PDHonline Course M216 (4 PDH)

HVAC Refresher - Facilities Standard for the Building Services (Part 2)

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2020

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Course Content

SECTION # 1: RULES OF THUMB

AIR-CONDITIONING CAPACITY

1) A ton of refrigeration (1TR) signifies the ability of air-conditioning equipment to extract heat @ 12000 Btu/hr. ASHARE (American Society of Heating, Refrigeration and Air-conditioning Engineers, Inc) has put together a table using national average data showing the Sq-ft/Ton as follows:

Sq-ft/Ton	High	Average	Low
Residential	600	500	380
Office	360	280	190

3) Each building is different and the design conditions differ greatly between regions to region. Factors to consider when figuring the sq-ft/ ton ratio include:

- Climate conditions (design temperatures)
- Expansive use of glass-particularly in the south and west orientations
- High ceilings-increasing the conditioned volume of the space
- Outside air requirements-especially important in high occupant load areas like conference rooms and classrooms.
- Heat generating equipment – example computers, copiers, laser printers, big screen TV's etc.
- Lighting-especially the extensive use of incandescent and metal halide lights. Fluorescent lights are more efficient and burn cooler-however; their ballasts generate a fair amount of heat.

Application	Average Load
Residence Apartment (1 or 2 room) Church	400-600 sq. ft. floor area per ton 400 sq. ft. of floor area per ton 20 people per ton
Office Building Large Interior Large Exterior Small Suite	 340 sq. ft. of floor area per ton 250 sq. ft. of floor area per ton 280 sq. ft. of floor area per ton

Application	Average Load
Restaurant Bar or Tavern Cocktail Lounge	200 sq. ft. of floor area per ton 9 people per ton 175 sq. ft. of floor area per ton
Computer Room Bank (main area) Barber Shop	50 – 150 sq. ft. of floor area per ton 225 sq. ft. of floor area per ton 250 sq. ft. of floor area per ton
Beauty Shop School Classroom Bowling Alley	180 sq. ft. of floor area per ton 250 sq. ft. of floor area per ton 1.5 – 2.5 tons per alley
Department Store Basement Main Floor Upper Floor Small Shop	 350 sq. ft. of floor area per ton 300 sq. ft. of floor area per ton 400 sq. ft. of floor area per ton 225sq. ft. of floor area per ton
Dress Shop Drug Store Factory (precision manufacturing)	280 sq. ft. of floor area per ton 150 sq. ft. of floor area per ton 275 sq. ft. of floor area per ton
Groceries – Supermarket Hospital Room Hotel Public Spaces	350 sq. ft. of floor area per ton 280 sq. ft. of floor area per ton 220sq. ft. of floor area per ton
Motel Auditorium or Theatre	400 sq. ft. of floor area per ton 20 people per ton
Shoe Store Specialty & Variety Store	220 sq. ft. of floor area per ton 200 sq. ft. of floor area per ton

In general air-conditioning requirements are higher (200 to 400 sq-ft/Ton) for hot & humid regions and lower (150 – 200 sq-ft/Ton) for cooler places.

Note: The figures above indicative only. It is recommended to always generate a detailed heating and cooling load calculation (such as using Manual J) for the building or space in question.

AIR CONDITIONER CAPACITY RANGES

The application and unit capacity ranges are as follows:

1) Room air conditioner - Capacity ranges 0.5 to 2 TR per unit, suitable for an area of not more than 1000 square feet
2) Packaged unit integral air-cooled condenser - Capacity ranges 3 to 50 TR, suitable for a maximum an area of 1000 – 10000 square feet
3) Split system with outdoor air-cooled condenser - Capacity ranges 0.5 to 50 TR, suitable for an area of 100 – 10000 square feet

4) Central air-conditioning chilled water system with air cooled condensers – Capacity ranges of 20 to 400 TR, suitable for an area of 4000 sq-ft and higher

5) Central air-conditioning chilled water systems with outdoor water cooled condenser - Capacity ranges 20 to 2000 TR, suitable for an area of 4000 sq-ft and higher.

COOLING CAPACITY SELECTER FOR HOMES

Air conditioners are sized by cooling capacity in BTU's per hour. To estimate the optimum capacity for any room, first calculate the size of the area to be conditioned by multiplying its width times its length, measured in feet. Then select the cooling capacity needed using the table below, The BTU's associated with the square footage will give an approximate optimum for the space.

Room Area	Square Feet	Cooling Capacity (BTU range)
10X15	150	up to 5200
10X20	200	6000
15X20	300	7500
17X20	340	8000
18X25	450	10000
22X25	550	12000
25X28	700	14000
25X32	800	15000
25X34	850	16000
25X40	1000	18000
27.5X40	1100	20000
35X40	1400	24000
37.5X40	1500	28000
40X40	1600	32000

Notes to using the table above

Cooling capacities are based on rooms occupied by two people and having average insulation, number of windows, and sun exposure.

To adapt the table for varying conditions, modify the capacity figures as follows:

Reduce capacity by 10% if area is heavily shaded.

Increase capacity by 10% for very sunny areas.

Add 600 Btu/hr for each additional person if area is occupied routinely by more than two people.

Add 4000 Btu/hr if area to be cooled is an average size kitchen.

Add 1000 Btu/hr for every 15 sq/ft of glass exposed to sun.

Add 3414 Btu/hr for every 1000 watts of electronic equipment.

SUPPLY AIR REQUIREMENTS (MECHANICAL COOLING & HEATING)

Equipment Type	Approximate Airflow Rate	Example
Gas/Oil Furnace	1 CFM per 100 Btu/hr output	64000 Btu/hr output furnace = 640CFM
Electric Furnace	50 – 70 CFM per kW input	10kW furnace = 10 x 70 = 700CFM 30kW furnace = 30 x 50 = 1500CFM
Electric Air-conditioning	400 CFM per ton	30000 Btu/hr cooling 30000/12000= 2.5tons 2.5 x 400 = 1000 CFM
Heat Pump	450 CFM per ton	30000 Btu/hr cooling 30000/12000= 2.5tons 2.5 x 450 = 1125 CFM

Note the values vary significantly with the equipment. CFM/kW tends to be higher with smallest equipment (5-15kW) and lower as equipment becomes larger.

In general, the following guidelines may be noted:

- 500 CFM/ton for Precision Air Conditioning
- 400 CFM/ton for Comfort Cooling Air Conditioning
- 200 CFM/ton Dehumidification

SELECTION OF CHILLERS

The following is used as a guide for determining the types of liquid chillers generally used for air conditioning

- Up to 25 tons (88kW) – Reciprocating
- 25 to 80 tons (88 to 280kW) – Reciprocating or Screw
- 80 to 200 tons (280 to 700kW) – Reciprocating, Screw or Centrifugal
- 200 to 800 tons (700 to 2800kW) – Screw or Centrifugal
- Above 800 tons (2800 kW) – Centrifugal

Circumstances Favouring Air-Cooled or Water Cooled Systems

Capacity Range (TR)	Favourable System
40 to 200	Air-cooled chilled water system (explore the pros and cons of using multiple DX systems if possible)
200 and above	Water-cooled chilled water system

CHARACTERISTICS & TYPICAL APPLICATIONS OF VARIOUS COOLING SYSTEMS

Characteristics	Air-Cooled Packaged Equipment	Water-Cooled Packaged Equipment	Air-Cooled Chilled-Water System	Water-Cooled Chilled-Water System
Building Height	Typically limited to 1- to 4-story buildings	Unlimited	Unlimited	Unlimited
Minimum Cooling Capacity	No limitation for modular systems	Typically cost-effective for projects >20 tons	Typically cost-effective for projects >100 tons	Typically cost-effective for projects >200 tons
Cooling Control	Low	Low-moderate	High	High
Maintenance	Low	Moderate-high	Moderate	High
Installed Cost	Low	Moderate-high	High	High
Operating Costs (energy and water)	Moderate	Low-moderate (climate dependent)	Moderate-high	Low
Typical Applications	1- to 2-story buildings	1- to 2-story buildings in hot/dry climates	Medium to large facilities with limited access to water or maintenance	Medium to very large facilities and campuses

CONVERTING KW/TON TO COP or EER

If a chiller's efficiency is rated at 1 KW/ton, the COP=3.5 and the EER=12

kW/ton	=	12 / EER
kW/ton	=	12 /(COP x 3.412)
EER	=	12 / (kW/ton)
EER	=	COP x 3.412
COP	=	EER / 3.412
COP	=	12 / (kW/ton x 3.412)

RECOMMENDED EFFICIENCY VALUES FOR UNITARY & APPLIED HEAT PUMPS

Equipment Type	Size Category	Sub-Category or Rating Condition	Required Efficiency
Air Cooled	< 65,000 Btuh	Split System	13.0 SEER

Equipment Type	Size Category	Sub-Category or Rating Condition	Required Efficiency
(Cooling Mode)		Single Package	13.0 SEER
	≥ 65,000 Btuh and < 135,000 Btuh	Split System and Single Package	11.0 EER 11.4 IPLV
	≥ 135,000 Btuh and <240,000 Btuh	Split System and Single Package	10.8 EER 11.2 IPLV
	≥ 240,000 Btuh	Split System and Single Package	10.0 EER 10.4 IPLV
Air Cooled (Heating Mode)	< 65,000 Btuh (Cooling Capacity)	Split System	8.0 HSPF
		Single Package	7.7 HSPF
	≥ 65,000 Btuh and < 135,000 Btuh (Cooling Capacity)	47°F db/43°F wb Outdoor Air	3.4 COP
		17°F db/15°F wb Outdoor Air	2.4 COP
	>135,000 Btuh (Cooling Capacity)	47°F db/43°F wb Outdoor Air	3.3 COP
		17°F db/15°F wb Outdoor Air	2.2 COP
Water Source (Cooling Mode)	< 135,000 Btuh (Cooling Capacity)	85°F Entering Water	14.0 EER
Water-Source (Heating Mode)	< 135,000 Btuh (Cooling Capacity)	70°F Entering Water	4.6 COP

RECOMMENDED CHILLER PERFORMANCE LEVELS

Equipment Type	Size Category	Required Efficiency- chillers without ASDs		Required Efficiency- Chillers with ASDs		ASHRAE 90.1-2001 (kW/ton)	
		Full Load (kW/ton)	IPLV (kW/ton)	Full Load (kW/ton)	IPLV (kW/ton)	Full Load	IPLV
Air cooled w/ condenser	All	1.2	1	N/A	N/A	1.26	1.15
Air cooled w/o condenser	All	1.08	1.08	N/A	N/A	1.13	1.02
Water cooled, reciprocating	All	0.84	0.63	N/A	N/A	0.84	0.70
Water cooled, rotary screw, and scroll	< 100 tons	0.78	0.6	N/A	N/A	0.72	0.63
	³100 tons and < 150 tons	0.73	0.55	N/A	N/A	0.72	0.63
	³150 tons and ≤ 300 tons	0.61	0.51	N/A	N/A		
	> 300 tons	0.6	0.49	N/A	N/A	0.64	0.57
Water cooled, centrifugal	< 150 tons	0.61	0.62	0.63	0.4	0.70	0.67
	³150 tons and ≤ 300 tons	0.59	0.56	0.6	0.4	0.63	0.60
	> 300 tons and ≤ 600 tons	0.57	0.51	0.58	0.4	0.58	0.55
	> 600 tons	0.55	0.51	0.55	0.4		

ELECTRIC UTILIZATION INDEX (EUI)

Electric utilization index (EUI) is the ratio of annual electricity consumption in kWh to the facility's square footage.

Type of Building	Common EUI
Grocery	61.0
Restaurant	38.9
Hospital / Health	16.4
Retail	12.1
School / College	10.3
Hotel / Motel	8.2
Office	7.5
Misc. Commercial	6.4
Warehouse	6.1

HEAT GAIN FROM OCCUPANTS AT VARIOUS ACTIVITIES (At Indoor Air Temperature of 78°F)

Activity	Total heat, Btu/h		Sensible heat, Btu/h	Latent heat, Btu/h
	Adult, male	Adjusted		
Seated at rest	400	350	210	140
Seated, very light work, writing	480	420	230	190
Seated, eating	520	580	255	325
Seated, light work, typing	640	510	255	255
Standing, light work or walking slowly	800	640	315	325
Light bench work	880	780	345	435
Light machine work, walking 3miles/hr	1040	1040	345	695
Moderate dancing	1360	1280	405	875
Heavy work, lifting	1600	1600	565	1035
Athletics	2000	1800	635	1165

The values are for 78°F room dry bulb temperature. For 80°F dry bulb temperature, the total heat remains the same, but the sensible heat value should be decreased by approximately 8% and the latent heat values increased accordingly.

