13).

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†The *CLTD* may be converted to deg C by multiplying by 5/9

The solar heat gain is much more complex because the sun's rays change direction minute by minute. The solar gain through glass is given by:

$$q_\theta = \dot{A}(SC)SHGF(CLF)\theta$$

where: A = Area, ft^2
SC = Shading coefficient (internal)
SHGF = Solar heat gain factor, btu/(hr ft^2)
CLF = Cooling load factor for time θ

The shading coefficient is given by:

$$SC = \frac{\text{Solar heat gain of fenestration}}{\text{Solar heat gain of double-strength glass (reference glass)}}$$

Reference glass - .87 transmittance
.08 reflectance
.05 absorptance

Table 15-8, Shading Coefficient for Single Glass and Insulating Glass
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Type of Glass	Nominal Thickness in.	Nominal Thickness mm	Solar Trans.[b]	Shading Coefficient $h_o = 4.0$ Btu/(hr-ft^2-F)	Shading Coefficient $h_o = 3.0$ Btu/(hr-ft^2-F)
A. Single Glass					
Clear	$\frac{1}{8}$	3.2	0.84	1.00	1.00
	$\frac{1}{4}$	6.4	0.78	0.94	0.95
	$\frac{3}{8}$	9.5	0.72	0.90	0.92
	$\frac{1}{2}$	12.7	0.67	0.87	0.88
Heat Absorbing	$\frac{1}{8}$	3.2	0.64	0.83	0.85
	$\frac{1}{4}$	6.4	0.46	0.69	0.73
	$\frac{3}{8}$	9.5	0.33	0.60	0.64
	$\frac{1}{2}$	12.7	0.24	0.53	0.58
B. Insulating Glass[a]					
Clear Out, Clear In	$\frac{1}{8}$[c]	3.2	0.71[e]	0.88	0.88
Clear Out, Clear In	$\frac{1}{4}$	6.4	0.61	0.81	0.82
Heat Absorbing[d] Out, Clear In	$\frac{1}{4}$	6.4	0.36	0.55	0.58

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[a]Refers to factory-fabricated units with $\frac{3}{16}$, $\frac{1}{4}$, or $\frac{1}{2}$-in. air space or to prime windows plus storm sash
[b]Refer to manufacturer's literature for values.
[c]Thickness of each pane of glass, not thicknes of assembled unit.
[d]Refers to gray, bronze, and green tinted heat-absorbing float glass
[e]Combined transmittance for assembled unit

Blinds, shades and drapes or curtains that are often installed on the inside next to the windows decrease the solar heat gain. The shading coefficient is also used to express this effect. Tables 15-9 and 15-10 show these coefficients.

The shading coefficients for draperies are a complex function of color and weave of fabric. Figure 15-4 and table 15-11 give a brief summary of shading coefficients given in the ASHRAE Fundamentals book.

Table 15-9, Shading Coefficients For Single Glass With Indoor Shading by Venetian Blinds and Roller Shades Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

				Type of Shading				
				Venetian Blinds		Roller Shade		
	Nominal Thickness					Opaque		Translucent
Type of Glass	in.	mm	Solar Transmittance	Medium	Light	Dark	White	Light
Regular sheet	[illegible] to [illegible]	2–6	0.87–0.80					
Regular plate/float	[illegible] to [illegible]	6–13	0.80–0.71					
Regular pattern	[illegible] to [illegible]	3–6	0.87–0.79	0.64	0.55	0.59	0.25	0.39
Heat-absorbing pattern	[illegible]	3	—					
Gray sheet	[illegible], [illegible]	5–6	0.74, 0.71					
Heat-absorbing plate/float	[illegible], $\frac{1}{4}$	5–6	0.46					
Heat-absorbing pattern	[illegible], $\frac{1}{4}$	5–6	—	0.57	0.53	0.45	0.30	0.36
Gray sheet	[illegible], [illegible]	3–6	0.59, 0.45					
Heat-absorbing plate/float or pattern	—		0.44–0.30	0.54	0.52	0.40	0.28	0.32
Heat-absorbing plate/float	[illegible]	10	0.34					
Heat-absorbing plate or pattern	—		0.29–0.15 0.24	0.42	0.40	0.36	0.28	0.31
Reflective coated glass (no inside shade)								
Shading coefficient = 0.30				0.25	0.23			
Shading coefficient = 0.40		--		0.33	0.29			
Shading coefficient = 0.50				0.42	0.38			
Shading coefficient = 0.60				0.50	0.44			

*Adapted by permission from *ASHRAE Handbook of Fundamentals*, 1977

Table 15-10, Shading Coefficients For Insulating Glass With Indoor Shading by Venetian Blinds and Roller Shades
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

					Type of Shading				
	Nominal Thickness, Each light		Solar Transmittance		Venetian Blinds		Roller Shade		
							Opaque		Translucent
Type of Glass	in.	mm	Outer Pane	Inner Pane	Medium	Light	Dark	White	Light
Regular sheet out Regular sheet in	[illegible]	2–3	0.87	0.87	0.57	0.51	0.60	0.25	0.37
Regular plate/float out Regular plate/float in	[illegible]	6	0.80	0.80					
Heat-absorbing plate/float out Regular plate/float in	[illegible]	6	0.46	0.80	0.39	0.36	0.40	0.22	0.30
Reflective coated glass (no inside shade)									
Shading coefficient = 0.20					0.19	0.18			
Shading coefficient = 0.30					0.27	0.26			
Shading coefficient = 0.40					0.34	0.33			

*Adapted by permission from *ASHRAE Handbook Fundamentals* 1977

Figure 15-4, Indoor Shading Properties of Drapery Fabrics
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

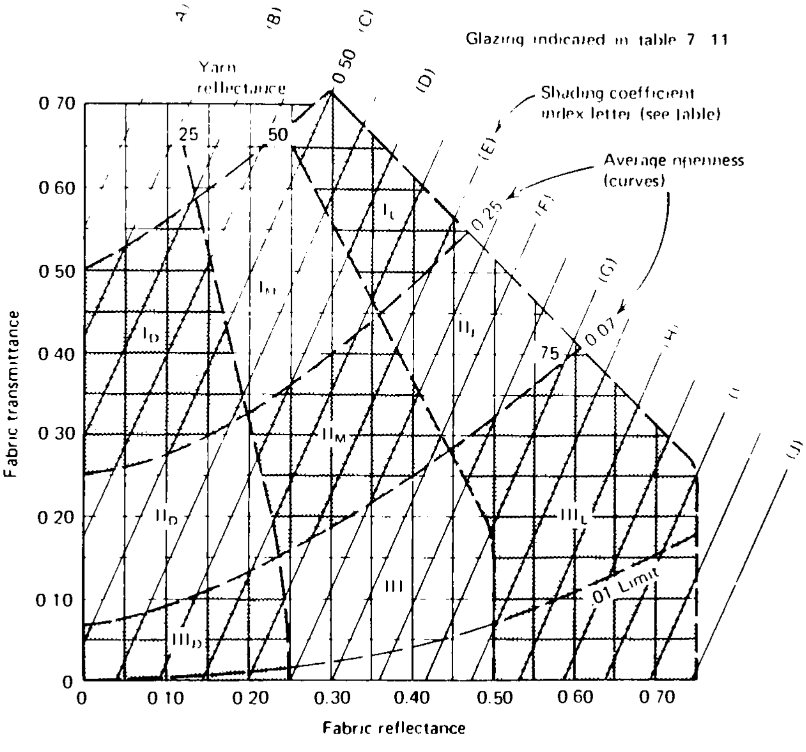

Table 15-11, Shading Coefficients For Single and Insulating Glass With Draperies
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Glazing	Glass Transmittance	Glass Shading Coefficient	J	I	H	G	F	E	D	C	B	A
Single Glass												
¼ in. or 6 mm Regular	0.80	0.95	0.35	0.40	0.45	0.50	0.55	0.60	0.65	0.70	0.75	0.80
½ in. or 13 mm Regular	0.71	0.88	0.35	0.39	0.43	0.48	0.52	0.56	0.61	0.66	0.70	0.74
¼ in. or 6 mm heat absorbing	0.46	0.67	0.33	0.36	0.38	0.41	0.44	0.46	0.49	0.52	0.54	0.57
½ in. or 13 mm heat absorbing	0.24	0.50	0.30	0.32	0.33	0.34	0.36	0.38	0.39	0.40	0.42	0.43
Reflective coated	—	0.60	0.33	0.36	0.38	0.41	0.43	0.46	0.49	0.51	0.54	0.57
(See manufacturers	—	0.50	0.31	0.33	0.34	0.36	0.38	0.39	0.41	0.42	0.44	0.46
literature for exact	—	0.40	0.26	0.27	0.28	0.29	0.30	0.32	0.33	0.34	0.35	0.36
values)	—	0.30	0.20	0.21	0.21	0.22	0.23	0.23	0.23	0.24	0.24	0.25
Insulating glass (½ in. or 13 mm air space) Regular out and regular in	0.64	0.83	0.35	0.37	0.42	0.45	0.48	0.52	0.56	0.58	0.62	0.66
Heat absorbing out and regular in	0.37	0.56	0.32	0.33	0.35	0.37	0.39	0.41	0.43	0.45	0.47	0.49
Reflective coated	—	0.40	0.28	0.28	0.29	0.31	0.32	0.34	0.36	0.37	0.37	0.38
(See manufacturers'	—	0.30	0.24	0.24	0.25	0.25	0.26	0.26	0.27	0.27	0.28	0.29
literature for exact values.)	—	0.20	0.15	0.15	0.16	0.16	0.17	0.17	0.18	0.18	0.19	0.19

*Adapted by permission from *ASHRAE Handbook of Fundamentals* 1977

The solar heat gain factor (SHGF) is the maximum value for the month for a given orientation and latitude. Tables 15-12 and 15-13 give some of these factors.

The cooling load factor (CLF) then represents the ratio of the actual solar heat gain, which becomes the cooling load, to the maximum solar heat gain. The cooling load factors depend on the actual solar heat gain for a particular time, the internal shading and the building construction when there is no internal shade.

Table 15-12, Maximum Solar Heat Gain Factor For Externally Shaded Glass, BTU/(HR-FT^2) (Based on Ground Reflectance of .2)
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Use for latitudes 0 24 deg
For latitudes greater than 24, use north orientation, Table 7-12
For horizontal glass in shade, use the tabulated values for all latitudes

	N	NNE/ NNW	NE/ NW	ENE/ WNW	E/ W	ESE/ WSW	SE/ SW	SSE/ SSW	S	(All Latit.) HOR
Jan	31	31	31	32	34	36	37	37	38	16
Feb	34	34	34	35	36	37	38	38	39	16
Mar	36	36	37	38	39	40	40	39	39	19
Apr	40	40	41	42	42	42	41	40	40	24
May	43	44	45	46	45	43	41	40	40	28
June	45	46	47	47	46	44	41	40	40	31
July	45	45	46	47	47	45	42	41	41	31
Aug	42	42	43	45	46	45	43	42	42	28
Sept	37	37	38	40	41	42	42	41	41	23
Oct	34	34	34	36	38	39	40	40	40	19
Nov	32	32	32	32	34	36	38	38	39	17
Dec.	30	30	30	31	32	34	36	37	37	15

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To convert to W/m² multiply by 3.155

Table 15-13, Maximum Solar Heat Gain Factor, BTU/(HR-FT^2), for Sunlit Glass, North Latitudes

Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

32 Deg

	N (Shade)	NNE/ NNW	NE/ NW	ENE/ WNW	E/ W	ESE/ WSW	SE/ SW	SSE/ SSW	S	HOR
Jan	24	24	29	105	175	229	249	250	246	176
Feb	27	27	65	149	205	242	248	232	221	217
Mar	32	37	107	183	227	237	227	195	176	252
Apr	36	80	146	200	227	219	187	141	115	271
May	38	111	170	208	220	199	155	99	74	277
June	44	122	176	208	214	189	139	83	60	276
July	40	111	167	204	215	194	150	96	72	273
Aug	37	79	141	195	219	210	181	136	111	265
Sep	33	35	103	173	215	227	218	189	171	244
Oct	28	28	63	143	195	234	239	225	215	213
Nov	24	24	29	103	173	225	245	246	243	175
Dec	22	22	22	84	162	218	246	252	252	158

40 Deg

	N (Shade)	NNE/ NNW	NE/ NW	ENE/ WNW	E/ W	ESE/ WSW	SE/ SW	SSE/ SSW	S	HOR
Jan	20	20	20	74	154	205	241	252	254	133
Feb	24	24	50	129	186	234	246	244	241	180
Mar	29	29	93	169	218	238	236	216	206	223
Apr	34	71	140	190	224	223	203	170	154	252
May	37	102	165	202	220	208	175	133	113	265
June	48	113	172	205	216	199	161	116	95	267
July	38	102	163	198	216	203	170	129	109	262

40 Deg

	N (Shade)	NNE/ NNW	NE/ NW	ENE/ WNW	E/ W	ESE/ WSW	SE/ SW	SSE/ SSW	S	HOR
Aug	35	71	135	185	216	214	196	165	149	247
Sep	30	30	87	160	203	227	226	209	200	215
Oct	25	25	49	123	180	225	238	236	234	177
Nov	20	20	20	73	151	201	237	248	250	132
Dec.	18	18	18	60	135	188	232	249	253	113

48 Deg

	N (Shade)	NNE/ NNW	NE/ NW	ENE/ WNW	E/ W	ESE/ WSW	SE/ SW	SSE/ SSW	S	HOR
Jan	15	15	15	53	118	175	216	239	245	85
Feb	20	20	36	103	168	216	242	249	250	138
Mar	26	26	80	154	204	234	239	232	228	188
Apr	31	61	132	180	219	225	215	194	186	226
May	35	97	158	200	218	214	192	163	150	247
June	46	110	165	204	215	206	180	148	134	252
July	37	96	156	196	214	209	187	158	146	244
Aug	33	61	128	174	211	216	208	188	180	223
Sep	27	27	72	144	191	223	228	223	220	182
Oct	21	21	35	96	161	207	233	241	242	136
Nov	15	15	15	52	115	172	212	234	240	85
Dec	13	13	13	36	91	156	195	225	233	65

[a]Reprinted by permission from *ASHRAE Handbook of Fundamentals*, 1977

[b]To convert to W/m^2 multiply by 3.155

Table 15-14, Cooling Load Factors for Glass Without Interior Shading, North Latitudes Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Glass Facing	Room Construction	Solar Time, hr									
		1	2	3	4	5	6	7	8	9	10
N (Shaded)	L	0.17	0.14	0.11	0.09	0.08	0.33	0.42	0.48	0.56	0.63
	M	0.23	0.20	0.18	0.16	0.14	0.34	0.41	0.46	0.53	0.59
	H	0.25	0.23	0.21	0.20	0.19	0.38	0.45	0.49	0.55	0.60
NE	L	0.01	0.04	0.01	0.02	0.02	0.23	0.41	0.51	0.51	0.45
	M	0.07	0.06	0.06	0.05	0.04	0.21	0.36	0.44	0.45	0.40
	H	0.09	0.08	0.08	0.07	0.07	0.23	0.37	0.44	0.44	0.39
E	L	0.04	0.03	0.01	0.02	0.02	0.19	0.37	0.51	0.57	0.57
	M	0.07	0.06	0.06	0.05	0.05	0.18	0.33	0.44	0.50	0.51
	H	0.09	0.09	0.08	0.08	0.07	0.20	0.34	0.45	0.49	0.49
SE	L	0.05	0.04	0.04	0.03	0.03	0.13	0.28	0.43	0.55	0.62
	M	0.09	0.08	0.07	0.06	0.05	0.14	0.26	0.38	0.48	0.54
	H	0.11	0.10	0.10	0.09	0.08	0.17	0.28	0.40	0.49	0.53
S	L	0.08	0.07	0.05	0.04	0.04	0.06	0.09	0.14	0.22	0.34
	M	0.12	0.11	0.09	0.08	0.07	0.08	0.11	0.14	0.21	0.31
	H	0.13	0.12	0.12	0.11	0.10	0.11	0.14	0.17	0.24	0.33
SW	L	0.12	0.10	0.08	0.06	0.05	0.06	0.08	0.10	0.12	0.14
	M	0.15	0.14	0.12	0.10	0.09	0.09	0.10	0.12	0.13	0.15
	H	0.15	0.14	0.13	0.12	0.11	0.12	0.13	0.14	0.16	0.17
W	L	0.12	0.10	0.08	0.06	0.05	0.06	0.07	0.08	0.10	0.11
	M	0.15	0.13	0.11	0.10	0.09	0.09	0.09	0.10	0.11	0.12
	H	0.14	0.13	0.12	0.11	0.10	0.11	0.12	0.13	0.14	0.14
NW	L	0.11	0.09	0.08	0.06	0.05	0.06	0.08	0.10	0.12	0.14
	M	0.14	0.12	0.11	0.09	0.08	0.09	0.10	0.11	0.13	0.15
	H	0.14	0.12	0.11	0.10	0.10	0.10	0.12	0.13	0.15	0.16
HOR	L	0.11	0.09	0.07	0.06	0.05	0.07	0.14	0.24	0.36	0.48
	M	0.16	0.14	0.12	0.11	0.09	0.11	0.16	0.24	0.33	0.43
	H	0.17	0.16	0.15	0.14	0.13	0.15	0.20	0.28	0.36	0.45

Glass Facing	Room Construction	11	12	13	14	15	16	17	18	19	20	21	22	23	24
N (Shaded)	L	0.71	0.76	0.80	0.82	0.82	0.79	0.79	0.84	0.61	0.48	0.38	0.31	0.25	0.20
	M	0.65	0.70	0.74	0.75	0.76	0.74	0.75	0.79	0.61	0.50	0.42	0.36	0.31	0.27
	H	0.65	0.69	0.72	0.72	0.72	0.70	0.70	0.75	0.57	0.46	0.39	0.34	0.31	0.28
NE	L	0.39	0.36	0.33	0.31	0.28	0.26	0.23	0.19	0.15	0.12	0.10	0.08	0.06	0.05
	M	0.36	0.33	0.31	0.30	0.28	0.26	0.24	0.21	0.17	0.15	0.13	0.11	0.09	0.08
	H	0.34	0.31	0.29	0.27	0.26	0.24	0.22	0.20	0.17	0.14	0.13	0.12	0.11	0.10
E	L	0.50	0.42	0.37	0.32	0.29	0.25	0.22	0.19	0.15	0.12	0.10	0.08	0.06	0.05
	M	0.46	0.39	0.35	0.31	0.29	0.26	0.23	0.21	0.17	0.15	0.13	0.11	0.10	0.08
	H	0.43	0.36	0.32	0.29	0.26	0.24	0.22	0.19	0.17	0.15	0.13	0.12	0.11	0.10
SE	L	0.63	0.57	0.48	0.42	0.37	0.31	0.28	0.24	0.19	0.15	0.12	0.10	0.08	0.07
	M	0.56	0.51	0.45	0.40	0.36	0.33	0.29	0.25	0.21	0.18	0.16	0.14	0.12	0.10
	H	0.53	0.48	0.41	0.36	0.31	0.30	0.27	0.24	0.20	0.18	0.16	0.14	0.13	0.12
S	L	0.48	0.59	0.65	0.65	0.59	0.50	0.43	0.36	0.28	0.22	0.18	0.15	0.12	0.10
	M	0.42	0.52	0.57	0.58	0.53	0.47	0.41	0.35	0.29	0.25	0.21	0.18	0.16	0.14
	H	0.43	0.51	0.56	0.55	0.50	0.41	0.37	0.32	0.26	0.22	0.20	0.18	0.16	0.15
SW	L	0.16	0.24	0.36	0.49	0.60	0.66	0.66	0.58	0.43	0.33	0.27	0.22	0.18	0.14
	M	0.17	0.23	0.33	0.44	0.53	0.58	0.59	0.53	0.41	0.33	0.28	0.24	0.21	0.18
	H	0.19	0.25	0.34	0.44	0.52	0.56	0.56	0.49	0.37	0.30	0.25	0.21	0.19	0.17
W	L	0.12	0.14	0.20	0.32	0.45	0.57	0.64	0.61	0.44	0.34	0.27	0.22	0.18	0.14
	M	0.13	0.14	0.19	0.29	0.40	0.50	0.56	0.55	0.41	0.33	0.27	0.23	0.20	0.17
	H	0.15	0.16	0.21	0.30	0.40	0.49	0.54	0.52	0.38	0.30	0.24	0.21	0.18	0.16
NW	L	0.16	0.17	0.19	0.23	0.33	0.47	0.59	0.60	0.42	0.33	0.26	0.21	0.17	0.14
	M	0.16	0.17	0.18	0.21	0.30	0.42	0.51	0.54	0.39	0.32	0.26	0.22	0.19	0.16
	H	0.18	0.18	0.19	0.22	0.30	0.41	0.50	0.51	0.36	0.29	0.23	0.20	0.17	0.15
HOR	L	0.58	0.66	0.72	0.74	0.73	0.67	0.59	0.47	0.37	0.29	0.24	0.19	0.16	0.13
	M	0.52	0.59	0.64	0.67	0.66	0.62	0.56	0.47	0.38	0.32	0.28	0.24	0.21	0.18
	H	0.52	0.59	0.62	0.64	0.62	0.58	0.51	0.42	0.35	0.29	0.26	0.23	0.21	0.19

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Table 15-15. Cooling Load Factors for Glass With Interior Shading, North Latitudes Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Glass Facing	Solar Time, hr																							
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
N	0.08	0.07	0.06	0.06	0.07	0.73	0.66	0.65	0.73	0.80	0.86	0.89	0.89	0.86	0.82	0.75	0.78	0.91	0.24	0.18	0.15	0.13	0.11	0.10
NE	0.03	0.02	0.02	0.02	0.02	0.56	0.76	0.74	0.58	0.37	0.29	0.22	0.26	0.24	0.22	0.20	0.16	0.12	0.06	0.05	0.04	0.04	0.03	0.03
E	0.03	0.02	0.02	0.02	0.02	0.47	0.72	0.80	0.76	0.62	0.41	0.27	0.24	0.22	0.20	0.17	0.14	0.11	0.06	0.05	0.05	0.04	0.03	0.03
SE	0.03	0.03	0.02	0.02	0.02	0.30	0.57	0.74	0.81	0.79	0.68	0.49	0.33	0.28	0.25	0.22	0.18	0.13	0.08	0.07	0.06	0.05	0.04	0.04
S	0.04	0.04	0.03	0.03	0.03	0.09	0.16	0.23	0.38	0.58	0.75	0.83	0.80	0.68	0.50	0.35	0.27	0.19	0.11	0.09	0.08	0.07	0.06	0.05
SW	0.05	0.05	0.04	0.04	0.03	0.07	0.11	0.14	0.16	0.19	0.22	0.38	0.59	0.75	0.83	0.81	0.69	0.45	0.16	0.12	0.10	0.09	0.07	0.06
W	0.05	0.05	0.04	0.04	0.03	0.06	0.09	0.11	0.13	0.15	0.16	0.17	0.31	0.53	0.72	0.82	0.81	0.61	0.16	0.12	0.10	0.08	0.07	0.06
NW	0.05	0.04	0.04	0.03	0.03	0.07	0.11	0.14	0.17	0.19	0.20	0.21	0.22	0.30	0.52	0.73	0.82	0.69	0.16	0.12	0.10	0.08	0.07	0.06
HOR	0.06	0.05	0.04	0.04	0.03	0.12	0.27	0.44	0.59	0.72	0.81	0.85	0.85	0.81	0.71	0.58	0.42	0.25	0.14	0.12	0.10	0.08	0.07	0.06

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15.5 Cooling Load - Internal Sources

For internal heat sources:

$$\dot{q}_\theta = \dot{q}_i (CLF)_\theta$$

where: $\dot{q}_i$ = Instantaneous heat gain from lights, people and equipment, btu/hr or watts.
CLF_θ = Cooling load factor for time θ.

15.5.1 People - Table 15-16 shows the latent and sensible heat gains caused by people at different activity levels. The latent heat gain goes directly into the air in the space, so this heat gain component immediately becomes part of the cooling load. However, the sensible component from a person is delayed due to storage of part of the energy in the room furnishings. The CLF depends on the number of hours a person spends in the space and the time of entry. These factors are found in table 15-17. For large people densities CLF = 1.0.

Table 15-16, Rates of Heat Gain from Occupants of Conditioned Spaces
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

		Total Heat Adults, Male		Total Heat Adjusted[c]		Sensible Heat		Latent Heat	
Degree of Activity	Typical Application	Watts	Btu/hr	Watts	Btu/hr	Watts	Btu/hr	Watts	Btu/hr
Seated at rest	Theater, movie	115	400	100	350	60	210	40	140
Seated, very light work writing	Offices, hotels, apts	140	480	120	420	65	230	55	190
Seated, eating	Restaurant[d]	150	520	170	580[d]	75	255	95	325
Seated, light work, typing	Offices, hotels, apts	185	640	150	510	75	255	75	255
Standing, light work or walking slowly	Retail Store, Bank	235	800	185	640	90	315	95	325
Light bench work	Factory	255	880	230	780	100	345	130	435
Walking, 3 mph, light machine work	Factory	305	1040	305	1040	100	345	205	695
Bowling[e]	Bowling alley	350	1200	280	960	100	345	180	615
Moderate dancing	Dance hall	400	1360	375	1280	120	405	255	875
Heavy work, heavy machine work, lifting	Factory	470	1600	470	1600	165	565	300	1035
Heavy work, athletics	Gymnasium	585	2000	525	1800	185	635	340	1165

[a]Reprinted by permission from *ASHRAE Handbook of Fundamentals*, 1977.
[b]Note Tabulated values are based on 78 F room dry-bulb temperature For 80 F room dry-bulb, the total heat remains the same, but the sensible heat value should be decreased by approximately 8% and the latent heat values increased accordingly.
[c]Adjusted total heat gain is based on normal percentage of men, women, and children for the application listed, with the postulate that the gain from an adult female is 85% of that for an adult male, and that the gain from a child is 75% of that for an adult male
[d]Adjusted total heat value for eating in a restaurant, includes 60 Btu/hr for food per individual (30 Btu sensible and 30 Btu latent)
[e]For bowling figure one person per alley actually bowling, and all others as sitting (400 Btu/hr) or standing and walking slowly (790 Btu/hr)

Table 15-17, Sensible Heat Cooling Load Factors for People
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Total Hours in Space	Hours after Each Entry into Space																							
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
2	0.49	0.58	0.17	0.13	0.10	0.08	0.07	0.06	0.05	0.04	0.04	0.03	0.03	0.02	0.02	0.02	0.02	0.01	0.01	0.01	0.01	0.01	0.01	0.01
4	0.49	0.59	0.66	0.71	0.27	0.21	0.16	0.14	0.11	0.10	0.08	0.07	0.06	0.06	0.05	0.04	0.04	0.03	0.03	0.03	0.02	0.02	0.02	0.01
6	0.50	0.60	0.67	0.72	0.76	0.79	0.34	0.26	0.21	0.18	0.15	0.13	0.11	0.10	0.08	0.07	0.06	0.06	0.05	0.04	0.04	0.03	0.03	0.03
8	0.51	0.61	0.67	0.72	0.76	0.80	0.82	0.84	0.38	0.30	0.25	0.21	0.18	0.15	0.13	0.12	0.10	0.09	0.08	0.07	0.06	0.05	0.05	0.04
10	0.53	0.62	0.69	0.74	0.77	0.80	0.83	0.85	0.87	0.89	0.42	0.34	0.28	0.23	0.20	0.17	0.15	0.13	0.11	0.10	0.09	0.08	0.07	0.06
12	0.55	0.64	0.70	0.75	0.79	0.81	0.84	0.86	0.88	0.89	0.91	0.92	0.45	0.36	0.30	0.25	0.21	0.19	0.16	0.14	0.12	0.11	0.09	0.08
14	0.58	0.66	0.72	0.77	0.80	0.83	0.85	0.87	0.89	0.90	0.91	0.92	0.93	0.94	0.47	0.38	0.31	0.26	0.23	0.20	0.17	0.15	0.13	0.11
16	0.62	0.70	0.75	0.79	0.82	0.85	0.87	0.88	0.90	0.91	0.92	0.93	0.94	0.95	0.95	0.96	0.49	0.39	0.33	0.28	0.24	0.20	0.18	0.16
18	0.66	0.74	0.79	0.82	0.85	0.87	0.89	0.90	0.92	0.93	0.94	0.94	0.95	0.96	0.96	0.97	0.97	0.97	0.50	0.40	0.33	0.28	0.24	0.21

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15.5.2 Lighting - The lighting load has a major effect on the cooling load in a building. There are a number of factors that must be considered to actually predict the heat gain.

A. Some energy is emitted in the form of radiation and absorbed by the space.

B. The absorbed energy is later transferred to the air by convection.

C. The manner in which lights are installed.

D. Mass structure of the building.

The instantaneous heat gain for lights may be expressed by:

$$\dot{q}_i = 3.412WF_uF_s$$

where: W = Summation of installed lights wattage, watts
F_u = Use factor - ratio of wattage in use to wattage installed
F_s = Special allowance factor for lights requiring more power than their rated wattage. For typical 40 W florescent lamps, F_s = 1.20

The cooling load factor lights is given by:

$$\dot{q} = \dot{q}_i(CLF)$$

Tables 15-18, 15-19 and 15-20 determine CLF factors.

