PDHonline Course M216 (4 PDH)

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

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An Approved Continuing Education Provider
HVAC Refresher – Facilities Standard for the Building Services (Part 2)

A. Bhatia

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:

<table>
<thead>
<tr>
<th>Sq-ft/Ton</th>
<th>High</th>
<th>Average</th>
<th>Low</th>
</tr>
</thead>
<tbody>
<tr>
<td>Residential</td>
<td>600</td>
<td>500</td>
<td>380</td>
</tr>
<tr>
<td>Office</td>
<td>360</td>
<td>280</td>
<td>190</td>
</tr>
</tbody>
</table>

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.

<table>
<thead>
<tr>
<th>Application</th>
<th>Average Load</th>
</tr>
</thead>
<tbody>
<tr>
<td>Residence</td>
<td>400-600 sq. ft. floor area per ton</td>
</tr>
<tr>
<td>Apartment (1 or 2 room)</td>
<td>400 sq. ft. of floor area per ton</td>
</tr>
<tr>
<td>Church</td>
<td>20 people per ton</td>
</tr>
<tr>
<td>Office Building</td>
<td></td>
</tr>
<tr>
<td>Large Interior</td>
<td>340 sq. ft. of floor area per ton</td>
</tr>
<tr>
<td>Large Exterior</td>
<td>250 sq. ft. of floor area per ton</td>
</tr>
<tr>
<td>Small Suite</td>
<td>280 sq. ft. of floor area per ton</td>
</tr>
<tr>
<td>Application</td>
<td>Average Load</td>
</tr>
<tr>
<td>------------------------------</td>
<td>----------------------------------</td>
</tr>
<tr>
<td>Restaurant</td>
<td>200 sq. ft. of floor area per ton</td>
</tr>
<tr>
<td>Bar or Tavern</td>
<td>9 people per ton</td>
</tr>
<tr>
<td>Cocktail Lounge</td>
<td>175 sq. ft. of floor area per ton</td>
</tr>
<tr>
<td>Computer Room</td>
<td>50 – 150 sq. ft. of floor area per ton</td>
</tr>
<tr>
<td>Bank (main area)</td>
<td>225 sq. ft. of floor area per ton</td>
</tr>
<tr>
<td>Barber Shop</td>
<td>250 sq. ft. of floor area per ton</td>
</tr>
<tr>
<td>Beauty Shop</td>
<td>180 sq. ft. of floor area per ton</td>
</tr>
<tr>
<td>School Classroom</td>
<td>250 sq. ft. of floor area per ton</td>
</tr>
<tr>
<td>Bowling Alley</td>
<td>1.5 – 2.5 tons per alley</td>
</tr>
<tr>
<td>Department Store</td>
<td></td>
</tr>
<tr>
<td>Basement</td>
<td>350 sq. ft. of floor area per ton</td>
</tr>
<tr>
<td>Main Floor</td>
<td>300 sq. ft. of floor area per ton</td>
</tr>
<tr>
<td>Upper Floor</td>
<td>400 sq. ft. of floor area per ton</td>
</tr>
<tr>
<td>Small Shop</td>
<td>225 sq. ft. of floor area per ton</td>
</tr>
<tr>
<td>Dress Shop</td>
<td>280 sq. ft. of floor area per ton</td>
</tr>
<tr>
<td>Drug Store</td>
<td>150 sq. ft. of floor area per ton</td>
</tr>
<tr>
<td>Factory (precision manufacturing)</td>
<td>275 sq. ft. of floor area per ton</td>
</tr>
<tr>
<td>Groceries – Supermarket</td>
<td>350 sq. ft. of floor area per ton</td>
</tr>
<tr>
<td>Hospital Room</td>
<td>280 sq. ft. of floor area per ton</td>
</tr>
<tr>
<td>Hotel Public Spaces</td>
<td>220 sq. ft. of floor area per ton</td>
</tr>
<tr>
<td>Motel</td>
<td>400 sq. ft. of floor area per ton</td>
</tr>
<tr>
<td>Auditorium or Theatre</td>
<td>20 people per ton</td>
</tr>
<tr>
<td>Shoe Store</td>
<td>220 sq. ft. of floor area per ton</td>
</tr>
<tr>
<td>Specialty &amp; Variety Store</td>
<td>200 sq. ft. of floor area per ton</td>
</tr>
</tbody>
</table>

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.