VENTILATION RECOMMENDATIONS

Application		*Occupancy (people/1000ft^2)*	*CFM/person*	*CFM/ft^2*
Food and Beverage Service	Dining rooms	70	20	
	Cafeteria, fast food	100	20	
	Bars, cocktail lounges	100	30	
	Kitchen (cooking)	20	15	
Offices	Office space	7	20	
	Reception areas	60	15	
	Conference rooms	50	20	
Public Spaces	Smoking lounge	70	60	
	Elevators			1.00
Retail stores, Showrooms	Basement & Street	30		0.30
	Upper floors	20		0.20
	Malls and arcades	20		0.20
	Smoking lounges	70	60	
	Beauty shops	25	25	
	Hardware stores	8	15	
Sports and Amusements	Spectator areas	150	15	
	Games rooms	70	25	
	Playing rooms	30	20	
	Ballrooms and discos	100	25	

Application		Occupancy (people/1000ft^2)	CFM/person	CFM/ft^2
Theatres	Lobbies	150	20	
	Auditorium	150	15	
Education	Classroom	50	15	
	Music rooms	50	15	
	Libraries	20	15	
	Auditoriums	150	15	
Hotels, Motels Resorts, Dormitories	Bedrooms			30CFM/room
	Living rooms			30CFM/room
	Lobbies	30	15	
	Conference rooms	50	20	
	Assembly rooms	120	15	
	Dry cleaning, laundry	30	30	
	Gambling casinos	120	30	
Health Care Facilities	Operating rooms	20	30	
	Patient rooms	10	25	
	Laboratories	30	20	
	Procedure rooms	20	15	
	Pharmacies	20	15	
	Physical therapy	20	15	

EXHAUST AIR REQUIREMENTS

Exhaust Air Requirements	
Janitor Closets	10 Air changes/hr
Locker Rooms	10 Air changes/hr
Toilets	10 Air changes/hr
Mechanical/Electrical Rooms	12 Air changes/hr
Rooms with Steam System (Laundry)	25 Air changes/hr
Battery Rooms	10 Air changes/hr

TYPICAL DESIGN VELOCITIES FOR HVAC COMPONENTS

Equipment	Velocity, Feet per minute (FPM)
Intake Louvers Velocity (7000 CFM and greater)	400 FPM

Equipment	Velocity, Feet per minute (FPM)
Exhaust Louvers (5000 CFM and greater)	500 FPM
Panel Filters • Viscous Impingement	 200 to 800 FPM
Panel Filters (Dry-Type, Pleated Media): • Low Efficiency • Medium Efficiency • High Efficiency • HEPA	 350 FPM 500 FPM 500 FPM 250 FPM
Renewable Media Filters • Moving-Curtain Viscous Impingement • Moving-Curtain Dry-Media	 500 FPM 200 FPM
Electronic Air Cleaners • Ionizing-Plate-Type • Charged-Media Non-ionizing • Charged-Media Ionizing	 300 to 500 FPM 250 FPM 150 to 350 FPM
Steam and Hot Water Coils	200 min - 1500 max
Electric Coils • Open Wire • Finned Tubular	Refer to Mfg. Data
Dehumidifying Coils	500 FPM
Spray-Type Air Washers	300 to 600 FPM
Cell-Type Air Washers	Refer to Mfg. Data
High-Velocity, Spray-Type Air Washers	1200 to 1800 FPM

CENTRIFUGAL FAN PARAMETERS

Centrifugal fans are by far the most prevalent type of fan used in the HVAC industry today. They are usually cheaper than axial fans and simpler in construction, but generally do not achieve the same efficiency. Centrifugal fans consist of a rotating wheel, or "impeller," mounted eccentrically inside a round housing. The impeller is electrically driven by a motor connected via a belt drive.

Parameters	Backward Curve			Forward Curve
	BC	BI	AF	FC
Blades	6-16	6-16	6-16	24-64
Maximum Efficiency (%)	78	85	90	70
Speed	High	High	High	Low

Parameters	Backward Curve			Forward Curve
	BC	BI	AF	FC
Cost	Medium	Medium	High	Med-Low
Static Pressure	Very high	High	Very high (40in-wg)	Low (5 inch- w.g)
Power Curve	Non-overloading	Non-overloading	Non-overloading	Overloading
Housing	Scroll	Scroll	Scroll	Scroll

AXIAL FAN PARAMETERS

Axial fans consist of a cylindrical housing, with the impeller mounted inside along the axis of the housing. In an axial fan, the impeller consists of blades mounted around a central hub similar to those of an airplane propeller. Typically, axial fans are more efficient than centrifugal fans.

Parameters	Propellers	Tube Axial	Vane axial
Blades	2 to 8	4 to 8	5 to 20
Maximum Efficiency (%)	50	75	85
Speed	Medium	High	Very high
Cost	Low	Medium	High
Static Pressure	Low (up to ¾ in)	Medium	High (up to 8 in)
Power Curve	Non-overloading	Non-overloading	Non- overloading
Housing	Annular ring	Cylindrical	Cylindrical with guide vanes on downstream side

FAN PERFORMANCE RELATIONSHIPS

Variable	Constant	Law	Equation
Rotational Speed	Fan Size Air Density Duct System	Flow is directly proportional to speed	$(Q_1 / Q_2) = (N_1 / N_2)$
		Pressure is directly proportional to speed^2	$(P_1 / P_2) = [(N_1 / N_2)]^2$
		Power is directly proportional to speed^3	$(HP_1 / HP_2) = [(N_1 / N_2)]^3$
Fan Size and Rotational Speed	Tip Speed	Flow and power is directly proportional to diameter^2	$(Q_1 / Q_2) = (HP_1 / HP_2) = [(D_1 / D_2)]^2$

Variable	Constant	Law	Equation
	Air Density	Speed is inversely proportional to diameter	$(N_1 / N_2) = (D_2 / D_1)$
		Pressure remains constant	$P_1 = P_2$
Fan Size	Rotational Speed Air Density	Flow is directly proportional to Diameter^2	$(Q_1 / Q_2) = [(D_1 / D_2)]^2$
		Flow is directly proportional to Diameter^2	$(P_1 / P_2) = [(D_1 / D_2)]^2$
		Power is directly proportional to Diameter^3	$(HP_1 / HP_2) = [(D_1 / D_2)]^3$
Rotational Speed and Air Density	Fan Size Pressure	Speed, flow and power are inversely proportional to square root of density	$(N_1 / N_2) = (Q_1 / Q_2) =$ $(HP_1 / HP_2) = [(\rho_1 / \rho_2)]^{1/2}$
Air Density	Rotational Speed Fan Size Duct System	Pressure and power are directly proportional to density	$(P_1 / P_2) = (HP_1 / HP_2) =$ $(\rho_1 / \rho_2) =$
		Flow remains constant	$Q_1 = Q_2$

GUIDE TO AIR OUTLET SELECTION

Tables below provide a general guide for the proper selection of outlets based on design requirements of CFM per square foot and air changes per hour (SMACNA 1990).

Type of Outlet	Floor Space		Approximate maximum air changes/hour for 10 feet ceiling
	CFM per Sq Feet	Lps per Sq-m	
Grilles & Registers	0.6 to 1.2	3 to 6	7
Slot Diffusers	0.8 to 2.0	4 to 10	12
Perforated Panel	0.9 to 3.0	5 to 15	18
Ceiling Diffuser	0.9 to 5.0	5 to 25	30
Perforated Ceiling	1.0 to 10.0	5 to 50	60

REFRIGERANTS & ENVIRONMENTAL FACTORS

In general the comparison of 4 most common refrigerants employed today on environmental factors is as below:

Criteria	HCFC-123	HCFC-22	HFC-134a	Ammonia
Ozone Depletion Potential	0.016	0.05	0	0

Criteria	HCFC-123	HCFC-22	HFC-134a	Ammonia
Global Warming Potential (relative to CO_2)	85	1500	1200	0
Phase out Date	2030	2020	N/A	N/A
Occupation Risk	Low	Low	Low	Low
Flammable	No	No	No	Yes

CURRENT & FUTURE REFRIGERANTS

Equipment Type	Traditional Refrigerant	Replacement Refrigerants
Rotary Screw- Chiller	HCFC-22	R407C, HFC-134a
Scroll Chiller	HCFC-22	R407C, R-410A
Reciprocating Chiller	HCFC-22	R-407C, R-410A
Absorption Chiller	R-718 (water)	R-718
Centrifugal Chiller	CFC-11, CFC-12	HFC-134a, HCFC-123
Packaged Air Conditioners	HCFC-22	R-407C, R-410A
Heat Pump	HCFC-22	R407C, R-410A
PTAC, PTHP	HCFC-22	R-407C, R-410A
Room Air Conditioning	HCFC-22	R-407C, R-410A

RECOMMENDED SHEET METAL THICKNESS FOR DUCTS

Rectangular Duct			Round Duct		
Greatest Dimension	Galvanized Steel (gauge)	Aluminum (gauge)	Diameter	Galvanized Steel (gauge)	Aluminum (gauge)
Up to 30 inch	24	22	Up to 8 inch	24	22
31 – 60 inches	22	20	9 – 24 inches	22	20
61 – 90 inches	20	18	25 – 48 inches	20	18
91inches and above	18	16	49 – 72 inches	18	16

SHEET METAL THICKNESS & WEIGHTS

Gauge No.	Steel (Manuf. Std. Ga.)		Galvanized (Manuf. Std. Ga.)	
	Thick. in.	Lb./ft.^{2}	Thick.in.	Lb./ft.^{2}
26	.0179	.750	.0217	.906
24	.0239	1.00	.0276	1.156
22	.0299	1.25	.0336	1.406
20	.0359	1.50	.0396	1.656
18	.0478	2.00	.0516	2.156
16	.0598	2.50	.0635	2.656
14	.0747	3.125	.0785	3.281
12	.1046	4.375	.1084	4.531
10	.1345	5.625	.1382	5.781
8	.1644	6.875	.1681	7.031
7	.1793	7.50	—	—

Gauge No.	Mill Std. Thick Aluminum*		Stainless Steel (U.S. Standard Gauge)	
	Thick. in.	Lb./ft.^{2}	Thick.in.	Lb./ft.^{2}
26	.020	.282	.0188	.7875
24	.025	.353	.0250	1.050
22	.032	.452	.0312	1.313
20	.040	.564	.0375	1.575
18	.050	.706	.050	2.100
16	.064	.889	.062	2.625
14	.080	1.13	.078	3.281
12	.100	1.41	.109	4.594
10	.125	1.76	.141	5.906
8	.160	2.26	.172	7.218
7	.190	2.68	.188	7.752

Note: Aluminium is specified and purchased by material thickness rather than gauge.

DUCTWORK AIR CARRYING CAPACITY

Branch Duct Size	Avg. CFM @ Static Pressure	Duct Cross-section
4" Round	30 CFM	12.57 Sq-in
5" Round	60 CFM	19.64 Sq-in
2 ¼" x 10"	60 CFM	23.00 Sq-in
2 ¼" x 12"	70 CFM	27.00 Sq-in
6" Round	100 CFM	28.27 Sq-in
3 ¼" x 10"	100 CFM	33.00 Sq-in
3 ¼" x 12"	120 CFM	39.00 Sq-in
7" Round	150 CFM	38.48 Sq-in

3 ¼" x 14"	140 CFM	46.00 Sq-in
8" Round	200 CFM	50.27 Sq-in
8" x 8"	260 CFM	64.00 Sq-in
10" Round	400 CFM	78.54 Sq-in
12 " x 8"	440 CFM	96.00 Sq-in
12"	620 CFM	113.09 Sq-in
16" x 8"	660 CFM	128.00 Sq-in
14" Round	930 CFM	153.93 Sq-in
16" Round	1300 CFM	201.06 Sq-in

PIPE SELECTION

Steel Pipe				Copper Pipe		
Pipe Size	**Flow Rate**	**Heating BTUH**	**Cooling Tons**	**Flow Rate**	**Heating BTUH**	**Cooling Tons**
1/2"	1.8 GPM	18,000 BTUH	1.5 Tons	1.5 GPM	15,000 BTUH	1.3 Tons
3/4"	4 GPM	40,000 BTUH	3.3 Tons	3.5 GPM	35,000 BTUH	2.9 Tons
1"	8 GPM	80,000 BTUH	6.7 Tons	7.5 GPM	75,000 BTUH	6.3 Tons
1 1/4"	16 GPM	160,000 BTUH	13.3 Tons	13 GPM	130,000 BTUH	10.8 Tons
1 1/2"	24 GPM	240,000 BTUH	20 Tons	20 GPM	200,000 BTUH	16.7 Tons
2"	47 GPM	470,000 BTUH	39 Tons	45 GPM	450,000 BTUH	38 Tons
2 1/2"	75 GPM	750,000 BTUH	63 Tons	80 GPM	800,000 BTUH	67 Tons
3"	130 GPM	1,300,000 BTUH	108 Tons	130 GPM	1,300,000 BTUH	108 Tons
4"	270 GPM	2,700,000 BTUH	225 Tons	260 GPM	2,600,000 BTUH	217 Tons
5"	530 GPM	5,300,000 BTUH	442 Tons			

6"	850 GPM	8,500,000 BTUH	708 Tons			
	• Heating capacity BTUH based on a 20 degree F temperature differential. Cooling capacity BTUH based on 10 to 16ºF temperature differential. • Cooling capacity Tons based on a 10 degree F temperature differential • Selection guide for water systems • Pipe sized for a maximum of 4 feet/100 feet pressure drop • GPM = BTUH / 10,000 (for heating units designed for 20ºF) • Temperature differential = MBH / GPM / 500 • MBH = BTUH X 1,000 • Ton of cooling = 12,000 BTUH					

CLEANROOM DESIGN

Cleanroom airflow design conventionally follows the table below to decide on the airflow pattern, average velocities and air changes per hour. One has to first identify the level of cleanliness required and apply the table below. Please note that there is no scientific or statutory basis for this inference other than the explanation that the table is derived from experience over past two decades.

Clean room Class	Airflow Type	Av. Airflow Velocity, fpm	Air changes/hr
1	Unidirectional	70-100	350-650
10	Unidirectional	60-110	300-600
100	Unidirectional	50-90	300-480
1,000	Mixed	40-90	150-250
10,000	Mixed	25-40	60-120
100,000	Mixed	10-30	10-40

SOUND & ACOUSTICS

When trying to calculate the additive effect of two sound sources, use the approximation as below (note that the logarithms cannot be added directly).

Adding Equal Sound Pressure Levels

Number of Sources	Increase in Sound Power Level (dB)	Increase in Sound Pressure Level dB
2	3	6
3	4.8	9.6
4	6	12
5	7	14
10	10	20
15	11.8	23.6
20	13	26

Adding Sound Power from Sources at different Levels

Sound Power Level Difference between two Sound Sources (dB)	Added Decibel to the Highest Sound Power Level (dB)
0	3
1	2.5
2	2
3	2
4	1.5
5	1
6	1
7	1
8	0.5
9	0.5
10 or more	0

NOISE CRITERIA – OCCUPIED SPACES

Noise Criteria (NC) are the curves based on different dB levels at different octave bands. Highest curve intercepted is NC level of sound source. See table below

Occupied Spaces	
Area	**Maximum NC**
Conference Rooms	NC 35
Corridors	NC 40
Lobby	NC 40
Large Offices & Computer Rooms	NC 40
Small Private Office	NC 35

Notes:

- The above NC levels must be attained in all octave bands.
- The above NC levels may be increased for the areas equipped with fan coil units. The designer shall submit an analysis showing the expected noise levels for the prior approval of VA.
- The systems must be engineered and the use of acoustic sound lining and sound attenuators should be considered to achieve the design sound levels.