Table 15-18, Cooling Load Factors When Lights are on for 10 Hours
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

"a" Classification	"b" Classification	Number of Hours After Lights Are Turned on: 0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
	A	0.03	0.47	0.58	0.66	0.73	0.78	0.82	0.86	0.88	0.91	0.93	0.49	0.39	0.32	0.26	0.21	0.17	0.13	0.11	0.09	0.07	0.06	0.05	0.04
	B	0.10	0.54	0.59	0.63	0.66	0.70	0.73	0.76	0.78	0.80	0.82	0.39	0.35	0.32	0.28	0.26	0.23	0.21	0.19	0.17	0.15	0.14	0.12	0.11
0.45	C	0.15	0.59	0.61	0.64	0.66	0.68	0.70	0.72	0.73	0.75	0.76	0.33	0.31	0.29	0.27	0.26	0.24	0.23	0.21	0.20	0.19	0.18	0.17	0.16
	D	0.18	0.62	0.63	0.64	0.66	0.67	0.68	0.69	0.69	0.70	0.71	0.27	0.26	0.26	0.25	0.24	0.23	0.23	0.22	0.21	0.21	0.20	0.19	0.19
	A	0.02	0.57	0.65	0.72	0.78	0.82	0.85	0.88	0.91	0.92	0.94	0.40	0.32	0.26	0.21	0.17	0.14	0.11	0.09	0.07	0.06	0.05	0.04	0.03
	B	0.08	0.62	0.66	0.69	0.73	0.75	0.78	0.80	0.82	0.84	0.85	0.32	0.29	0.26	0.23	0.21	0.19	0.17	0.15	0.14	0.12	0.11	0.10	0.09
0.55	C	0.12	0.66	0.68	0.70	0.72	0.74	0.75	0.77	0.78	0.79	0.81	0.27	0.25	0.24	0.22	0.21	0.20	0.19	0.17	0.16	0.15	0.14	0.14	0.13
	D	0.15	0.69	0.70	0.71	0.72	0.73	0.73	0.74	0.75	0.76	0.76	0.22	0.22	0.21	0.20	0.20	0.19	0.18	0.18	0.17	0.17	0.16	0.16	0.15
	A	0.02	0.66	0.73	0.78	0.83	0.86	0.89	0.91	0.93	0.94	0.95	0.31	0.25	0.20	0.16	0.13	0.11	0.08	0.07	0.05	0.04	0.04	0.03	0.02
	B	0.06	0.71	0.74	0.76	0.79	0.81	0.83	0.84	0.86	0.87	0.89	0.25	0.22	0.20	0.18	0.16	0.15	0.13	0.12	0.11	0.10	0.09	0.08	0.07
0.65	C	0.09	0.74	0.75	0.77	0.78	0.80	0.81	0.82	0.83	0.84	0.85	0.21	0.20	0.18	0.17	0.16	0.15	0.14	0.14	0.13	0.12	0.11	0.11	0.10
	D	0.11	0.76	0.77	0.77	0.78	0.79	0.79	0.80	0.81	0.81	0.82	0.17	0.17	0.16	0.16	0.15	0.15	0.14	0.14	0.14	0.13	0.13	0.12	0.12
	A	0.01	0.76	0.81	0.84	0.88	0.90	0.92	0.93	0.95	0.96	0.97	0.22	0.18	0.14	0.12	0.09	0.08	0.06	0.05	0.04	0.03	0.03	0.02	0.02
	B	0.04	0.79	0.81	0.83	0.85	0.86	0.88	0.89	0.90	0.91	0.92	0.18	0.16	0.14	0.13	0.12	0.10	0.09	0.08	0.08	0.07	0.06	0.06	0.05
0.75	C	0.07	0.81	0.82	0.83	0.84	0.85	0.86	0.87	0.88	0.89	0.89	0.15	0.14	0.13	0.12	0.12	0.11	0.10	0.10	0.09	0.09	0.08	0.08	0.07
	D	0.08	0.83	0.83	0.84	0.84	0.85	0.85	0.86	0.86	0.87	0.87	0.12	0.12	0.12	0.11	0.11	0.11	0.10	0.10	0.10	0.09	0.09	0.09	0.09

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Table 15-19, "a" Classification for Lights Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

"a"	Light Fixture and Ventilation Arrangements
0.45	Recessed lights which are not vented Low air supply rate—less than 0.5 cfm/ft^2 of floor area Supply and return diffusers below ceiling
0.55	Recessed lights which are not vented Medium to high air supply rate—more than 0.5 cfm/ft^2 of floor area Supply and return diffusers below ceiling or through ceiling space and grill
0.65	Vented light fixtures Medium to high air supply rate—more than 0.5 cfm/ft^2 of floor area Supply air through ceiling or wall but return air flows around light fixtures and through ceiling space
0.75	Vented or free hanging lights Supply air through ceiling or wall but return air flows around light fixtures and through a ducted return

[a]Reprinted by permission from *ASHRAE Handbook of Fundamentals*, 1977.
[b]Based on rooms having an average amount of furnishings.

Table 15-20, "b" Classification for Lights
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Room Air Circulation and Type of Supply and Return	Floor Construction and Floor Weight in Pounds Per Square Foot of Floor Area				
	2 in. Wooden Floor 10 lb/ft²	3 in. Concrete Floor 40 lb/ft²	6 in. Concrete Floor 75 lb/ft²	8 in. Concrete Floor 120 lb/ft²	12 in. Concrete Floor 160 lb/ft²
Low ventilation rate — minimum required to handle cooling load. Supply through floor, wall or ceiling diffuser. Ceiling space not vented.	B	B	C	D	D
Medium ventilation rate. Supply through floor, wall or ceiling diffuser. Ceiling space not vented.	A	B	C	D	D
High room air circulation induced by primary air of induction unit or by fan coil unit. Return through ceiling space.	A	B	C	C	D
Very high room air circulation used to minimize room temperature gradients. Return through ceiling space	A	A	B	C	D

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ᵇBased on floor covered with carpet and rubber pad. For floor covered with floor tile use letter designation in next row down with the same floor weight

15.5.3 Unhooded Appliances - Tables 15-21 are CLF factors for unhooded appliances.

Table 15-21, Sensible Heat Cooling Load Factors for Appliances-Unhooded
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

FACTORS FOR APPLIANCES—UNHOODED

Total Operational Hours	Hours after Appliances Are On																							
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
2	0.56	0.64	0.15	0.11	0.08	0.07	0.06	0.05	0.04	0.04	0.03	0.03	0.02	0.02	0.02	0.02	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01
4	0.57	0.65	0.71	0.75	0.23	0.18	0.14	0.12	0.10	0.08	0.07	0.06	0.05	0.05	0.04	0.04	0.03	0.03	0.02	0.02	0.02	0.02	0.01	0.01
6	0.57	0.65	0.71	0.76	0.79	0.82	0.29	0.22	0.18	0.15	0.13	0.11	0.10	0.08	0.07	0.06	0.06	0.05	0.04	0.04	0.03	0.03	0.03	0.02
8	0.58	0.66	0.72	0.76	0.80	0.82	0.85	0.87	0.33	0.26	0.21	0.18	0.15	0.13	0.11	0.10	0.09	0.08	0.07	0.06	0.05	0.04	0.04	0.03
10	0.60	0.68	0.73	0.77	0.81	0.83	0.85	0.87	0.89	0.90	0.36	0.29	0.24	0.20	0.17	0.15	0.13	0.11	0.10	0.08	0.07	0.07	0.06	0.05
12	0.62	0.69	0.75	0.79	0.82	0.84	0.86	0.88	0.89	0.91	0.92	0.93	0.38	0.31	0.25	0.21	0.18	0.16	0.14	0.12	0.11	0.09	0.08	0.07
14	0.64	0.71	0.76	0.80	0.83	0.85	0.87	0.89	0.90	0.92	0.93	0.93	0.94	0.95	0.40	0.32	0.27	0.23	0.19	0.17	0.15	0.13	0.11	0.10
16	0.67	0.74	0.79	0.82	0.85	0.87	0.89	0.90	0.91	0.92	0.93	0.94	0.95	0.96	0.96	0.97	0.42	0.34	0.28	0.24	0.20	0.18	0.15	0.13
18	0.71	0.78	0.82	0.85	0.87	0.89	0.90	0.92	0.93	0.94	0.94	0.95	0.96	0.96	0.97	0.97	0.97	0.98	0.43	0.35	0.29	0.24	0.21	0.18

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15.6 Heat Gain From Infiltration and Outdoor Ventilation Air

The method to determine infiltration was discussed in the Heating Load section.

15.7 Summation of Heat Gains and Cooling Loads

Figure 6-5 shows a convenient and systematic approach to the calculation and summation of the cooling load for a structure.

15.8 Cooling Loads by the Transfer Function Method

This method, through the use of computers, is able to solve the heat gian-cooling load problem, taking into account more variables to obtain solutions for every hour of the day with greater accuracy than hand calculations. Also, this method allows for detailed studies of the energy requirements of the structure. This method was developed by ASHRAE Task Group on Energy Requirements.

A transfer function is a set of coefficients that relate an output function at some specific time to the value of one or more driving functions that time and previous values of both input and output functions.

15.8.1 Heat Gains Though Exterior Walls and Roofs - The heat transfer from the inside surface of a wall at a particular time is related through the use of appropriate coefficients to the sol-air temperature at the same time and previous times, to the space temperature and to the heat

Figure 15-5, Cooling Load Worksheet
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

1	Identification of Room					1			2			3		
2	Length of Exposed Wall - Feet													
3	Room Dimensions - Feet													
4	Comments													
	Load Component	Direction	U or SC	CLTD or SHGF Hour	CLF or Δt Hour	Area, cfm, etc.	Heat Load Btuh	Cooling Load-Btuh Hour	Area, cfm, etc	Heat Load Btuh	Cooling Load-Btuh	Area, cfm, etc	Heat Load Btuh	Cooling Load-Btuh Hour
5	Gross Exposed Walls	N												
		S												
		E												
		W												
6	Windows and Doors-Convective	N												
		S												
		E												
		W												
7	Windows and Glass Doors-Solar	N												
		S												
		E												
		W												
8	Net Exposed Walls and Partitions	N												
		S												
		E												
		W												
9	Roof													
10	Floors													
11	Infiltration, Sensible			1 1										
12	Infiltration, Latent			4840										
13	People, Sensible													
14	People, Latent													
15	Internal, Sensible													
16	Internal, Latent													
17	Lights													
18	Duct Loss													
19	Total Btuh													

transfer rate at previous times. This may be written as:

$$\dot{q}_{i,\theta} = A[\sum_{n=0} b_n(t_{e,\theta-n\blacktriangle}) - \sum_{n=1} d_n(\dot{q}_{i,\theta-n\blacktriangle})/A - t_i \sum_{n=0} c_n]$$

Where b_n, d_n and c_n are transfer function coefficients that depend on the construction of the wall or roof section. ▲ is the size of the time element.

These coefficients are given in tables 15-22 , 15-23 and 15-24.

To solve the heat gain, the sol-air temperature as a function of time is needed. This is given by:

$$t_e = t_o + \alpha G_t/h_o - \varepsilon \blacktriangle R/h_o$$

The dry bulb temperature, t_o, is computed by the following:

$$t_o = t_d - \blacktriangle R(x)$$

where: t_d = Design dry bulb temperature, °F
$\blacktriangle R$ = Daily range, °F
x = Percentage of daily range divided by 100. Table 15-25 gives this percentage.

* Note: The Heat gains from solar radiation on windows, heat sources within the space, ventilation and infiltration are computed as previously discussed.

Table 15-22, Transfer Function Coefficients for Exterior Walls (Time Interval = 1.0 hr)
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Construction Description	Code Numbers of Layers		Coefficients[a] b_n and d_n: $n = 0$	$n = 1$	$n = 2$	$n = 3$	$n = 4$	$n = 5$	$n = 6$	U	$\sum_{n=0} c_n$
4 in or 100 mm face brick, 2 in or 50 mm insulation, and 4 in. or 100 mm lightweight concrete block	A0, A2, B3	b	0.00000	0.00046	0.00225	0.00150	0.00016				
	C2, E1, E0	d	1.00000	−1.73771	0.90936	−0.11373	0.00496	−0.00001		0.102	0.00437
4 in. or 100 mm face brick, air space, and 4 in. or 100 mm common brick	A0, A2, B1	b	0.00000	0.00086	0.00485	0.00378	0.00050	0.00001			
	C4, E1, E0	d	1.00000	−1.79201	0.98014	−0.16102	0.00609	−0.00003		0.301	0.01000
4 in. or 100 mm face brick, air space, and 4 in. or 100 mm lightweight concrete block	A0, A2, B1	b	0.00003	0.00286	0.01029	0.00504	0.00037				
	C2, E1, E0	d	1.00000	−1.50943	0.65654	−0.07415	0.00212			0.248	0.01859
12 in or 300 mm heavyweight concrete	A0, A1, C11	b	0.00000	0.00029	0.00303	0.00412	0.00105	0.00005			
	E1, E0	d	1.00000	−1.86853	1.09284	−0.21487	0.01094	−0.00009		0.421	0.00854
Frame wall with 4 in or 100 mm brick veneer	A0, A2, B6	b	0.00037	0.00823	0.00983	0.00125	0.00001				
	A6, E0	d	1.00000	−1.03045	0.20108	−0.00726				0.121	0.01969
Frame wall	A0, A6, B6	b	0.01977	0.06317	0.01064	0.00006					
	A6, E0	d	1.00000	−0.25848	0.01072					0.124	0.09364
Frame wall with 3 in. or 25 mm insulation	A0, A1, B1	b	0.00509	0.02644	0.00838	0.00010					
	B4, E1, E0	d	1.00000	−0.59602	0.08757	−0.00002				0.081	0.04001

Construction Description	Code Numbers of Layers		Coefficients[a] b_n and d_n: $n = 0$	$n = 1$	$n = 2$	$n = 3$	$n = 4$	$n = 5$	$n = 6$	U	$\sum_{n=0} c_n$
4 in or 100 mm face brick, 2 in. or 50 mm insulation and 4 in. or 100 mm common brick	A0, A2, B3	b	0.00000	0.00012	0.00084	0.00082	0.00014				
	C4, E1, E0	d	1.00000	−1.96722	1.20279	−0.22850	0.01033	−0.00006		0.111	0.00192
8 in. or 200 mm heavyweight concrete block	A0, A1, C8	b	0.0004	0.0171	0.0310	0.0065	0.0001				
	E1, E0	d	1.0000	−1.1621	0.3132	−0.0139				0.402	0.0551
4 in. or 100 mm face brick, 8 in. or 200 mm clay tile and airspace	A0, A2, C6	b	0.0000	0.0000	0.0007	0.0019	0.0011	0.0001			
	B1, E1, E0	d	1.0000	−2.1290	1.5667	−0.4781	0.0605	−0.0029		0.221	0.0038
4 in. or 100 mm heavyweight concrete with 1 in. or 25 mm insulation	A0, A1, C5	b	0.0005	0.0094	0.0106	0.0013					
	B2, E1, E0	d	1.0000	−1.1763	0.3011	−0.0157				0.200	0.0218
4 in. or 100 mm heavyweight concrete	A0, A1, C5	b	0.0078	0.0705	0.0355	0.0011					
	E1, E0	d	1.0000	−0.8789	0.0753	−0.0001				0.585	0.1149
Sheet metal with 1 in. or 25 mm insulation	A0, A3, B2	b	0.1424	0.0479							
	B1, A3, E0	d	1.0000	−0.0013						0.191	0.1903

[a] Adapted by permission from *ASHRAE Handbook of Fundamentals*, 1977

[a] U, b's, and c's are in Btu/(hr-ft²-F), and d is dimensionless. To convert U, b's, and c's to W/(m²-C) multiply by 5.6783

Table 15-23, Transfer Function Coefficients for Roofs (Time Interval = 1.0 hr)
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Construction Description	Code Numbers of Layers		Coefficients* b_n and d_n								
			$n = 0$	$n = 1$	$n = 2$	$n = 3$	$n = 4$	$n = 5$	$n = 6$	U†	$\sum_{n=0} c_n$
Roof terrace system	A0, C12, B1, B6, E2, E3, C5, E4, E5, E0	b	0.00000	0.00008	0.00048	0.00039	0.00006			0.082	0.00101
		d	1.0000	−1.7304	0.8564	−0.1611	0.0024				
1 in. or 25 mm wood with 1 in. or 25 mm insulation	A0, E2, E3, B5, B7, E4, E5, E0	b	0.0003	0.0082	0.0103	0.0014				0.115	0.0202
		d	1.0000	−1.0046	0.1845	−0.0046					
8 in. or 200 mm lightweight concrete	A0, E2, E3, C16, E4, E5, E0	b	0.00000	0.00002	0.00046	0.00133	0.00079	0.00011		0.092	0.00271
		d	1.00000	−1.91091	1.22135	−0.31019	0.03001	−0.00095	0.00001		
4 in or 100 mm lightweight concrete	A0, E2, E3, C14, E4, E5, E0	b	0.0001	0.0055	0.0141	0.0045	0.0002			0.134	0.0244
		d	1.0000	−1.0698	0.2665	−0.0143	0.0001				
Steel sheet with 1 in. or 25 mm insulation	A0, E2, E3, B5, A3, E4, E5, E0	b	0.0085	0.0505	0.0179	0.0004				0.134	0.0773
		d	1.0000	−0.4700	0.0476						
2.5 in. or 64 mm wood with 1 in. or 25 mm insulation	A0, E2, E3, B5, B8, E0	b	0.0000	0.0017	0.0068	0.0035	0.0003			0.130	0.0123
		d	1.0000	−1.3557	0.5121	−0.0634	0.0015				
6 in. or 150 mm heavyweight concrete with 1 in. or 25 mm insulation	A0, E2, E3, B5, C13, E0	b	0.0001	0.0036	0.0068	0.0016				0.192	0.121
		d	1.0000	−1.3001	0.3991	−0.0361	0.0001				
4 in. or 100 mm heavyweight concrete with 1 in. or 25 mm insulation	A0, E2, E3, B5, C5, E0	b	0.0008	0.0117	0.0100	0.0008				0.200	0.0233
		d	1.0000	−1.0800	0.2015	−0.0051					

*Adapted by permission from *ASHRAE Handbook of Fundamentals* 1977

†U, b's and c's are in Btu/(hr-ft²-F), and d is dimensionless. To convert U, b's, and c's to W/(m²-C), multiply by 5.6783

Table 15-24, Transfer Function Coefficients for Interior Partitions, Floors, and Ceilings (Time Interval = 1.0 hr)
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Construction Description	Code Numbers of Layers	Coefficients[a] b_n and d_n	$n = 0$	$n = 1$	$n = 2$	$n = 3$	$n = 4$	$n = 5$	$n = 6$	U	$\sum c_n$
4 in. or 100 mm lightweight concrete block with $\frac{3}{4}$ in. or 19 mm plaster	E0, E1, C2, E1, E0	b	0.0048	0.0514	0.0339	0.0018				0.314	0.0919
		d	1.0000	−0.8456	0.1397	−0.0015					
4 in. or 100 mm heavyweight concrete block with $\frac{3}{4}$ in. or 19 mm plaster	E0, E1, C3, E1, E0	b	0.0092	0.0705	0.0318	0.0008				0.421	0.1123
		d	1.0000	−0.8203	0.0874	−0.0001					
8 in. or 200 mm lightweight concrete block, plastered both sides	E0, E1, C7, E1, E0	b	0.0002	0.0106	0.0214	0.0054	0.0001			0.271	0.0377
		d	1.0000	−1.2098	0.3736	−0.0246	0.0001				
8 in. or 200 mm heavyweight concrete block, plastered both sides	E0, E1, C8, E1, E0	b	0.0004	0.0141	0.0231	0.0043	0.0001			0.360	0.0420
		d	1.0000	−1.1995	0.3293	−0.0132					
12 in. or 300 mm heavyweight concrete plastered both sides	E0, E1, C11, E1, E0	b	0.0000	0.0002	0.0023	0.0029	0.0007			0.372	0.0006
		d	1.0000	−1.8959	1.1220	−0.2203	0.0107	−0.0001			
Frame, partition with $\frac{3}{4}$ in. or 19 mm gypsum board	E0, E1, B1, E1, E0	b	0.0729	0.1526	0.0159					0.388	0.2414
		d	1.0000	−0.3986	0.0208						
2 in. or 50 mm heavyweight concrete floor deck	E0, A5, C12, E0	b	0.0505	0.0691	0.0011					0.362	0.1207
		d	1.0000	−0.6662							
4 in. or 100 mm heavyweight concrete floor deck	E0, A5, C5, E0	b	0.0111	0.0405	0.0072					0.341	0.0588
		d	1.0000	−0.8507	0.0229						
4 in. or 100 mm lightweight concrete floor deck	E0, A5, C2, E0	b	0.0200	0.0710	0.0120					0.243	0.1030
		d	1.0000	−0.5878	0.0116						
2 in. or 50 mm wood deck	E0, A5, B10, E0	b	0.0020	0.0245	0.0174	0.0010				0.201	0.0449
		d	1.0000	−0.9025	0.1271	−0.0008					
4 in. or 100 mm lightweight concrete deck with false ceiling	E0, A5, C2, E4, E5, E0	b	0.0020	0.0195	0.0104	0.0004				0.144	0.0323
		d	1.0000	−0.8295	0.0534	−0.0003					
2 in. or 50 mm wood deck with false ceiling	E0, A5, B10, E4, E5, E0	b	0.0001	0.0048	0.0078	0.0014				0.129	0.0141
		d	1.0000	−1.1372	0.2530	−0.0061					
Steel deck with false ceiling	E0, A5, A3, E4, E5, E0	b	0.0749	0.0853	0.0022					0.186	0.1624
		d	1.0000	−0.1257	0.0001						

[a] Adapted by permission from *ASHRAE Handbook of Fundamentals* 1977.
[b] U, b's and c's are in Btu/(hr-ft²-F) and d is dimensionless. To convert U, b's and c's to W/(m²-C) multiply by 5.6783.

Table 15-25, Percentage of the Daily Range Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Time, hr	Percent	Time, hr	Percent	Time, hr	Percent	Time, hr	Percent
1	87	7	93	13	11	19	34
2	92	8	84	14	3	20	47
3	96	9	71	15	0	21	58
4	99	10	56	16	3	22	68
5	100	11	39	17	10	23	76
6	98	12	23	18	21	24	82

*Reprinted by permission from *ASHRAE Handbook of Fundamentals*, 1977.
Hour 1 is 1:00 A.M. solar time

15.9.2 Cooling Load by The Transfer Function Method -

The cooling load for a space at a particular time may be related through the use of appropriate coefficients to the heat gain at that time and to previous times and to the cooling load at previous times. The cooling load is given by:

$$\dot{q}_{c,\theta} = \sum_{i=1}(v_0\dot{q}_{i,\theta} + v_1\dot{q}_{i,\theta-\Delta} + v_2\dot{q}_{i,\theta-2\Delta} + \cdots) - w_1\dot{q}_{c,\theta-\Delta} - w_2\dot{q}_{c,\theta-2\Delta} - w_3\dot{q}_{c,\theta-3\Delta} - \cdots$$

where: v = Coefficient dependent on the nature of the heat gain and mass of the structure.
w = Coefficient dependent on the mass of the structure.

The equation below is used to estimate the fraction of the heat gain F_c that results in the cooling load.

$$F_c = 1 - .02K_t$$

The v coefficients are then modified by being multiplied by this coefficient. K_t is the unit length conductance and is given by:

$$K_t = 1/L\ [(UA)_w + (UA)_{ow} + (UA)_c]$$

where: L = Length of entrance wall, ft.
U = Overall heat transfer coefficient of room element, btu/(hr ft^2 °F) (w-window, ow-outside wall, c-corridor)
A = Area of the room element, ft^2

Table 15-26 and 15-27 show values for v and w coefficients.

Table 15-26, Coefficients of Room Transfer Functions
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Heat Gain Component	Room Envelope Construction	v_0	v_1 Dimensionless	v_2
Solar heat gain through glass with no interior shading and heat generated by equipment and people which is dissipated by radiation	Light	0.224	$= 1 + w_1 - v_0$	0.0
	Medium	0.197	$= 1 + w_1 - v_0$	0.0
	Heavy	0.187	$= 1 + w_1 - v_0$	0.0
Conduction heat gain through exterior walls, roofs, partitions and doors, and windows with blinds or drapes	Light	0.703	$= 1 + w_1 - v_0$	0.0
	Medium	0.681	$= 1 + w_1 - v_0$	0.0
	Heavy	0.676	$= 1 + w_1 - v_0$	0.0
Heat generated by lights	Light	0.0	= "a" in Table 7-19	$= 1 + w_1 - v_1$
	Medium	0.0	= "a" in Table 7-19	$= 1 + w_1 - v_1$
	Heavy	0.0	= "a" in Table 7-19	$= 1 + w_1 - v_1$
Heat generated by equipment and people which is dissipated by convection and energy gain due to ventilation and infiltration air	Light	1.0	0.0	0.0
	Medium	1.0	0.0	0.0
	Heavy	1.0	0.0	0.0

*Reprinted by permission from *ASHRAE Handbook of Fundamentals* 1977

Table 15-27, The Value of w_1 for Different Room Air Circulation Rates and Envelope Construction
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Room Envelope Construction[b]	2-in. Wood Floor	3-in. Concrete Floor	6-in. Concrete Floor	8-in. Concrete Floor	12-in. Concrete Floor	Room Air[c] Circulation & Type of Supply and Return
Specific Mass, lb/ft² of floor area	10	40	75	120	160	
	−0.88	−0.92	−0.95	−0.97	−0.98	Low
	−0.84	−0.90	−0.94	−0.96	−0.97	Medium
	−0.81	−0.88	−0.93	−0.95	−0.97	High
	−0.77	−0.85	−0.92	−0.95	−0.97	Very high
	−0.73	−0.83	−0.91	−0.94	−0.96	

[a]Reprinted by permission from *ASHRAE Handbook of Fundamentals*, 1977
[c]*Low*: Low ventilation rate-minimum required to cope with cooling load due to lights and occupants in interior zone. Supply through floor, wall, or ceiling diffuser. Ceiling space not vented
Medium: Medium ventilation rate, supply through floor, wall, or ceiling diffuser. Ceiling space not vented
High: Room air circulation induced by primary air of induction unit or by fan coil unit. Return through ceiling space
Very high: High room circulation used to minimize temperature gradient in a room. Return through ceiling space
[b]Floor covered with carpet and rubber pad; for a floor covered only with floor tile take next w_1 value down the column

"Summary of the Transfer Function Method - The heat gain and cooling load calculations using the transfer function method should be carried out by a digital computer. The general procedure is outlined as follows:

1. Derive hourly values of the outdoor dry bulb temperature.
2. Compute the sol-air temperatures for each surface and each hour.
3. Compute the instantaneous sensible heat gain for each wall, partition, and roof. There is a heat gain for each hour of the day for each surface.

4. Compute the instantaneous sensible heat gain for the doors. Doors are assumed to have negligible energy storage.
5. Compute the convective heat gain for the windows.
6. Compute the solar radiation heat gain for the windows.
7. Compute the heat gain due to the lights, which is simply the power input to the lights for the times they are on.
8. Compute the sensible heat gain due to people for the hours the space is occupied.
9. Compute the sensible heat gain due to infiltration for each hour. This may not be constant for each hour.
10. Compute the latent heat gain due to infiltration and people for each hour.
11. Sum the instantaneous heat gains that will appear immediately as cooling load.
12. Sum the solar heat gains for glass that have interior shading.
13. Sum the conduction-convection heat gains through the roof, walls, windows and solar gains from widows with inside blinds or drapes.
14. Sum the heat gain due to lights that are not on all the time.

15. Sum the heat gains due to equipment and people and dissipated by radiation.
16. Transform the separate heat gain totals of items 12, 13, 14 and 15 above to the cooling load.
17. Obtain the total space cooling load for each hour by summing items 10, 11 and 16."3

15.9 Heat Extraction Rate and Room Temperature

Simple energy balance:

$$\dot{q}_x - \dot{q}_c = (mc)_{air}\, {}^{dt}r/_{d\theta}$$

where: $\dot{q}_x$ = heat extraction rate, btu/hr

$\dot{q}_c$ = Cooling load, btu/hr

$(mc)air\, {}^{dt}r/_{d\theta}$ = space heat increase rate, btu/hr

The above equation is represented below as a transfer function, Air Transfer Function:

$$\sum_{i=0} P_i(\dot{q}_{x,\theta-i\blacktriangle} - \dot{q}_{c,\theta-2\blacktriangle}) = \sum_{i=0} g_i(t_i - t_{r,\theta-i\blacktriangle})$$

where: P_i, g_i = Transfer function coefficient

$\dot{q}_c$ = Cooling load at the various times

t_i = Room temperature used for cooling loads

t_r = Actual room temperature at various times

$$g_0 = g^*_0\, A_{f1} + [UA + \underline{p}C_p(\dot{Q}_i + \dot{Q}_v)]P_0$$

$$g_1 = g^*_1\, A_{f1} + [UA + \underline{p}C_p(\dot{Q}_i + \dot{Q}_v)]P_0$$

$$g_2 = g^*_2\, A_{f1}$$

where: g^*_i = Normalized, table 15-28, btu/(hr ft^2 °F)

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3 Faye C. Mc. Quiston and Jerald D. Parker. Heating Ventilation and Air Conditioning Analysis and Design, 2nd ed. New York: John Wiley and Sons, 1982

A_{fl} = Floor area, ft^2

UA = Conductance

$\dot{Q}_i$ = Infiltration rate, ft^3/hr

$\dot{Q}_v$ = Ventilation rate, ft^3/hr

P_i = Coefficient, table 15-28

Table 15-28, Normalized Coefficients of the Room Air Transfer Function
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Room Envelope Construction	Btu/(hr-ft²-F)			Dimensionless	
	g_0^*	g_1^*	g_2^*	p_0	p_1
Light	+1.68	−1.73	+0.05	1.0	−0.82
Medium	+1.81	−1.89	+0.08	1.0	−0.87
Heavy	+1.85	−1.95	+0.10	1.0	−0.93

*Reprinted with permission from *ASHRAE Handbook of Fundamentals*, 1977.
[a]For all cases, the room is assumed furnished.

The heat extraction rate can be represented by the linear function:

$$\dot{q}_{x,\theta} = W + St_{r,\theta}$$

where: W and S are parameters that characterize the equipment at time θ.

$$S = [\dot{q}_{x,max} - \dot{q}_{x,min}]/\Delta t_r$$

$$W = [\dot{q}_{x,max} + \dot{q}_{x,min}]/2 - St^*_{r,\theta}$$

Where: Δt_r = Throttle range

$\dot{q}_{x,min}$, $\dot{q}_{x,max}$ = Extraction rate

t^*_r = Set point

t_r = Room Temperature

Figure 15-6 shows this relationship.