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

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)

<table>
<thead>
<tr>
<th>Equipment Type</th>
<th>Approximate Airflow Rate</th>
<th>Example</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas/Oil Furnace</td>
<td>1 CFM per 100 Btu/hr output</td>
<td>64000 Btu/hr output furnace = 640CFM</td>
</tr>
</tbody>
</table>
| 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

<table>
<thead>
<tr>
<th>Capacity Range (TR)</th>
<th>Favourable System</th>
</tr>
</thead>
<tbody>
<tr>
<td>40 to 200</td>
<td>Air-cooled chilled water system (explore the pros and cons of using multiple DX systems if possible)</td>
</tr>
<tr>
<td>200 and above</td>
<td>Water-cooled chilled water system</td>
</tr>
</tbody>
</table>

CHARACTERISTICS & TYPICAL APPLICATIONS OF VARIOUS COOLING SYSTEMS
<table>
<thead>
<tr>
<th>Characteristics</th>
<th>Air-Cooled Packaged Equipment</th>
<th>Water-Cooled Packaged Equipment</th>
<th>Air-Cooled Chilled-Water System</th>
<th>Water-Cooled Chilled-Water System</th>
</tr>
</thead>
<tbody>
<tr>
<td>Building Height</td>
<td>Typically limited to 1- to 4-story buildings</td>
<td>Unlimited</td>
<td>Unlimited</td>
<td>Unlimited</td>
</tr>
<tr>
<td>Minimum Cooling Capacity</td>
<td>No limitation for modular systems</td>
<td>Typically cost-effective for projects &gt;20 tons</td>
<td>Typically cost-effective for projects &gt;100 tons</td>
<td>Typically cost-effective for projects &gt;200 tons</td>
</tr>
<tr>
<td>Cooling Control</td>
<td>Low</td>
<td>Low-moderate</td>
<td>High</td>
<td>High</td>
</tr>
<tr>
<td>Maintenance</td>
<td>Low</td>
<td>Moderate-high</td>
<td>Moderate</td>
<td>High</td>
</tr>
<tr>
<td>Installed Cost</td>
<td>Low</td>
<td>Moderate-high</td>
<td>High</td>
<td>High</td>
</tr>
<tr>
<td>Operating Costs</td>
<td>Moderate</td>
<td>Low-moderate (climate dependent)</td>
<td>Moderate-high</td>
<td>Low</td>
</tr>
<tr>
<td>Typical Applications</td>
<td>1- to 2-story buildings</td>
<td>1- to 2-story buildings in hot/dry climates</td>
<td>Medium to large facilities with limited access to water or maintenance</td>
<td>Medium to very large facilities and campuses</td>
</tr>
</tbody>
</table>

**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

<table>
<thead>
<tr>
<th>kW/ton = 12 / EER</th>
</tr>
</thead>
<tbody>
<tr>
<td>kW/ton = 12 / (COP x 3.412)</td>
</tr>
<tr>
<td>EER = 12 / (kW/ton)</td>
</tr>
<tr>
<td>EER = COP x 3.412</td>
</tr>
<tr>
<td>COP = EER / 3.412</td>
</tr>
<tr>
<td>COP = 12 / (kW/ton x 3.412)</td>
</tr>
</tbody>
</table>

**RECOMMENDED EFFICIENCY VALUES FOR UNITARY & APPLIED HEAT PUMPS**

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

**RECOMMENDED CHILLER PERFORMANCE LEVELS**
### ELECTRIC UTILIZATION INDEX (EUI)

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

<table>
<thead>
<tr>
<th>Type of Building</th>
<th>Common EUI</th>
</tr>
</thead>
<tbody>
<tr>
<td>Grocery</td>
<td>61.0</td>
</tr>
<tr>
<td>Restaurant</td>
<td>38.9</td>
</tr>
<tr>
<td>Hospital / Health</td>
<td>16.4</td>
</tr>
<tr>
<td>Retail</td>
<td>12.1</td>
</tr>
<tr>
<td>School / College</td>
<td>10.3</td>
</tr>
<tr>
<td>Hotel / Motel</td>
<td>8.2</td>
</tr>
<tr>
<td>Office</td>
<td>7.5</td>
</tr>
<tr>
<td>Misc. Commercial</td>
<td>6.4</td>
</tr>
<tr>
<td>Warehouse</td>
<td>6.1</td>
</tr>
</tbody>
</table>
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