AVERAGE HEAT CONTENT (BTU) OF FUELS

Fuel Type	No. of Btu/Unit
#2 Fuel Oil	140,000/gallon
#6 Fuel Oil	150,500 /gallon
Diesel	137,750/gallon
Kerosene	134,000/gallon
Electricity	3,412/kWh
Natural Gas*	1,025,000/thousand cubic feet
Propane	91,330/gallon
Wood (air dried)*	20,000,000/cord or 8,000/pound
Pellets (for pellet stoves; premium)	16,500,000/ton
Kerosene	135,000/gallon
Coal	28,000,000/ton

GLAZING PROPERTIES

Material	"U" Value (Btu / hr-ft^2-°F)
Glass, single	1.13
Glass, double glazing	.70
Single film plastic	1.20
Double film plastic	.70
Corrugated FRP panels	1.20
Corrugated polycarbonate	1.20
Plastic structured sheet;	
16 mm thick	.58
8 mm thick	.65
6 mm thick	.72
Concrete block, 8 inch	.51

ROOF INSULATION

The following table provides some rules-of-thumb on the cost effectiveness of adding roof insulation to an existing building.

Existing Condition	Is it cost effective to add insulation?
No insulation to R-6	Yes, always
R-7 to R-19	Yes, if attic is accessible or if built-up roof is replaced
Greater than R-19	Not usually cost effective

ENERGY STAR BUILDING LABEL

The U.S. Environmental Protection Agency (EPA) and the U.S. Department of Energy (DOE) joined forces in establishing the Energy Star Building Label, a voluntary, performance based, benchmarking and recognition initiative. In February 1998, DOE published Energy Star target performance levels for thermal transmittance and solar heat gain factors for windows, doors and skylights.

Region	Item	Energy Star
North (Mostly Heating)	Windows and Doors U factor / SHGC	0.35 / -
	Skylights, U factor / SHGC	0.45 / -

Region	Item	Energy Star
Central (Heating and Cooling)	Windows and Doors U factor / SHGC	0.40 / 0.55
	Skylights, U factor / SHGC	0.45 / 0.55
South (Mostly Cooling)	Windows and Doors U factor / SHGC	0.75 / 0.40
	Skylights, U factor / SHGC	0.75 / 0.40

LIGHTING WATTAGE ESTIMATION

Location	Rule of thumb (Watts/sq-ft)
General Office Areas	1.5 to 3.0
Private	2.0 -5.0
Conference Rooms	2.0 – 6.0
Public Places (Banks, Post offices, Courts etc)	2.0 – 5.0
Precision Manufacturing	3.0 – 10.0
Computer Rooms/Data Processing Facilities	2.0 – 5.0
Restaurants	1.5 – 3.0
Kitchens	1.5 – 2.5
Pubs, Bars, Clubhouses, Taverns etc	1.5 – 2.0
Hospital Patient Rooms	1.0 – 2.0
Hospital General Areas	1.5 – 2.5
Medical /Dental Centres, Clinics	1.5 – 2.5
Residences	1.0 – 4.0
Hotel & Motels (public places and guest rooms)	1.0 – 3.0
School Classrooms	2.0 – 6.0
Dining halls, Lunch Rooms, Cafeterias	1.5 – 2.5
Library, Museums	1.0 – 3.0

Location	Rule of thumb (Watts/sq-ft)
Retail, Department & Pharmacist Stores	2.0 – 6.0
Jewellery Showrooms, Shoes, Boutiques etc	2.0 – 4.0
Shopping Malls	2.0 – 4.0
Auditoriums, Theatres	1.0 – 3.0
Religious Places (Churches)	1.0 – 3.0
Bowling Alleys	1.0 – 2.5

HEAT LOAD FROM OFFICE EQUIPMENT

Heat Gain Rates for Equipment - Watts (1 Watt = 3.41 Btu/h)

	Continuous	Average	Idle
Computer – 15" Monitor	110	-	20
- 17" Monitor	125	-	25
- 19" Monitor	135	-	30
Laser Printer - Desktop	130	100	10
- Small Office	320	160	70
- Large Office	550	275	125
Fax Machine		30	
Other Office Equipment		25% Nameplate (Watts ≈ volts×amps)	
Coffee Maker – 10 cup		1050 + 1540 Btu/h Latent	
Microwave Oven – 1 ft^3		400	
Refrigerator – 15 ft^3		300	
Water Cooler – 8 gal/hr		350	
¼ hp Motor	270		
¾ hp Motor	750		
1 hp motor	930		
10 hp Motor	8500		
($Watts = 746 \times hp/\eta_{Motor}$)			

RATE OF HEAT GAIN FROM MISCELLANEOUS APPLIANCES

Electrical Appliances	Manufacturer's Rating		Recommended Rate of Heat Gain, Btuh		
	Watts	Btuh	*Sensible	Latent	Total
Hair dryer	1,580	5,400	2,300	400	2,700
Hair dryer	705	2,400	1,870	330	2,200
Neon sign,			30		30
per linear ft of tube			60		60
Sterilizer, instrument	1,100	3,750	650	1,200	1,850
Gas-Burning Appliances					
Lab burners Bunsen		3,000	1,680	420	2,100
Fishtail		5,000	2,800	700	3,500
Meeker		6,000	3,360	840	4,200
Gas Light, per burner		2,000	1,800	200	2,000
Cigar lighter		2,500	900	100	1,000

SYNCHRONOUS SPEED BY NUMBER OF POLES

POLES	60 CYCLES	50 CYCLES
2	3600	3000
4	1800	1500
6	1200	1000
8	900	750
10	720	600

SECTION -2 USEFUL EQUATIONS

COOLING & HEATING EQUATIONS

Roofs, External Walls & Conduction through Glass

The equation used for sensible loads from the opaque elements such as walls, roof, partitions and the conduction through glass is:

*H = U * A * (CLTD)*

Where

- H describes Sensible heat flow (Btu/Hr)
- U = Thermal Transmittance for roof or wall or glass. See 1997 ASHRAE Fundamentals, Chapter 24 or 2001 ASHRAE Fundamentals, chapter 25. (Unit- Btu/Hr Sq-ft °F)
- A = area of roof, wall or glass calculated from building plans (sq-ft)
- CLTD = Cooling Load Temperature Difference (in °F) for roof, wall or glass. For winter months CLTD is (T_i - T_o) which is temperature difference between inside and outside. For summer cooling load, this temperature differential is affected by thermal mass, daily temperature range, orientation, tilt, month, day, hour, latitude, solar absorbance, wall facing direction and other variables and therefore adjusted CLTD values are used. Refer 1997 ASHRAE Fundamentals, Chapter 28, tables 30, 31, 32, 33 and 34.

Solar Load through Glass, Skylights and Plastic Sheets

Heat transfer through glazing is both conductive and transmission. It is calculated in two steps:

Step # 1

The equation used for sensible loads from the conduction through glass is:

*H = U * A * (CLTD)*

Where

- H = Sensible heat gain (Btu/Hr)
- U = Thermal Transmittance for roof or wall or glass. See 1997 ASHRAE Fundamentals, Chapter 24 or 2001 ASHRAE Fundamentals, chapter 25. (Unit- Btu/Hr Sq-ft °F)
- A = area of roof, wall or glass calculated from building plans (sq-ft)
- CLTD = Cooling Load Temperature Difference (in °F) for glass. Refer 1997 ASHRAE Fundamentals, Chapter 28, tables 30, 31, 32, 33 and 34.

Step # 2

The equation used for radiant sensible loads from the transparent/translucent elements such as window glass, skylights and plastic sheets is:

*H = A * (SHGC) * (SC) * (CLF)*

Where

- H = Sensible heat gain (Btu/Hr)
- A = area of roof, wall or glass calculated from building plans (sq-ft)
- SHGC = Solar Heat Gain Coefficient. See 1997 ASHRAE Fundamentals, Chapter 28, table 35
- CLF = Solar Cooling Load Factor. See 1997 ASHRAE Fundamentals, Chapter 28, table 36.

Partitions, Ceilings & Floors

The equation used for sensible loads from the partitions, ceilings and floors:

*H = U * A * (Ta - Tr)*

- H = Sensible heat gain (Btu/Hr)
- U = Thermal Transmittance for roof or wall or glass. See 1997 ASHRAE Fundamentals, Chapter 24 or 2001 ASHRAE Fundamentals, and Chapter 25. (Unit- Btu/Hr Sq-ft °F)
- A = area of partition, ceiling or floor calculated from building plans (sq-ft)
- Ta = Temperature of adjacent space in °F (Note: If adjacent space is not conditioned and temperature is not available, use outdoor air temperature less 5°F)
- Tr = Inside room design temperature of conditioned space in °F (assumed constant usually 75°F)

Ventilation & Infiltration Air

Ventilation air is the amount of outdoor air required to maintain Indoor Air Quality for the occupants (refer ASHRAE Standard 62 for minimum ventilation requirements) and makeup for air leaving the space due to equipment exhaust, exfiltration and pressurization.

$H_{sensible} = 1.08 * CFM * (T_o - T_c)$

$H_{latent} = 0.68 \times CFM \times \Delta W_{GR}$

$H_{latent} = 4840 \times CFM \times \Delta W_{Lb}$

$H_{total} = 4.5 * CFM * (h_o - h_c)$

$H_{total} = H_{sensible} + H_{latent}$

Where

- $H_{sensible}$ = Sensible heat gain (Btu/hr)
- H_{latent} = Latent heat gain (Btu/hr)
- H_{total} = Total heat gain (Btu/hr)
- CFM = Ventilation airflow rate in cubic feet per minute
- T_o = Outside dry bulb temperature, °F
- T_c = Dry bulb temperature of air leaving the cooling coil, °F
- ΔW_{GR} = Humidity Ratio Difference (Gr H_2O/Lb of dry air) = ($W_o - W_c$)
- ΔW_{LB} = Humidity Ratio Difference (Lb H_2O /Lb of dry air) and = ($W_o - W_c$)
- W_o = Outside humidity ratio, Lb H_2O per Lb (dry air)
- W_c = Humidity ratio of air leaving the cooling coil, Lb H_2O per Lb (dry air)
- h_o = Outside/Inside air enthalpy, Btu per lb (dry air)
- h_c = Enthalpy of air leaving the cooling coil Btu per lb (dry air)

Refer to 1997 ASHRAE Fundamentals, Chapter 25, for determining infiltration

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People

The heat load from people is both sensible load and the latent load. Sensible heat is transferred through conduction, convection and radiation while latent heat from persons is transferred through water vapor released in breathing and/or perspiration. The total heat transferred depends on the activity, clothing, air temperature and the number of persons in the building.

$H_{sensible} = N * (H_S) * (CLF)$

$H_{latent} = N * (H_L)$

Where

- $H_{sensible}$ = Total Sensible heat gain (Btu/hr)
- H_{latent} = Total latent heat gain (Btu/hr)
- N = number of people in space.
- H_S, H_L = Sensible and Latent heat gain from occupancy is given in 1997 ASHRAE Fundamentals Chapter 28, Table 3 (Btu/hr per person depending on nature of activity)
- CLF = Cooling Load Factor, by hour of occupancy. See 1997 ASHRAE Fundamentals, Chapter 28, table 37.Note: CLF = 1.0, if operation is 24 hours or of cooling is off at night or during weekends.

The sensible heat influence on the air temperature and latent heat influence the moisture content of indoor space.

Lights

The lights result in sensible heat gain.

$H = 3.41 * W * F_{UT} * F_{BF} * (CLF)$

Where

- H = Sensible heat gain (Btu/hr)
- W = Installed lamp watts input from electrical lighting plan or lighting load data
- F_{UT} = Lighting use factor, as appropriate
- F_{BF} = Blast factor allowance, as appropriate
- CLF = Cooling Load Factor, by hour of occupancy. See 1997 ASHRAE Fundamentals, Chapter 28, Table 38. *Note: CLF = 1.0, if operation is 24 hours or if cooling is off at night or during weekends.*

Power Loads & Motors

Three different equations are used under different scenarios:

a. Heat gain of power driven equipment and motor when both are located inside the space to be conditioned

   $H = 2545 * (P / Eff) * F_{UM} * F_{LM}$

   Where

   - H = Sensible heat gain (Btu/hr)
   - P = Horsepower rating from electrical power plans or manufacturer's data (HP)
   - Eff = Equipment motor efficiency, as decimal fraction
   - F_{UM} = Motor use factor (normally = 1.0)
   - F_{LM} = Motor load factor (normally = 1.0)
   - Note: F_{UM} = 1.0, if operation is 24 hours

b. Heat gain of when driven equipment is located inside the space to be conditioned space and the motor is outside the space or air stream

   $H = 2545 * P * F_{UM} * F_{LM}$

   Where

   - H = Sensible heat gain (Btu/hr)
   - P = Horsepower rating from electrical power plans or manufacturer's data (in HP)

- Eff = Equipment motor efficiency, as decimal fraction
- F_{UM} = Motor use factor
- F_{LM} = Motor load factor
- Note: F_{UM} = 1.0, if operation is 24 hours

c. Heat gain of when driven equipment is located outside the space to be conditioned space and the motor is inside the space or air stream

*H = 2545 * P * [(1.0-Eff)/Eff] * F_{UM} * F_{LM}*

Where

- H = Sensible heat gain (Btu/hr)
- P = Horsepower rating from electrical power plans or manufacturer's data (HP)
- Eff = Equipment motor efficiency, as decimal fraction
- F_{UM} = Motor use factor
- F_{LM} = Motor load factor
- Note: F_{UM} = 1.0, if operation is 24 hours

Appliances

*H = 3.41 * W * F_u * F_r * (CLF)*

Where

- H = Sensible heat gain (Btu/hr)
- W = Installed rating of appliances in watts. See 1997 ASHRAE Fundamentals, Chapter 28; Table 5 thru 9 or use manufacturer's data. For computers, monitors, printers and miscellaneous office equipment, see 2001 ASHRAE Fundamentals, Chapter 29, Tables 8, 9, & 10.
- F_u = Usage factor. See 1997 ASHRAE Fundamentals, Chapter 28, Table 6 and 7
- F_r = Radiation factor. See 1997 ASHRAE Fundamentals, Chapter 28, Table 6 and 7
- CLF = Cooling Load Factor, by hour of occupancy. See 1997 ASHRAE Fundamentals, Chapter 28, Table 37 and 39. Note: CLF = 1.0, if operation is 24 hours or of cooling is off at night or during weekends.