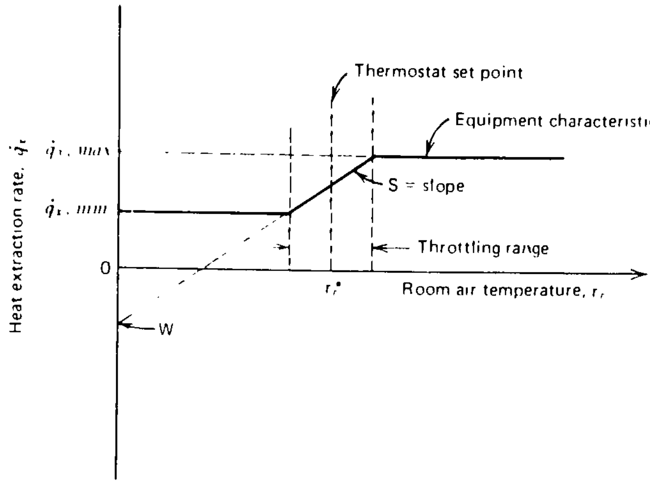

Figure 15-6, Cooling Equipment Characteristic
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

From the above equation we get:

$$\dot{q}_{x,o} = \frac{g_0 W}{S + g_0} + \frac{S}{S + g_0} G_\theta$$

$$\text{where: } G_\theta = t_i \sum_{i=0}^{2} g_i - \sum_{i=1}^{2} g_i (t_{i,\theta-i\Delta}) + \sum_{i=0}^{1} P_i(\dot{q}_{x,\theta-i\Delta}) - \sum_{i=0}^{i} P_i(\dot{q}_{x,\theta-i\Delta})$$

Combining the previous equations the following relationship was derived:

$$t_{r,\theta} = (G_\theta - \dot{q}_{x,\theta})/g_0$$

Figure 15-7 illustrates results obtained by using the above method.

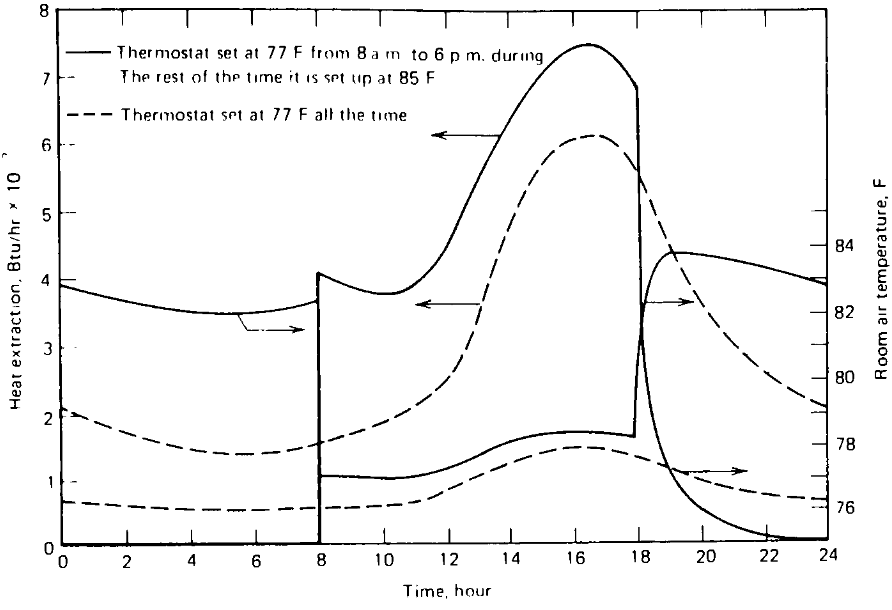

Figure 15-7, Room Air Temperature and Heat Extraction Rates for Continuous and Intermittent Operation
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

16. Water Piping Systems

Introduction - In HVAC design it is often necessary to supply chilled and/or heated water to parts of a building to control the comfort in that building. The first six sections discuss the basic type of piping systems. These systems are: series loop, Conventional one-pipe, two-pipe direct return, two-pipe reverse return, three-pipe, and four pipe systems. The seventh and eighth sections discusses pipe expansion, and pipe anchors and supports. The ninth section discusses pumps and the tenth section discusses special fittings and devices required in piping systems.

7.1 The Series Loop Heating System

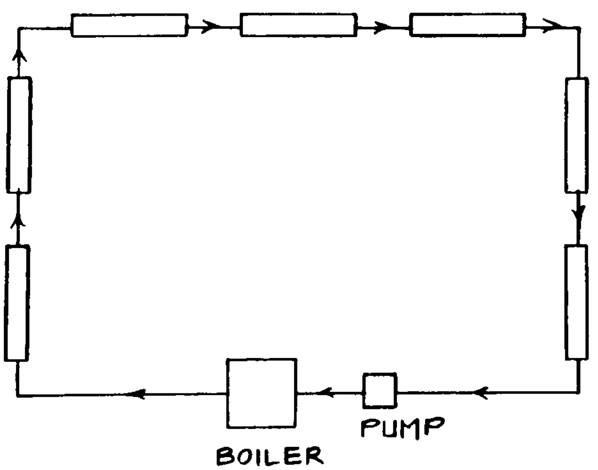

Figure 16-1, Series Loop Heating System

The simplest of all heating systems and is used almost exclusively for residential heating systems because of its limited capacity.

16.1.1 Advantages

A. Simplicity

B. Low cost

16.1.2 Disadvantages

A. If one unit is shut down then all the units are shut down.

B. Pipe size limited to 1 inch in diameter.

C. Units down stream receive cooler water so unit sizes may have to be increased to give the same heat output.

16.1 The Conventional One-Pipe System

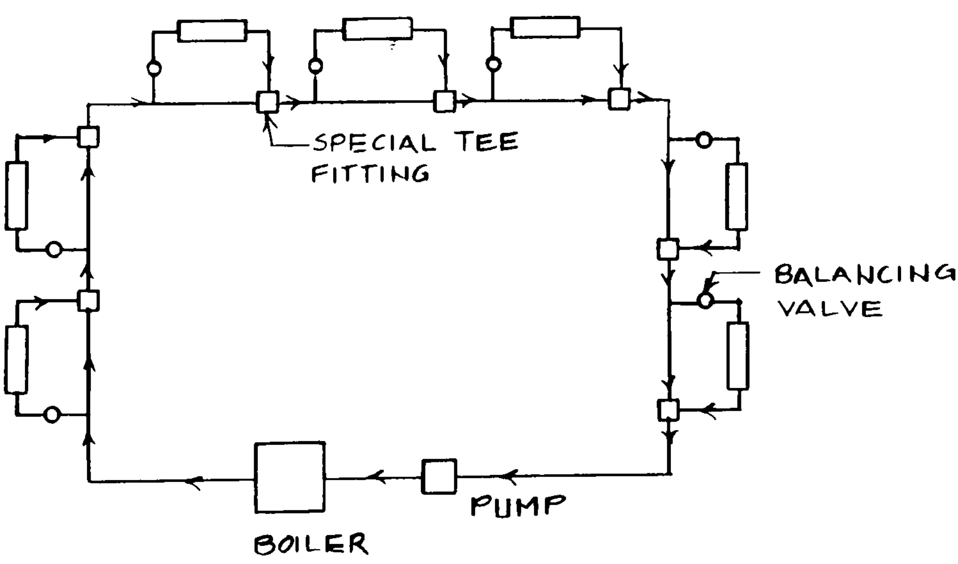

Figure 16-2, Conventional One-Pipe System

This is the most common type of one-pipe systems. The system's name sometimes depends on the manufacturer's trade name for the special tee fitting required for each of the heating or cooling units. This special fitting is designed to divert a portion of the flow in the main through the heat transfer device.

The special tee fitting causes a pressure drop which is equal to or greater than the pressure loss through the risers plus the heater, thus causing some water to flow through the short branch circuits.

The pressure drop across the heaters has to be kept reasonably low to keep the overall head loss to a minimum. The water temperature will decrease down stream causing down stream units to be larger for the same heat output.

16.2 The Two-Pipe Direct-Return System

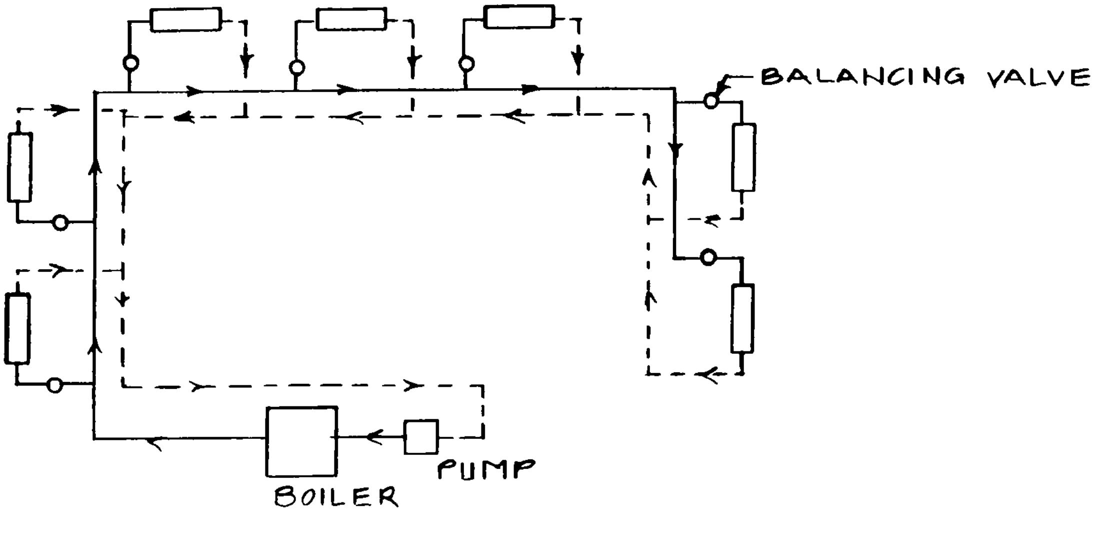

Figure 16-3, Two-Pipe Direct-Return System

In observing the diagram above, the distance along the piping through each heat-transfer device is different. This can cause problems in the balancing of this type of system.

The two-pipe arrangement requires more pipe for a given system than the one-pipe systems, therefor they are more expensive. The flexibility of the water distribution, temperature control, capacity and simplicity make this system and the reverse return system well suited for larger systems.

16.3 The Two-Pipe Reverse-Return System

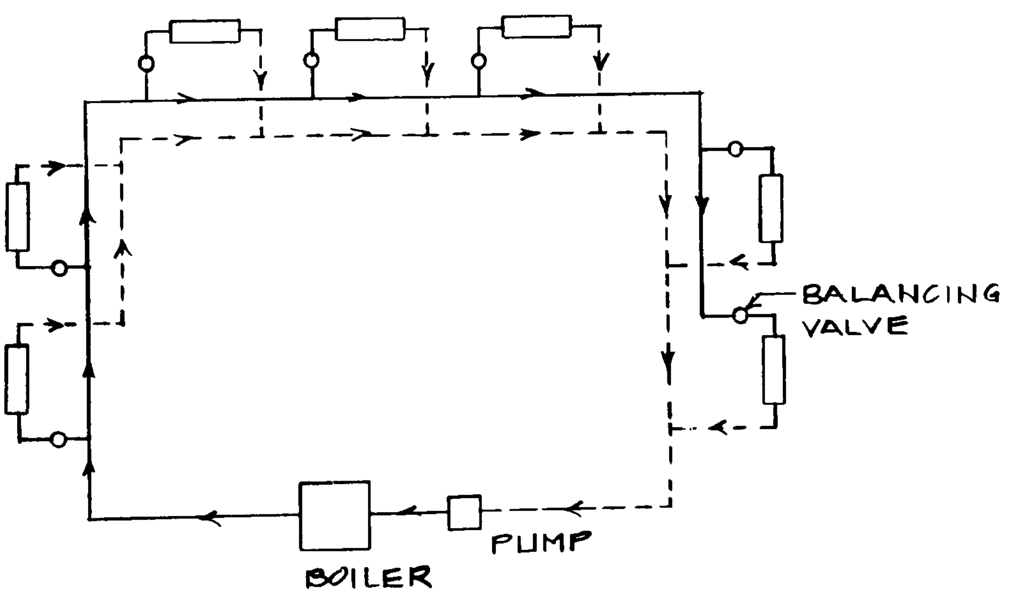

Figure 16-4, Two-Pipe Reverse-Return System

In this system the distance traveled by the water from the boiler or chiller, through each unit, and back is essentially the same. This helps the system to be self balancing. In practice it is necessary to put balancing valves in at each unit.

16.4 The Three-Pipe Arrangement

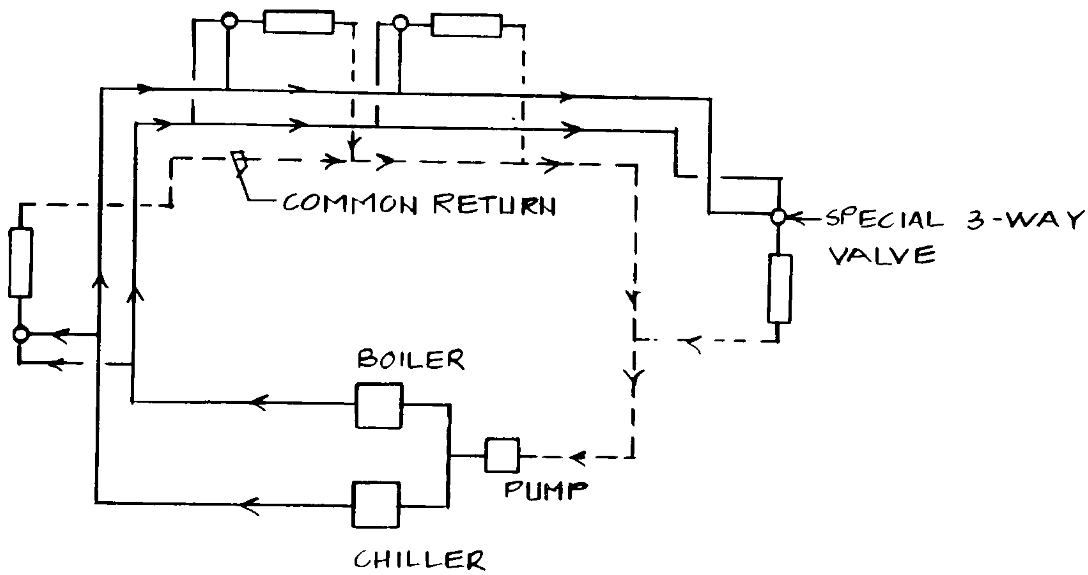

Figure 16-5, Three-Pipe System

The system has one hot water supply main, one chilled water supply main and one return. The special three-way valve only lets hot water or chilled water through the unit and will not temper the water.

Some units may be using hot water while other units chilled. This causes a mixing of the water in the return line. To avoid excessive mixing and reduce operating cost, it is usually necessary to zone return lines and to direct the chilled water through the chiller and hot water through the heat exchanger.

16.5 Four-Pipe Arrangement

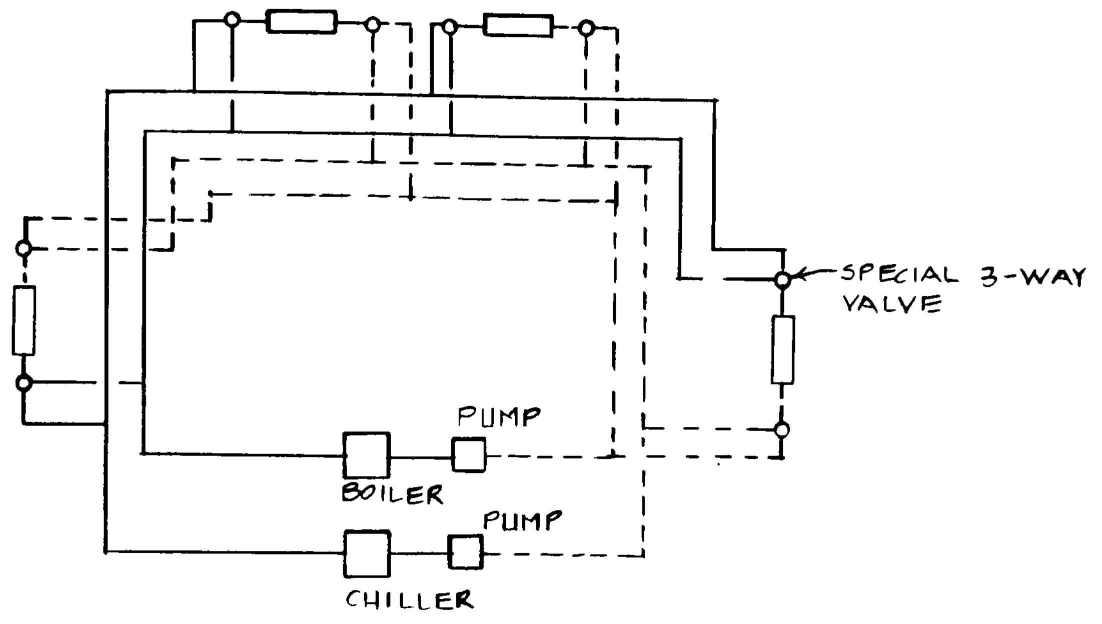

Figure 16-6, Four-Pipe System

With a four pipe system, the ultimate in simplicity of design, room temperature control and economy of operation is accomplished with only a slight increase in first cost over the three-pipe arrangement.

16.6 Pipe Expansion

Pipe expansion is caused by the change in temperature in a piping system. If the pipe is not free to expand or contract, pipe joints may leak, pipe fittings may crack or buckle and damage may be done to the structure.

The problem relates to the temperature of the system from cold start to operating temperature. The length change in a pipe, from this temperature change, can be calculated by the following:

$$\Delta L = L_o \alpha (t - t_o)$$

where: ΔL - Change in pipe length
L_o = Original pipe length
α = Linear coefficient of expansion of the pipe material
$t-t_o$ = Temperature change

Below is a table showing expansion of pipe per 100 feet for different temperature ranges.

Table 16-1, Linear Expansion of Pipe in Inches per 100 Feet

Temperature Change,* (°F)	Steel	Brass and Copper
50	0.38	0.57
60	0.45	0.64
70	0.53	0.74
80	0.61	0.85
90	0.68	0.95
100	0.76	1.06
110	0.84	1.17
120	0.91	1.28
130	0.99	1.38
140	1.07	1.49
150	1.15	1.60
160	1.22	1.70
170	1.30	1.81
180	1.37	1.91
190	1.45	2.02
200	1.57	2.13
250	1.99	2.66
300	2.47	3.19
350	2.94	3.72
400	3.46	4.25

*Between fluid in pipe and surrounding air

The increase in pipe length must be absorbed either by the design of the system or by the installation of devices designed to absorbed this expansion. Expansion joints composed of pipe fittings and commercially available expansion bends may also be used. Below are examples these joints and bends.

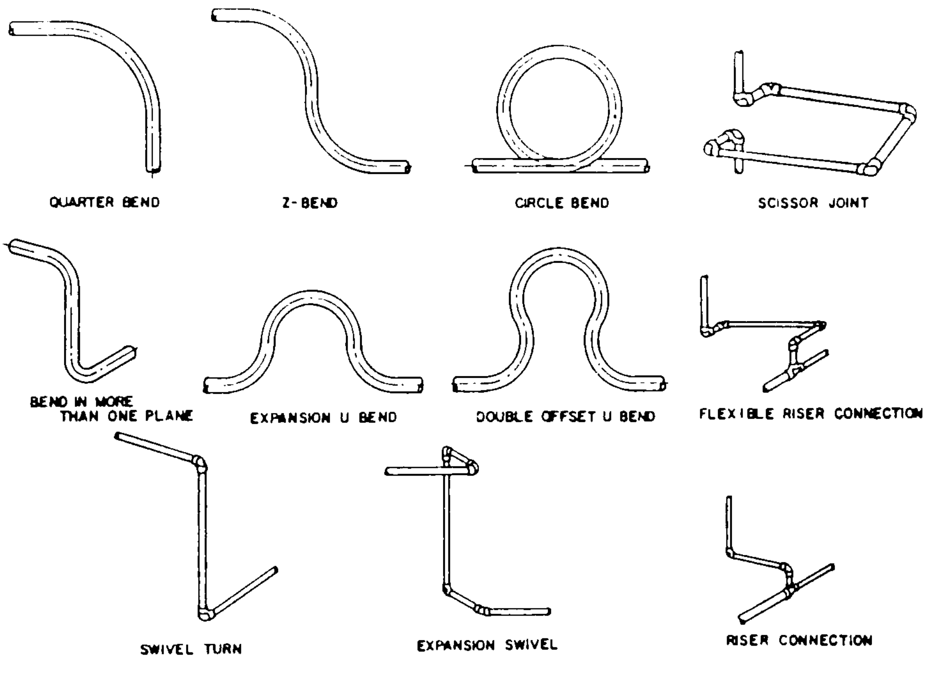

Figure 16-7, Common Expansion Joints and Expansion Bends

16.7 Pipe Anchors and Supports

Pipes must be securely anchored and properly supported. Anchoring of the pipe causes the expansion of the pipe to be absorbed at the proper point in the system. Adequate supports must be provided by specially designed hangers. If the pipe is heavy and it contains water, it will tend to sag between short distances as short as 10 feet. The proper spacing for supports, anchors and expansion joints is dependent on the pipe size, type of fluid and operating temperature. Manufactures should always be consulted for proper pipe installation.

16.8 Pumps

There are two pump classifications: positive displacement and nonpositive displacement.

Positive displacement pumps are usually in the form of reciprocating piston pumps. This type of pump creates lift and pressure by displacing liquid with a piston moving in a cylinder. The only limitation on pressure that may be developed by this pump is the strength of its structural parts. This type of pump has a limited use in the HVAC industry. Another type of positive displacement is the rotary pump.

The centrifugal pump is a nonpositive displacement pump and is the most frequently used pump in the HVAC industry. This type of pump's capacity is dependent on the resistance offered to the flow by the system.

There are two different kinds of centrifugal pumps: inline pumps and end suction pumps. Inline pumps are normally used with fractional horse power electric motors up to one horse power and operate at speeds of 1,750 rpm. End suction pumps run from 1/4 hp and on up. These pumps are base mounted and operate speeds of 1,750 or 3,450 rpm.

16.8.1 Definitions - terms used in selecting pumps and in pump System design.

A. Suction lift - exist when the source of the supply is below the centerline of the pump.

B. Static suction lift - is the vertical distance in feet from the centerline of the pump to the free level of the liquid to be pumped.

C. Suction head - exist when the source of the supply is above the centerline of the pump.

D. Static suction head - is the vertical distance in feet from the centerline of the pump to the free level of liquid to be pumped.

E. Static discharge head - is the vertical distance in feet between the centerline of the pump and the point of the free discharge or the surface of the liquid in the discharge tank.

F. Total static head - is the vertical distance in feet between the free level of the source of the supply and the point of the free discharge or free surface of the discharge liquid.

G. Friction head (h_L) - is the head required to overcome the resistance to the flow in the pipe and the fittings.

H. Velocity head (h_v) - is the energy of the liquid as a result of its motion at the same velocity V. ($h_v = V^2/2g$)

I. Pressure head - must be considered when a pumping system either begins or terminates in a tank which is under some pressure other than atmospheric.

J. Total dynamic suction lift (TDSL) - is the static suction lift plus the velocity head at the pump suction flange plus the total friction head in the line.

K. Total dynamic suction head (TDSH) - is the static suction head minus the velocity head at the pump suction flange minus the total friction head in the suction line.

L. Total dynamic discharge head (TDDH) - is the static discharge head plus velocity head at the pump discharge flange plus the total friction head in the discharge line.

M. Total head or total dynamic head (TDH) - is the total dynamic discharge head minus the total dynamic suction head or plus the total dynamic suction lift.

$$hp = TDH = TDDH - TDSH$$

$$hp = TDH = TDDH + TDSL$$

16.8.2 Net Positive Suction Head (NPSH) and Cavitation

The Hydraulic Institute defines NPSH as the total suction head in feet of fluid absolute, determined at the suction nozzle of the pump and corrected to datum, less the vapor pressure of the liquid in feet absolute.

Simply stated, it is an analysis of energy conditions on the suction side of the pump to determine if the liquid will vaporize at the lowest pressure point in the pump.

A liquid exerts on its surroundings a pressure which is dependent on its temperature. This pressure is called vapor pressure and increases with increasing temperature. When this pressure is the same pressure of the fluid surroundings, the fluid begins to boil.

NPSH is symply a measure of the amount of suction head present to prevent this vaporization at the lowest pressure point in the pump. NPSH required ($NPSH_R$) is the positive head in feet absolute required at the pump's suction to overcome the pressure drops in the pump and maintain the liquid above its vapor pressure. Pump manufacturer's performance curves normally provide this information.

NPSH available ($NPSH_A$) is the excess pressure of the liquid in feet absolute over the vapor pressure as it arrives at the pump suction.

$NPSH_A$ can be calculated by the following:

$$NPSH_A = (P_B - P_V)/\alpha + (s - h_L)$$

where: P_B = Absolute pressure on the liquid surface in the suction tank, lbf/ft^2

P_V = Vapor pressure of the liquid at the liquid temperature, lbf/ft^2 absolute

α = Specific weight of the liquid, lbf/ft^3

s = Vertical distance from the pump centerline to the level of liquid in the suction tank, (tank above s is positive, tank below s is negative)

h_L = Total friction loss in the pump suction pipe and fittings

For existing systems $NPSH_A$ can be calculated by:

$$NPSH_A = (P_B - P_V)/\alpha \pm GR + h_V$$

where: GR = Gage reading of pump suction expressed in feet, (positive if above atmospheric and negative below atmospheric)

h_V = Velocity head in the suction pipe at the gage connection, ft

Cavitation - is the creation of bubbles in the pump which are the direct cause of insufficient $NPSH_A$. When these bubbles reach higher pressures in the pump they implode which causes a rumbling noise and results in lower pump output and pump damage.

16.9 Special Fittings and Devices Required in Hydronic Systems

Venturi Pipe Fitting - used in the conventional one pipe heating system to direct some flow through the heating unit (special tee fitting).

Air Scoop and Air Vent - designed to eliminate air from the hydronic system.

Expansion Tank - connects to the air scoop and acts as an air cushion for the system. It is mounted above the boiler and is an airtight cylinder. As the system water temperature increases, the water expands and the expansion tank absorbs this increase in volume. The tank also maintains a reasonable stable pressure in the system.

Flow Control Device - is a valve which prevents gravity circulation in the system when the pump is stopped.

Relief Valve - designed to prevent the boiler from rupturing should the pressure exceed the boilers working pressure.

Reducing Valve - this valve is designed to automatically feed makeup water to into the system when the system water pressure drops below the setting of the reducing valve.

17. Fans and Building Air Distribution

Introduction - In this chapter the first five sections discusses fans. These sections are as follows: fans, fan performance, fan selection, fan installation, and fans and the variable air volume system. The next five sections discusses duct design. These sections are as follows: air flow in ducts and fittings, duct design-general consideration, design of low velocity duct system, turning vanes and dampers, and high velocity duct design.

17.1 Fans

Fans are one of the most important part of an HVAC system since they act like a heart moving air through the system. There are two basic types of fans: centrifugal and axial.

17.1.1 Centrifugal Fans - This type of fan is the most commonly used fan in the industry because it can efficiently move large or small quantities of air over a wide range of pressures. The principle of operation is the same as the centrifugal pump. The figure below shows this type of fan. The impeller blades can either be forward curved, backward curve, or radial. Fan design influences fan characteristics.

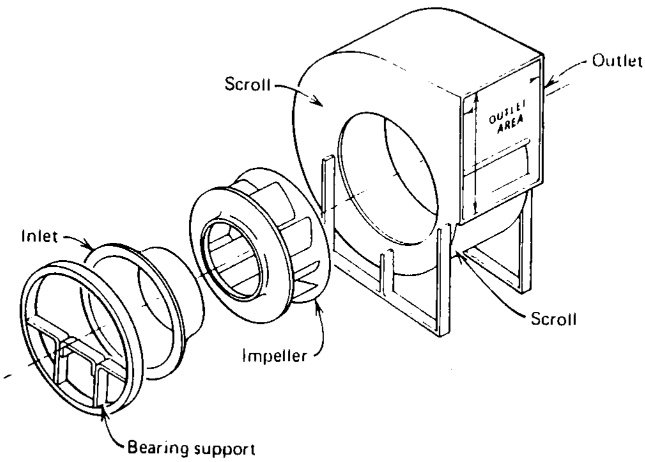

Figure 17-1, Exploded View of a Centrifugal Fan
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

17.1.1 Axial Fans - There are two types of axial fans: vaneaxial and tubeaxial. A Vaneaxial fan axial fan is mounted in the centerline of the duct and produces an axial flow of air. Guide vanes are provided before and after the wheel to reduce rotation of the air stream. A Tubeaxial fan is the same as the vaneaxial fan but without the guid vanes. The figure below illustrates both types.

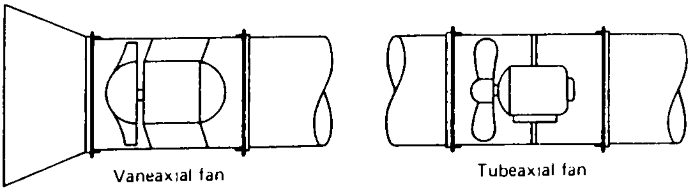

Figure 17-2, Axial Flow Fans
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Axial flow fans are not capable of producing pressures as high as those of the centrifugal type fan, but can move more large quantities of air at low pressures. Axial fans generally produce higher noise levels than centrifugal fans.

17.2 Fan Performance

The performance of fans is generally given in the form of a graph showing pressure, efficiency and power as a function of capacity.

The energy transferred to the air by the impeller results in an increase in static and velocity pressure. The sum of these two pressures gives the total pressure. These pressures are normally in terms of inches of water.

Applying Bernoulli's equation to a fan yields the following:

$$\frac{g_c W}{g} = \frac{g_c}{g}\frac{P_1 - P_2}{\rho} + \frac{1}{2g}(\bar{V}_1^2 - \bar{V}_2^2) = \frac{g_c}{g}\frac{P_{01} - P_{o2}}{\rho}$$

where: W = The energy imparted to the air.

Total power imparted to the air is given by:

$$\dot{W}_t = \dot{m}(P_{01} - P_{02})/\rho$$

The static power is the part of the total power that is used to produce the change in static head. It is given by the following:

$$\dot{W}_s = \dot{m}(P_1 - P_2)/\underline{P}$$

The fan efficiency can be expressed in two ways.