<table>
<thead>
<tr>
<th>Application</th>
<th>Occupancy (people/1000ft²)</th>
<th>CFM/person</th>
<th>CFM/ft²</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Food and Beverage Service</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Dining rooms</td>
<td>70</td>
<td>20</td>
<td></td>
</tr>
<tr>
<td>Cafeteria, fast food</td>
<td>100</td>
<td>20</td>
<td></td>
</tr>
<tr>
<td>Bars, cocktail lounges</td>
<td>100</td>
<td>30</td>
<td></td>
</tr>
<tr>
<td>Kitchen (cooking)</td>
<td>20</td>
<td>15</td>
<td></td>
</tr>
<tr>
<td><strong>Offices</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Office space</td>
<td>7</td>
<td>20</td>
<td></td>
</tr>
<tr>
<td>Reception areas</td>
<td>60</td>
<td>15</td>
<td></td>
</tr>
<tr>
<td>Conference rooms</td>
<td>50</td>
<td>20</td>
<td></td>
</tr>
<tr>
<td><strong>Public Spaces</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Smoking lounge</td>
<td>70</td>
<td>60</td>
<td>1.00</td>
</tr>
<tr>
<td>Elevators</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Retail stores, Showrooms</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Basement &amp; Street</td>
<td>30</td>
<td>30</td>
<td>0.30</td>
</tr>
<tr>
<td>Upper floors</td>
<td>20</td>
<td>20</td>
<td>0.20</td>
</tr>
<tr>
<td>Malls and arcades</td>
<td>20</td>
<td>20</td>
<td>0.20</td>
</tr>
<tr>
<td>Smoking lounges</td>
<td>70</td>
<td>60</td>
<td></td>
</tr>
<tr>
<td>Beauty shops</td>
<td>25</td>
<td>25</td>
<td></td>
</tr>
<tr>
<td>Hardware stores</td>
<td>8</td>
<td>15</td>
<td></td>
</tr>
<tr>
<td><strong>Sports and Amusements</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Spectator areas</td>
<td>150</td>
<td>15</td>
<td></td>
</tr>
<tr>
<td>Games rooms</td>
<td>70</td>
<td>25</td>
<td></td>
</tr>
<tr>
<td>Playing rooms</td>
<td>30</td>
<td>20</td>
<td></td>
</tr>
<tr>
<td>Ballrooms and discos</td>
<td>100</td>
<td>25</td>
<td></td>
</tr>
<tr>
<td>Application</td>
<td>Occupancy (people/1000ft²)</td>
<td>CFM/person</td>
<td>CFM/ft²</td>
</tr>
<tr>
<td>---------------------------------</td>
<td>-----------------------------</td>
<td>------------</td>
<td>---------</td>
</tr>
<tr>
<td><strong>Theatres</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Lobbies</td>
<td>150</td>
<td>20</td>
<td></td>
</tr>
<tr>
<td>Auditorium</td>
<td>150</td>
<td>15</td>
<td></td>
</tr>
<tr>
<td><strong>Education</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Classroom</td>
<td>50</td>
<td>15</td>
<td></td>
</tr>
<tr>
<td>Music rooms</td>
<td>50</td>
<td>15</td>
<td></td>
</tr>
<tr>
<td>Libraries</td>
<td>20</td>
<td>15</td>
<td></td>
</tr>
<tr>
<td>Auditoriums</td>
<td>150</td>
<td>15</td>
<td></td>
</tr>
<tr>
<td><strong>Hotels, Motels, Resorts, Dormitories</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Bedrooms</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Living rooms</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Lobbies</td>
<td>30</td>
<td>15</td>
<td></td>
</tr>
<tr>
<td>Conference rooms</td>
<td>50</td>
<td>20</td>
<td></td>
</tr>
<tr>
<td>Assembly rooms</td>
<td>120</td>
<td>15</td>
<td></td>
</tr>
<tr>
<td>Dry cleaning, laundry</td>
<td>30</td>
<td>30</td>
<td></td>
</tr>
<tr>
<td>Gambling casinos</td>
<td>120</td>
<td>30</td>
<td></td>
</tr>
<tr>
<td><strong>Health Care Facilities</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Operating rooms</td>
<td>20</td>
<td>30</td>
<td></td>
</tr>
<tr>
<td>Patient rooms</td>
<td>10</td>
<td>25</td>
<td></td>
</tr>
<tr>
<td>Laboratories</td>
<td>30</td>
<td>20</td>
<td></td>
</tr>
<tr>
<td>Procedure rooms</td>
<td>20</td>
<td>15</td>
<td></td>
</tr>
<tr>
<td>Pharmacies</td>
<td>20</td>
<td>15</td>
<td></td>
</tr>
<tr>
<td>Physical therapy</td>
<td>20</td>
<td>15</td>
<td></td>
</tr>
</tbody>
</table>