Conductive Heat Transfer

Conductive heat flow occurs in the direction of decreasing temperature and takes place when a temperature gradient exists in a solid (or stationary fluid) medium. The equation used to express heat transfer by conduction is known as Fourier's Law and is expressed as:

$H = k \times A \times \Delta T / t$

Where

- H = Hat transferred per unit time (Btu/hr)
- A = Heat transfer area (ft^2)
- k = Thermal conductivity of the material (Btu/ (hr °F ft^2/ft))
- ΔT = Temperature difference across the material (°F)
- t = material thickness (ft)

R-Values/U-Values

$R = 1/ C = 1/K \times t$

$U = 1/ \Sigma R$

Where

- R = R-Value (Hr Sq-ft °F/Btu)
- U = U-Value (Btu/Hr Sq-ft °F)
- C = Conductance (Btu/hr Sq-ft °F)
- K = Conductivity (Btu in/ hr Sq-ft °F)
- ΣR = Sum of the thermal resistances for each component used in the construction of the wall or roof section.
- t = thickness (ft)

Notes:

✓ The lower the U-factor, the greater the material's resistance to heat flow and the better is the insulating value. U-value is the inverse of R-value (hr sq-ft °F /Btu).

Heat Loss by Conduction & Convection through Roof & Walls

Heat loss by conduction and convection heat transfer through any surface is given by:

$H_{sensible} = A * U * (Ti – To)$

Where

- H = heat transfer through walls, roof, glass, etc. (Btu/hr)
- A = surface areas (sq-ft)

- U = air-to-air heat transfer coefficient (Unit- Btu/Hr Sq-ft °F)
- Ti = indoor air temperature (°F)
- To = outdoor air temperature (°F)

Heat Loss through Floors on Slab

The slab heat loss is calculated by using the following equation:

$H = F * P * (T_i - T_o)$

Where:

- H = Sensible heat loss (Btu/hr)
- F = Heat Loss Coefficient for the particular construction and is a function of the degree days of heating. (Unit- Btu/Hr Sq-ft °F)
- P = Perimeter of slab (ft)
- T_i = Inside temperature (°F)
- T_o = Outside temperature (°F)

Heat loss from slab-on- grade foundations is a function of the slab perimeter rather than the floor area. The losses are from the edges of the slab and insulation on these edges will significantly reduce the heat losses.

Heat loss through Infiltration and Ventilation

The heat loss due to infiltration and controlled natural ventilation is divided into sensible and latent losses. The energy associated with having to raise the temperature of infiltrating or ventilating air up to indoor air temperature is the sensible heat loss, which is estimated by:

$H_{sensible} = V * \rho_{air} * C_p * (Ti - To)$

Where:

- $H_{sensible}$ = Sensible heat loss
- V = volumetric air flow rate
- ρ_{air} is the density of the air
- C_p = specific heat capacity of air at constant pressure
- Ti = indoor air temperature
- To = outdoor air temperature

The energy quantity associated with net loss of moisture from the space is latent heat loss which is given by:

$H_{latent} = V * \rho_{air} * h_{fg} * (Wi – Wo)$

Where

- H_{latent} = Latent heat loss
- V = volumetric air flow rate
- ρ_{air} is the density of the air
- Wi = humidity ratio of indoor air
- Wo = humidity ratio of outdoor air
- hfg = latent heat of evaporation at indoor air temperature

SENSIBLE HEAT FACTOR or RATIO (SHR)

$SHR = H_s / H_t = H_s / (H_s + H_L)$

Where

- SHR = Sensible heat ratio
- H_s = Sensible heat gain
- H_L = Latent heat gain
- H_t = Total heat gain

Notes:

✓ SHR from 0.95 - 1.00 for Precision air conditioning (computers and data centres)

✓ SHR from 0.65 - 0.75 for Comfort cooling (people)

✓ SHR from 0.50 - 0.60 for Dehumidification (pools and outside air)

Lower SHR value indicates that the dehumidification requirement will be high and the supply air leaving the cooling coil shall be at lower temperature to meet the dehumidification needs. The supply airflow rate will be less. (See below for further justification)

SUPPLY AIR FLOW RATE

Supply air flow rate to a space is based only on the total space sensible heat load, thus

$Q = 1.08 \times [H_s / (T_R – T_S)]$

Where

- Q = air flow in cubic feet per minute (CFM)

- 1.08 = conversion constant = 0.244 X (60/13.5); 0.244 = specific heat of moist air, Btu/lb of dry air and 13.5 = specific volume of moist air, cu-ft. per lb of dry air (@70°F, 50% RH) and 60 is conversion from hour to minute.
- H_s = total room sensible heat gain, BTU per hr.
- T_R = Room dry bulb temperature, °F usually 75ºF
- T_S = Supply air or leaving air temperature from the cooling coil in ºF

The selection of temperature differential ($T_R – T_S$) is stated for simplicity above but actually it is little tricky as the real operating temperature differential is determined by the laws of "Psychrometrics" governing the performance of air system. As a rule of thumb following table provides an indicative relationship between the sensible heat ratio and leaving air temperature. Once the TD is known, the supply airflow rate (CFM) can be calculated:

Sensible Heat ratio Versus TD Value			
SHR	**LAT (*T_S*)**	**Room DB (*T_R*)**	**TD (*$T_R – T_S$*)**
X < .80	54	75	21
.80<X>.85	56	75	19
X >.85	58	75	17

Where:

- The sensible heat ratio (SHR) is: = (Sensible load)/(Sensible load + Latent load)
- LAT is the leaving air temperature
- TD is the temperature difference between the room temperature and the leaving air temperature (LAT from the air handler).

AIR BALANCE EQUATIONS

SA = RA + OA = RA + EA + RFA

Where

- SA = Supply Air
- RA = Return Air
- OA = Outside Air
- EA = Exhaust Air

- RFA = Relief Air

If minimum OA (ventilation air) is greater than EA, then the space will be positive pressurized and

OA = EA + RFA

If EA is greater than minimum OA (ventilation air), then the space will be under negative pressure and

OA = EA; RFA = 0

For Economizer Cycle

OA = SA = EA + RFA; RA = 0

CALCULATION OF HEATING DEGREE DAYS

Heating Degree Days (HDD) for a particular climate is obtained by subtracting each day's mean outdoor dry bulb temperature from the balance point temperature; this result is the number of HDDs for that day. For example, if the maximum and minimum outdoor dry bulb temperatures of a place were 80°F and 20°F respectively, and the balance point temperature were 65°F, then HDD of the place for that particular day would have been 65-[(80+20)/2] = 15. If the mean outdoor dry bulb temperature is equal to or higher than the balance point temperature, then the HDD would be equal to 0.

Degree Days and Annual Heating loss

A preliminary estimate of annual heating load, using degree day method, can be obtained by the following formula:

$H = PHL \times 24 \times HDD / \Delta T$

Where

- H = Annual heating load in Btu
- PHL = peak heating load (heat loss) in Btu/hr
- HDD = heating degree days
- ΔT = temperature difference, °F

CALCULATION OF COOLING DEGREE DAYS

Cooling Degree Days (CDD) for a particular climate is obtained by subtracting each day's mean outdoor dry bulb temperature from the balance point temperature; this result is the number of CDDs for that day. For example, if the maximum and minimum outdoor dry bulb temperatures of a place were 90°F and 60°F respectively, and the balance point temperature were 65°F, then CDD of the place for that particular day would have been [(90+60)/2]-65 = 10. If the mean outdoor dry bulb temperature is equal to or lower than the balance point temperature, then the CDD would be equal to 0.

Annual cooling load

A preliminary estimate of annual heating load, using degree day method, can be obtained by the following formula:

C = PCL x 24 x CDD / ΔT

Where

- C = Annual cooling load in Btu
- PCL = peak cooling load (heat gain) in Btu/hr
- CDD = cooling degree days
- ΔT = temperature difference, °F

FUEL CONSUMPTION BY HEAING UNITS

The amount of fuel required for annual heating can be calculated using the formula:

F = H / (e x FHC)

Where

- F = Quantity of fuel consumed (in gallon, MCF, kW, etc.) per year for heating
- H = Annual heating load in BTUH
- e = efficiency of the heating unit in decimal fraction (coefficient of performance in case of a heat pump)
- FHC = fuel heat content Btu per gal or kW

FUEL CONSUMPTION BY COOLING UNITS

The amount of fuel required for annual cooling can be calculated using the formula:

F = C / (SEER x 1000)

Where

- F = Electrical energy required in kWh per year for cooling
- C = Annual cooling load in Btu
- SEER = Seasonal Energy Efficiency Ratio of the cooling unit (Quantity of heat removed in BTUH for an input of 1 watt)

Fuel Cost

USD ($) = F x cost per unit (unit being gallon, mcf, kW, etc.)

AIR CHANGE RATE EQUATIONS

The most generic method used to calculate ventilation air requirements is based on complete changes of air in a structure or room in a given time period. To determine the airflow required to adequately ventilate an area, calculate the Room Volume to be ventilated Width x Length x Height = ft^3 (cubic feet) and than calculate the Air Volume requirement by multiplying the Room Volume by the Air Change Rate per hour.

ACH = Cu-ft / min x 60 min/hr / room volume

CFM = ACH x room volume / 60 min/hr

Where

- ACH = Air Change Rate per Hour
- CFM = Air Flow Rate (Cubic Feet per Minute)
- Room volume = Space Volume (Cubic Feet)

ESTIMATING AIR LEAKAGE

Leakage through the fixed openings should be restricted as much as possible. The amount of expected leakage can be calculated from the following:

$Lv = 4005 \times (R_p)^{½}$

$L = 4005 \times A \times (R_p)^{½}$

Where

- Lv = Leakage velocity in FPM
- L = Air leakage in CFM
- R_p = Room pressure in in-WG
- A = Opening area

Assuming 0.05 in-WG room pressures has an opening of 2 sq-ft opening

Leakage velocity = 0.223 x 4005

= 895 feet per minute

With a total of 2 square feet opening size

Leakage air volume = 2 x 895 = 1800 CFM

MIXED AIR TEMPERATURE

$T_{MA} = (T_{RA} \times Q_{RA} / Q_{SA}) + (T_{OA} \times Q_{OA} / Q_{SA})$

Where

- Q_{SA} = Supply Air flow rate (CFM)
- Q_{RA} = Return Air flow rate (CFM)
- Q_{OA} = Outside Air flow rate (CFM)
- T_{MA} = Mixed Air Temperature (°F)
- T_{RA} = Return Air or room design Temperature (°F)
- T_{OA} = Outside Air Temperature (°F)

NATURAL VENTILATION

The equation below is used in calculating ventilation (or infiltration) due to the stack effect.

$Q = C \times A \times [h \times (t_i - t_o)] / t_i$

In this equation:

- Q = Air Flow Rate (CFM)
- C = constant of proportionality = 313 (This assumes a value of 65 percent of the maximum theoretical flow, due to limited effectiveness of actual openings. With less favourable conditions, due to indirect paths from openings to the stack, etc., the effectiveness drops to 50 percent, and C = 240.)
- A = area of cross-section through stack or outlets in *sq ft. (*Note: Inlet area must be at least equal to this amount)
- t_i = (higher) temperature inside (°F), within the height h
- t_o = (lower) temperature outside (°F)
- h = height difference between inlets and outlets (ft)

OUTDOOR AIR

The equation for calculating outdoor quantities using carbon dioxide measurements is:

Outdoor air (in percent) = (Cr – Cs) x 100 / (Cr – Co)

Where:

Cs= ppm of carbon dioxide in the mixed air (if measured at an air handler) or in supply air (if measured in a room)

Cr= ppm of carbon dioxide in the return air

Co= ppm of carbon dioxide in the outdoor air

The auto-controller ensures that the increased ventilation is supplied only when required or needed for higher occupancies. This benefit in the energy cost savings because of reduced cooling and heating of outdoor air during reduced occupancy rates.

DILUTION VENTILATION

Dilution ventilation is most often used to advantage to control the vapours from organic liquids such as the less toxic solvents. To determine the correct volume flow rate for dilution (Q_d), it is necessary to estimate the evaporation rate of the contaminant (q_d) according to the following equation:

$q_d = 387 \text{ (lbs)} / (MW) \times (T) \times (\rho)$

Where

- q_d = Evaporation rate in CFM
- 387 = Volume in cubic feet formed by the evaporation of one lb-mole of a substance, e.g. a solvent
- MW = Molecular weight of the emitted material
- lbs = Pounds of evaporated material
- T = Time of evaporation in minutes
- ρ = density correction factor

The appropriate dilution volume flow rate for toxics is:

$Q_d = q_d \times K_m \times 10^6 / C_a$

Where

- Q_d = Volume flow rate of air, in CFM
- q_d = Evaporation rate in CFM
- K_m = Mixing factor to account for poor or random mixing (note K_m= 2 to 5; K_m = 2 is optimum)
- C_a = Accessible airborne concentration of the material

ESTIMATING AIR VOLUME FOR HOODS

Enclosed Hoods

For enclosed hoods, the exhaust volumetric flow rate can be calculated by the equation:

Q = V x A

Where

- Q = Volumetric flow rate (CFM)
- V = Average flow velocity (FPM)
- A = Flow cross-sectional area (Sq-ft)

The inflow velocity is usually around 100 FPM.