1) The ratio of the total air power to shaft input power.

$$\underline{n}_t = \dot{W}_t/\dot{W}_{sh} = \frac{\dot{m}(P_{01} - P_{02})}{\underline{P}\dot{W}_{sh}}$$

2) In terms of the volume flow rate:

$$\underline{n}_t = \frac{\dot{Q}(P_{01} - P_{02})}{\dot{W}_{sh}}$$

where: $\dot{Q}$ = Volume flow rate, ft^3/min
$P_{01} - P_{02}$ = Change in total pressure, lbf/ft^2
$\dot{W}_{sh}$ = Shaft power, ft-lbf

The static fan efficiency is the ratio of static air power to shaft input power and is given by:

$$\underline{n}_t = \frac{\dot{W}_s}{\dot{W}_{sh}} = \frac{\dot{m}(P_1 - P_2)}{\underline{P}\dot{W}_{sh}} = \frac{\dot{Q}(P_1 - P_2)}{\dot{W}_{sh}}$$

The figures below show typical fan performance curves for centrifugal fans: forward curve, backward curve and radial.

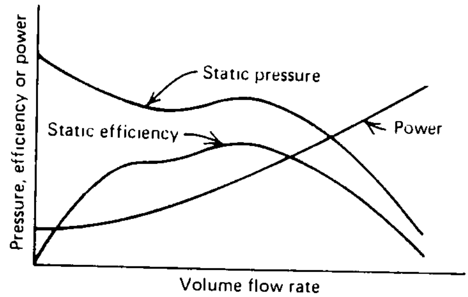

Figure 17-3, Forward-tip Fan Characteristic

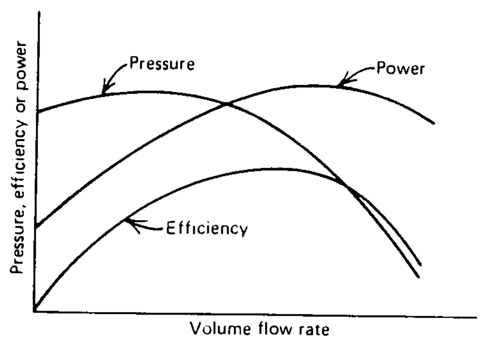

Figure 17-4, Backward-tip Fan Characteristic

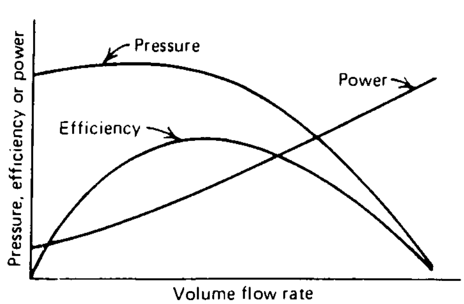

Figure 17-5, Radial-tip Fan Characteristic
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A conventional representation of fan performance is shown below for a back-ward curve fan. This graph shows total pressure and total efficiency.

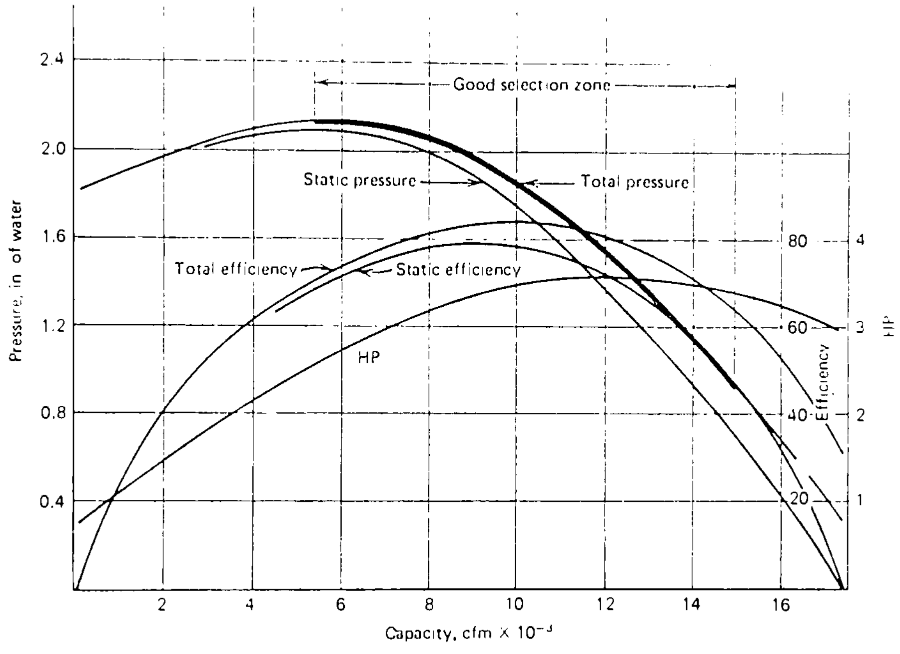

Figure 17-6, Conventional Curves for Backward-curved Blade Fan
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Note that the area for desired application is marked on the graph above. When this data from this zone are plotted on a logarithmic scale, the curves appear as shown on the graph below. The plot has some advantages over the conventional representation. Many different fan speeds can be conveniently shown and the System characteristics is a straight line parallel to the efficiency lines.

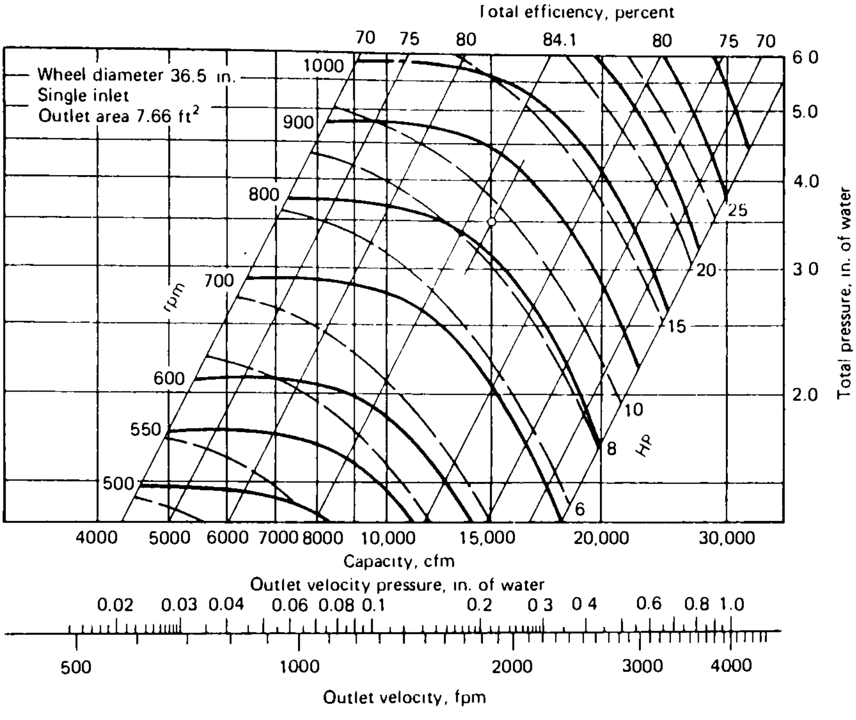

Figure 17-7, Performance Data for a Typical Backward-curved Blade Fan
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The below table compares some of the more important characteristics of centrifugal fans.

Table 17-1, Comparison of Centrifugal Fan Types
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Item	Forward-Curved Blades	Radial Blades	Backward-Curved Blades
Efficiency	Medium	Medium	High
Space required	Small	Medium	Medium
Speed for given pressure rise	Low	Medium	High
Noise	Poor	Fair	Good

Fan noise is a very important consideration in HVAC design. The noise level a fan produces is proportional to its tip velocity of its impeller and the air velocity leaving the wheel. Also, fan noise is Roughly proportional to the pressure developed regardless of the blade type. However, backward curved fans generally have better noise characteristics.

Combining both the system and fan characteristics on one plot is vary useful in matching a fan to a system and to insure fan operation at the desired conditions. The figure below illustrates this plot.

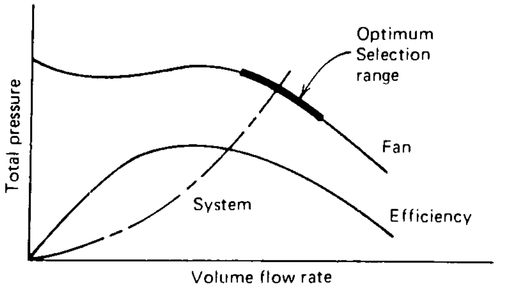

Figure 17-8, Optimum Match Between System and Forward-curved Blade Fan
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

The figure below shows the system plot on the fan logarithmic characteristic plot. The line S-S is the system characteristic line.

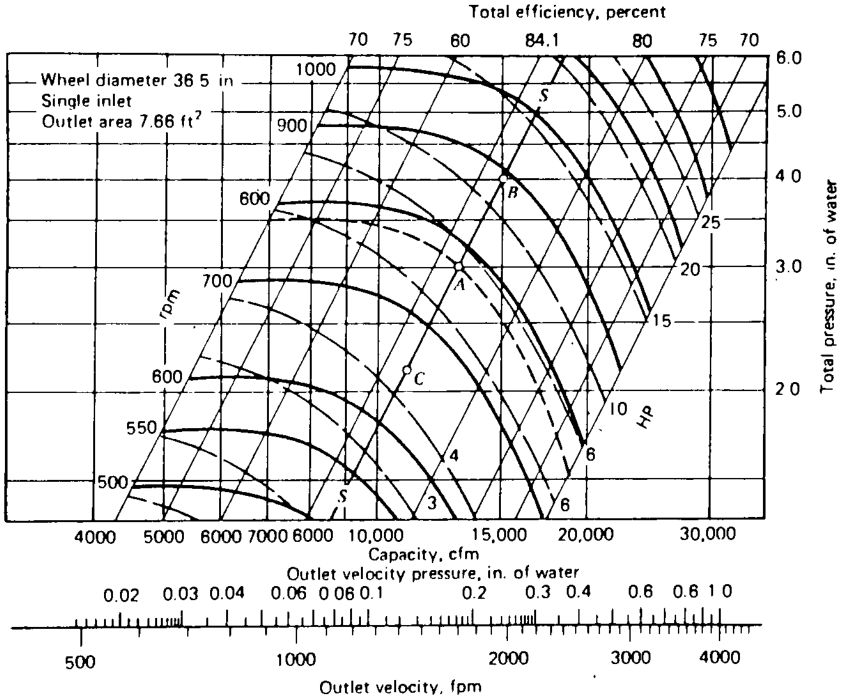

Figure 17-9, Performance Chart Showing Combination of Fan and System
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

17.2.1 Fan Laws - There are several relationships between fan capacity, pressure, speed and power which are referred as the Fan Laws. The first three laws are the most useful.

1. The capacity is directly Proportional to the fan speed.
2. The pressure (static, total, or velocity) is proportional to the square of the fan speed.

3. The power required is proportional to the cube of the fan speed.
4. The pressure and power are proportional to the density of the air at constant speed and capacity.
5. Speed, capacity and power are inversely proportional to the square root of the density at constant pressure.
6. The capacity, speed and pressure are inversely proportional to the density and the power is inversely proportional to the square of the density at a constant mass flow rate.

17.3 Fan Selection

To select a fan the total capacity of the system plus the total pressure of the system must be known. The type of fan arrangement, the possibility of fans in parallel or series, the nature of the load (variable or steady) and the noise constraints must be considered in selecting the right type of fan. Once the system characteristics are determined the fan selection the becomes based on efficiency, reliability, size, weight, speed, and cost.

For fan selection manufacturers show fan characteristics by graphs previously seen. To assist in fan selection manufactures give tables of fan characteristics as seen by the examples below.

Table 17-2, Pressure-Capacity Table for a Forward-curved Blade Fan
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

cfm	Outlet Velocity	½ in. of water[a]		⅝ in. of water		¾ in. of water		1 in. of water		1¼ in. of water		1½ in. of water	
		rpm	bhp[b]	rpm	bhp	rpm	bhp	rpm	bhp	rpm	bhp	rpm	bhp
851	1200	848	0.13	933	0.16	1018	0.19	—	—	—	—	—	—
922	1300	866	0.15	945	0.18	1019	0.21	—	—	—	—	—	—
993	1400	884	0.17	957	0.20	1030	0.23	1175	0.30	—	—	—	—
1064	1500	901	0.19	973	0.22	1039	0.26	1182	0.32	—	—	—	—
1134	1600	926	0.22	997	0.24	1057	0.29	1190	0.35	1320	0.43	—	—
1205	1700	954	0.25	1020	0.27	1078	0.31	1200	0.38	1325	0.46	1436	0.55
1276	1800	983	0.28	1044	0.31	1100	0.34	1210	0.42	1330	0.50	1440	0.59
1347	1900	1011	0.31	1068	0.35	1126	0.38	1230	0.46	1341	0.54	1447	0.63
1418	2000	1039	0.35	1092	0.39	1152	0.42	1250	0.50	1352	0.59	1458	0.66
1489	2100	1068	0.39	1115	0.43	1178	0.47	1275	0.54	1370	0.62	1470	0.72
1560	2200	1096	0.44	1147	0.47	1204	0.51	1300	0.59	1390	0.67	1482	0.77
1631	2300	1124	0.48	1179	0.52	1230	0.56	1325	0.64	1420	0.73	1500	0.83
1702	2400	1152	0.53	1210	0.58	1256	0.62	1350	0.70	1448	0.78	1525	0.88

Note. Data are for a 9 in. wheel diameter and an outlet of 0.71 ft^2

[a]Static pressure

[b]bhp = shaft power in horsepower.

17.4 Fan Installation

The performance of a fan can be drastically effected by improper installation. Duct connections should be such that the air may enter and leave the fan as uniformly as possible with no abrupt change in direction. The figure below shows some good and poor installations.

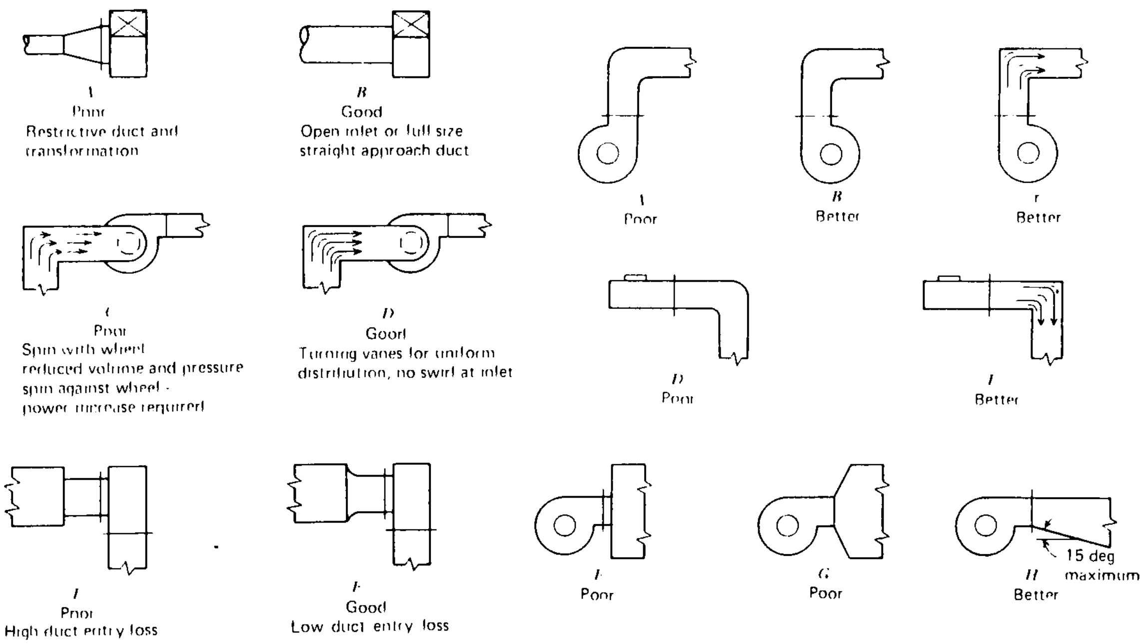

Figure 17-10, Comparison of Good and Poor Fan Connections
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Some manufactures furnish application factors from which modified fan curve can be computed in case the fan can be mounted in only one position. The figure below shows this modified fan curve.

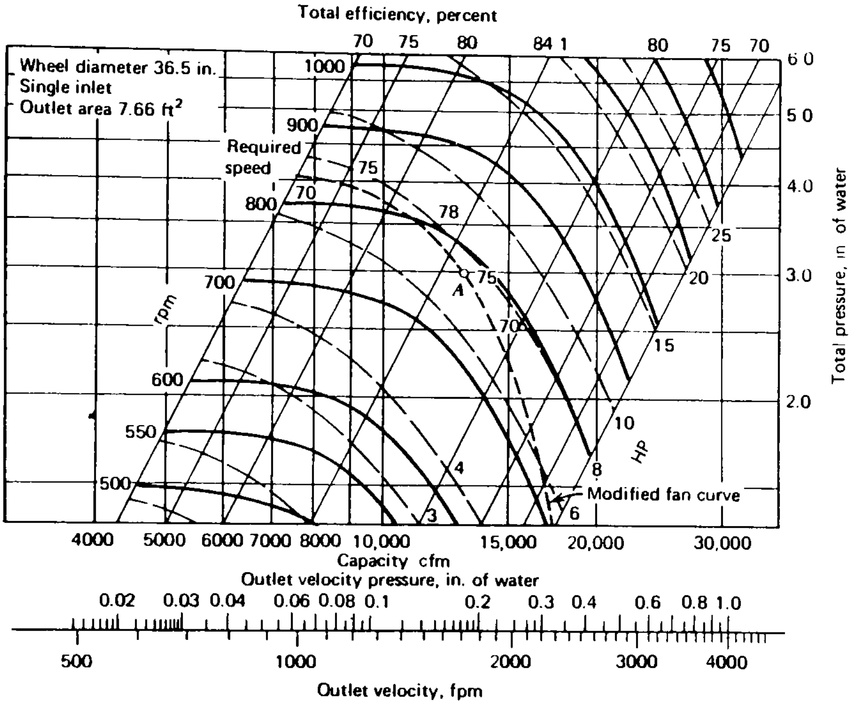

Figure 17-11, Performance Curves Showing Modified Fan Curve
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

17.5 Fans and Variable Air Volume Systems

The variable volume air distribution system is usually designed to supply air to a large number of spaces with the total amount of circulated air varying between some minimum and full load design quantity. Normally the minimum is about 20 to 25 percent of the maximum. The volume flow rate of the air is controlled independently from the fan by terminal boxes and the fan must respond to the system.

The fan's capacity is directly proportional to the fan speed and power is proportional to the cube of the speed. It is obvious that the fan speed should be decreased as the volume flow rate decreases. There are three ways in adjusting the fan speed: 1) Eddy currant drives, 2) Variable pulley drives and 3) Variable speed electric motors.

The eddy currant drive makes use of a magnetic coupling. They have an infinite adjustment of fan speed and are excellent devices. The only draw back of these devices is their high cost.

The variable pulley drives use belts and pulleys which change fan speed as the fan is operating. The main disadvantage of this type of drive is maintenance.

The variable speed electric motors would be ideal but they very low efficiencies which offsets the benefit of lowering the fan speed.

Another approach to control of the fan is to throttle and introduce a swirling component to the air entering the fan. This method alters the fan characteristics in such a way that less power is required at lower flow rates. This method is accomplished through the use of variable inlet vanes which are a radial camper system located at the inlet to the fan. This system is usually controlled by maintaining a constant static pressure at the most remote terminal box. This method is not as effective in reducing fan power as fan speed reduction, but the cost and maintenance are low.

The next two figures show this result of using a variable speed fan and a variable inlet vane far.

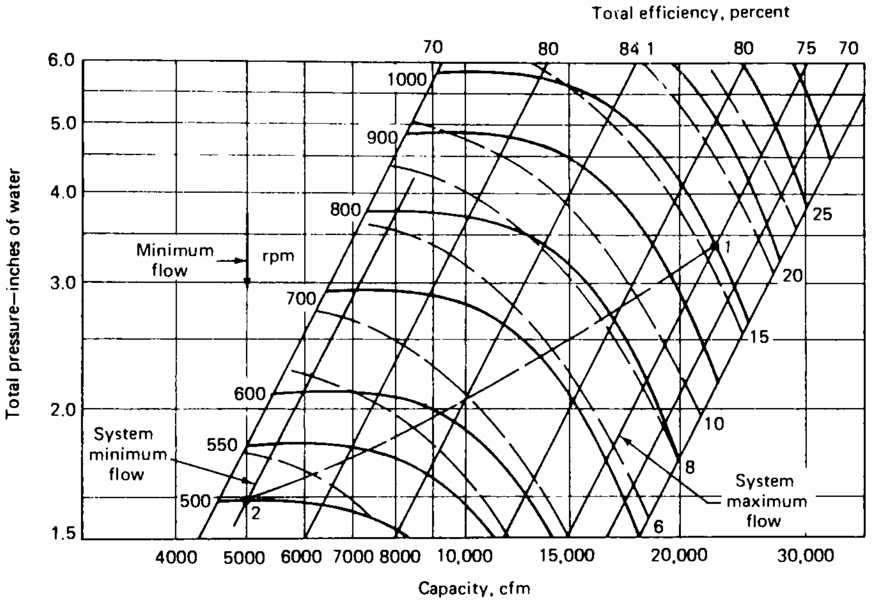

Figure 17-12, Variable Inlet Vane Fan in a Variable Volume System
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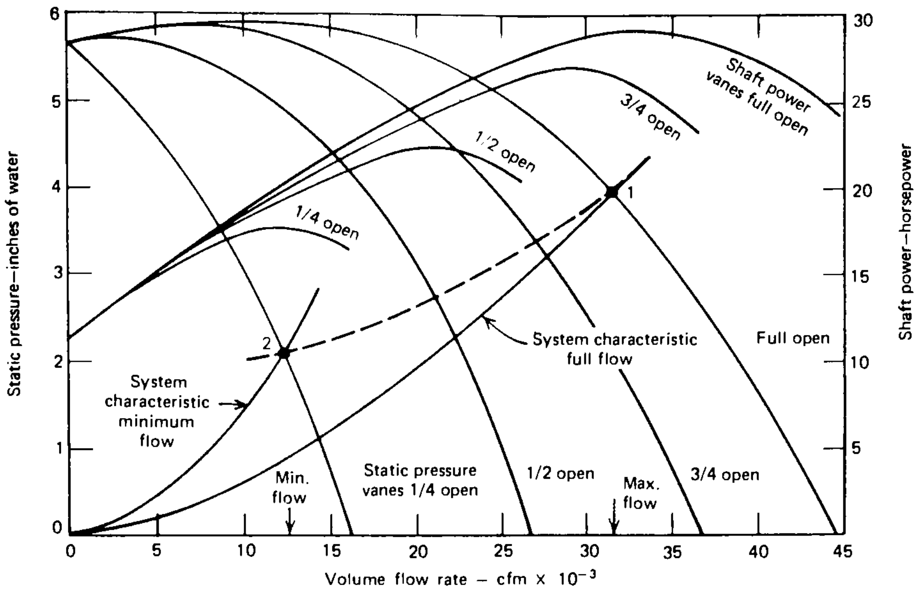

Figure 17-13, Variable Inlet Vane Fan in a Variable Volume System
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17.6 Air Flow In Ducts and Fittings

Air flow in ducts and fittings is discussed in the Fluid section.

17.7 Duct Design - General Considerations

The purpose of the duct system is to deliver the specified amount of air at the desired temperature and pressure to offset the given load in the space its serving. The duct system must be designed in such a way that only slight adjustments are made to achieve desired distribution of air. Duct systems must be also designed to have low noise levels. This is done by limiting air velocities, by using sound-absorbing duct materials or liners and by avoiding drastic restrictions in the duct such as nearly closed dampers.

There are two basic kinds of duct systems: low velocity and high velocity. Low velocity systems will generally have a pressure loss of less than .15 in of water per 100 feet, where as high pressure losses ar up to .7 in of water per 100 feet. The next two tables give recommended maximum duct velocities for high and low velocity systems.

Table 17-3, Recommended Maximum Duct Velocities for High Velocity Systems
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

cfm (m^3/min) Carried by the Duct	Maximum Velocities fpm (m/s)
60,000 to 40,000 (1,700 to 1,133)	6000 (30.5)
40,000 to 25,000 (1,333 to 708)	5000 (25.4)
25,000 to 15,000 (708 to 425)	4500 (22.9)
15,000 to 10,000 (425 to 283)	4000 (20.3)
10,000 to 6,000 (283 to 170)	3500 (17.8)
6,000 to 3,000 (170 to 85)	3000 (15.2)
3,000 to 1,000 (85 to 28)	2500 (12.7)

*Reprinted by permission from *ASHRAE Handbook of Fundamentals*, 1977

Table 17-4, Recommended and Maximum Duct Velocities for Low Velocity Systems
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

Designation	Residences	Schools, Theaters, Public Buildings	Industrial Buildings
	Recommended Velocities, fpm (m/s)		
Outdoor air intakes[a]	500 (2.54)	500 (2.54)	500 (2.54)
Filters[a]	250 (1.27)	300 (1.52)	350 (1.78)
Heating coils[a]	450 (2.29)	500 (2.54)	600 (3.05)
Cooling coils[a]	450 (2.29)	500 (2.54)	600 (3.05)
Air washers[a]	500 (2.54)	500 (2.54)	500 (2.54)
Fan outlets	1000-1600 (5.08-8.13)	1300-2000 (6.60-10.16)	1600-2400 (8.13-12.19)
Main ducts	700-900 (3.56-4.57)	1000-1300 (5.08-6.60)	1200-1800 (6.1-9.14)
Branch ducts	600 (3.05)	600-900 (3.05-4.57)	800-1000 (4.06-5.08)
Branch risers	500 (2.54)	600-700 (3.05-3.56)	800 (4.06)
	Maximum Velocities, fpm (m/s)		
Outdoor air intakes[a]	800 (4.06)	900 (4.57)	1200 (6.10)
Filters[a]	300 (1.52)	350 (1.78)	350 (1.78)
Heating coils[a]	500 (2.54)	600 (3.05)	700 (3.56)
Cooling coils[a]	450 (2.29)	500 (2.54)	600 (3.05)
Air washers[a]	500 (2.54)	500 (2.54)	500 (2.54)
Fan outlets	1700 (8.64)	1500-2200 (7.62-11.18)	1700-2800 (8.64-14.22)
Main ducts	800-1200 (4.06-6.10)	1100-1600 (5.59-8.13)	1300-2200 (6.60-11.18)
Branch ducts	700-1000 (3.56-5.08)	800-1300 (4.06-6.60)	1000-1800 (5.08-9.14)
Branch risers	650-800 (3.30-4.06)	800-1200 (4.06-6.10)	1000-1600 (5.08-8.13)

Metal ducts are the most commonly used duct material. They are usually acoustically lined near the air distribution equipment. The outside of the duct is covered with insulation and a vapor barrier

Also, fibrous glass ducts are used for their sound absorption characteristics and their ease of installation because they are already insulated and contain a vapor barrier. Duct systems should be well sealed to prevent air leaks.

17.8 Design of Low Velocity Duct Systems

The methods discussed in this section pertain to low velocity systems where the average velocity is less than 1000 ft/min. These methods can be used for high velocity system design, but the results would not be satisfactory in most cases. The two methods discussed in this section are the equal friction method and the balance capacity method.

17.8.1 Equal Friction Method

The principle of this method is to make the pressure loss per foot of length the same for the entire system. If the system layout is symmetrical with all runs from the fan to the diffusers are about the same length, then this method will produce a good balanced system. However, most systems will have a variety of duct runs and the runs will have to be dampened. Dampers can cause noise in the system.

The usual procedure is to select a velocity in the main duct next to the fan which has an acceptable noise level. The pressure loss per length for that duct is used to size the rest of the system. A desirable feature of this method is the gradual reduction of air velocity from the fan outlet, there for reducing noise problems. After sizing the system designer must compute the total pressure loss of the longest run (largest flow resistance).

17.8.2 Balance Capacity Method

This method of duct design has been referred as the "Balance Pressure Loss Method", however it si the floe rate or capacity of each outlet that is balanced and not the pressure. The basic principle of this method of design is to make the loss in total pressure equal for all duct runs from the fan to the outlets when the required amount of air is flowing in each. In general all runs will have a different equivalent length and the pressure loss per unit length for each run will be different.

The design procedure for the balanced capacity method is the same as the equal friction method. First, the total loss in pressure per unit length for the longest run is determined. Secondly the other run's pressure losses per unit length are determined to balance the flow. Dampers might be needed to balance the flow if duct velocities are low.

The only limitation of this method is the use of equivalent lengths for fittings in the system. There is a minor error when duct velocities are less than 1000 ft/min for the duct fittings.

17.8.3 Return Air Systems

The same methods used in low velocity duct supplies can be used in return systems.

17.9 Turning Vanes and Dampers

These devices are the two main accessory devices used in duct systems.

17.9.1 Turning Vanes

Turning vanes have the purpose of preventing turbulence and consequent high losses in total pressure where turns are necessary in rectangular ducts. There are two types of vanes: air foil and flat. Air foil vanes are more efficient but also more expensive to fabricate. Below shows the typical use of vanes in a rectangular duct.

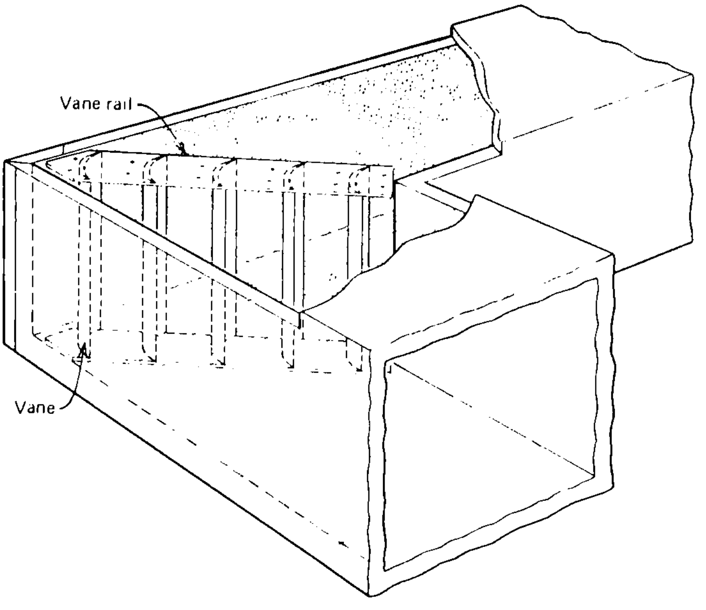

Figure 17-14, Typical Use of Vanes in a Rectangular Duct
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

17.9.2 Dampers

Dampers are used to balance a system and to control ventilation and exhaust air. The dampers may be hand operated and locked in position after adjustment or may be motor operated and controlled by a temperature sensor or by other remote signals. They are also used to control the spread of fire and smoke. Below is a figure of a typical opposed blade damper.