**EXHAUST AIR REQUIREMENTS**

<table>
<thead>
<tr>
<th>Exhaust Air Requirements</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Janitor Closets</td>
<td>10 Air changes/hr</td>
</tr>
<tr>
<td>Locker Rooms</td>
<td>10 Air changes/hr</td>
</tr>
<tr>
<td>Toilets</td>
<td>10 Air changes/hr</td>
</tr>
<tr>
<td>Mechanical/Electrical Rooms</td>
<td>12 Air changes/hr</td>
</tr>
<tr>
<td>Rooms with Steam System (Laundry)</td>
<td>25 Air changes/hr</td>
</tr>
<tr>
<td>Battery Rooms</td>
<td>10 Air changes/hr</td>
</tr>
</tbody>
</table>

**TYPICAL DESIGN VELOCITIES FOR HVAC COMPONENTS**

<table>
<thead>
<tr>
<th>Equipment</th>
<th>Velocity, Feet per minute (FPM)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Intake Louvers Velocity (7000 CFM and greater)</td>
<td>400 FPM</td>
</tr>
</tbody>
</table>
### Equipment

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

---

**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.

### CENTRIFUGAL FAN PARAMETERS

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Backward Curve</th>
<th>Forward Curve</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>BC</td>
<td>BI</td>
</tr>
<tr>
<td>Blades</td>
<td>6-16</td>
<td>6-16</td>
</tr>
<tr>
<td>Maximum Efficiency (%)</td>
<td>78</td>
<td>85</td>
</tr>
<tr>
<td>Speed</td>
<td>High</td>
<td>High</td>
</tr>
</tbody>
</table>
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.

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

FAN PERFORMANCE RELATIONSHIPS

<table>
<thead>
<tr>
<th>Variable</th>
<th>Constant</th>
<th>Law</th>
<th>Equation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotational Speed</td>
<td>Fan Size</td>
<td>Flow is directly proportional to speed</td>
<td>( \frac{Q_1}{Q_2} = \frac{N_1}{N_2} )</td>
</tr>
<tr>
<td></td>
<td>Air Density</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Duct System</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pressure is directly proportional to speed²</td>
<td>( \frac{P_1}{P_2} = \left[\frac{N_1}{N_2}\right]^2 )</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Power is directly proportional to speed³</td>
<td>( \frac{HP_1}{HP_2} = \left[\frac{N_1}{N_2}\right]^3 )</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Fan Size and Rotational Speed</td>
<td>Tip Speed</td>
<td>Flow and power is directly proportional to diameter²</td>
<td>( \frac{Q_1}{Q_2} = \frac{HP_1}{HP_2} = \left[\frac{D_1}{D_2}\right]^2 )</td>
</tr>
</tbody>
</table>
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 | Flow is directly proportional to Diameter² | $(Q_1 / Q_2) = [(D_1 / D_2)^2$}
| Air Density | Flow is directly proportional to Diameter² | $(P_1 / P_2) = [(D_1 / D_2)^2$}
| Flow is directly proportional to Diameter³ | $(HP_1 / HP_2) = [(D_1 / D_2)^3$}
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 | Pressure and power are directly proportional to density | $(P_1 / P_2) = (HP_1 / HP_2) = (\rho_1 / \rho_2)$
| Fan Size | Flow remains constant | $Q_1 = Q_2$
| Duct System | 