Non-enclosed Hoods

For non enclosed hoods, the capture velocity and the air velocity at the point of contaminant release must be equal and be directed so that the contaminant enters the hood. This results in different volumetric flow rate equations for different type of hoods. For un-flanged round and rectangular openings, the required flow rate equation is:

Q = V x [(10X x 10X) + A]

Where

- Q = Flow rate (CFM)
- V = Capture velocity (FPM)
- X = Centreline distance from the hood face to the point of contaminant generation (ft)
- A = Hood face area (Sq-ft)

L = Long dimension of the slot (ft)

For slot hoods, the required flow rte is predicted by an equation for openings between 0.5 to 2" in width:

Q = 3.7 x L x V x X

Where

- Q = Flow rate (CFM)
- V = Capture velocity (FPM)
- X = Centreline distance from the hood face to the point of contaminant generation (ft)
- L = Long dimension of the slot (ft)

If a flange is installed around the hood opening, the required flow rate for plain openings is reduced to 75 percent of that for the corresponding un-flanged opening. The flange size should be approximately equal to four times the area divided by the perimeter of the face hood. For flanged slots with aspect rations less than 0.15 and flanges greater than three times the slot width, the equation is

Q = 2.6 x L x V x X

A baffle is sold barrier that prevents airflow from unwanted areas in front of the hood. For hoods that include baffles, the DallaValle half hood equation is used to approximate the required flow rate:

$Q = v \times [(5X \times 5X) + A]$

HOOD SYSTEM PRESSURE

The hood converts duct static pressure to velocity pressure. The hood's ability to convert static pressure to velocity pressure is given by the coefficient of entry (C_e), as follows:

$C_e = (VP/SP_h)^{0.5} = [1/(1+K)]^{0.5}$

Where

- K = Loss factor
- VP = Velocity pressure in duct
- SP_h = Absolute static pressure about 5 duct diameters down the duct from the hood

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DUCTWORK EQUATIONS

Velocity in Duct

Velocity in duct can be expressed as

$V = Q / A = 144 \times Q / a \times b$

Where

- V = air velocity in ft per minute (FPM)
- Q = air flow through duct in cubic ft per minute (CFM)
- A = cross-section of duct in sq-ft

For rectangular ducts

- a = Width of duct side (inches)
- b = Height of other duct side (inches)

Equivalent Round Duct Size for a Rectangular Duct

Equivalent round duct size for a rectangular duct can be expressed as

$D_{eq} = 1.3 \times (a \times b)^{0.625} / (a + b)^{0.25}$

Where

- D_{eq} = equivalent diameter
- a = one dimension of rectangular duct (inches)

- b = adjacent side of rectangular duct (inches)

Equations for Flat Oval Ductwork

$$A = \frac{[(DL - DS) \times DS] + \dfrac{(\pi \times DS^2)}{4}}{144}$$

$$P = \frac{(\pi \times DS) + 2 \times (DL - DS)}{12}$$

$$Deq = \frac{1.55 \times (A)^{0.625}}{(P)^{0.25}}$$

Where

- DL = Major Axis Dimension (Inches)
- DS = Minor Axis Dimension (Inches)
- A = Cross-Sectional Area (Sq-ft)
- P = Perimeter or Surface Area (Sq-ft per linear feet)
- D_{eq} = Equivalent Round Duct Diameter

Duct Air Pressure Equations

TP = SP + VP

Where

- TP = Total Pressure
- SP = Static Pressure, friction losses
- VP = Velocity Pressure, dynamic losses

Velocity Pressure

$VP = (V / 4005)^2$

Where

- VP = Velocity pressure
- V = Air velocity in FPM

FAN EQUATIONS

Fan Output Power

BHP = Q x SP / (6356 x Fan $_{Eff}$.)

Where

- BHP = Break Horsepower
- P_t = Total pressure, in-WG
- Q = Air flow rate in CFM
- SP = Static pressure in-WG
- FAN_{EFF} = Fan efficiency usually in 65–85% range

Fan Motor Horsepower

Motor HP = BHP / Motor $_{Eff}$

Where

- BHP = Break Horsepower
- $Motor_{Eff}$ = Motor drive efficiency usually 80-95%

Tip Speed

Ts = 3.14 x D x N

Where

- Ts = Fan tip speed, FPM
- D = Fan diameter, ft
- N = Fan speed, RPM

Effect of Air Density

1) Air flow rate remains the same with a change in air density

2) Pressure varies in proportion to the air density

$$P_2 = P_1 \times \left(\frac{\rho_2}{\rho_1}\right)$$

3) Fan input power varies in proportion to the air density

$$H_2 = H_1 \times \left(\frac{\rho_2}{\rho_1}\right)$$

Where

- P_2 = Pressure at duty point 2, in-WG
- P_1 = Pressure at duty point 1, in-WG
- H_2 = Fan input power at duty point 2, in-WG
- H_1 = Fan input power at duty point 1, in-WG
- ρ_2 = Density at duty point 2, lb/ft^3
- ρ_1 = Density at duty point 1, lb/ft^3

Effect of Fan Speed

1) Fan air flow rate varies directly with fan speed ratio

$$Q_2 = Q_1 \times \left(\frac{N_2}{N_1}\right)$$

2) Fan pressure varies with square of fan speed ratio

$$P_2 = P_1 \times \left(\frac{N_2}{N_1}\right)^2$$

3) Fan power varies with cube of fan speed ratio

$$H_2 = H_1 \times \left(\frac{N_2}{N_1}\right)^3$$

Where

- Q_2 = Air flow rate at duty point 2, CFM
- Q_1 = Air flow rate at duty point 1, CFM
- P_2 = Pressure at duty point 2, in-WG
- P_1 = Pressure at duty point 1, in-WG
- H_2 = Fan input power at duty point 2, in-WG
- H_1 = Fan input power at duty point 1, in-WG
- N_2 = Fan speed at duty point 2, RPM
- N_1 = Fan speed at duty point 1, RPM

V-belt Length Formula

Once a sheave combination is selected we can calculate approximate belt length. Calculate the approximate V-belt length using the following formula:

$L = 2C + 1.57 * (D+d) + (D-d)^2 / 4C$

Where

- L = Pitch Length of Belt
- C = Center Distance of Sheaves
- D = Pitch Diameter of Large Sheave
- d = Pitch Diameter of Small Sheave

PUMP EQUATIONS

Pump Energy Consumption

The energy consumption of the pumps depends on two factors:

Pump BHP = GPM x TDH x SG / (3960 x Efficiency)

Pump BHP = GPM x PSI x SG / (1713 x Efficiency)

Where

- BHP = brake horse power
- Q = water flow, gallons per minute (GPM)
- TDH = Total Dynamic Head, ft
- SG = Specific Gravity, for water it is 1
- Efficiency = Pump efficiency from its pump curves for the water flow and TDH

Power consumption, KWH = KW input x operating hours

The KW input will depend on the motor efficiency and pump power requirement. (1 KW = 0.746 HP)

Pump Motor Horsepower

Motor HP = BHP / Motor $_{Eff}$

Where

- BHP = Break Horsepower
- Motor $_{Eff}$ = Motor drive efficiency usually 80-95%

Pump Affinity Laws

Effect on centrifugal pumps of change of speed or impeller diameter

Capacity varies directly as the speed or impeller diameter (GPM x rpm x D)

Head varies as the square of speed or impeller diameter (GPM x rpm^2 x D^2)

BHP varies as the cube of the speed or impeller diameter (BHP x rpm^3 x D^3)

For a fixed diameter impeller the capacity (GPM), Head (Ft) and BHP varies as follows:

$$\frac{GPM_2}{GPM_1} = \frac{RPM_2}{RPM_1}$$

$$\frac{HD_2}{HD_1} = \left[\frac{GPM_2}{GPM_1}\right]^2 = \left[\frac{RPM_2}{RPM_1}\right]^2$$

$$\frac{BHP_2}{BHP_1} = \left[\frac{GPM_2}{GPM_1}\right]^3 = \left[\frac{RPM_2}{RPM_1}\right]^3 = \left[\frac{HD_2}{HD_1}\right]^{1.5}$$

Specific Gravity

Specific gravity is direct ratio of any liquid's weight to the weight of water at 62 deg F. Water at 62 deg F weighs 8.33 lbs per gallon and is designated as 1.0 specific gravity. By definition, the specific gravity of a fluid is: SG = P_F / P_W

Where P_F is the fluid density and P_W is water density at standard conditions.

Example Specific Gravity of HCl = Weight of HCl / Weight of Water = 10.0 / 8.34 = 1.2

Head and Pressure

To start, head is not equivalent to pressure. Since the pump is a dynamic device, it is convenient to consider the head generated rather than the pressure. The term "Head" is usually expressed in feet whereas pressure is usually expressed in pounds per square inch. The relationship between two is

PSI = Head (feet) x Specific Gravity / 2.31

Velocity Head

$$VH = \frac{V^2}{2g}$$

Where

- VH = Velocity head in ft
- V = Velocity in ft/s
- g = Acceleration due to gravity (32.17 ft/s^2)

Bernoulli's equation

The pump generates the same head of liquid whatever the density of the liquid being pumped.

In the following equation (Bernoulli's equation) each of the terms is a head term: elevation head h, pressure head $\tilde{p}$ and velocity head $v^2/2g$. Head is equal to specific energy, of which the units are lbf-ft/lbf. Therefore, the elevation head is actually the specific potential energy, the pressure head, the specific pressure energy and the velocity head is the specific kinetic energy (specific means per unit weight).

$$h + p/y + v^2/2g = E = Constant$$

Where

- h: elevation in ft;
- p: pressure lb/sq-in;
- y: fluid specific weight
- v: velocity in ft/s
- g: acceleration due to gravity (32.17 ft/s^2);
- E: specific energy or energy per unit mass.

Note: A centrifugal pump develops head not pressure. All pressure figures should be converted to feet of head taking into considerations the specific gravity.

Pump NPSH

To determine the NPSH available, the following formula may be used

$$NPSHA = HA \pm HS - HF - HVP$$

Where

- NPSHA = Net Positive Suction Available at Pump expressed in feet of fluid
- NPSHR = Net Positive Suction Required at Pump (Feet)
- HA = Absolute pressure on the surface of the liquid where the pump takes suction, expressed in feet. This could be atmospheric pressure or vessel pressure (pressurized tank). It is a positive factor (34 Feet for Water at Atmospheric Pressure)
- HS =Static elevation of the liquid above or below the centerline of the impeller, expressed in feet. Static suction head is positive factor while static suction lift is a negative factor.
- HF = Friction and velocity head loss in the piping, also expressed in feet. It is a negative factor.

- HVP = Absolute vapor pressure of the fluid at the pumping temperature, expressed in feet of liquid. It is a negative pressure.

The Net Positive Suction Head (N.P.S.H.) is the pressure head at the suction flange of the pump less the vapour pressure converted to fluid column height of the fluid. The N.P.S.H. is always positive since it is expressed in terms of absolute fluid column height. The value, by which the pressure in the pump suction exceeds the liquid vapour pressure and is expressed as a head of liquid and referred to as Net Positive Suction Head Available – (NPSHA). This is a characteristic of the suction system design. The value of NPSH needed at the pump suction to prevent the pump from cavitating is known as NPSH Required – (NPSHR). This is a characteristic of the pump design.

Note that NPSHA > NPSHR i.e. the N.P.S.H. available must always be greater than the N.P.S.H. required for the pump to operate properly.

Pump Specific Speed

Equation below gives the value for the pump specific speed;

$Ns = (N_r \times Q) / (H)^{3/4}$

Where

- Ns = Specific speed
- Q = Flow in US gallons per minute (GPM)
- Nr = Pump speed, RPM
- H = Head, ft

Specific speed is a dimensionless quantity.

Specific speed is indicative of the shape and characteristics of an impeller. Impeller form and proportions vary with specific speed but not the size. It can be seen that there is a gradual change in the profiles from radial to axial flow configuration. Studies indicate that a pump efficiency at the best efficiency point (BEP) depends mainly on the specific speed, and a pump with specific speed of 1500 is more efficient then the one with specific speed of 1000.

CHILLER HEAT LOAD & WATER FLOW

The flow rate necessary to deliver the full output of the heat source at a specific temperature drop can be found using equation below:

$Q = H / (8.01 \times \rho \times c \times \Delta T)$

Where:

- Q= Water volume flow rate (GPM)

- H = Heat load (Btu/hr)
- ΔT = Intended temperature drop (°F)
- ρ = Fluid's density at the average system temperature (lb/ft^3)
- c = the fluid's specific heat at the average system temperature (Btu/lb/°F)
- 8.01 = a constant

In small to medium size hydronic systems, the product of (8.01 x ρ x c) can be taken as 500 for water, 479 for 30% glycol, and 450 for 50% glycol. The total heat removed by air condition chilled-water installation can thus be expressed as

$H = 500 \times Q \times \Delta T$

Where

- H = total heat removed (Btu/h)
- Q = water flow rate (gal/min)
- ΔT = temperature difference (°F)

Evaporator Flow Rate

The evaporator water flow rate can be expressed as

$Q_e = H_{tons} \times 24 / \Delta T$

Where

- Q_e = Evaporator water flow rate (GPM)
- H_{tons} = Air conditioning cooling load (tons)
- ΔT = Temperature differential between inlet and outlet (°F)

Condenser Flow Rate

The condenser water flow rate can be expressed as

$Q_c = H_{tons} \times 30 / \Delta T$

Where

- Q_c = Condenser water flow rate (GPM)
- H_{tons} = Air conditioning cooling load (tons)
- ΔT = Temperature differential between inlet and outlet (°F)

Note the equation above assumes 25% heat of compression.