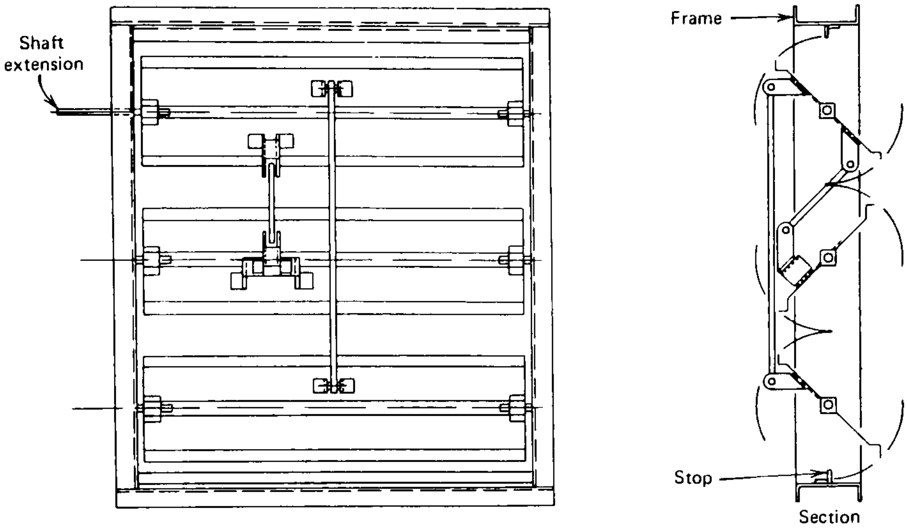

Figure 17-15, Typical Opposed Blade Damper Assembly
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

An extractor is a combination damper and turning assembly. The figure below shows a typical extractor.

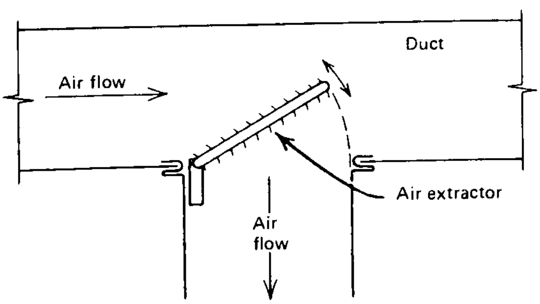

Figure 17-16, Combination Turning Vane and Damper Assembly
Abridge from "Heating Ventilation and Air Conditioning Analysis and Design", by Faye C. Mc. Quiston and Jerald D. Parker. Copyright 1982 by John Wiley and Sons

17.10 High Velocity Duct Design

In large commercial structures space allocation is vary important. One major reduction in space is to reduce the size of the ducts. The reduction in duct sizes causes much higher velocities which introduces some new problems in high velocity duct design.

Noise is probably the most serious problem with high velocity air movement. Special attention must be given to the design and installation of sound-attenuating equipment in the system. Generally a sound absorber is installed just down stream of the fan an acts like an automobiles muffler.

Since air cannot be used at high velocities, a device called the terminal box is used to throttle the air to a low velocity, control the flow rate and to attenuate the noise. One terminal box may have several outlets.

There are much greater pressures in high velocity ducts and ordinary ducts can not take the excessive pressures. The type of duct used for this application is know as spiral duct and comes in either round or oval cross-sections.

The criterion for designing high velocity duct systems is somewhat different from that used for low velocity systems. Emphasis is shifted from a self-balancing system to one that has minimum losses in total pressure. The high velocity system achieves the proper flow rate at each outlet through the use of the terminal box. There ar two methods used to size high velocity duct systems: static regain method and assumed velocity method.

17.10.1 Static Regain Method

This method systematically reduces the air velocity in the direction of flow in such a way that the increase (regain) in static pressure in each transition just balances the pressure losses in the following section. The main disadvantages of this method are: 1) The very low velocities and large duct sizes that may result at the end of long duct runs, 2) The tedious book keeping and trial-and-error aspects of the method, and 3) The total pressure requirements of each part of the duct system are not readily apparent.

17.10.2 Assumed Velocity Method

This method evolved following the general acceptance of the total pressure concept in duct design. Appreciation of the total energy requirements of the duct system has placed emphasis on controlling the local velocities and total pressure losses during the design of the system. The method is based on the selection of acceptable air velocities at the inlet and in the branches of the system and then on the gradual reduction of the air velocities in between. The total pressure losses are then calculated for each section proceeding from either end.

This design method is recommended for variable air volume duct systems with zones that peak at widely varying times. Because the air flow rates in the various parts of the system change continually, there is no advantage to using a sophisticated design method.

18. Air-Conditioning and Heating Systems

Introduction - This chapter is divided into 4 major sections: All-Air Systems, All-Water Systems, Air and Water Systems, and Heat Pump Systems. The All-Air System discusses single-path, constant volume, reheat, variable air volume, variable air volume reheat, and dual path systems.

There are many different types of systems. This chapter discusses the most common types used in commercial and residential applications.

18.1 All-Air Systems

18.1.1 General

An All-Air System provides complete sensible and latent cooling through the air supplied to each space or zone. Heating is be accomplished by the same air stream.

There are two basic types of All-Air systems: single path and dual path systems. For single path systems this chapter will discuss:1) constant volume, 2) single-zone, 3) reheat, 4) simple variable air volume, 5) variable air volume - fan powered, and 6) constant and intermittent fans. For dual path systems this chapter will discus their general configuration.

18.1.2 System Considerations

In designing an All-Air system the following considerations must be addressed: the type of duct system, heating and cooling loads, room air volume, air volume for ventilation and system controls.

18.1.3 Evaluation (Advantages and Disadvantages)

18.1.3.1 Advantages

1. The central location of major equipment keeps operation and maintenance to unoccupied areas and permits maximum choice of filtration, order and noise control, and high quality durable equipment.
2. Complete absence of equipment in conditioned space which reduces the possible damage to furnishings and minimizes service in these areas.
3. Allows the use of the greatest number of potential cooling season hours with outside air in place of mechanical refrigeration.
4. Seasonal changeover is simple and readily adaptable to automatic controls.
5. Gives a wide choice of zonability, flexibility and humidity control under all operating conditions, with simultaneous heating and cooling available (dual path systems).
6. Heat-recovery systems may be readily incorporated.
7. Allows good design flexibility for optimum air distribution, draft control and local

requirements. Interferes least with draperies at windows.

8. Well suited to applications requiring unusual exhaust makeup.

9. Infringes least on perimeter floor space.

10. Adapts to winter humidification.

18.1.3.2 Disadvantages

1. Requires additional duct clearances, which can reduce the usable floor space and increase building height.
2. In areas with low outside temperatures where air is used for perimeter heating causes fans to operate longer during unoccupied periods.
3. Air-balance is difficult and requires great care.
4. All-air perimeter systems are usually not available for use during building construction as soon as perimeter hydronic systems.
5. Accessibility to terminals demands close cooperation between architectural, mechanical and structural designers.
6. Applications with high internal cooling loads require greater air flow, so the designer should work closely with the architect to solve possible architectural problems.

18.1.4 Single-Path Systems

18.1.4.1 Single Duct, Constant Volume - These systems are controlled by the drybulb temperature of the zone they are cooling or heating. This type of system varies the temperature of the constant air volume to offset the load in the zone

18.1.4.2 Single-Zone System - The simplest of all-air systems, this system uses one supply unit to serve a single temperature control zone. These systems can maintain the temperature and humidity closely and efficiently. Below shows a schematic of this type of system and the next page shows this system's process on the psychrometric chart.

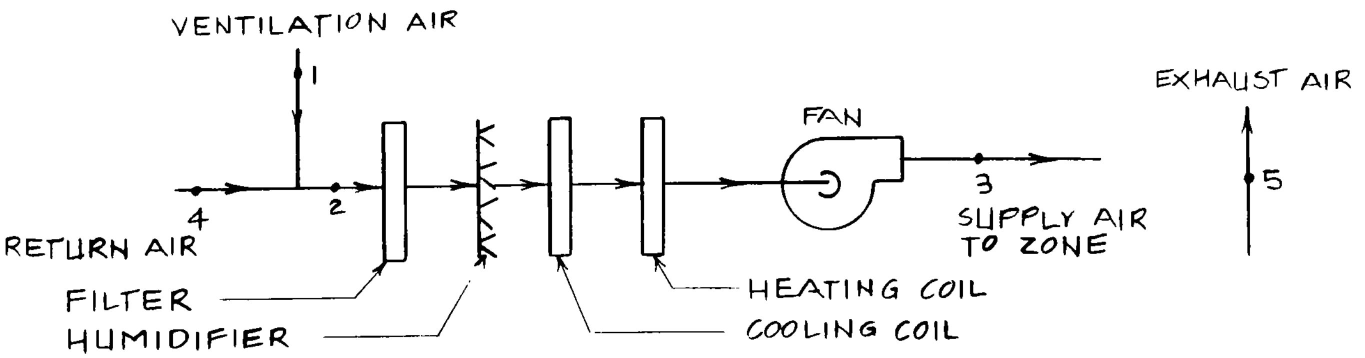

Figure 18-1, Single-Zone System

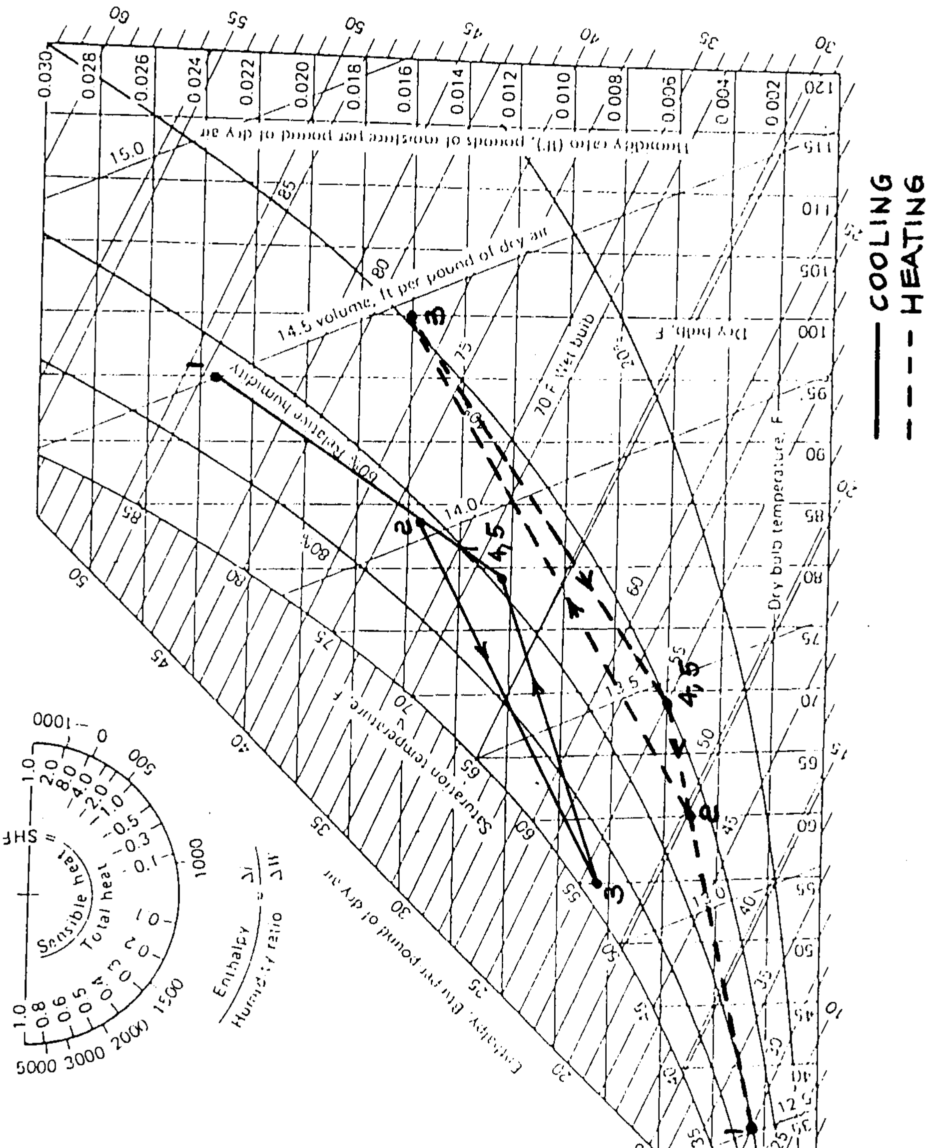

Figure 18-2, Single-Zone System - Psychrometric Chart

18.1.4.3 Reheat Systems - The reheat system is a modification of the single-zone system. This type of system provides: 1) Zone or space control for areas of unequal loading, 2) Heating or cooling of perimeter areas with different exposures and 3) Close control for process or comfort applications. Reheat implies that heat is added as a secondary to either precondition primary air or recirculate room air. The figure below shows a schematic of this system and on the next page shows this system on the psychrometric chart.

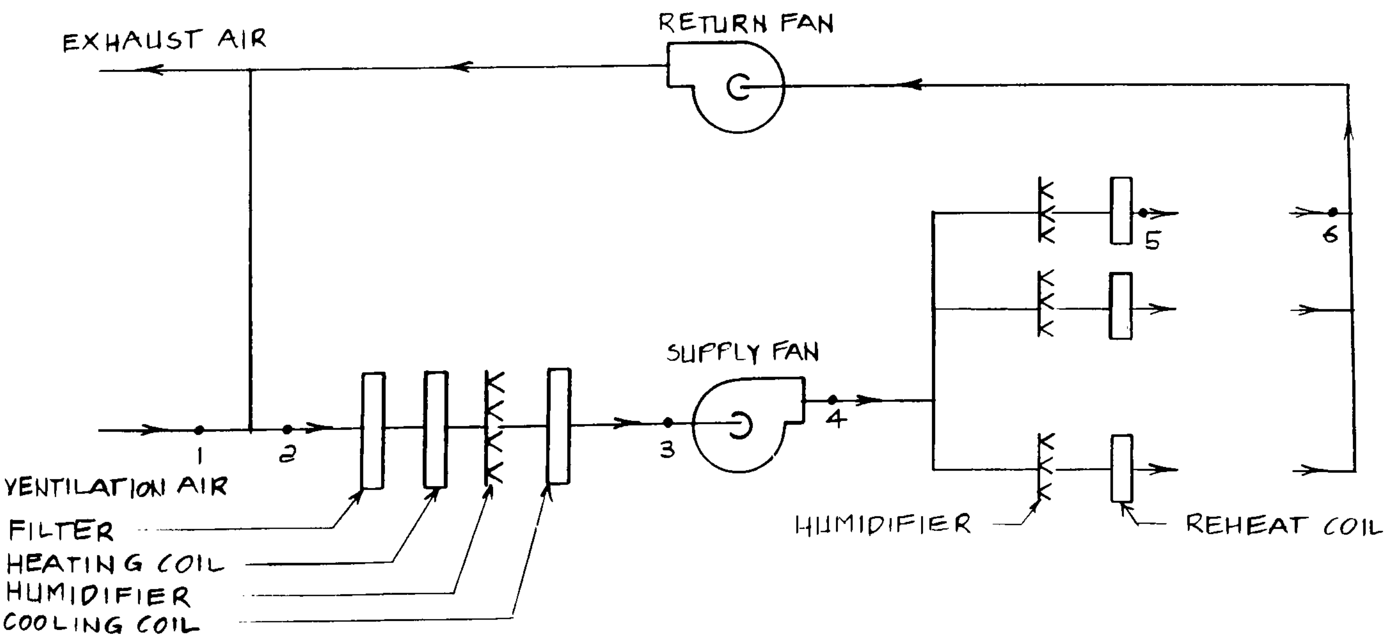

Figure 18-3, Reheat System

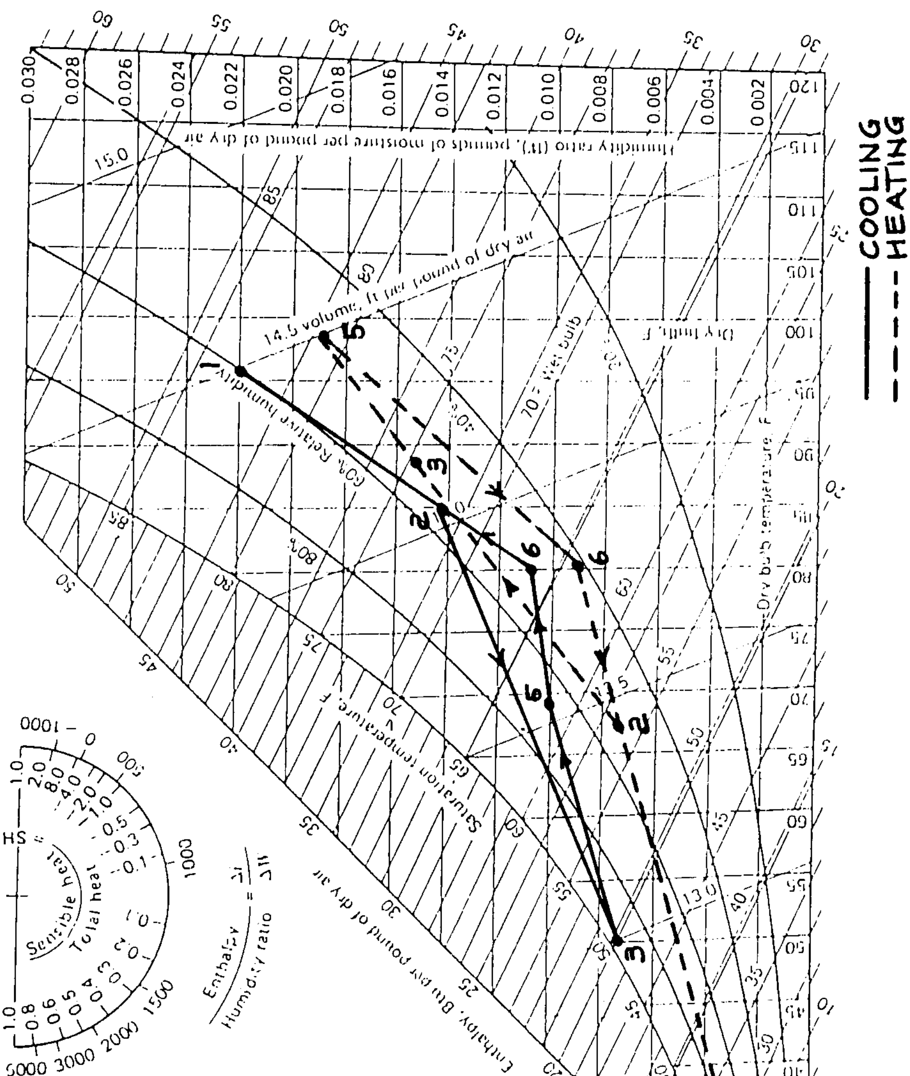

Figure 18-4, Reheat System - Psychrometric Chart

18.1.4.4 Advantages and Disadvantages of Single-Duct Constant Volume Systems.

Advantages - The reheat system closely controls space conditions. It particularly applies to laboratories or where close control of space conditions is vital.

Disadvantages - Reheat systems are expensive to operate and does not meet most energy codes for ordinary comfort conditioning applications.

18.1.4.5 Single Duct, Variable Air Volume (VAV)

A variable air volume (VAV) system controls the drybulb temperature within a space by varying the volume of supply air rather than the supply air temperature. VAV systems can be applied to interior or perimeter zones, with common or separate fan systems, common or separate air temperature controls and with or without auxiliary heating devices. Flow into the space is controlled by varying the position of a simple damper or a volume regulator device in a duct, a pressure reducing box, or at a terminal diffuser.

Simple VAV systems typically cool only and have no requirement for simultaneous heating and cooling in various zones. Perimeter radiation, radiant heat, or an independent constant volume, variable temperature air system normally handles heating requirements.

The fan of a VAV system is designed to handle the largest simultaneous block load, not the some of the individual peaks.

The figure below shows the typical arrangement of a variable air volume system with separate perimeter heating. The next page shows the system operation on the psychrometric chart.

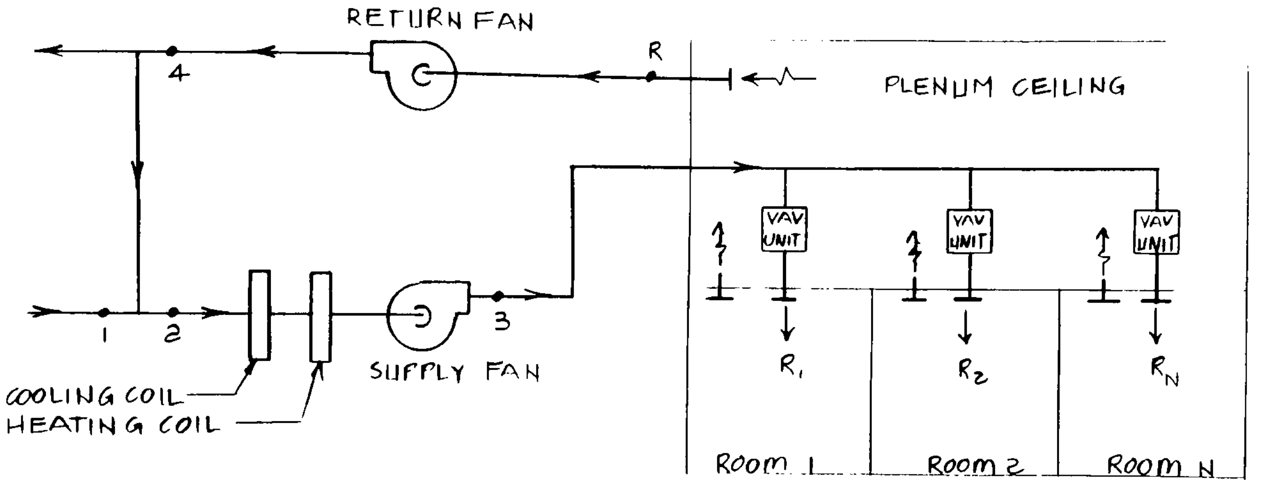

Figure 18-5, Variable Air Volume System

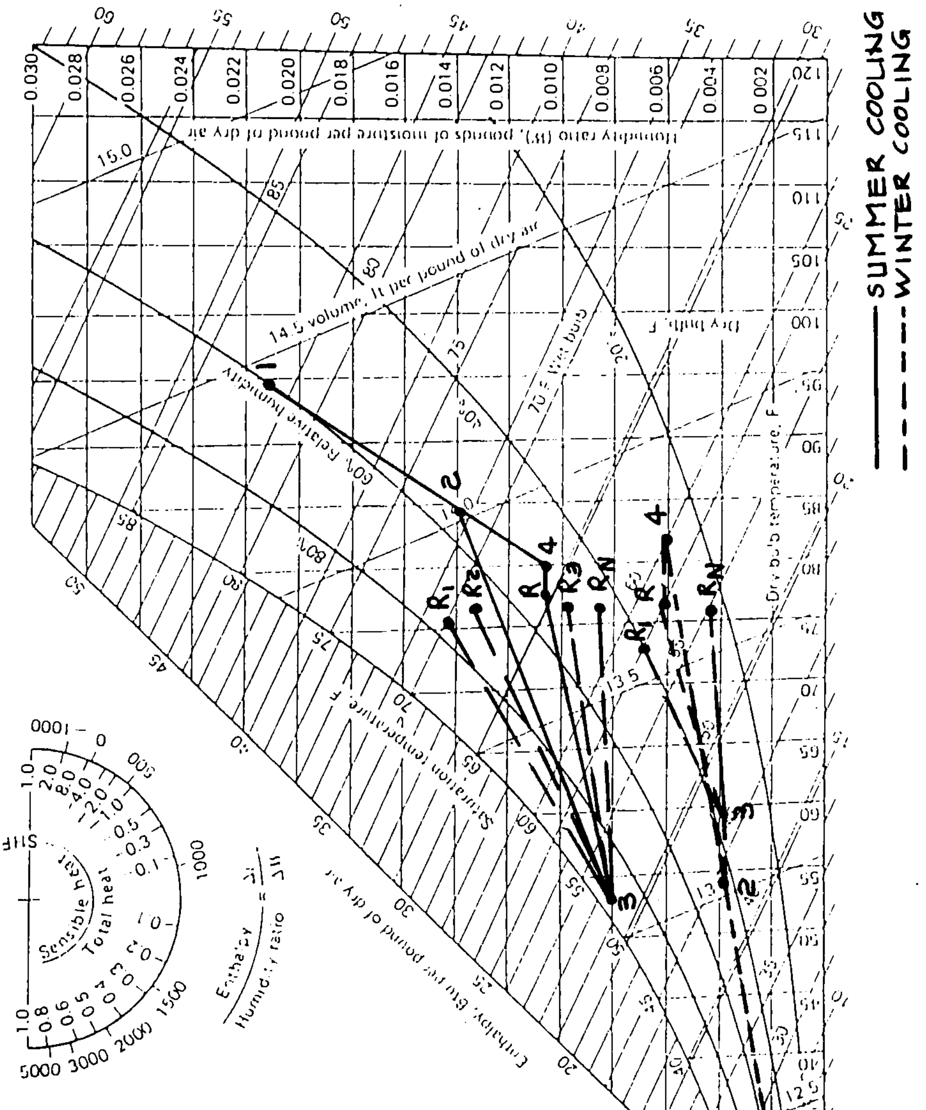

Figure 18-6, Variable Air Volume System - Psychrometric Chart

18.1.4.6 VAV - Reheat

Reheat in the simple VAV system incorporates heating at or near the terminal units. It is applied to systems requiring full heating and cooling flexibility in interior and exterior zones.

VAV with reheat permits flow reduction as the first step in control. Heat then turns on when the air flow reaches a predetermined minimum. A summer abort feature is used to reduce costs.

18.1.4.7 VAV - Components and Controls

1. VAV Units - Flow is controlled by duct mounted units serving air outlets in a control zone or by control units integral to each supply unit.
2. Pressure Independent Volume Regulator Units - Regulate the flow rate in response to the the thermostat's call for heating or cooling. The required flow rate is maintained regardless of fluctuating system pressure.
3. Pressure-Dependent, Air-Flow Limiting, Maximum-Volume Units - Regulate maximum volume, but the flow rate below maximum varies with inlet pressure variation. These units are less expensive than the pressure independent units.

4. Pressure Dependent Units - These units do not regulate the flow rate but position the volume regulating device in response to the thermostat.
5. Bypass (dumping units) - VAV room supply is accomplished in a constant volume system by returning excess supply air into the return ceiling plenum or return duct, thus bypassing the room.
6. Supply Outlet Throttle Units - The area of the throat or the discharge opening of these supply outlets, usually linear diffusers, is thermostatically varied. Noise control might be a problem with these units.
7. Controls - The type of controls for VAV units varies with the terminal device. Most systems use either pneumatic or electric controls that may either be self-powered or system air actuated.

18.1.4.8 "VAV - Advantages and Design Precautions

Advantages:

1. The variable volume concept, when combined with one of the perimeter heating systems, offers inexpensive temperature control for multiple zoning and a high degree of simultaneous heating and cooling flexibility.

2. In true VAV systems, full load advantage may be taken of changing loads from lights, occupancy, solar, and equipment; diversities of as much as 30%, compared with systems based on the sum of the peaks, are permitted.
3. True VAV air systems, except pressure-dependent systems, are virtually self balancing, impaired only by inadequate static pressure control of volume regulation.
4. It is easy and inexpensive to subdivide into new zones and to handle increased loads with new tenancy or usage if the overall system has the reserve for the load increase or if the load does not exceed the original design simultaneous peak.
5. Operating cost savings are accrued from the following building characteristics. These savings do not apply to return air systems.

   a. Fans run long hours at reduced volumes, so they use less energy.

   b. Refrigeration, heating and pumping matches diversity of loads, so energy is saved.

   c. Outside air cooling, where applicable, gives better economy.

   d. Unoccupied areas may be fully cut-off to decrease both refrigeration and ventilation requirements

Design Precautions:

1. Air Distribution

   a. Install high entrainment types of outlets to achieve higher air velocity at minimum flow.

   b. Elvaluate performance at minimum, as well as maximum flow.

   c. Evaluate the effect of minimum volume on space air movements.

2. Fans and Controls

   a. Use fan controls to save power to operate at minimum system pressure for noise control. They may not be economical for systems of 10,000 cfm and below.

   b. Note that on cooling start-up with all variable volume controls wide open, system static pressure will be abnormally low and system volume abnormally high."4

18.1.4.9 Single-Duct VAV-Fan Powered

Fan-powered systems are available in parallel or series flow. In parallel units, the fan sits outside the primary air stream to allow intermittent fan operation. In series units, the fan sits in the primary air stream and runs continuously when the zone is

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4 ASHRAE Handbook, 1987 HVAC Systems and Applications, American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc. 1987

occupied. Fan-powered systems, both series and parallel, are often selected because they move more air through a room at low cooling loads and during reheating compared to VAV reheat or perimeter radiation systems.

1. Series Arrangement - This type of unit continuously operates and varies the amount of cold primary air. The less primary air the more plenum air is used. This unit provides a relatively constant volume of air. Below is a plan view of this type of unit.

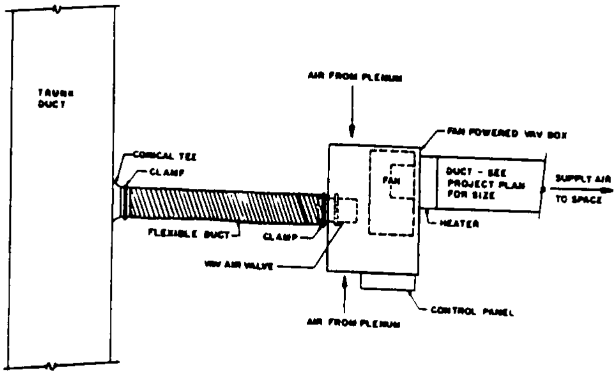


Figure 13-7, Series Arrangement - Plan View
Abridge from "How to Design and Specify a Good
Variable Air Volume Heating, Ventilation and
Air Conditioning System", by Lee Kendric, P.E..
Merrifield, Va.