**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).

<table>
<thead>
<tr>
<th>Type of Outlet</th>
<th>Floor Space</th>
<th>Approximate maximum air changes/hour for 10 feet ceiling</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>CFM per Sq Feet</strong></td>
<td>Lps per Sq-m</td>
<td></td>
</tr>
<tr>
<td>Grilles &amp; Registers</td>
<td>0.6 to 1.2</td>
<td>3 to 6</td>
</tr>
<tr>
<td>Slot Diffusers</td>
<td>0.8 to 2.0</td>
<td>4 to 10</td>
</tr>
<tr>
<td>Perforated Panel</td>
<td>0.9 to 3.0</td>
<td>5 to 15</td>
</tr>
<tr>
<td>Ceiling Diffuser</td>
<td>0.9 to 5.0</td>
<td>5 to 25</td>
</tr>
<tr>
<td>Perforated Ceiling</td>
<td>1.0 to 10.0</td>
<td>5 to 50</td>
</tr>
</tbody>
</table>

**REFRIGERANTS & ENVIRONMENTAL FACTORS**

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

<table>
<thead>
<tr>
<th>Criteria</th>
<th>HCFC-123</th>
<th>HCFC-22</th>
<th>HFC-134a</th>
<th>Ammonia</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ozone Depletion Potential</td>
<td>0.016</td>
<td>0.05</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>
### CURRENT & FUTURE REFRIGERANTS

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

### RECOMMENDED SHEET METAL THICKNESS FOR DUCTS

<table>
<thead>
<tr>
<th>Greatest Dimension</th>
<th>Rectangular Duct</th>
<th>Round Duct</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Galvanized Steel (gauge)</td>
<td>Aluminum (gauge)</td>
</tr>
<tr>
<td>Up to 30 inch</td>
<td>24</td>
<td>22</td>
</tr>
<tr>
<td>31 – 60 inches</td>
<td>22</td>
<td>20</td>
</tr>
<tr>
<td>61 – 90 inches</td>
<td>20</td>
<td>18</td>
</tr>
<tr>
<td>91 inches and above</td>
<td>18</td>
<td>16</td>
</tr>
</tbody>
</table>

### SHEET METAL THICKNESS & WEIGHTS
Note: Aluminium is specified and purchased by material thickness rather than gauge.

**DUCTWORK AIR CARRYING CAPACITY**

<table>
<thead>
<tr>
<th>Branch Duct Size</th>
<th>Avg. CFM @ Static Pressure</th>
<th>Duct Cross-section</th>
</tr>
</thead>
<tbody>
<tr>
<td>4&quot; Round</td>
<td>30 CFM</td>
<td>12.57 Sq-in</td>
</tr>
<tr>
<td>5&quot; Round</td>
<td>60 CFM</td>
<td>19.64 Sq-in</td>
</tr>
<tr>
<td>2 ¼&quot; x 10&quot;</td>
<td>60 CFM</td>
<td>23.00 Sq-in</td>
</tr>
<tr>
<td>2 ¼&quot; x 12&quot;</td>
<td>70 CFM</td>
<td>27.00 Sq-in</td>
</tr>
<tr>
<td>6&quot; Round</td>
<td>100 CFM</td>
<td>28.27 Sq-in</td>
</tr>
<tr>
<td>3 ½&quot; x 10&quot;</td>
<td>100 CFM</td>
<td>33.00 Sq-in</td>
</tr>
<tr>
<td>3 ½&quot; x 12&quot;</td>
<td>120 CFM</td>
<td>39.00 Sq-in</td>
</tr>
<tr>
<td>7&quot; Round</td>
<td>150 CFM</td>
<td>38.48 Sq-in</td>
</tr>
<tr>
<td>Pipe Size</td>
<td>Flow Rate</td>
<td>Heating BTUH</td>
</tr>
<tr>
<td>------------</td>
<td>-----------</td>
<td>--------------</td>
</tr>
<tr>
<td>3 ¼&quot; x 14&quot;</td>
<td>140 CFM</td>
<td>46.00 Sq-in</td>
</tr>
<tr>
<td>8&quot; Round</td>
<td>200 CFM</td>
<td>50.27 Sq-in</td>
</tr>
<tr>
<td>8&quot; x 8&quot;</td>
<td>260 CFM</td>
<td>64.00 Sq-in</td>
</tr>
<tr>
<td>10&quot; Round</td>
<td>400 CFM</td>
<td>78.54 Sq-in</td>
</tr>
<tr>
<td>12&quot; x 8&quot;</td>
<td>440 CFM</td>
<td>96.00 Sq-in</td>
</tr>
<tr>
<td>12&quot;</td>
<td>620 CFM</td>
<td>113.09 Sq-in</td>
</tr>
<tr>
<td>16&quot; x 8&quot;</td>
<td>660 CFM</td>
<td>128.00 Sq-in</td>
</tr>
<tr>
<td>14&quot; Round</td>
<td>930 CFM</td>
<td>153.93 Sq-in</td>
</tr>
<tr>
<td>16&quot; Round</td>
<td>1300 CFM</td>
<td>201.06 Sq-in</td>
</tr>
</tbody>
</table>