CONDENSATE GENERATION

Condensate generation in an air condition system where specific humidity before and after are known can be expressed as

$Q_{Cond} = Q_{air} \times \Delta W_{Lb} / (SpV \times 8.33)$

$Q_{Cond} = Q_{air} \times \Delta W_{GR} / (SpV \times 8.33 \times 7000)$

Where

- Q_{Cond} = Air Conditioning condensate generated (GPM)
- Q_{Air} = Air Flow Rate through the air-handling unit cooling coil (Cu-ft / minute)
- SpV = Specific Volume of Air (Cu-ft per lb of dry air)
- ΔW_{lb} = Specific Humidity diff. between inlet and outlet of air stream across coil (lb-H_2O per lb of dry air)
- ΔW_{Gr} = Specific Humidity diff. between inlet and outlet of air stream across coil (Gr. H_2O per lb of dry air)

FLOW RATES IN HEATING SYSTEMS

The volumetric flow rate in a heating system can be expressed by the basic equation:

$Q = H / (Cp \times \rho \times \Delta T)$

Where

- Q = volumetric flow rate (GPM)
- H = heat flow rate (Btu/hr)
- C_p = specific heat capacity (Btu/lb-°F)
- ρ = density (lb/ft^3)
- ΔT = temperature difference (°F)

The basic equation can be expressed for water with temperature 60°F flow rate as:

$Q = H\ (7.48\ gal/ft^3) / ((1\ Btu/lb\text{-}^{\circ}F)\ (62.34\ lb/ft^3)\ (60\ min/h)\ \Delta T)$

Or

$Q = h / (500 \times \Delta T)$

Where

- *Q* = Water flow rate (GPM)
- *H* = Heat flow rate (Btu/hr)

- ΔT = Temperature difference (°F) (usually 20ºF)

For more exact volumetric flow rates for hot water the properties of hot water should be used.

Water Mass Flow Rate

Water mass flow can be expressed as:

m = h / ((1.2 Btu/lb-°F) x ΔT)

Where

- m = mass flow (lb/hr)

FRICTION LOSS IN WATER PIPES

Friction loss in water pipes can be obtained by using the empirical Hazen-Williams equation

Hazen-Williams Equation

$f = 0.2083\ (100/C)^{1.852}\ Q^{1.852} / d^{4.8655}$

Where

- f = friction head loss in feet of water per 100 feet of pipe
- C = Hazen-Williams roughness constant
- Q = volume flow (gal/min)
- d = inside diameter (inches)

Note that the Hazen-Williams formula is empirical and lacks physical basis. Be aware that the roughness constants are based on "normal" condition with approximately 3 ft/sec.

Coefficients (C) for some common materials used in ducts and pipes can be found in the table below:

Material	**Hazen-Williams Coefficient - C -**
Brass	130 - 140
Cast-Iron - new unlined (CIP)	130
Cast-Iron 10 years old	107 - 113
Cast-Iron 20 years old	89 - 100
Cast-Iron 30 years old	75 - 90
Concrete	120 - 140
Copper	130 - 140
Ductile Iron Pipe (DIP)	120
Galvanized iron	120

Plastic	140 - 150
Steel	130

Darcy-Weisbach equation and Head Loss

The Darcy-Weisbach equation is valid for fully developed, steady, incompressible flow.

The friction factor or coefficient - λ -depends on the flow - if it is laminar, transient or turbulent (the Reynolds Number) - and the roughness of the tube or duct. The friction coefficient can be calculated by the Colebrooke Equation or by using the Moody Diagram.

Alternatively the Darcy-Weisbach equation can be expressed as head loss:

$H_{loss} = \lambda (L / d_h) [v^2 / (2 \times g)]$

Where

- H_{loss} = head loss (ft)
- λ = friction coefficient
- *L* = length of duct or pipe (ft)
- g = acceleration of gravity (32.2 ft/s^2)
- d_h = The hydraulic diameter - d_h - is used for calculating the dimensionless Reynolds Number (Re) to determine if the flow is turbulent or laminar.

The Reynolds Number (Re) is important in analyzing any type of flow when there is substantial velocity gradient - shear. The Reynolds Number indicates the relative significance of the viscous effect compared to the inertia effect. The Reynolds number is proportional to inertial force divided by viscous force. The flow is

- laminar if Re < 2300
- transient for 2300 < Re < 4000
- turbulent if Re > 4000

Reynolds Number can be expressed as:

$Re = D \times v \times \rho / \mu$

Where

- D = characteristic length (For a pipe or duct the characteristic length is the pipe or duct diameter, in m)
- v = velocity (m/s)
- ρ = density (kg/m^3)
- μ = dynamic (absolute) viscosity (Ns/m^2)

The table below shows Reynolds Number for one gallon of water flowing through pipes of different dimensions:

Pipe Size										
(inches)	**1**	**1 ½**	**2**	**3**	**4**	**6**	**8**	**10**	**12**	**18**
Reynolds number with one (1) gal/min	3800	2500	1900	1270	950	630	475	380	320	210

Hydraulic Diameter: The hydraulic diameter is not the same as the geometrical diameter in non-circular ducts or pipes and can be calculated from the generic equation:

$d_h = 4\, A / P$

Where:

- d_h = hydraulic diameter (in)
- A = area section of the pipe (in^2)
- P = wetted perimeter of the pipe (in)

Friction Coefficient *(λ)* for fully developed laminar flow the roughness of the duct or pipe can be neglected. The friction coefficient depends only the Reynolds Number - *Re* - and can be expressed as:

$\lambda = 64 / Re$

Where

- *Re* = the dimensionless Reynolds number

COOLING TOWERS

Cooling towers are rated in terms of *approach* and *range*.

The approach is the difference in temperature between the cooled-water temperature and the entering-air wet bulb temperature.

Approach = LWT – WBT

The range is the temperature difference between the water inlet and exit states.

Range = EWT – LWT

Where

- EWT = Entering hot water temperature (°F)
- LWT = Leaving cold water temperature (°F)

- WBT = Ambient wet bulb temperature (Design WB, °F)

Water Circulation through Cooling Tower

Q = H / (Range X 500)

Where

- H = Cooling tower heat rejection in Btu/hr
- Q = Water flow rate in GPM

Cooling Tower Water Balance

The amount of water that enters as make-up must be equal to the total water that exits the system or

MR = water lost through evaporation (ER) + bleed (BR) + drift (DR)}

Where

- MR = Makeup water requirement in GPM
- DR = Typical drift rate in GPM
- ER = Evaporation rate in GPM
- BR = Bleed rate in GPM

The evaporation rate of cooling tower is

ER = Q X Range / 1,000

The drift loss is roughly 0.2 to 0.5%

DR = 0.002% X Q

When we ignore the insignificant drift losses

Then, MR = ER + BR …… (eq.1)

Recognizing that in order to keep off from making scale, all of the solids that enter as make- up must exit as bleed, it follows that:

MR = COC x BR …… (eq.2)

And that:

MR = ER [(COC)/ (COC -1)] ……. (eq.3)

Combine (eq.2) and (eq.3) to get:

BR = ER / (COC -1)

Where

- COC = Cycles of concentration.

Ideally the COC is maximized to 5 to 7 by addition of water treatment chemicals.

Cooling Tower Efficiency

Since the cooling towers are based on the principles of evaporative cooling, the maximum cooling tower efficiency depends on the *wet bulb temperature* (WBT) of the air.

The cooling tower efficiency can be expressed as:

Efficiency = (EWT - LWT) x 100 / (EWT - WBT)

Where

- Efficiency = cooling tower efficiency - common range between 70 - 75%
- EWT = inlet temperature of water to the tower (°F)
- LWT = outlet temperature of water from the tower (°F)
- WBT = wet bulb temperature of air (°F)

The temperature difference between inlet and outlet water *(EWT - LWT)* is normally in the range 10 - 15°C. The water consumption - the make up water - of a cooling tower is about 0.2-0.3 liter per minute per ton of refrigeration.

Cooling towers use the principle of evaporative cooling in order to cool water. They can achieve water temperatures below the dry bulb temperature (DBT) of the air used to cool it. They are in general smaller and cheaper for the same cooling load than other cooling systems.

Cooling Tower Tons

A cooling tower ton is defined as:

1 cooling tower ton = 15,000 Btu/hr (3782 kCal /hr)

This is roughly 25% more than chiller ton because the heat of compression of the refrigeration compressor is added to the condenser/cooling tower.

CONTROL VALVE SIZING

A control valve is the single most important element in any fluid handling system. The basic plan in sizing a valve is to determine the flow coefficient (Cv) of a valve.

Flow Coefficient

Most suppliers publish valve capacity tables based on the flow coefficient Cv. This is defined as the volume flow rate in gallons of 60°F water that will pass through the valve in one minute at one pound pressure drop. By definition

$$C_{v\ net} = \frac{Qgpm}{\sqrt{\Delta P_{net}\ /\ Sg}}$$

Where

1) Sg is the specific gravity of the fluid, water = 1
2) ∆P is in psi
3) Cv = Flow coefficient

The flow coefficient or pressure loss coefficient is used to relate the pressure loss of a valve to the discharge of the valve at a given valve opening. Valve capacity tables usually show Cv and then flow rate at various pressure drops. *The rated Cv is established with the valve fully open. As the valve partially closes to some intermediate position the Cv will decrease.* The rate at which it decreases determines the shape of the curve of flow rate through the valve and % valve stem movement.

Rangeability Factor: This describes the ability of a valve to stay on its theoretical characteristic at the bottom end near the closed position. This is the minimum value that should be considered if good control on light load is to be achieved. The ratio between the full flow and the minimum controllable flow is the rangeability factor.

Rangeability factor (RF) = Maximum flow / minimum controllable flow

The rangeability factor is measured under laboratory conditions, with a constant differential pressure applied across the valve. Rangeability is a characteristic of the valve itself and it depends on its design and tolerances.

Turndown Ratio: This is the ratio between the maximum normal flow and the minimum controllable flow. It is substantially less than the range ability if the valve is oversized, either by error or deliberately to allow for an occasional peak load.

Turndown Ratio (TR) = Maximum flow (installed) / minimum controllable flow

The higher the turndown ratio is, the better the controllability will be.

Valve Authority: As a control valve closes, the pressure drop across the valve increases so that when the valve is completely closed, the differential pressure drop across the valve matches the pressure drop from the supply to the return line. This pressure drop is known as ∆P-max. When the valve is completely open, the pressure drop across the valve is at its lowest point and is referred to ∆P-min. The ratio (N) ∆P min ÷ ∆P max is the valve authority.

The increase in pressure drop across the valve as it closes is important to note. Valves are rated based on a constant pressure drop. As the pressure drop shifts, the performance of the valve changes. The method to minimize the change in valve performance is to maintain the Valve Authority (N) above 0.5.

Authority is related to turndown. Turndown ratio = Rangeability X (Authority) $^{0.5}$.

HEAT EXCHANGERS

Range = EWT – LWT

Approach = $EWT_{HS} - LWT_{CS}$

Where

- EWT = Entering Water Temperature (°F)
- LWT = Leaving Water Temperature (°F)
- EWT_{HS} = Entering Water Temperature – Hot Side (°F)
- LWT_{CS} = Leaving Water Temperature – Cold Side (°F)

Logarithmic Mean Temperature Difference – LMTD

In a heat transfer process, the temperature difference varies with position and time. The determination of the mean temperature difference in a heat transfer process depends upon the direction of fluid flow involved in the process. The primary and secondary fluid in a heat exchanger process may:

1) flow in the same direction - parallel flow/co-current flow
2) in the opposite direction - counter current flow
3) or perpendicular to each other - cross flow

When the secondary fluid passes over the heat transfer surface, the highest rate of heat transfer occurs at the inlet and progressively decays with higher secondary fluid temperature along its way to the outlet.

The rise in secondary temperature is non-linear and is best represented by a logarithmic calculation. For this purpose the mean temperature difference chosen is termed the Logarithmic Mean Temperature Difference or LMTD and can be expressed as:

$$LMTD = (TD_2 - TD_1) / ln (TD_2 / TD_1)$$

Where

- LMTD = Logarithmic Mean Temperature Difference °F
- $TD_1 = T_{P1} – T_{S2}$ - Entering primary fluid and leaving secondary fluid temperature difference °F
- $TD_2 = T_{P2} – T_{S1}$ - Leaving primary fluid and entering secondary fluid temperature difference °F

Arithmetic Mean Temperature Difference – AMTD

An easier but less accurate way to calculate the mean temperature difference is to consider the Arithmetic Mean Temperature Difference or AMTD and can be expressed as:

$$AMTD = (T_{P1} + T_{P2}) / 2 - (T_{S1} + T_{S2}) / 2$$

Where

- AMTD = Arithmetic Mean Temperature Difference °F

- T_{P1} = primary inlet temperature $^{\circ}F$
- T_{P2} = primary outlet temperature $^{\circ}F$
- T_{S1} = secondary inlet temperature $^{\circ}F$
- T_{S2} = secondary outlet temperature $^{\circ}F$

A linear increase in the secondary fluid temperature makes it easier to do manual calculations. AMTD will in general give a satisfactory approximation for the mean temperature difference.

When heat is transferred as a result of a change of phase in condensation or evaporation heat exchangers, the temperature of the primary or secondary fluid remains constant. For example, with saturation of steam the primary fluid temperature can be taken as a constant because heat is transferred as a result of a change of phase only. The equation can then be simplified by setting

$T_{P1} = T_{P2}$ or $T_{S1} = T_{S2}$ $^{\circ}F$

Thus in the equation above, the temperature profile in the primary fluid is not dependent on the direction of flow.