2. Parallel Arrangement - This fan modulates primary air in response to cooling demand and energizes the integral fan in sequence to deliver induced plenum air to meet heating demand. Below is a plan view of this type of unit.

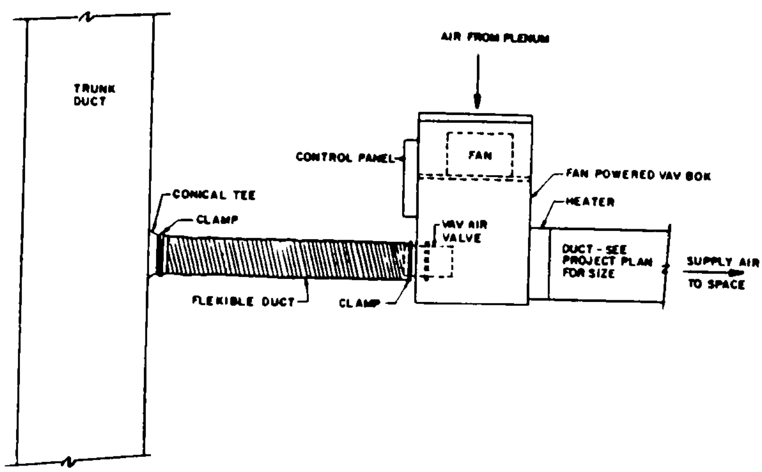


Figure 18-8, Parallel Arrangement - Plan View
Abridge from "How to Design and Specify a Good
Variable Air Volume Heating, Ventilation and
Air Conditioning System", by Lee Kendric, P.E..
Merrifield, Va.

18.1.5 Dual-Path Systems

These systems condition all the air in a central apparatus and distribute it to the conditioned spaces through two parallel mains or ducts. One duct carries cold air, the other warm air. At each space the two ducts connect to a mixing box and the mixing box tempers the air to the desired space requirements. These systems give good results when designed and operated under the following set of conditions: 1) In moderately humid climates, where outdoor design conditions do not exceed 78°F, 2) When minimum outdoor air does not exceed 35 to 40% of the total air flow. 3) When heat is available under the energy codes, and 4) When summer cold-duct temperatures does not exceed 55°F.

The figure below shows the simplest dual duct system.

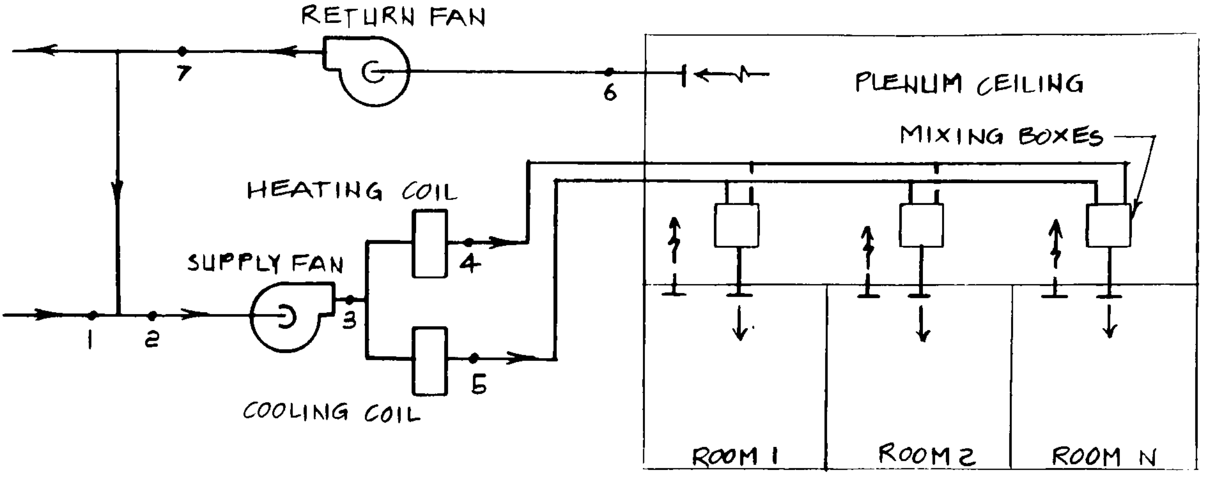


Figure 18-9, Dual Path System

Thermodynamically this cycle is equivalent to a single-duct system with face-and-bypass dampers at the cooling coil, arranged to bypass a mixture of outdoor and recirculated air in response to a zone thermostat as the internal heat load fluctuates. The next page shows this cycle on the psychrometric chart.

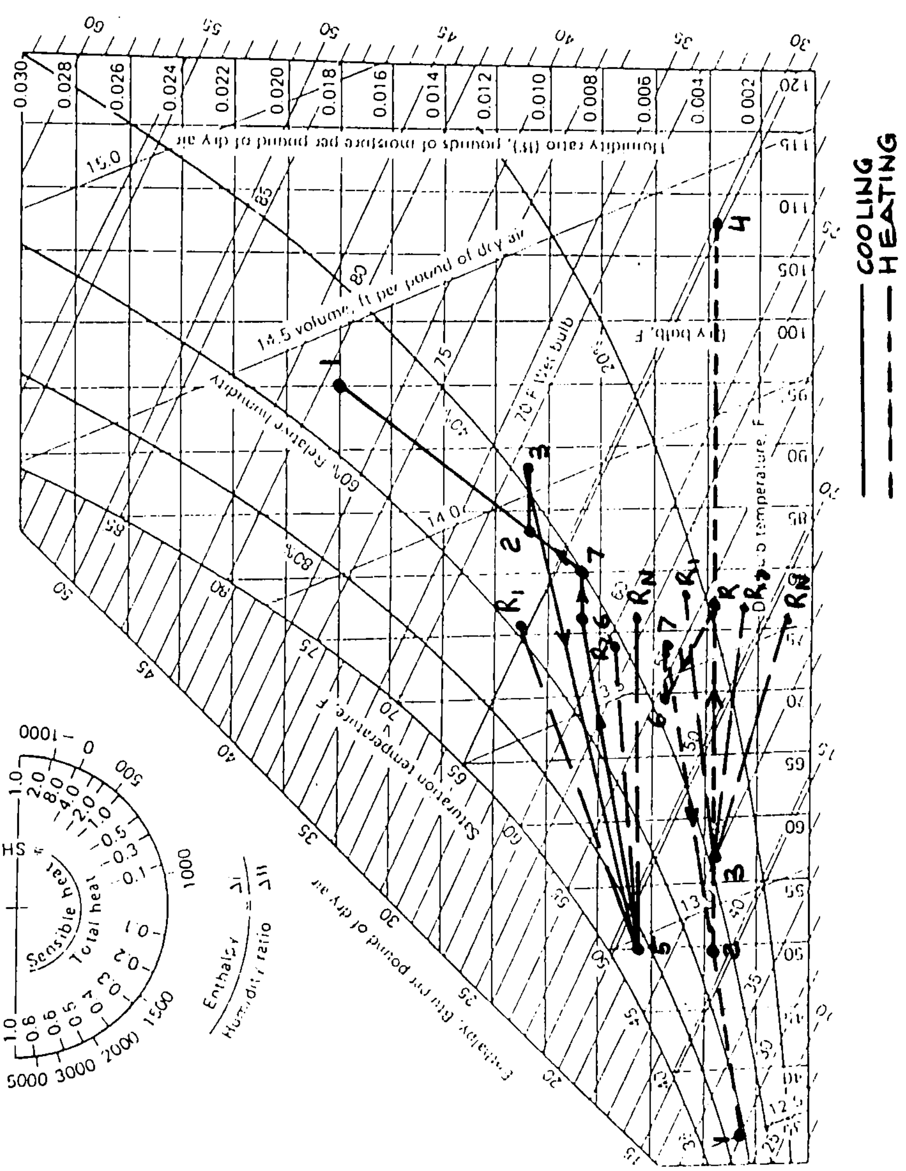


Figure 18-10, Dual Path System - Psychrometric Chart

18.2 All-Water Systems

These systems heat and/or cool a space by direct heat transfer between water and circulating air. Hot water systems deliver heat to a space by water that is hotter than the air in contact with the heat transfer surface. Examples of such systems include the following: 1) baseboard radiation, 2) free-standing radiators, 3) wall or floor radiant, 4) bare pipe, and 5) Other configurations.

These types of heating devices are classified as gravity convection. These systems are not state of the art and are seldom seen in new building applications.

18.2.1 Fan Coil Units

There are four basic principles of air conditioning and heating: 1) temperature control, 2) humidity control, 3) air movement, and 4) Air purity.

The unit that provides for the four basic principles above is called the fan coil unit. This unit forces air over heating or cooling coils to get the desired air temperatures. The typical fan coil with its basic parts is shown below:

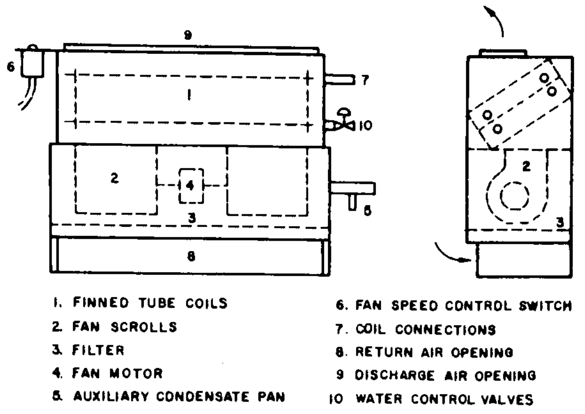


Figure 18-11, Typical Fan Coil Unit
Abridge from "ASHRAE Handbook, 1987 HVAC Systems and Applications", by The American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. 1987

Fan coil units come in variety of configurations. The figure below shows some of these configurations.

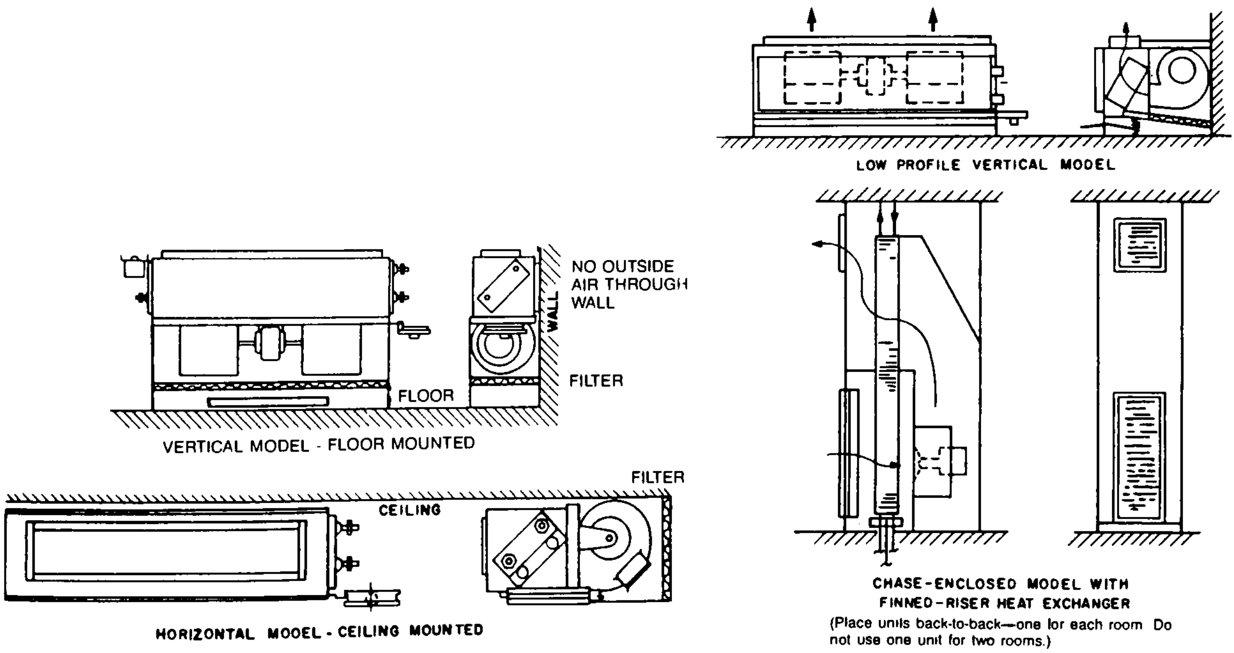


Figure 18-12, Typical Fan-Coil Unit Arrangements
Abridge from "ASHRAE Handbook, 1987 HVAC Systems and Applications", by The American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. 1987

18.2.2 Water Distribution

The various types of water distribution systems can be found in the Water Piping System section.

18.2.3 Applications

Fan-coil systems are best applied to individual space temperature control. Fan-coil systems also prevent cross contamination from one room to another. Suitable applications are hotels, motels, apartment buildings and office buildings.

18.2.4 Advantages and Disadvantages

Advantages:

1. Piping systems take up less space than duct systems.
2. Central water chilling or heating plants, while retaining the ability to shut off local terminals in unused areas.
3. Gives individual room control without cross-contamination.

Disadvantages:

1. All water systems require much more maintenance than central all air systems.
2. Units require condensate piping which has to be flushed periodically.
3. Filters need frequent changing.
4. Ventilation is more difficult.

18.2.5 Ventilation

Ventilation air is generally the most difficult factor to control and represents a major load component. Usually their is a central ventilation system that maintains a neutral air temperature of about 70°F. This system best controls ventilation air with the greatest freedom from problems related to stack effect and infiltration.

18.3 Air-and-Water Systems

Air-and-water systems condition spaces by distributing air and water sources to terminal units installed in habitable space throughout a building. The air and water are cooled or heated in central mechanical equipment rooms. The air supplied is called primary air and the water supplied is called secondary water. Sometimes a separate electric heating coil is included instead of a hot water coil.

Air-and-water systems apply primarily to exterior spaces of buildings with high sensible loads and where close control of humidity is not required. These systems work well in office buildings, hospitals, hotels, hotels, schools, apartment houses, research laboratories, and other buildings where their performance criteria meets the buildings needs.

These systems are usually installed in exterior building spaces and are designed to provide: 1) All required space heating and cooling needs, and 2) Simultaneous heating and cooling in different parts of the building.

18.3.1 System Description

An air-and-water system includes central air-conditioning equipment, duct and water distribution systems, and a room terminal. The room terminal may be an induction unit, fan coil unit, or a conventional supply air outlet combined with a radiant panel.

Generally, the air supply has a constant volume and clean outside air for ventilation. In the cooling season, the air is dehumidified sufficiently in the central conditioning unit to achieve comfort humidity conditions throughout the spaces served and to avoid condensation resulting from normal room latent load on the room coiling coil. In winter, moisture can be added centrally to limit the dryness.

The water side system is the basic type either two-pipe, three-pipe or four-pipe system. The water is usually heated by a control boiler and cooled by a central chiller.

18.3.2 Air-Water Induction Systems

The figure below shows the basic arrangement of an air-water induction terminal.

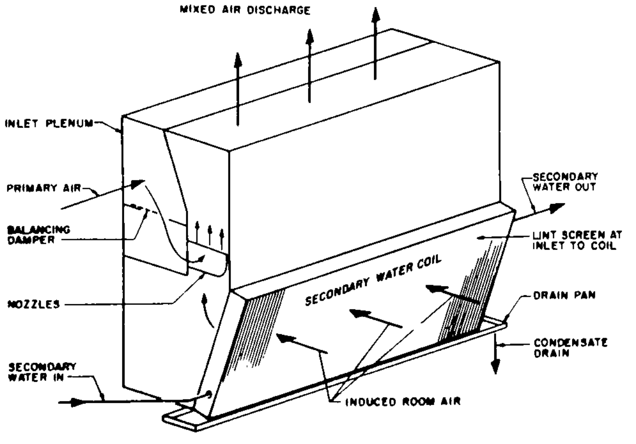


Figure 18-13, Air-Water Induction Unit
Abridge from "ASHRAE Handbook, 1987 HVAC Systems and Applications", by The American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. 1987

Centrally conditioned primary air is supplied to the unit plenum at high pressure. The acoustically treated plenum attenuates part of the noise generated in the unit and duct system. A balancing damper adjusts the primary air quantity within limits. The high-pressure air flows through the induction nozzles and induces secondary air from the room through the secondary coil. This secondary air is either heated or cooled, depending on the season, the room requirement or both.

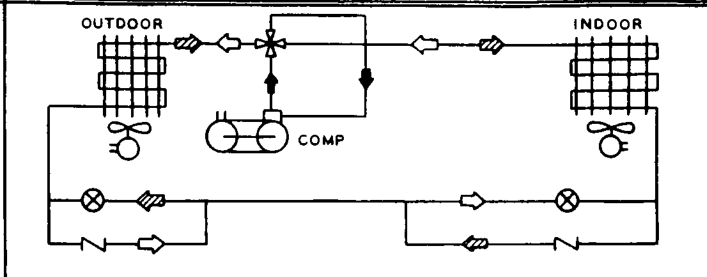
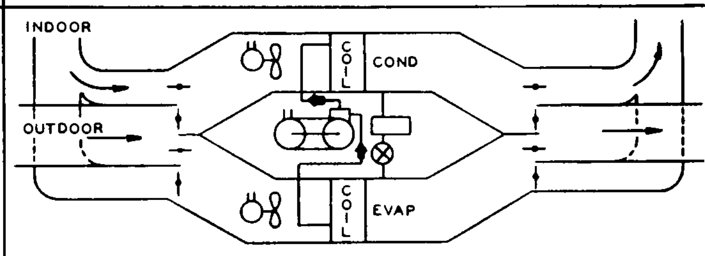
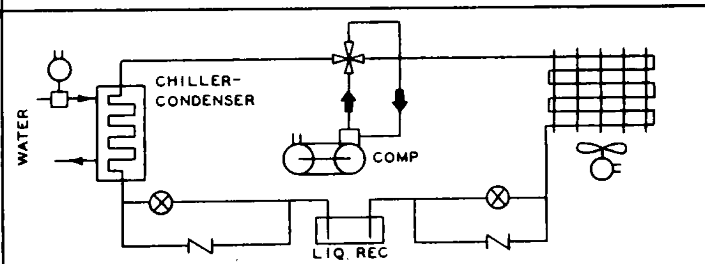
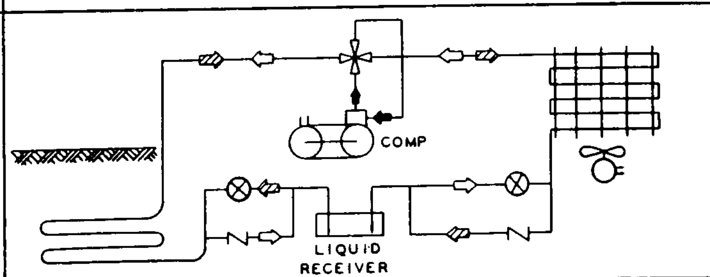
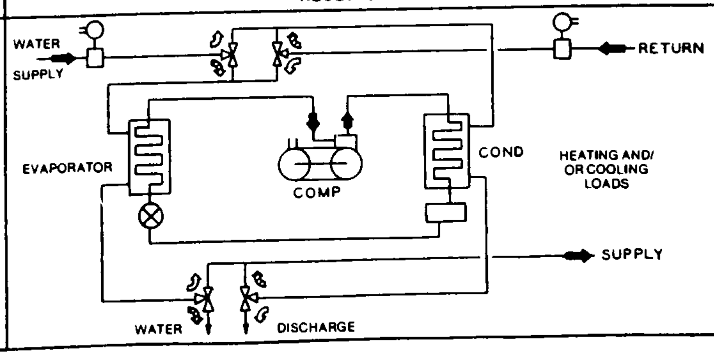
18.4 Heat Pumps - Applied

The thermodynamics located in the appendix discusses the heat pump cycle. This section discusses the application of the heat pump system. This section will discuss the most common type of heat pump, the closed vapor compression cycle.

18.4.1 Heat Pump Classification

Heat pumps are classified by 1) Heat source and sink, 2) Heating and cooling distribution fluid, 3) Building structure, 4) Size and configuration and 5) Limitation of the source and sink. Below is a chart of common heat pumps types.

Table 18-1, Common Heat Pump Types
Abridge from "ASHRAE Handbook, 1987 HVAC Systems and Applications", by The American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. 1987

HEAT SOURCE AND SINK	DISTR. FLUID	THERMAL CYCLE	DIAGRAM HEATING COOLING HEATING AND COOLING
AIR	AIR	REFRIGERANT CHANGEOVER	
AIR	AIR	AIR CHANGEOVER	
WATER	AIR	REFRIGERANT CHANGEOVER	
AIR	WATER		
EARTH	AIR	REFRIGERANT CHANGEOVER	
WATER	WATER	WATER CHANGEOVER	

There are many types of sources and sinks. The chart below discusses these sources and sinks.

Table 18-2, Heat Pump Sources and Sinks Abridge from "ASHRAE Handbook, 1987 HVAC Systems and Applications", by The American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. 1987

		Suitability		Availability		Cost		Temperature		Common Practice	
Source or Sink	Examples	Heat Source	Heat Sink	Location Relative to Need	Coincidence with Need	Installed	Operation and Maintenance	Level	Variation	Use	Limitations
AIR outdoor	ambient air	good, but performance and capacity fall when very cold	good, but performance and capacity fall when very hot	universal	continuous	low	low	variable	generally extreme	most common, many standard products	defrosting and supplemental heat usually required
exhaust	building ventilation	excellent, but insufficient	excellent, but insufficient	excellent if planned for in building design	excellent	low to moderate	low unless exhaust is dirt or grease laden	excellent	very low	emerging as conservation measure	insufficient for typical loads
WATER well	ground-water, well often shared with potable water source	excellent	excellent	poor to excellent, practical depth varies by location	continuous	low if existing well used or shallow wells suitable, can be high otherwise	low, but periodic maintenance required	generally excellent, varies by location	extremely stable	common	water disposal and required permits may limit, double wall exchangers may be required, may foul or scale
surface	lakes, rivers, ocean	excellent with large water bodies or high flow rates	excellent with large water bodies or high flow rates	limited, depends on proximity	usually continuous	depends on proximity and water quality	depends on proximity and water quality	usually satisfactory	depends on source	available, particularly for fresh water	often regulated or prohibited; may clog, foul, or scale
tap (city)	municipal water supply	excellent	excellent	excellent	continuous	low	low unless water use or disposal is charged	excellent	usually very low	excellent	use or disposal may be regulated or prohibited, may corrode or scale
condensing	cooling towers, refrigeration systems	excellent	poor to good	varies	varies with cooling loads	usually low	moderate	favorable as heat source	depends on source	available	suitable only if heating need is coincident with heat rejection
closed loops	building water-loop heat pump systems	good, loop may need supplemental heat	favorable, loop heat rejection may be needed	excellent if designed as such	as needed	low	moderate	as designed	as designed	very common	may be impractical as retrofit
waste	raw or treated sewage, grey water	fair to excellent	fair, varies with source	varies	varies, may be adequate	depends on proximity, high for raw sewage	varies, high for raw sewage	good	usually low	uncommon, practical only in large systems	usually regulated, may clog, foul, scale, or corrode

18.4.2 Operating Cycles

18.4.2.1 Water-to Water Heat Pump Cycle

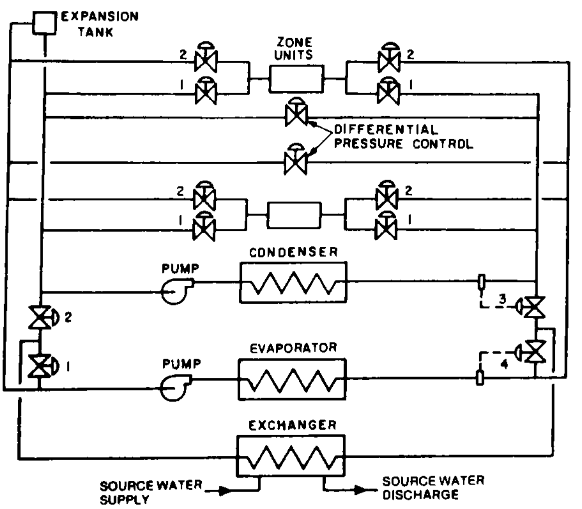


Figure 18-14, Water-to-Water Heat Pump System Abridge from "ASHRAE Handbook, 1987 HVAC Systems and Applications", by The American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. 1987

This system can provide simultaneous heating and cooling. This system uses water as the heat source or sinks and as the heating and cooling medium. Heating is provided to the zone units by closing valves 2 and 3 and expansion valves 1 and 4. Cooling is provided to the zone units by closing valves 1 and 4 and opening valves 2 and 3.

18.4.2.2 Air to Water Single-Stage Heat pump

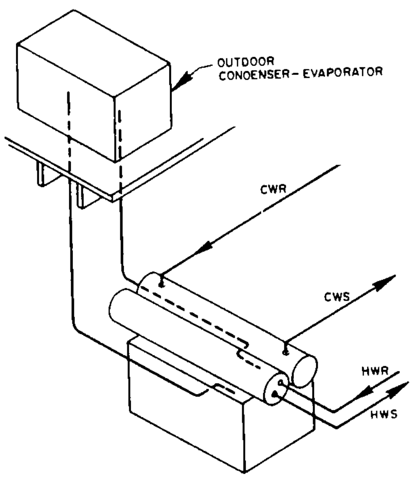


Figure 18-15, Air-to-Water Single Stage Heat Pump
Abridge from "ASHRAE Handbook, 1987 HVAC Systems and Applications", by The American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. 1987

This type of heat pump uses a 4 pipe system to provide simultaneous heating and cooling. The only operational reversal takes place in the outdoor unit.

18.4.2.3 Air-to-AIR Heat Pump

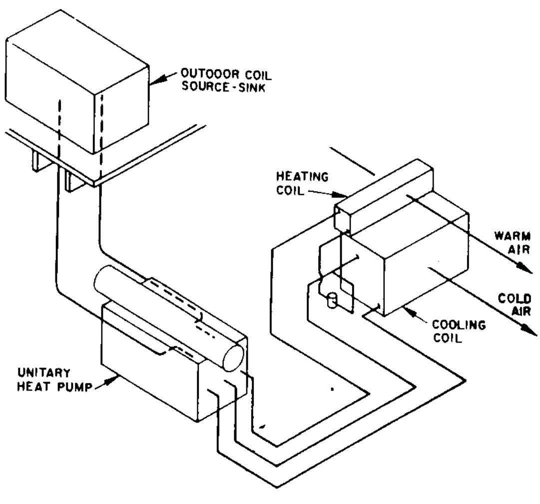


Figure 18-16, Air-to-Air Heat Pump
Abridge from "ASHRAE Handbook, 1987 HVAC Systems and Applications", by The American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. 1987

More efficient than the air to water heat pump because only one heating and cooling medium is used. There is also no heat exchangers to contend with.

18.4.2.4 Systems For Heat Recovery

These systems are used when commercial structures need simultaneous heating and cooling for prolong periods of time. These systems permit the transfer of surplus heat from one area of the building to another. There are three basic types of systems: 1) the heat transfer pump with a double-bundle condenser, 2) the heat transfer system with storage tank to store hot water at night, and 3) the multistage (cascade) heat transfer system.

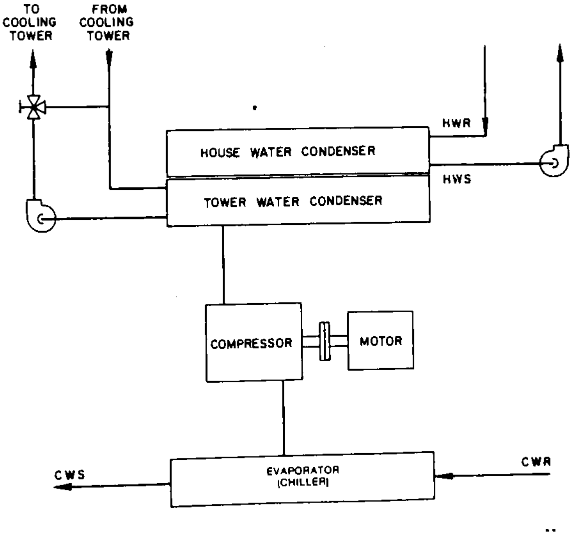


Figure 18-17, Heat Transfer Heat Pump with Double-Bundle Condenser

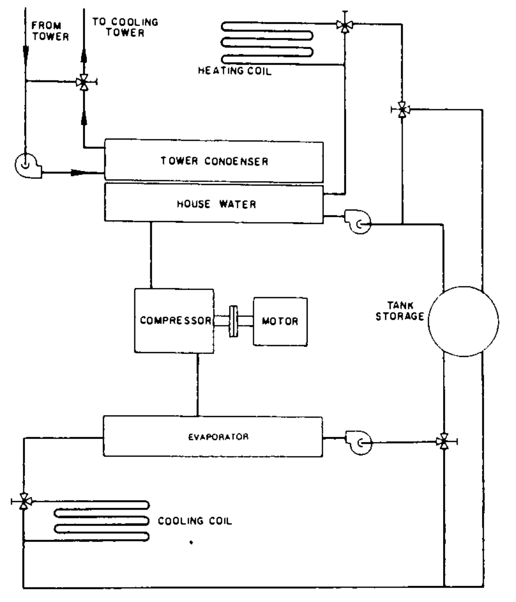


Figure 18-18, Heat Transfer System with Storage Tank

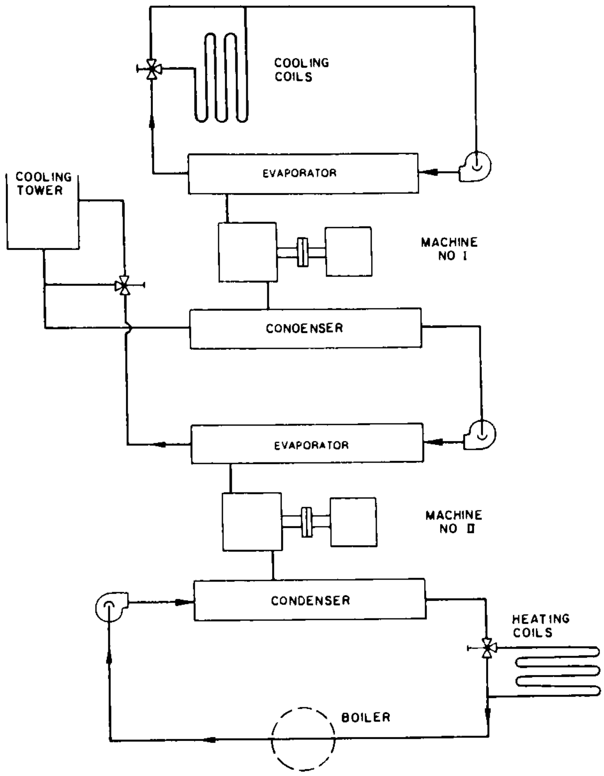


Figure 18-19, Multistage (Cascade) Heat Transfer System
Abridge from "ASHRAE Handbook, 1987 HVAC Systems and Applications", by The American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. 1987

18.4.2.5 Water-Loop System

In water loop systems individual heat pumps can either reject heat or absorbed heat from the water loop.