---

**PIPE SELECTION**

<table>
<thead>
<tr>
<th>Pipe Size</th>
<th>Flow Rate</th>
<th>Heating BTUH</th>
<th>Cooling Tons</th>
<th>Flow Rate</th>
<th>Heating BTUH</th>
<th>Cooling Tons</th>
</tr>
</thead>
<tbody>
<tr>
<td>1/2&quot;</td>
<td>1.8 GPM</td>
<td>18,000 BTUH</td>
<td>1.5 Tons</td>
<td>1.5 GPM</td>
<td>15,000 BTUH</td>
<td>1.3 Tons</td>
</tr>
<tr>
<td>3/4&quot;</td>
<td>4 GPM</td>
<td>40,000 BTUH</td>
<td>3.3 Tons</td>
<td>3.5 GPM</td>
<td>35,000 BTUH</td>
<td>2.9 Tons</td>
</tr>
<tr>
<td>1&quot;</td>
<td>8 GPM</td>
<td>80,000 BTUH</td>
<td>6.7 Tons</td>
<td>7.5 GPM</td>
<td>75,000 BTUH</td>
<td>6.3 Tons</td>
</tr>
<tr>
<td>1 1/4&quot;</td>
<td>16 GPM</td>
<td>160,000 BTUH</td>
<td>13.3 Tons</td>
<td>13 GPM</td>
<td>130,000 BTUH</td>
<td>10.8 Tons</td>
</tr>
<tr>
<td>1 1/2&quot;</td>
<td>24 GPM</td>
<td>240,000 BTUH</td>
<td>20 Tons</td>
<td>20 GPM</td>
<td>200,000 BTUH</td>
<td>16.7 Tons</td>
</tr>
<tr>
<td>2&quot;</td>
<td>47 GPM</td>
<td>470,000 BTUH</td>
<td>39 Tons</td>
<td>45 GPM</td>
<td>450,000 BTUH</td>
<td>38 Tons</td>
</tr>
<tr>
<td>2 1/2&quot;</td>
<td>75 GPM</td>
<td>750,000 BTUH</td>
<td>63 Tons</td>
<td>80 GPM</td>
<td>800,000 BTUH</td>
<td>67 Tons</td>
</tr>
<tr>
<td>3&quot;</td>
<td>130 GPM</td>
<td>1,300,000 BTUH</td>
<td>108 Tons</td>
<td>130 GPM</td>
<td>1,300,000 BTUH</td>
<td>108 Tons</td>
</tr>
<tr>
<td>4&quot;</td>
<td>270 GPM</td>
<td>2,700,000 BTUH</td>
<td>225 Tons</td>
<td>260 GPM</td>
<td>2,600,000 BTUH</td>
<td>217 Tons</td>
</tr>
<tr>
<td>5&quot;</td>
<td>530 GPM</td>
<td>5,300,000 BTUH</td>
<td>442 Tons</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
- Heating capacity BTUH based on a 20 degree F temperature differential. Cooling capacity BTUH based on 10 to 16°F 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 20°F)
- 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.

<table>
<thead>
<tr>
<th>Cleanroom Class</th>
<th>Airflow Type</th>
<th>Av. Airflow Velocity, fpm</th>
<th>Air changes/hr</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Unidirectional</td>
<td>70-100</td>
<td>350-650</td>
</tr>
<tr>
<td>10</td>
<td>Unidirectional</td>
<td>60-110</td>
<td>300-600</td>
</tr>
<tr>
<td>100</td>
<td>Unidirectional</td>
<td>50-90</td>
<td>300-480</td>
</tr>
<tr>
<td>1,000</td>
<td>Mixed</td>
<td>40-90</td>
<td>150-250</td>
</tr>
<tr>
<td>10,000</td>
<td>Mixed</td>
<td>25-40</td>
<td>60-120</td>
</tr>
<tr>
<td>100,000</td>
<td>Mixed</td>
<td>10-30</td>
<td>10-40</td>
</tr>
</tbody>
</table>