HUMIDIFICATION

$GR_{reqd} = (W_{GR} / SpV)_{Room\ air} - (W_{GR} / SpV)_{Supply\ air}$

$Lb_{reqd} = (W_{Lb} / SpV)_{Room\ air} - (W_{Lb} / SpV)_{Supply\ air}$

$Q_{steam} = (Q_{air} \times GR_{reqd} \times 60) / 7000$

$Q_{steam} = Q_{air} \times Lb_{reqd} \times 60$

Where

- GR_{reqd} = Grains of Moisture Required (Gr H_2O per Cu-ft of air)
- Lb_{reqd} = Pounds of Moisture Required (Lb H_2O per Cu-ft of air)
- Q_{air} = Air Flow Rate (CFM)
- Q_{steam} = Steam Flow Rate (Lb per hour)
- SpV = Specific Volume of Air (Cu-ft per lb of dry air)
- ΔW_{lb} = Specific Humidity (lb-H_2O per lb of dry air)
- ΔW_{Gr} = Specific Humidity (Gr. H_2O per lb of dry air)

Humidifier Sensible Heat Gain

$H = (0.244 \times Q \times \Delta T) + (L \times 380)$

Where

- H = Sensible Heat Gain (Btu/Hr)
- Q = Steam Flow (Lb-Steam/Hr)
- ΔT = Steam Temperature - Supply Air Temperature (°F)
- L = Length of Humidifier Manifold (ft)

DEHUMIDIFIER EQUATIONS

A measure of the capacity of a dehumidifier is expressed in lbs per hour of moisture removal and is estimated by equation:

$MRC = [Q_{air} \times (60 \text{ min/hr} / Vs] \times (GPP_{in} - GPP_{out})) / 7000 \text{ grains/lb}$

Where,

- MRC = Moisture removal capacity (in Lb /hr)
- Q_{air} = Volumetric rate of air (CFM)
- Vs = Specific volume of air (Cu-ft / lb)
- GPP_{in} = Grains of moisture per pound of dry air in the inlet air stream
- GPP_{out} = Grains of moisture per pound of dry air in the outlet air stream

The difference in ($GPP_{in} – GPP_{out}$) represents the grain "depression" or removal across the dehumidifier.

EXPANSION TANKS

An expansion tank is necessary in a heating, cooling or air condition system to avoid an unacceptable increase of the system pressure during heat-up. Expansion tanks are in general designed as open, closed or diaphragm tanks.

Open Expansion Tanks

The necessary volume of an open expansion tank can be expressed as

$V_{et} = 2 \times \{V_w \times [(v_1 / v_0) - 1] – 3 \times a \times (T_1 – T_0)\}$

Closed Expansion Tanks

The necessary volume of a closed expansion tank can be expressed as

$V_{et} = 2 \{V_w [(v_1 / v_0) - 1] – 3 \times a \times (T_1 – T_0)\} / [(p_a / p_0) - (p_a / p_1)]$

Diaphragm Expansion Tanks

The necessary volume of a diaphragm expansion tank can be expressed as

$V_{et} = 2 \{V_w [(v_1 / v_0) - 1] - 3 \times a \times (T_1 – T_0)\} / [1 - (p_0 / p_1)]$

Where

1) V_{et} = Necessary expansion tank volume (Gallons)

2) V_w = Water volume in the system piping (Gallons)

3) v_0 = Specific volume of water at initial (cold) temperature T_0 in (ft^3/lb), refer *1989 ASHRAE Fundamentals,* Chapter 2, Table 25 or Part 27, Properties of Air and Water

4) v_1 = Specific volume of water at operating (hot) temperature T_1 (ft^3/lb), refer *1989 ASHRAE Fundamentals,* Chapter 2, Table 26 or Part 27, Properties of Air and Water

5) T_1 = Higher System Temperature (°F)

   - Heating Water: T_2 = Supply Water Temperature
   - Chilled Water: T_2 = 95°F Ambient Temperature (Design Weather Data)
   - Dual Temperature: T_2 = Heating Water Supply Temperature

6) T_0 = Lower System Temperature (°F)

   - Heating Water: T_1 = 45–50°F Temperature at Fill Condition
   - Chilled Water: T_1 = Supply Water Temperature
   - Dual Temperature: T_1 = Chilled Water Supply Temperature

7) p_a = Atmospheric pressure - 14.7psia

8) p_0 = System initial (fill) pressure - cold pressure (psia)

9) p_1 = System operating pressure - hot pressure (psia)

10) a = Linear Coefficient of Expansion ($ά_{STEEL} = 6.5 \times 10^{-6}$ and $ά_{COPPER} = 9.5 \times 10^{-6}$)

Notes:

a. System Volume Estimate: 12 Gal /Ton and 35 Gal /BHP

b. System Fill Pressure/Minimum System Pressure Estimate: Height of System +5 to 10 PSI OR - 5–10 PSI, whichever is greater.

c. System Operating Pressure/Maximum Operating Pressure Estimate:

   - 150 Lb Systems 45–125 PSI
   - 250 Lb Systems 125–225 PSI

TONS OF REFRIGERATION (TR)

A ton of refrigeration is the amount of heat removed by an air conditioning system that would melt 1 ton of ice in 24 hours.

1TR = 12,000 Btu/hr

One ton of refrigeration is equal to heat extraction @ of 200 BTUs per minute, 12,000 Btu per hour or 3025.9 Kcal/hr. This is based on the latent heat of fusion for ice which is 144 Btu per pound.

ENERGY EFFICIENCY TERMS OF REFRIGERATION SYSTEMS

KW per ton

The term kW/ton is common used for large commercial and industrial air-conditioning, heat pump and refrigeration systems. The term is defined as the ratio of the rate of energy consumption in kW to the rate of heat removal in tons at the rated condition. The lower the kW/ton the more efficient is the system.

$KW/ton = P_c / H_r$

Where

- P_c = energy consumption (kW)
- H_r = heat removed (ton)

Coefficient of Performance-COP

The Coefficient of Performance - COP - is the basic unit less parameter used to report the efficiency of refrigerant based systems. The Coefficient of Performance - COP - is the ratio between useful energy acquired and energy applied and can be expressed as:

$COP = H_u / H_a$

Where

- COP = Coefficient of performance
- H_u = Useful energy acquired (Btu)
- H_a = Energy applied (Btu)

COP can be used to define either cooling efficiency or heating efficiency as for a heat pump.

- For cooling, COP is defined as the ratio of the rate of heat removal to the rate of energy input to the compressor.
- For heating, COP is defined as the ratio of rate of heat delivered to the rate of energy input to the compressor.

COP can be used to define the efficiency at a single standard or non-standard rated condition or a weighted average seasonal condition. The term may or may not include the energy consumption of auxiliary systems such as indoor or outdoor fans, chilled water pumps, or cooling tower systems. For purposes of comparison, the higher the COP the more efficient the system.

COP can be treated as an efficiency where COP of 2.00 = 200% efficient For unitary heat pumps, ratings at two standard outdoor temperatures of 47°F and 17°F (8.3°C and -8.3°C) are typically used.

Energy Efficiency Ratio – EER

The Energy Efficiency Ratio - EER - is a term generally used to define the cooling efficiency of unitary air-conditioning and heat pump systems. The efficiency is determined at a single rated condition specified by the appropriate equipment standard and is defined as the ratio of net cooling capacity - or heat removed in Btu/h (not in tons) - to the total input rate of electric energy applied - in watt hour (not in kW). The units of EER are Btu/W-hr.

$EER = H_c / P_a$

Where

- EER = Energy efficient ratio (Btu/W-hr)
- H_c = net cooling capacity (Btu/hr)
- P_a = applied energy (Watts)

This efficiency term typically includes the energy requirement of auxiliary systems such as the indoor and outdoor fans and the higher the EER the more efficient is the system. Energy efficiency ratio is further categorized as Energy efficiency ratio (EER) and Seasonal energy efficiency ratio (SEER):

- The cooling equipment systems such as room air conditioners, heat pumps etc used in residential and small commercial buildings often express cooling system efficiency in terms of the Energy Efficiency Ratio (EER).
- The central air-conditioning equipment used in large residential and commercial buildings expresses cooling system efficiency in terms of the Seasonal Energy Efficiency Ratio (SEER).

Recommended selection of room air conditioners is EER of at least 9.0 for mild climates and over 10 for hot climates and for central air conditioning system it is tested to be as high as 17 units. The U.S. Government's minimum efficiency level is 10 SEER for split systems and 9.7 for packaged units.

Efficiency - Heating Systems

Turndown Ratio = Maximum Firing Rate: Minimum Firing Rate (i.e., 5:1, 10:1, 25:1)

Overall Thermal Efficiency = (Gross Btu Output / Gross Btu Input) x 100%

- Overall Thermal Efficiency Range 75%–90%

Combustion Efficiency = {(Btu Input – Btu Stack Loss) / Btu Input} x 100%

- Combustion Efficiency Range 85%–95%

CALCULATE DECIBEL

Decibel is a logarithmic unit used to describe the ratio of the signal level - power, sound pressure, voltage, intensity, etc. Decibel is a logarithmic unit used to describe the ratio of the signal level - power, sound pressure, intensity or several other things.

The decibel can be expressed as:

$Decibel = 10 \log (P / P_{ref})$

Where

- P = signal power (W)
- P_{ref} = reference power (W)

Note! Doubling the signal level increases the decibel with 3 dB (10 log (2)).

Adding Equal Sound Power Sources

The sound power and sound power level is commonly used to specify the emitted noise or sound from technical equipment as fans, pump and other machines.

The logarithmic decibel scale is convenient when calculating resulting sound power levels and sound pressure levels for two or more sound or noise sources.

$L_{wt} = 10 \log (n\, N / N_o) = 10 \log (N / N_o) + 10 \log (n)$

$= L_{ws} + 10 \log (n)$

Where

- L_{wt} = Total sound power level (dB)
- L_{ws} = Sound power level from each single source (dB)
- N = sound power (W)
- $N_o = 10^{-12}$ - reference sound power (W)
- n = number of sources

Note: Adding two identical sources will increase the total sound power level with 3 dB (10 log (2)).

Adding Equal Sound Pressure Levels

The resulting sound pressure level when adding equal sound pressure can be expressed as:

$L_{pt} = L_{ps} + 20 \log (n)$

Where

- L_{pt} = total sound pressure (dB)
- L_{ps} = Sound pressure level from each single source (dB)
- n = number of sources

Adding Sound Power from Sources at different Levels

The sound power level from more than one source can be calculated as:

$L_{wt} = 10 \log ((N_1 + N_2 ... + N_n) / N_o)$

STEAM & CONDENSATE EQUATIONS

Some common steam and condensate equations can be expressed as

Steam Heating

$m_s = H / 960$

Where

- m_s = steam mass flow rate (Lbs /hr)
- H = heat flow rate (Btu/hr)

Steam Heating Liquid Flow

$m_s = Q_L \times 500 \times SG_L \times C_{pL} \times \Delta T_L / L_S$

Where

- m_s = steam mass flow rate (Lbs /hr)
- Q_L = volume flow liquid (GPM)
- SG_L = specific heat capacity of the liquid (Btu/lb °F)
- C_{pL} = specific gravity of the fluid
- ΔT_L = temperature difference liquid (°F)
- L_s =latent heat of steam at steam design pressure (Btu/lb)

Steam Heating Air or Gas Flow

$m_s = Q_G \times 60 \times \rho_G \times C_{pG} \times \Delta T_G / L_S$

Where

- m_s = steam mass flow rate (lbs /hr)
- Q_G = volume flow gas (CFM)
- ρ_G = density of the gas (lb/ft^3)
- C_{pG} = specific gravity of the gas (Air C_{pG} = 0.24 Btu/Lb)
- ΔT_G = temperature difference gas (°F)
- L_s = latent heat of steam at steam design pressure (Btu/lb)

Steam Pipe Sizing Equations

$\Delta P = [(0.01306 \times W^2 \times (1+ 3.6/ID)] / (3600 \times D \times ID^5)$

$W = 60 \times \{(P \times D \times ID^5) / [0.01306 \times (1+3.6 / ID)]\}^{1/2}$

$W = 0.41667 \times V \times A_{inches} \times D = 60 \times V \times A_{Feet} \times D$

$V = 2.4 \times W / A_{inches} \times D = W / (60 \times A_{feet} \times D)$

Where

- ΔP = Pressure Drop per 100 Feet of Pipe (Psig/100 feet)
- W = Steam Flow Rate (Lbs /Hr)
- ID = Actual Inside Diameter of Pipe (Inches)
- D = Average Density of Steam at System Pressure (Lbs/Cu-ft)
- V = Velocity of Steam in Pipe (Feet/Minute)
- A_{INCHES} = Actual Cross Sectional Area of Pipe (Sq-inches)
- A_{FEET} = Actual Cross Sectional Area of Pipe (Sq-ft)

Steam Condensate Pipe Sizing Equations

$FS = (H_{SS} - H_{SC}) / H_{LC} \times 100$

$W_{CR} = FS / 100 \times W$

Where

- FS = Flash Steam (Percentage %)
- H_{SS} = Sensible Heat at Steam Supply Pressure (Btu/Lb)
- H_{SC} = Sensible Heat at Condensate Return Pressure (Btu/Lb)
- H_{LC} = Latent Heat at Condensate Return Pressure (Btu/Lb)

- W = Steam Flow Rate (Lbs/Hr)
- W_{CR} = Condensate Flow based on percentage of Flash Steam created during condensing process (Lbs/hr)

Use this flow rate in steam equations above to determine condensate return pipe size.