The closed loop solar heat pump system uses a storage tank to store access heat during the day time to be used at night. The solar collectors are used to add heat to the system in the winter time. This system with the condenser and evaporator eliminates the need for a boiler in the system. A diagram of this system is shown below.

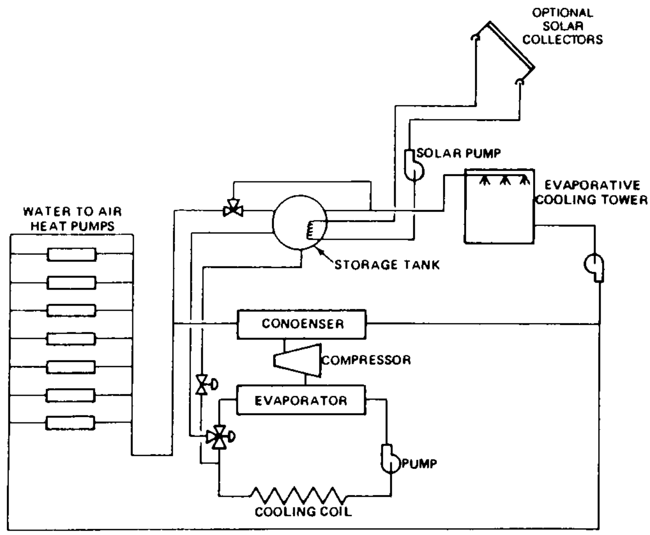


Figure 18-20, Closed Loop Solar Heat Pump System
Abridge from "ASHRAE Handbook, 1987 HVAC Systems and Applications", by The American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. 1987

The heat transfer system using water-to-air unitary heat pumps provides the transfer of heat from one part of a building to another through the use of unitary heat pumps and a two two-pipe water loop system. A boiler and a evaporative cooler are used to maintain the water loop temperature to some specific temperature. The diagram below shows this system.

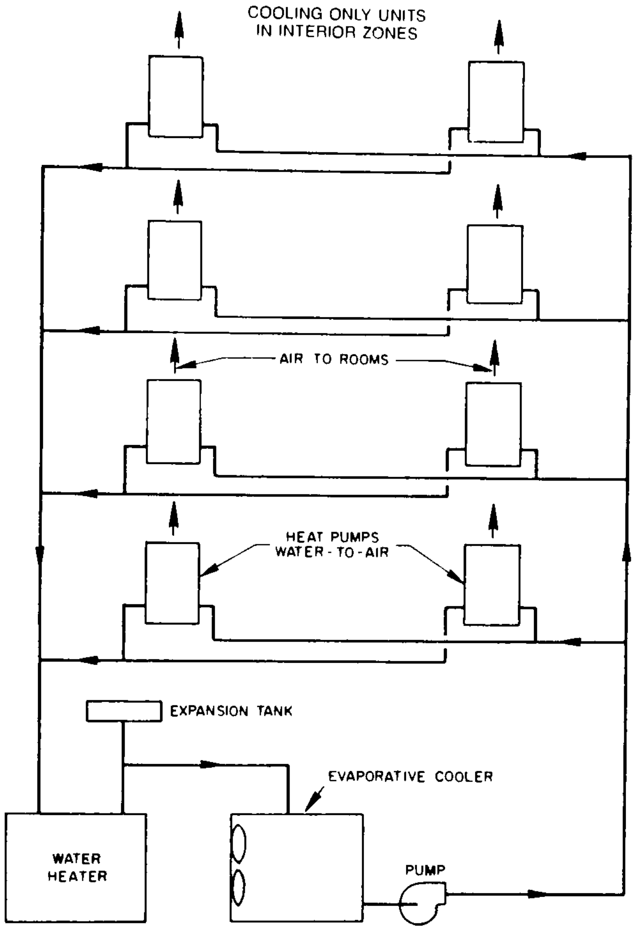


Figure 18-21, Heat Transfer System Using Water-to-Air Unitary Heat Pump
Abridge from "ASHRAE Handbook, 1987 HVAC Systems and Applications", by The American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. 1987

19. Special Topics

19.1 Noise and Vibration

In HVAC and refrigeration systems design it vary important to keep in mind the effects of noise and vibration. In these systems their are many sources of noise and vibration. Some examples are: High velocity turbulent flow in air ducts, High velocity air flow through louvers and diffusers, Reciprocating equipment, connections between equipment, piping, ducts and mounts, and the list goes on and on.

To eliminate th problems of noise and vibration in the cases stated above: Ducts should be acoustically lined where noise transmitted through the duct is likely, Air flows in ducts should be kept to a level where they do not cause excess noise, All equipment should be mounted with vibration isolators, there should be flexible connections between equipment, pipes and ducts, and equipment should be placed in areas where its noise is kept to a minimum.

There are many other contributors to noise and vibrations but they are too lengthy to be discussed for the scope of this paper.

19.2 Codes

There are many codes which regulate the designs of HVAC systems. These codes are enforced to ensure the safety of the people using the HVAC system. There are two basic groups of codes, state and local (town or county). There are several types of codes in each group. Some examples are: mechanical, electrical, plumbing, structural and fire codes. Before the design of an HVAC system, a code search must be preformed to find the codes that apply the system being designed.

19.3 Controls

Controls are used to run and regulate a HVAC system. Controls are usually regulated by temperature but they also can be regulated by humidity, pressure, flow rate, and other equipment. Controls can be in the form of analog or digital.

It is important to have the right controls and control sequence because controls to a HVAC system is like the central nervous system in a human being. Vary complex control systems are usually handled by control engineers and specialists. Usually in a HVAC design the designer will specify the control sequence or order of operation and the control engineer will design the controls to meet that sequence.

References

1. Faye C. McQuiston and Jerald D. Parker, "Heating Ventilation and Air Conditioning, Analysis and Design," 2^{and} ed., John Whiley and Sons, 1982

2. "ASHRAE Handbook, 1985 Fundamentals, inch-pound Edition," American Society of Heating, Refrigeration and Air Conditioning Engineers, Inc., 1985

3. George E. Clifford, "Heating Ventilation and Air Conditioning," 2^{and} ed., Reston Publishing Company, Inc., 1984

4. William S. Janna, "Engineering Heat Transfer," 1 st ed., PWS Publishers, 1986

5. John W. Fox and Alan T. McDonald, "Introduction to Fluid Mechanics," 3^{rd} ed., Wiley and Sons, 1985

6. B. V. Karlekar, "Thermodynamics for Engineers," 1^{st} ed., Prentice Hall, 1983

7. "ASHRAE Handbook, 1987 HVAC Systems and Applications", American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc., 1987

8. Lee Kendric, P.E. "How to Design and Specify a Good Variable Air Volume Heating, Ventilation and Air Conditioning System", Merrifield, Va.

APPENDIX

20. Calculations

20.1 Load Calculation Sample Output

DESIGN SPACE COOLING LOADS

Zone Name : HP-2 02-10-89
Location : Hartford, Connecticut Aug 1500h
Job Name : THESIS
Prepared By : FISHER & KUEGLER, P.C. 6121587110
Carrier Hourly Analysis Program Page 1

****

Space Name	Mult	Space Sensible (tons /space)	Supply Air (CFM/space)
2-S1	x 1	0.67	416.6
2-S2.3.4.5	x 4	1.06	657.4
2-S6	x 1	0.70	430.9
2-N1	x 1	0.53	329.3
2-N2.3.4.5	x 4	0.78	484.4
2-N6	x 1	0.56	343.6
2-E1	x 1	0.60	372.6
2-E2.3	x 2	0.87	535.5
2-E4	x 1	0.49	302.1
2-W1	x 1	0.63	387.0
2-W2.3	x 2	0.49	301.3
2-W4	x 1	0.51	316.5

****

DESIGN SPACE HEATING LOADS

Zone Name : HP-2 02-10-29
Location : Hartford, Connecticut Winter db : 3.0 F
Job Name : THESIS Indoor db : 68.0 F
Prepared By : FISHER & KUEGLER, P.C. 6121587110
Carrier Hourly Analysis Program Page 1 of 1

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Space Name	Mult	Space Sensible (BTU/hr/space)	Supply Air (CFM/space)
2-S1	x 1	4,710.9	128.4
2-S2,3,4,5	x 4	5,660.4	154.2
2-S6	x 1	4,711.0	128.4
2-N1	x 1	4,711.0	128.4
2-N2,3,4,5	x 4	5,660.4	154.2
2-N6	x 1	4,710.9	128.4
2-E1	x 1	4,241.7	115.6
2-E2,3	x 2	5,402.6	147.2
2-E4	x 1	4,241.7	115.6
2-W1	x 1	4,241.6	115.6
2-W2,3	x 2	2,650.8	72.2
2-W4	x 1	4,241.6	115.6

****

ZONE DESIGN COOLING LOAD SUMMARY

Zone Name : HP-2
Location : Hartford, Connecticut
Job Name : THESIS
Prepared By : FISHER & KUEGLER, P.C.
Carrier Hourly Analysis Program

09-10-99
Aug 1500h
6121587110

LOAD COMPONENT	SENSIBLE (BTU/hr)	LATENT (BTU/hr)
SOLAR GAIN	52,387	0
GLASS TRANSMISSION	14,840	0
WALL TRANSMISSION	3,554	0
ROOF TRANSMISSION	0	0
TRANS. LOSS TO UNCOND. SPACE	0	0
LIGHTING (13,037 W TOTAL)	44,482	0
OTHER ELEC. (9,312 W TOTAL)	31,773	0
PEOPLE (42.33 PEOPLE TOTAL)	10,370	8,677
MISCELLANEOUS LOADS	0	0
COOLING INFILTRATION	4,021	5,385
COOLING SAFETY LOAD	16,143	1,406
SUB-TOTALS	177,569	15,468
NET VENTILATION LOAD (847 CFM)	14,428	19,316
SUPPLY FAN LOAD (BHP= 7.3)	18,476	0
ROOF LOAD TO PLENUM	0	0
LIGHTING LOAD TO PLENUM	19,064	0
TOTAL COOLING LOADS	229,536	34,784

TOTAL COOLING LOAD = 264,320 BTU/hr
or 22.03 Tons or 211.4 sqft/Tons
ZONE TOTAL FLOOR AREA = 4,656.00 sqft
ZONE OVERALL U-FACTOR = 0.189 BTU/hr/sqft/F

Transmission and Solar Gain by Exposure

LOAD COMPONENT		AREA (sqft)	TRANSMISSION (BTU/hr)	SOLAR GAIN (BTU/hr)
GLASS LOADS:	NE	0	0	0
	E	350	2,968	9,917
	SE	0	0	0
	S	580	4,918	25,291
	SW	0	0	0
	W	240	2,035	8,160
	NW	0	0	0
	N	580	4,918	9,019
	H	0	0	0
WALL LOADS:	NE	0	0	-
	E	826	484	-
	SE	0	0	-
	S	1,292	2,004	-
	SW	0	0	-
	W	696	589	-
	NW	0	0	
	N	1,292	478	

****

COIL SELECTION PARAMETERS

COIL ENTERING AIR TEMP. (DB/WB)	=	78.4/ 64.4 deg F
COIL LEAVING AIR TEMP. (DB/WB)	=	55.1/ 54.5 deg F
COIL SENSIBLE LOAD	=	229.536 BTU/hr
COIL TOTAL LOAD	=	264.320 BTU/hr
COOLING SUPPLY AIR TEMPERATURE	=	57.0 deg F
TOTAL COOLING CFM (actual)	=	9.139 CFM
TOTAL COOLING CFM (std. air)	=	9.134 CFM
COOLING CFM/sqft	=	1.96 CFM/sqft
RESULTING ROOM REL. HUMIDITY	=	49 %
COIL BYPASS FACTOR	=	0.050

****

ZONE DESIGN HEATING LOAD SUMMARY

Zone Name : HP-2
Location : Hartford. Connecticut
Job Name : THESIS
Prepared By : FISHER & KUEGLER. P.C.
Carrier Hourly Analysis Program

02-10-89
Winter db : 3.0 F
Indoor db : 69.0 F
6121587110

****

Note: Heating load is computed at winter design condition.

LOAD COMPONENT	LOAD (BTU/hr)
WALL TRANSMISSION	11.743
ROOF TRANSMISSION	0
GLASS TRANSMISSION	60.287
TRANSMISSION LOSS TO UNCOND. SPACES	0
INFILTRATION LOSS	16.334
SLAB FLOOR	0
HEATING SAFETY BTU/hr	8.836
SUB-TOTAL	97.201
NET VENTILATION LOSS	59.395
TOTAL HEATING LOAD	156.596

HEATING SUPPLY CFM	2.649 CFM
HEATING SUPPLY AIR TEMPERATURE	102.0 deg F
HEATING VENTILATION AIR CFM	847 CFM
HEATING SEASON ROOM DRY BULB TEMP.	68.0 deg F

****

20.2 Piping System

PIPE SYSTEM INPUT SHEET

System Name: FlOOR - 2

Sheet 1 of 3

Section Number	Length	Flow	Known Loss	Junction Start No.	Junction Start Type	Junction End No.	Junction End Type	Fitting Data: Additional Thru Tee	Fitting Data: Additional Branch Tee	(3)	(4) GATE	(5) 45° EL	(6) ANGLE	(7) 90° EL	(8)	(9)	(10)
1	5'	134	–	0	N	1	N				1			1			
2	50'	44	–	1	T	2	N							2			
3	27'	41.6	–	2	T	3	N										
4	8'	39.2	–	3	T	4	N										
5	4'	37.5	–	4	T	5	N							1			
6	9'	35.8	–	5	T	6	N										
7	46'	33.4	–	6	T	7	N										
8	8'	31	–	7	T	8	N										
9	4'	29.3	–	8	T	9	N							1			
10	8'	27.3	–	9	T	10	N										
11	27'	24.3	–	10	T	11	N										
12	47'	21.3	–	11	T	12	N										
13	27'	18.3	–	12	T	13	N										
14	8'	15.3	–	13	T	14	N										
15	4'	13.3	–	14	T	15	N							1			
16	8'	11.6	–	15	T	16	N										
17	57'	9.9	–	16	T	17	N							4			
18	8'	8.2	–	17	T	19	N										
19	4'	6.5	–	18	T	19	N							1			
20	8'	4.8	–	19	T	20	N										
21	46'	90	25	1	B	21	T				2			4			
22	27'	92.4	–	21	N	22	T										
23	8'	94.8	–	22	N	23	T										
24	4	96.5	–	23	N	24	T							1			
25	9	98.2	–	24	N	25	T										

PIPE SYSTEM INPUT SHEET

System Name: Floor-2

Sheet 2 of 3

Section Number	Length	Flow	Known Loss	Junction Start No.	Junction Start Type	Junction End No.	Junction End Type	Fitting Data: Additional Thru Tee	Fitting Data: Additional Branch Tee	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
26	46	100.6	–	25	N	26	T										
27	8	103	–	26	N	27	T										
28	4	104.7	–	27	N	28	T										
29	8	106.7	–	28	N	29	T							1			
30	27	109.7	–	29	N	30	T										
31	47	112.7	–	30	N	31	T										
32	27	115.7	–	31	N	32	T										
33	8	118.7	–	32	N	33	T										
34	4	120.7	–	33	N	34	T										
35	8	122.4	–	34	N	35	T							1			
36	57	124.1	–	35	N	36	T										
37	8	125.8	–	36	N	37	T							4			
38	4	127.5	–	37	N	38	T										
39	8'	129.2	–	38	N	39	T							1			
40	27'	131.6	–	39	N	40	T										
41	61'	134	–	40	N	41	N				1			4			
42		2.4	5.4	2	B	21	B				2						
43		2.4	5.4	3	B	22	B				2						
44		1.7	5	4	B	23	B				2						
45		1.7	5	5	B	24	B				2						
46		2.4	5.4	6	B	25	B				2						
47		2.4	5.4	7	B	26	B				2						
48		1.7	5	8	B	27	B				2						
49		2	3.8	9	B	28	B				2						
50		3	8.3	10	B	29	B				2						

PIPE SYSTEM INPUT SHEET

System Name: Floor-2

Sheet 3 of 3

Section Number	Length	Flow	Known Loss	Junction Start No.	Junction Start Type	Junction End No.	Junction End Type	Fitting Data: Additional Thru Tee	Fitting Data: Additional Branch Tee	(3)	(4) SATI	(5)	(6) ANGLE	(7) 90° EL	(8)	(9)	(10)
51		3	8,3	11	B	30	B				2						
52		3	8,3	12	B	31	B				2						
53		3	8,3	13	B	32	B				2						
54		2	3,8	14	B	32	B				2						
55		1,7	5	15	B	24	B				2						
56		1,7	5	16	B	35	B				2						
57		1,7	5	17	B	36	B				2						
58		1,7	5	18	B	37	B				2						
59		1,7	5	19	B	38	B				2						
60		2,4	5,4	20	B	39	B				2						
61		2,4	5,4	20	T	40	B				2						

Water Pipe Design

For

FLOOR 2

By

THESIS

02-17-89

FLOOR-2

WATER PIPE SIZING PARAMETERS

Material Selected: Schedule 40 Steel, Closed System
Fluid Selected: Ethylene Glycol Solution
Average Fluid Temperature: 75.00 F
Under 4 in., use maximum velocity of 2.5 (fps)
At or above 4 in., use maximum friction loss of 4 (Ft. H20/100 Ft.)

PIPE SECTION DATA

SECTION NO.	LENGTH (ft)	FLOW (gpm)	START JUNCTION	END JUNCTION
1	5.000	134.00	0	1
2	50.000	44.00	1	2
3	27.000	41.60	2	3
4	8.000	39.20	3	4
5	4.000	37.50	4	5
6	9.000	35.80	5	6
7	46.000	33.40	6	7
8	8.000	31.00	7	8
9	4.000	29.30	8	9
10	8.000	27.30	9	10
11	27.000	24.30	10	11
12	47.000	21.30	11	12
13	27.000	18.30	12	13
14	8.000	15.30	13	14
15	4.000	13.30	14	15
16	8.000	11.60	15	16
17	57.000	9.90	16	17
18	8.000	8.20	17	18
19	4.000	6.50	18	19
20	8.000	4.80	19	20
21	46.000	90.00	1	21
22	27.000	92.40	21	22
23	8.000	94.80	22	23
24	4.000	96.50	23	24
25	9.000	98.20	24	25
26	46.000	100.60	25	26
27	8.000	103.00	26	27
28	4.000	104.70	27	28
29	8.000	106.70	28	29
30	27.000	109.70	29	30
31	47.000	112.70	30	31
32	27.000	115.70	31	32
33	8.000	118.70	32	33
34	4.000	120.70	33	34
35	8.000	122.40	34	35
36	57.000	124.10	35	36
37	8.000	125.80	36	37
38	4.000	127.50	37	38
39	8.000	129.20	38	39

SECTION NO.	LENGTH (ft)	FLOW (gpm)	START JUNCTION	END JUNCTION
40	27.000	131.60	39	40
41	61.000	134.00	40	41
42	0.000	2.40	2	21
43	0.000	2.40	3	22
44	0.000	1.70	4	23
45	0.000	1.70	5	24
46	0.000	2.40	6	25
47	0.000	2.40	7	26
48	0.000	1.70	8	27
49	0.000	2.00	9	28
50	0.000	3.00	10	29
51	0.000	3.00	11	30
52	0.000	3.00	12	31
53	0.000	3.00	13	32
54	0.000	2.00	14	33
55	0.000	1.70	15	34
56	0.000	1.70	16	35
57	0.000	1.70	17	36
58	0.000	1.70	18	37
59	0.000	1.70	19	38
60	0.000	2.40	20	39
61	0.000	2.40	20	40

FITTING (1-5) LOSS DATA

SECT. NO.	THRU TEE	BRANCH TEE	GLOBE VALV	ANGLE VALV	GATE VALVE	KNOWN LOSS (Ft. H20)
1	0	0	0	0	1	0.000
2	1	0	0	0	0	0.000
3	1	0	0	0	0	0.000
4	1	0	0	0	0	0.000
5	1	0	0	0	0	0.000
6	1	0	0	0	0	0.000
7	1	0	0	0	0	0.000
8	1	0	0	0	0	0.000
9	1	0	0	0	0	0.000
10	1	0	0	0	0	0.000
11	1	0	0	0	0	0.000
12	1	0	0	0	0	0.000
13	1	0	0	0	0	0.000
14	1	0	0	0	0	0.000
15	1	0	0	0	0	0.000
16	1	0	0	0	0	0.000
17	1	0	0	0	0	0.000
18	1	0	0	0	0	0.000
19	1	0	0	0	0	0.000
20	1	0	0	0	0	0.000
21	1	1	0	0	2	25.000
22	1	0	0	0	0	0.000
23	1	0	0	0	0	0.000

SECT. NO.	THRU TEE	BRANCH TEE	GLOBE VALV	ANGLE VALV	GATE VALVE	KNOWN LOSS (Ft. H2O)
24	1	0	0	0	0	0.000
25	1	0	0	0	0	0.000
26	0	0	0	0	0	0.000
27	1	0	0	0	0	0.000
28	1	0	0	0	0	0.000
29	1	0	0	0	0	0.000
30	1	0	0	0	0	0.000
31	1	0	0	0	0	0.000
32	0	0	0	0	0	0.000
33	1	0	0	0	0	0.000
34	1	0	0	0	0	0.000
35	1	0	0	0	0	0.000
36	1	0	0	0	0	0.000
37	0	0	0	0	0	0.000
38	1	0	0	0	0	0.000
39	1	0	0	0	0	0.000
40	1	0	0	0	0	0.000
41	0	0	0	0	1	0.000
42	0	2	0	0	2	5.400
43	0	2	0	0	2	5.400
44	0	2	0	0	2	5.000
45	0	2	0	0	4	5.000
46	0	2	0	0	2	5.400
47	0	2	0	0	4	5.400
48	0	2	0	0	2	5.000
49	0	2	0	0	2	3.800
50	0	2	0	0	2	8.300
51	0	2	0	0	2	8.300
52	0	2	0	0	2	8.300
53	0	2	0	0	2	8.300
54	0	2	0	0	2	3.800
55	0	2	0	0	2	5.000
56	0	2	0	0	2	5.000
57	0	2	0	0	2	5.000
58	0	2	0	0	2	5.000
59	0	2	0	0	2	5.000
60	0	2	0	0	2	5.400
61	1	1	0	0	2	5.400

FITTING (6-10) LOSS DATA

SECT. NO.	90 DEG ELB	45 DEG ELB	KNOWN LOSS (Ft. H2O)
1	1	0	0.000
2	2	0	0.000
3	0	0	0.000
4	0	0	0.000
5	1	0	0.000
6	0	0	0.000
7	0	0	0.000

SECT. NO.	90 DEG ELB	45 DEG ELB	KNOWN LOSS (Ft. H20)
8	0	0	0.000
9	1	0	0.000
10	0	0	0.000
11	0	0	0.000
12	0	0	0.000
13	0	0	0.000
14	0	0	0.000
15	1	0	0.000
16	0	0	0.000
17	4	0	0.000
18	0	0	0.000
19	1	0	0.000
20	0	0	0.000
21	4	0	25.000
22	0	0	0.000
23	0	0	0.000
24	1	0	0.000
25	0	0	0.000
26	0	0	0.000
27	0	0	0.000
28	1	0	0.000
29	0	0	0.000
30	0	0	0.000
31	0	0	0.000
32	0	0	0.000
33	0	0	0.000
34	1	0	0.000
35	0	0	0.000
36	4	0	0.000
37	0	0	0.000
38	1	0	0.000
39	0	0	0.000
40	0	0	0.000
41	4	0	0.000
42	0	0	5.400
43	0	0	5.400
44	0	0	5.000
45	0	0	5.000
46	0	0	5.400
47	0	0	5.400
48	0	0	5.000
49	0	0	3.800
50	0	0	8.300
51	0	0	8.300
52	0	0	8.300
53	0	0	8.300
54	0	0	3.800
55	0	0	5.000
56	0	0	5.000
57	0	0	5.000
58	0	0	5.000
59	0	0	5.000
60	0	0	5.400

SECT. NO.	90 DEG ELB	45 DEG ELB	KNOWN LOSS (Ft. H2O)
61	1	0	5.400

SIZING RESULTS

SECT NO.	NOM. SIZE (in)	FLOW (gpm)	VELOCITY (fps)	EQUIV. LENGTH (ft)	FRICTION LOSS (Ft. H2O)	BALANCE REQ'D (Ft. H2O)
1	4	134.00	3.37	19.50	0.23	
2	3	44.00	1.91	70.00	0.40	16.49
3	3	41.60	1.80	32.00	0.17	
4	3	39.20	1.70	13.00	0.06	
5	3	37.50	1.63	16.50	0.07	
6	2 1/2	35.80	2.40	13.10	0.15	
7	2 1/2	33.40	2.24	50.10	0.51	
8	2 1/2	31.00	2.07	12.10	0.11	
9	2 1/2	29.30	1.96	14.10	0.11	
10	2 1/2	27.30	1.83	12.10	0.08	
11	2	24.30	2.32	30.30	0.41	
12	2	21.30	2.03	50.30	0.53	
13	2	18.30	1.75	30.30	0.24	0.00
14	1 1/2	15.30	2.41	10.60	0.21	2.42
15	1 1/2	13.30	2.09	10.60	0.16	
16	1 1/4	11.60	2.49	10.30	0.26	
17	1 1/4	9.90	2.12	72.50	1.36	
18	1 1/4	8.20	1.76	10.30	0.14	
19	1	6.50	2.41	8.30	0.28	
20	1	4.80	1.78	9.70	0.18	
21	4	90.00	2.27	122.70	25.69	
22	4	92.40	2.33	33.70	0.20	
23	4	94.80	2.39	14.70	0.09	
24	4	96.50	2.43	20.70	0.13	
25	4	98.20	2.47	15.70	0.10	
26	4	100.60	2.53	46.00	0.32	
27	4	103.00	2.59	14.70	0.11	
28	4	104.70	2.64	20.70	0.15	
29	4	106.70	2.69	14.70	0.11	
30	4	109.70	2.76	33.70	0.27	
31	4	112.70	2.84	53.70	0.46	
32	4	115.70	2.91	27.00	0.24	
33	4	118.70	2.99	14.70	0.14	
34	4	120.70	3.04	20.70	0.20	
35	4	122.40	3.08	14.70	0.15	
36	4	124.10	3.12	103.70	1.06	
37	4	125.80	3.17	8.00	0.08	
38	4	127.50	3.21	20.70	0.22	
39	4	129.20	3.25	14.70	0.16	
40	4	131.60	3.31	33.70	0.38	
41	4	134.00	3.37	105.50	1.24	
42	3/4	2.40	1.44	9.80	5.57	3.23
43	3/4	2.40	1.44	9.80	5.57	3.26

SECT NO.	NOM. SIZE (in)	FLOW (gpm)	VELOCITY (fps)	EQUIV. LENGTH (ft)	FRICTION LOSS (Ft. H20)	BALANCE REQ'D (Ft. H20)
44	1/2	1.70	1.79	7.40	5.27	3.59
45	1/2	1.70	1.79	8.80	5.32	3.60
46	3/4	2.40	1.44	9.80	5.57	3.31
47	3/4	2.40	1.44	11.60	5.60	3.09
48	1/2	1.70	1.79	7.40	5.27	3.42
49	1/2	2.00	2.11	7.40	4.17	4.57
50	3/4	3.00	1.80	9.80	8.56	0.21
51	3/4	3.00	1.80	9.80	8.56	0.07
52	3/4	3.00	1.80	9.80	8.56	
53	3/4	3.00	1.80	9.80	8.56	
54	1/2	2.00	2.11	7.40	4.17	1.21
55	1/2	1.70	1.79	7.40	5.27	0.83
56	1/2	1.70	1.79	7.40	5.27	0.73
57	1/2	1.70	1.79	7.40	5.27	0.43
58	1/2	1.70	1.79	7.40	5.27	0.38
59	1/2	1.70	1.79	7.40	5.27	0.32
60	3/4	2.40	1.44	9.80	5.57	
61	3/4	2.40	1.44	9.20	5.56	0.39

Total System Friction Head (Ft. H20) = 31.76
Total System Fluid Volume (gal) = 376.17

ESTIMATE OF MATERIALS

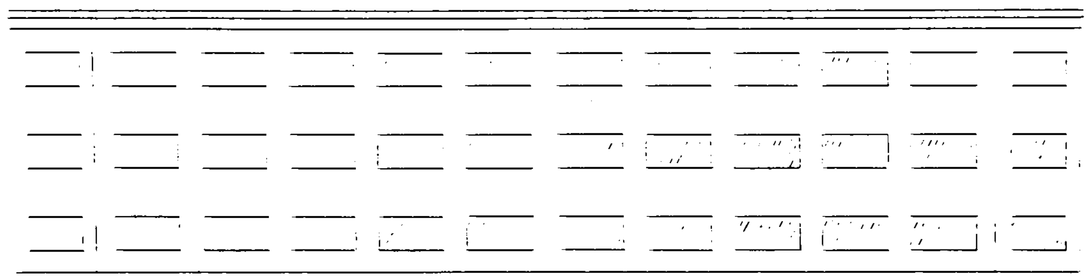
FITTINGS	Pipe Nominal Size (in) 1/2	3/4	1	1 1/4	1 1/2	2
GLOBE VALV	0	0	0	0	0	0
ANGLE VALV	0	0	0	0	0	0
GATE VALVE	22	22	0	0	0	0
90 DEG ELB	0	1	1	4	1	0
45 DEG ELB	0	0	0	0	0	0
TOTAL ft:	0.00	0.00	12.00	73.00	12.00	101.00