RELIEF VALVE SIZING

Liquid System Relief Valves and Spring Style Relief Valves:

$A = (GPM \times (G)^{1/2}) / [28.14 \times K_B \times K_V \times (\Delta P)^{1/2}]$

Liquid System Relief Valves and Pilot Operated Relief Valves:

$A = (GPM \times (G)^{1/2}) / [36.81 \times K_V \times (\Delta P)^{1/2}]$

Steam System Relief Valves:

$A = W / (51.5 \times K \times P \times K_{SH} \times K_N \times K_B)$

Gas and Vapor System Relief Valves (Lb/Hr.):

$A = (W \times (TZ)^{1/2}) / [C \times K \times P \times K_B \times (M)^{1/2}]$

Gas and Vapor System Relief Valves (SCFM):

$A = (SCFM \times (TGZ)^{1/2}) / (1.175 \times C \times K \times P \times K_B)$

Where

- A = Minimum Required Effective Relief Valve Discharge Area (Sq- inches)
- GPM = Required Relieving Capacity at Flow Conditions (Gallons per Minute)
- W = Required Relieving Capacity at Flow Conditions (Lbs / hr)
- SCFM = Required Relieving Capacity at Flow Conditions (Standard Cubic Feet per Minute)
- G = Specific Gravity of Liquid, Gas, or Vapor at Flow Conditions Water = 1.0 for most HVAC applications; Air = 1.0
- C = Coefficient Determined from Expression of Ratio of Specific Heats; C = 315 if Value is Unknown
- K = Effective Coefficient of Discharge; K = 0.975
- K_B = Capacity Correction Factor Due to Back Pressure; KB = 1.0 for Atmospheric Discharge Systems
- K_V = Flow Correction Factor Due to Viscosity; KV = 0.9 to 1.0 for most HVAC Applications with Water

- K_N = Capacity Correction Factor for Dry Saturated Steam at Set Pressures above 1500 Psia and up to 3200 Psia; KN = 1.0 for most HVAC Applications
- K_{SH} = Capacity Correction Factor Due to the Degree of Superheat; KSH = 1.0 for Saturated Steam
- Z = Compressibility Factor; Z = 1.0 If Value is Unknown
- P = Relieving Pressure (Psia); P = Set Pressure (Psig) + Over Pressure (10% Psig) + Atmospheric Pressure (14.7 Psia)
- ΔP = Differential Pressure (Psig); ΔP = Set Pressure (Psig) + Over Pressure (10% Psig) - Back Pressure (Psig)
- T = Absolute Temperature (°R = °F. + 460)
- M = Molecular Weight of the Gas or Vapor

Notes:

1) When multiple relief valves are used, one valve shall be set at or below the maximum allowable working pressure, and the remaining valves may be set up to 5 percent over the maximum allowable working pressure.
2) When sizing multiple relief valves, the total area required is calculated on an overpressure of 16 percent or 4 Psi, whichever is greater.
3) For superheated steam, the correction factor values listed below may be used:
   - Superheat up to 400 °F: 0.97 (Range 0.979–0.998)
   - Superheat up to 450 °F: 0.95 (Range 0.957–0.977)
   - Superheat up to 500 °F: 0.93 (Range 0.930–0.968)

Relief Valve Vent Line Maximum Length

$L = (9 \times P_1^2 \times D^5) / C^2 = (9 \times P_2^2 \times D^5) / (16 \times C^2)$

Where

- P_1 = 0.25 x [(PRESSURE SETTING x 1.1) + 14.7]
- P_2 = [(PRESSURE SETTING x 1.1) + 14.7]
- L = Maximum Length of Relief Vent Line (Feet)
- D = Inside Diameter of Pipe (Inches)
- C = Minimum Discharge of Air (Lbs/Min)

STEEL PIPE EQUATIONS

$A = 0.785 \times ID^2$

$W_P = 10.6802 \times T \times (OD - T)$

$W_W = 0.3405 \times ID^2$

$OSA = 0.2618 \times OD$

$ISA = 0.2618 \times ID$

$A_M = 0.785 \times (OD^2 - ID^2)$

Where

- A = Cross-Sectional Area (Sq- inches)
- W_P = Weight of Pipe per Foot (Lbs)
- W_W = Weight of Water per Foot (Lbs)
- T = Pipe Wall Thickness (Inches)
- ID = Inside Diameter (Inches)
- OD = Outside Diameter (Inches)
- OSA = Outside Surface Area per Foot (Sq-ft)
- ISA = Inside Surface Area per Foot (Sq-ft)
- A_M = Area of the Metal (Sq-inches)

FLOW COEFFICIENT (C_V) - Formulas for Liquids, Steam and Gases

The flow coefficient (C_v) - is important for proper design of control valves, which provides flow comparison of different sizes and types of valve of different manufacturer's. Cv is generally determined experimentally and express the flow capacity - GPM (gallons per minute) of water that a valve will pass for a pressure drop of 1 lb/in^2 (psi). The flow factor (Kv) - is also in common use, but express the capacity in SI-units.

Specific formulas used to estimate Cv for different fluids is indicated below:

Flow Coefficient - *Cv*- for Liquids

For liquids the flow coefficient - C_v - expresses the flow capacity in gallons per minute (GPM) of 60^{o}F water with a pressure drop of 1 psi (lb/in^2).

Flow expressed by volume

$C_v = Q \times (SG / \Delta p)^{1/2}$

Where

- *Q = water flow (US gallons per minute)*

- *SG = specific gravity (1 for water)*
- *Δp = pressure drop (psia)*

Flow is expressed by weight

$C_v = w / (500 \times (\Delta p \times SG)^{1/2})$

Where

- *w = water flow (lb/h)*
- *SG = specific gravity (1 for water)*
- *Δp = pressure drop (psia)*

Flow Coefficient - Cv - for Saturated Steam

Since steam and gases are compressible fluids, the formula must be altered to accommodate changes in the density.

Critical (Choked) Pressure Drop

At choked flow the critical pressure drop the outlet pressure - p_o - from the control valve is less than 58% of the inlet pressure - p_i. The flow coefficient can be expressed as:

$C_v = m / 1.61\, p_i$

Where

- m = steam flow (lb/h)
- p_i = inlet steam absolute pressure (psia)
- p_o = outlet steam absolute pressure (psia)

Non Critical Pressure Drop

For non critical pressure drop the outlet pressure - p_o - from the control valve is greater than 58% of the inlet pressure - p_i. The flow coefficient can be expressed as:

$C_v = m / 3.2\, ((p_i - p_o) \times p_o)^{1/2}$

Flow Coefficient - Cv- for Air and other Gases

For critical pressure drop the outlet pressure - p_o - from the control valve is less than 53% of the inlet pressure - p_i. The flow coefficient can be expressed as:

$C_v = Q \times [SG\, (T + 460)]^{1/2} / 660\, p_i$

Where

- Q = free gas per hour, standard cubic feet per hour (Cu-foot/h)
- SG = specific gravity of flowing gas gas relative to air at 14.7 psia and 60°F
- T = flowing air or gas temperature (°F)
- p_i = inlet gas absolute pressure (psia)

For non critical pressure drop the outlet pressure - p_o - from the control valve is greater than 53% of the inlet pressure - p_i. The flow coefficient can be expressed as:

$C_v = Q \times [SG\ (T + 460)]^{1/2} / [1360\ (\Delta p \times p_o)^{1/2}]$

Where

- $\Delta p = (p_i - p_o)$
- p_o = outlet gas absolute pressure (psia)

ELECTRICITY

- 1 HP (motor) = 0.746 KW (operating energy)
- 5 HP x 0.746 KW/HP x 3413 BTUH/KW = 12,700 BTUH = 1 Ton of Cooling
- Watts = Volts x Amps
- Efficiency = 746 x Output Horsepower (HP) / Input Watts
- KW (1 Phase) = Volts x Amps x Power Factor / 1000
- KW (3 Phase) = Volts x Amps x 1.732 x Power Factor / 1000
- KVA (3 Phase) = 1.732 x Volts x Amps /1000
- BHP (3 Phase) = 1.732 x Volts x Amps x Power Factor x Device Efficiency / 746
- Motor HP = BHP / Motor Efficiency

Motor Drive Formulas

$D_{FP} \times RPM_{FP} = D_{MP} \times RPM_{MP}$

$BL = [(D_{FP} + D_{MP}) \times 1.5708] + (2 \times L)$

Where

- D_{FP} = Fan Pulley Diameter
- D_{MP} = Motor Pulley Diameter
- RPM_{FP} = Fan Pulley RPM
- RPM_{MP} = Motor Pulley RPM

- BL = Belt Length
- L = Center-to-Center Distance of Fan and Motor Pulleys

STANDARD UNITS & CONVERSION

1) Air Flow (Q) - m^3/s
   - $m^3/h = 3600 \times m^3/s$
   - CFM = $0.59 \times m^3/h$
3) Area (A) – m^2
   - $1\ m^2 = 1550\ in^2 = 10.764\ ft^2 = 1.1968\ yd^2 = 3.861\times10^{-7}\ mile^2$
4) Density – kg/m^3
   - $1{,}000\ kg/m^3$ = 62.43 Lbs/Cu-foot = 8.33 Lbs/Gal. = 0.1337 Cu-foot/Gal
   - Standard air (ρ) = $1.2\ kg/m^3$
5) Energy - kJ
   - 1 kWh = 3.6×10^6 J = 859.9 kcal = 2.656×10^6 foot lb_f = 3.412×10^3 Btu
   - 1 kJ = 1 kNm = 1kWs = 10^3 J = 0.947813 Btu = 737.6 foot lb_f = 0.23884 kcal
6) Energy per unit mass
   - 1 kJ/kg = 0.4299 Btu/ lb_m = 0.23884 kcal/kg
7) Heat (H) – KJ/min
   - Sensible heat: Hs (Btu/hr) = 500 x GPM (gallons per min) x ΔT (temperature difference in deg F)
   - Sensible heat: Hs (KJ/hr) = 250.8 x LPM (liters per min) x ΔT (temperature difference in deg C)
   - Sensible heat: Hs (Btu/hr) = 1.08 x CFM (cu-ft per min of air flow) x ΔT (temperature difference in deg F)
   - Sensible heat: Hs (KJ/hr) = 72.42 x CMM (m^3 per min of air flow) x ΔT (temperature difference in deg C)
   - Latent heat: H_L (Btu/hr) = 0.68 x CFM (cu-ft per min of air flow) x ΔW (Humidity Ratio Difference (Grains of H_2O/Lb of dry air)
   - Latent heat: H_L (KJ/hr) = 177734.8 x CMM (m^3 per min of air flow) x ΔW (Humidity Ratio Difference (Kg H_2O/ Kg of dry air)
   - Total heat: H_T (Btu/hr) = 4.5 x CFM (cu-ft per min of air flow) x Δh (Enthalpy Difference Btu/Lb of dry air)

- Total heat: H_T (KJ/hr) = 72.09 x CMM (m^3 per min of air flow) x Δh (Enthalpy Difference KJ/ Kg of dry air)

7) Heat flow rate – kW, kJ/s, kcal/hr

- 1 kW (kJ/s) = 859.9 kcal/h = 3,413 Btu/h = 1.341 hp = 738 foot lb/s = 1,000 J/s = 3.6×10^6 J/h
- 1 kcal/hr = 1.163×10^{-3} kW = 3.969 Btu/h = 1.582×10^{-3} hk = 1.560×10^{-3} hp = 0.8583 foot lb/s

8) Heat flux

- 1 kcal/m^2K = 0.205 Btu/foot$^{2\circ}$F
- 1 kcal/m^2 = 0.369 Btu/foot^2
- 1 Btu/foot^2 = 2.713 kcal/m^2 = 2.043×10^4 J/m^2K

8) Heat Generation per unit Volume

- 1 kcal/m^3 = 0.112 Btu/foot^3
- 1 Btu/foot^3 = 8.9 kcal/m^3 = 3.73×10^4 J/m^3

9) Heat Generation per unit Mass

- 1 Btu/lb = 0.556 kcal/kg = 2,326 J/kg
- 1 kcal/kg = 1.800 Btu/lb

10) Heat transfer coefficient

- 1 Btu/foot^2 h °F = 5.678 W/m^2 K = 4.882 kcal/h m^2 °C
- 1 W/m^2K = 0.85984 kcal/h m^2 °C = 0.1761 Btu/ foot^2 h °F
- 1 kcal/h m^2 °C = 1.163 W/m^2K = 0.205 Btu/ foot^2 h °F

11) Length

- 1 m (meter) = 3.2808 foot = 39.37 in = 1.0936 yd = 6.214×10^{-4} mile

12) Mass, Weight

- 1 kg = 1,000 g = 2.2046 lb
- 1 lb = 16 oz = 0.4536 kg = 453.6 g = 7000 grains

13) Mass Flow Rate

- 1 kg/h = 2.778×10^{-4} kg/s = 3.67×10^{-2} lb/min

14) Power – Watts/kW

- 1 kW = 1,000 Watts = 3,412 Btu/h

- 1 hp (English horse power) = 745.7 W = 0.746 kW = 550 foot lb/s = 2,545 Btu/h = 33.000 foot lb/m

15) Power per unit Area

- 1 W/m^2 = 0.3170 Btu/(h $foot^2$) = 0.85984 kcal/(h m^2)

16) Pressure

- 1 atm = 101.325 kN/m^2 = 101.325 kPa = 1.013 bar = 14.696 psia (lb/in^2) = 0 psig = 29.92 in-Hg =760 mm mercury at 62 ^{0}F (16.7 °C) = 760 torr = 33.95 Foot.H_2O = 407.2 In-wg (Water Gauge) = 2116.8 Lbs/Sq-ft

17) Specific Energy

- 1 J/kg = 0.43 Btu/lb = 2.39×10^{-4} kcal/kg
- 1 kcal/kg = 1.80 Btu/lb = 4,187 J/kg

18) Specific Heat Capacity

- 1 J/ (kg K) = 2.389×10^{-4} kcal/ (kg °C) = 2.389×10^{-4} Btu/ (lb °F)
- 1 kcal/ (kg °C) = 4,186.8 J/ (kg K) = 1 Btu/ (lb °F)

19) Specific Volume

- 1 m^3/kg = 16.02 $foot^3/lb$

20) Temperature (T) - °C

- °C = (°F – 32)/1.8

21) Thermal Conductivity

- 1 W/ (m K) = 0.85984 kcal/ (h m °C) = 0.5779 Btu/ (foot h °F)
- 1 kcal/ (h m °C) = 1.163 W/ (m K) = 0.6720 Btu/ (foot h °F)

22) Thermal Resistance

- 1 (h °F)/Btu = 1.8958 K/W

23) Velocity, Speed- m/s

- 1 m/s = 3.6 km/h = 196.85 foot/min = 2.237 mph

24) Volume

- 1 Gallon (U.S.) = 3.785×10^{-3} m^3 = 3.785 dm^3 (litre) = 0.13368 $foot^3$ = 4.951×10^{-3} yd^3 = 0.8327 Imp. Gal (UK) = 4 Quarts = 8 Pints

25) Volume Flow

- 1 m^3/h = 0.5886 $foot^3$/min (CFM) = 3.667 Imp. Gal (UK)/min = 4.403 gal (US)/min
- 1 GPM = 3.785 LPM (litres per min)