FITTINGS	Pipe Nominal Size (in) 2 1/2	3	4
GLOBE VALV	0	0	0
ANGLE VALV	0	0	0
GATE VALVE	0	0	4
90 DEG ELB	1	3	17
45 DEG ELB	0	0	0
TOTAL ft:	25.00	89.00	451.00

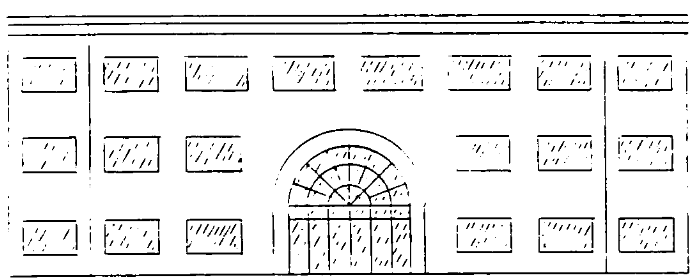
JUNCTION NUMBER	DIMENSIONS AT JUNCTION
1	4 x 4 x 3
2	3 x 3 x 3/4
3	3 x 3 x 3/4
4	3 x 3 x 1/2
5	3 x 2 1/2 x 1/2
6	2 1/2 x 2 1/2 x 3/4
7	2 1/2 x 2 1/2 x 3/4
8	2 1/2 x 2 1/2 x 1/2
9	2 1/2 x 2 1/2 x 1/2
10	2 1/2 x 2 x 3/4
11	2 x 2 x 3/4
12	2 x 2 x 3/4
13	2 x 1 1/2 x 3/4
14	1 1/2 x 1 1/2 x 1/2
15	1 1/2 x 1 1/4 x 1/2
16	1 1/4 x 1 1/4 x 1/2
17	1 1/4 x 1 1/4 x 1/2
18	1 1/4 x 1 x 1/2
19	1 x 1 x 1/2
20	1 x 3/4 x 3/4
21	4 x 4 x 3/4
22	4 x 4 x 3/4
23	4 x 4 x 1/2
24	4 x 4 x 1/2
25	4 x 4 x 3/4
26	4 x 4 x 3/4
27	4 x 4 x 1/2
28	4 x 4 x 1/2
29	4 x 4 x 3/4
30	4 x 4 x 3/4
31	4 x 4 x 3/4
32	4 x 4 x 3/4
33	4 x 4 x 1/2
34	4 x 4 x 1/2
35	4 x 4 x 1/2
36	4 x 4 x 1/2
37	4 x 4 x 1/2
38	4 x 4 x 1/2
39	4 x 4 x 3/4
40	4 x 4 x 3/4

21. Drawings

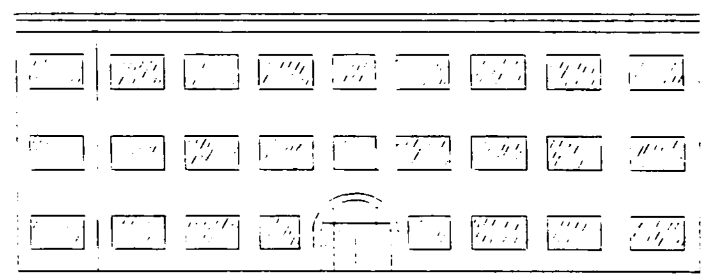
21.1 Architectural



NORTH AND SOUTH ELEVATION
SCALE: 1/8" = 1'-0"



WEST ELEVATION
SCALE: 1/8" = 1'-0"



EAST ELEVATION
SCALE: 1/8" = 1'-0"

HEATING, VENTILATION AND AIR CONDITIONING

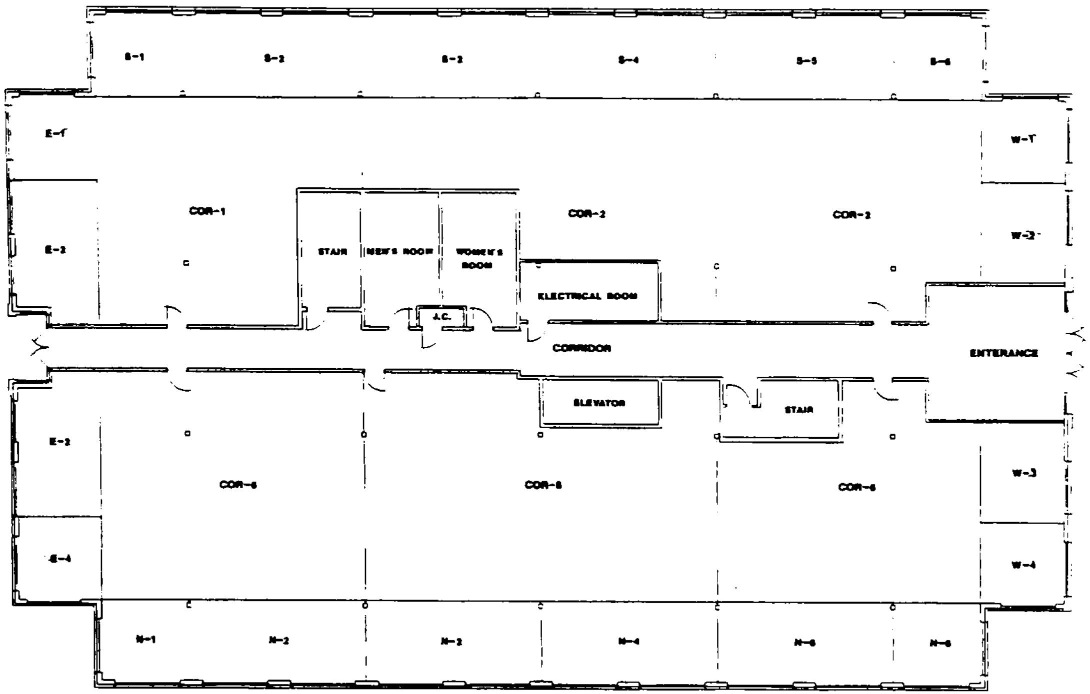
ARCHITECTURAL

PROJECT NUMBER

DRAWING NUMBER

A–1

BY: KURT KUEGLER



FIRST FLOOR
SCALE: 1/8" = 1'-0"

NORTH

HEATING, VENTILATION AND
AIR CONDITIONING

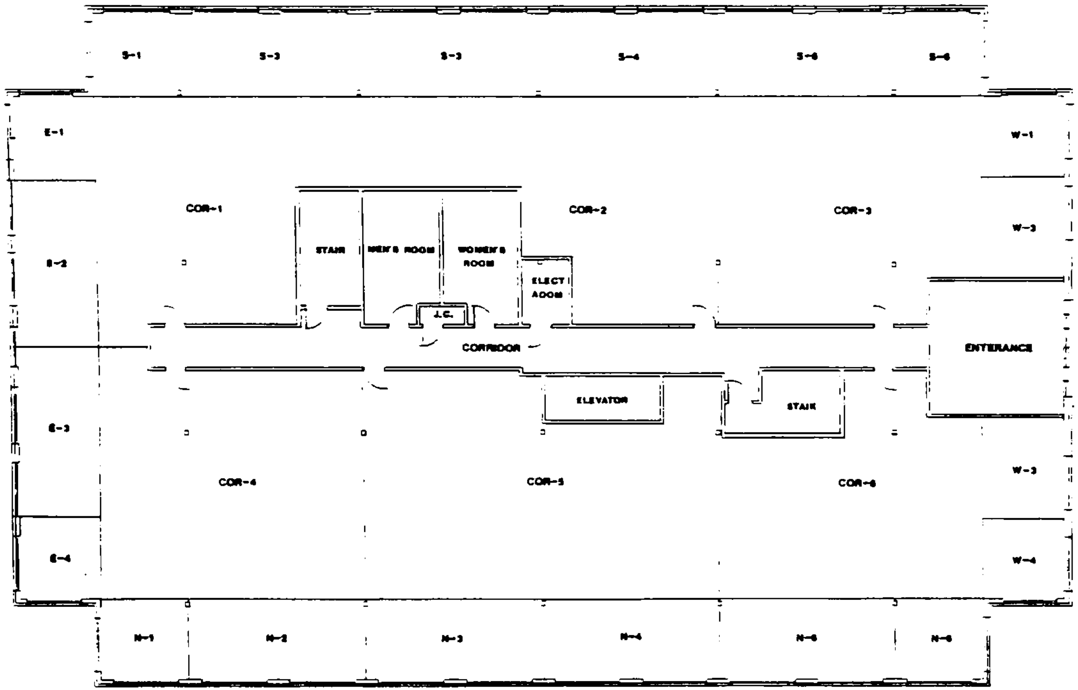
ARCHITECTURAL

PROJECT NUMBER

DRAWING NUMBER

A-2

BY: KURT KUEGLER



SECOND FLOOR
SCALE: 1/8" = 1'-0"

NORTH

HEATING, VENTILATION AND
AIR CONDITIONING

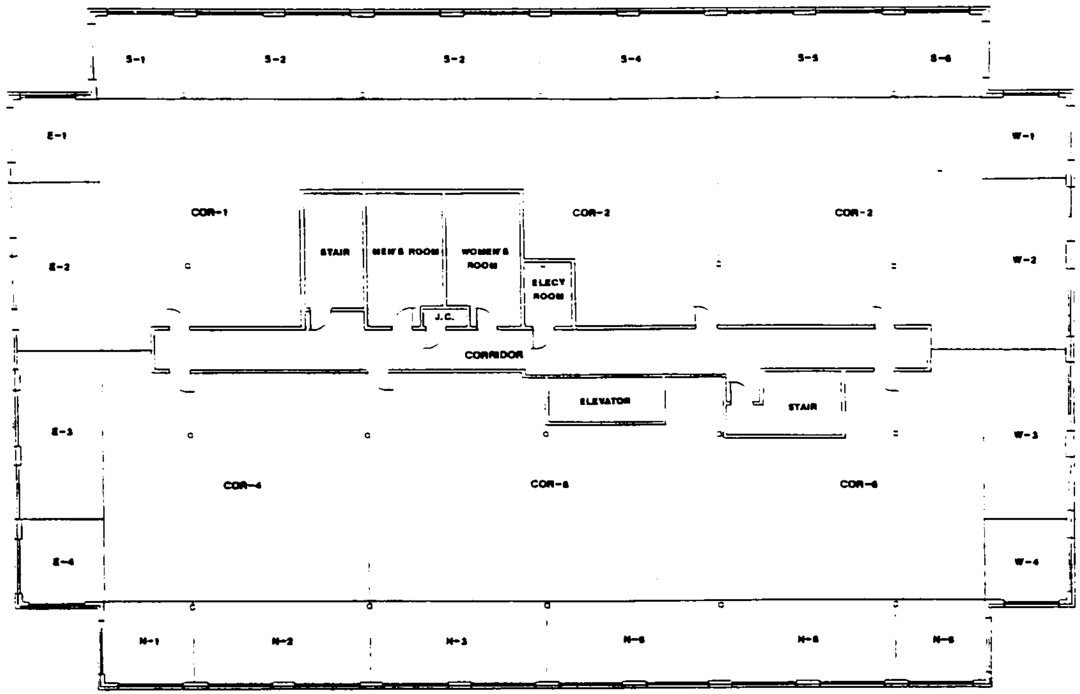
ARCHITECTURAL

PROJECT NUMBER

DRAWING NUMBER

A-3

BY: KURT KUEGLER



THIRD FLOOR
SCALE: 1/8" = 1'-0"

NORTH

HEATING, VENTILATION AND AIR CONDITIONING

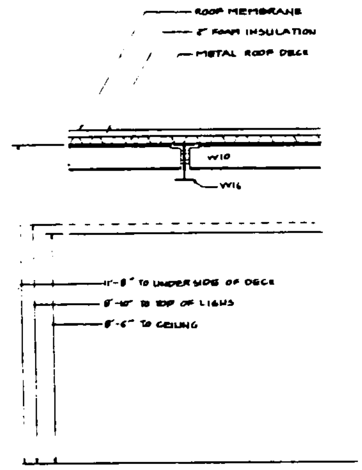
ARCHITECTURAL

PROJECT NUMBER

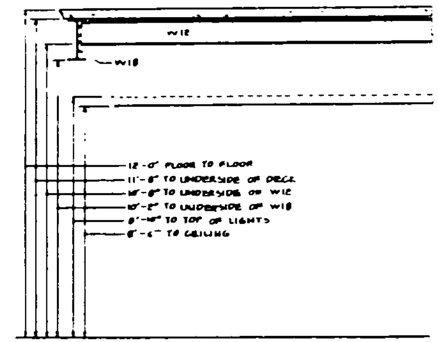
DRAWING NUMBER

A-4

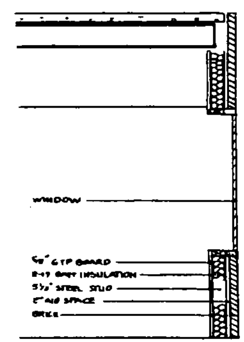
BY: KURT KUEGLER



ROOF SECTION



CEILING SECTION



WALL SECTION

HEATING, VENTILATION AND AIR CONDITIONING

ARCHITECTUAL

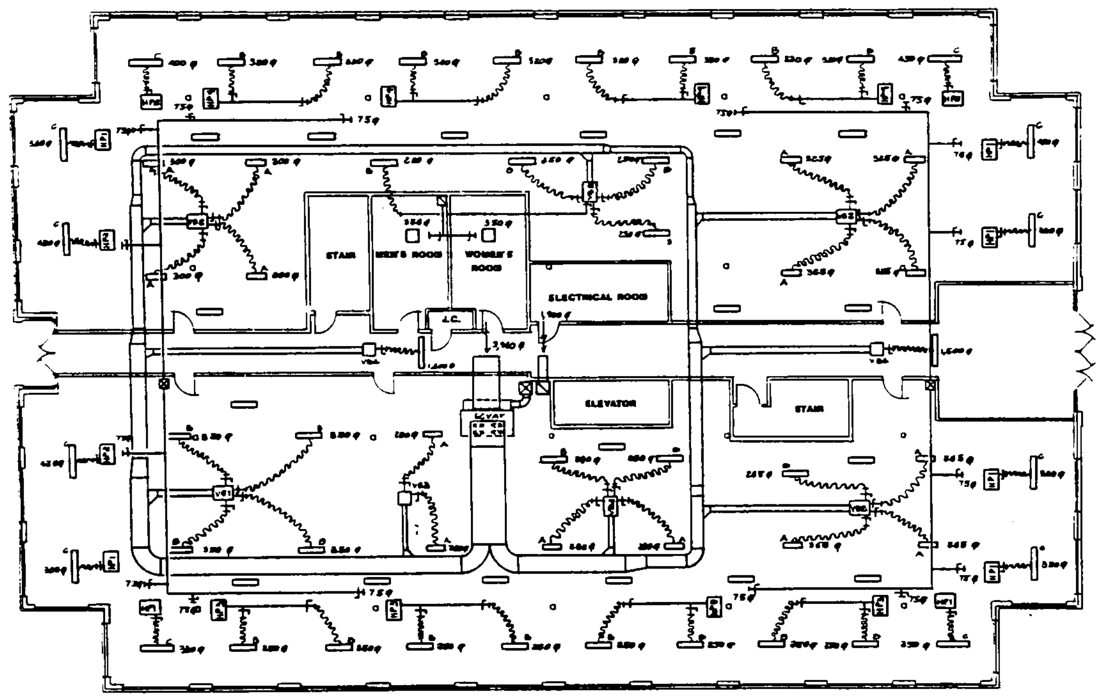
PROJECT NUMBER

DRAWING NUMBER

A–5

BY: KURT KUEGLER

21.2 Mechanical



FIRST FLOOR
SCALE: 1/8" = 1'-0"

NORTH

HEATING, VENTILATION AND AIR CONDITIONING

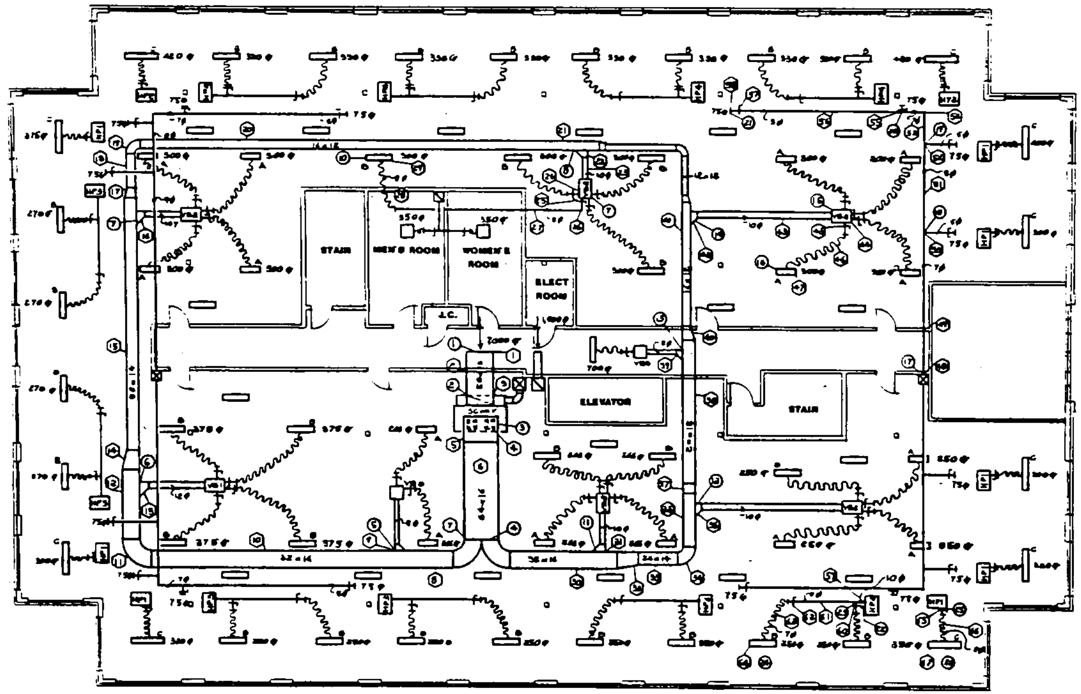
MECHANICAL

PROJECT NUMBER

DRAWING NUMBER

M–1

ST: KURT KUEGLER



SECOND FLOOR
SCALE: 1/8" = 1'-0"

NORTH

HEATING, VENTILATION AND AIR CONDITIONING

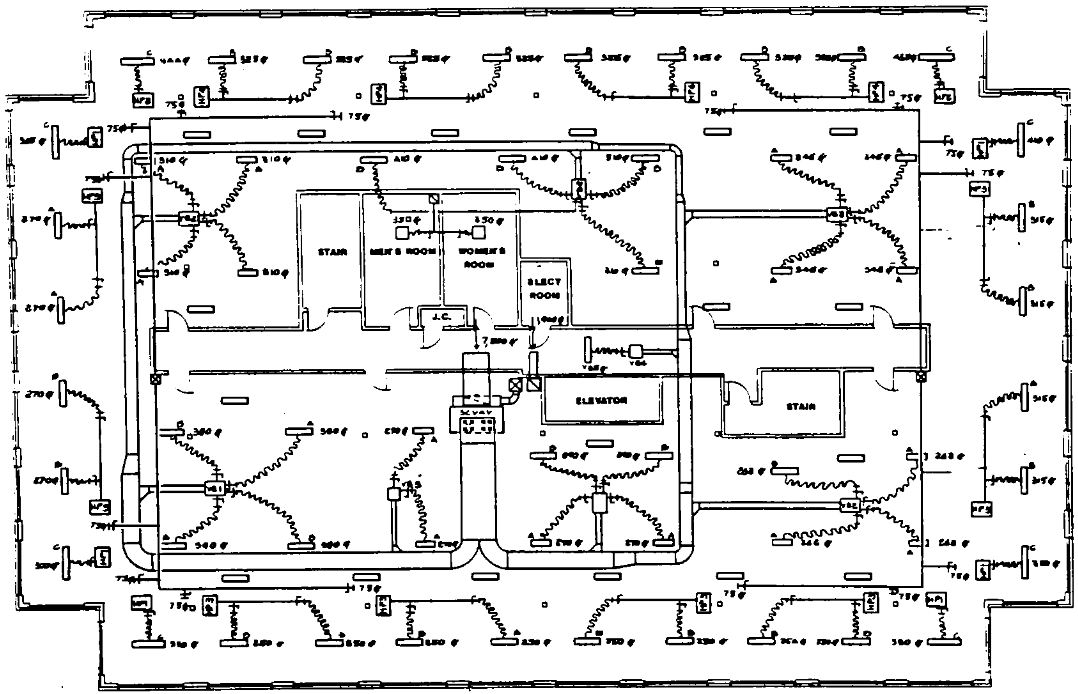
MECHANICAL

PROJECT NUMBER

DRAWING NUMBER

M–2

BY: BURT BUEGLER



THIRD FLOOR
SCALE: 1/8" = 1'-0"

NORTH

HEATING, VENTILATION AND
AIR CONDITIONING

MECHANICAL

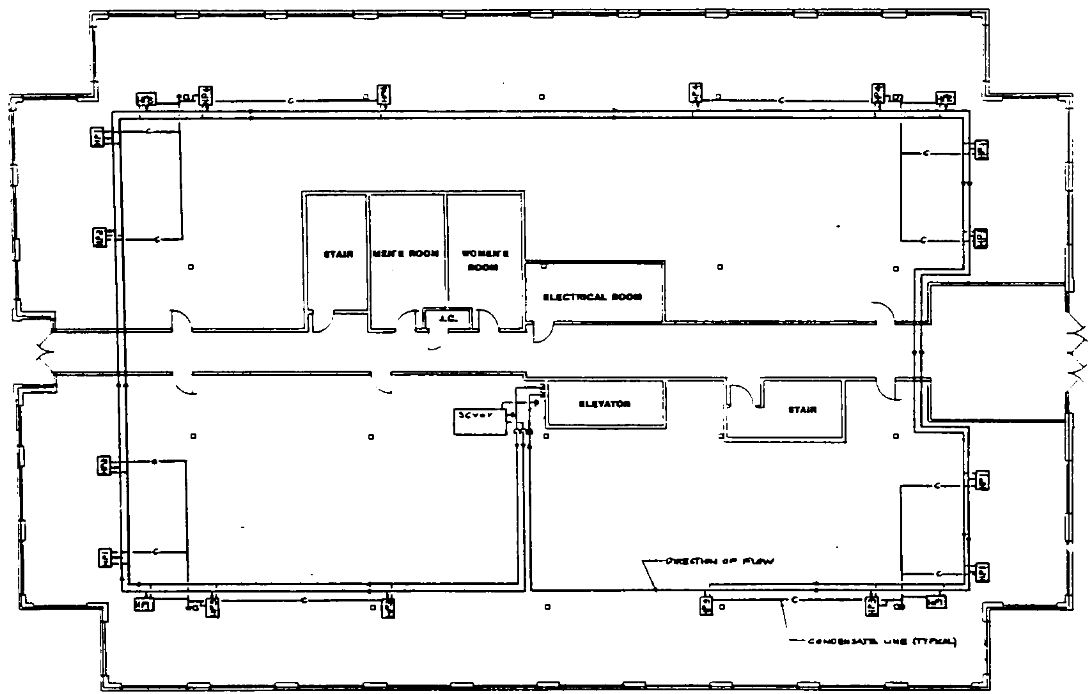
PROJECT NUMBER

DRAWING NUMBER

M-3

BY: KURT KUEGLER

21.3 Piping



FIRST FLOOR
SCALE: 1/8" = 1'-0"

NORTH

HEATING, VENTILATION AND AIR CONDITIONING

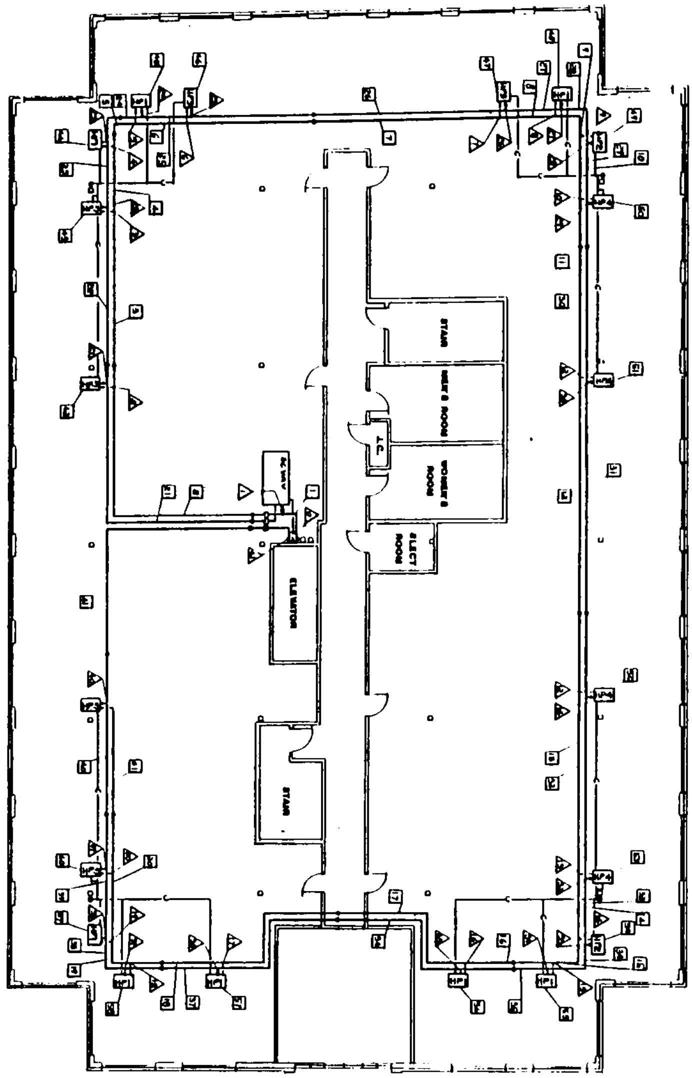
PIPING

PROJECT NUMBER

DRAWING NUMBER

P–1

BY: KURT KUEGLER



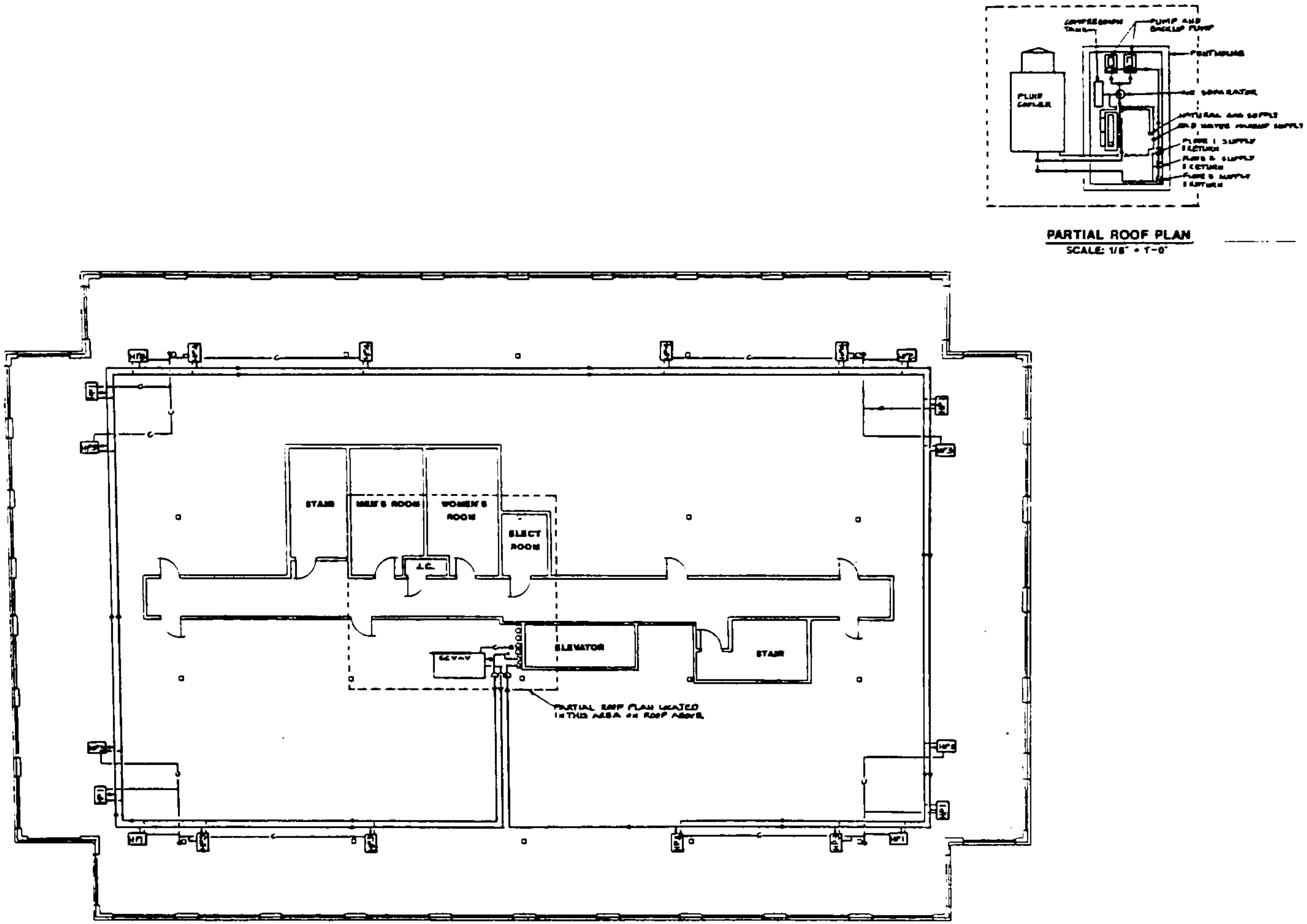
SECOND FLOOR
SCALE: 1/8" = 1'-0"

HEATING, VENTILATION AND
AIR CONDITIONING

PIPING

NORTH

P–2



PARTIAL ROOF PLAN
SCALE: 1/8" = 1'-0"

THIRD FLOOR
SCALE: 1/8" = 1'-0"

NORTH

HEATING, VENTILATION AND AIR CONDITIONING

PIPING

PROJECT NUMBER

DRAWING NUMBER

P–3

BY: KURT KUEGLER