## 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

<table>
<thead>
<tr>
<th>Number of Sources</th>
<th>Increase in Sound Power Level (dB)</th>
<th>Increase in Sound Pressure Level dB</th>
</tr>
</thead>
<tbody>
<tr>
<td>2</td>
<td>3</td>
<td>6</td>
</tr>
<tr>
<td>3</td>
<td>4.8</td>
<td>9.6</td>
</tr>
<tr>
<td>4</td>
<td>6</td>
<td>12</td>
</tr>
<tr>
<td>5</td>
<td>7</td>
<td>14</td>
</tr>
<tr>
<td>10</td>
<td>10</td>
<td>20</td>
</tr>
<tr>
<td>15</td>
<td>11.8</td>
<td>23.6</td>
</tr>
<tr>
<td>20</td>
<td>13</td>
<td>26</td>
</tr>
</tbody>
</table>

### Adding Sound Power from Sources at different Levels

<table>
<thead>
<tr>
<th>Sound Power Level Difference between two Sound Sources (dB)</th>
<th>Added Decibel to the Highest Sound Power Level (dB)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>3</td>
</tr>
<tr>
<td>1</td>
<td>2.5</td>
</tr>
<tr>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>3</td>
<td>2</td>
</tr>
<tr>
<td>4</td>
<td>1.5</td>
</tr>
<tr>
<td>5</td>
<td>1</td>
</tr>
<tr>
<td>6</td>
<td>1</td>
</tr>
<tr>
<td>7</td>
<td>1</td>
</tr>
<tr>
<td>8</td>
<td>0.5</td>
</tr>
<tr>
<td>9</td>
<td>0.5</td>
</tr>
<tr>
<td>10 or more</td>
<td>0</td>
</tr>
</tbody>
</table>

### 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

<table>
<thead>
<tr>
<th>Area</th>
<th>Maximum NC</th>
</tr>
</thead>
<tbody>
<tr>
<td>Conference Rooms</td>
<td>NC 35</td>
</tr>
<tr>
<td>Corridors</td>
<td>NC 40</td>
</tr>
<tr>
<td>Lobby</td>
<td>NC 40</td>
</tr>
<tr>
<td>Large Offices &amp; Computer Rooms</td>
<td>NC 40</td>
</tr>
<tr>
<td>Small Private Office</td>
<td>NC 35</td>
</tr>
</tbody>
</table>

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

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

GLAZING PROPERTIES
<table>
<thead>
<tr>
<th>Material</th>
<th>“U” Value (Btu / hr-ft²-°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Glass, single</td>
<td>1.13</td>
</tr>
<tr>
<td>Glass, double glazing</td>
<td>.70</td>
</tr>
<tr>
<td>Single film plastic</td>
<td>1.20</td>
</tr>
<tr>
<td>Double film plastic</td>
<td>.70</td>
</tr>
<tr>
<td>Corrugated FRP panels</td>
<td>1.20</td>
</tr>
<tr>
<td>Corrugated polycarbonate</td>
<td>1.20</td>
</tr>
<tr>
<td>Plastic structured sheet;</td>
<td></td>
</tr>
<tr>
<td>16 mm thick</td>
<td>.58</td>
</tr>
<tr>
<td>8 mm thick</td>
<td>.65</td>
</tr>
<tr>
<td>6 mm thick</td>
<td>.72</td>
</tr>
<tr>
<td>Concrete block, 8 inch</td>
<td>.51</td>
</tr>
</tbody>
</table>

**ROOF INSULATION**

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

<table>
<thead>
<tr>
<th>Existing Condition</th>
<th>Is it cost effective to add insulation?</th>
</tr>
</thead>
<tbody>
<tr>
<td>No insulation to R-6</td>
<td>Yes, always</td>
</tr>
<tr>
<td>R-7 to R-19</td>
<td>Yes, if attic is accessible or if built-up roof is replaced</td>
</tr>
<tr>
<td>Greater than R-19</td>
<td>Not usually cost effective</td>
</tr>
</tbody>
</table>

**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.

<table>
<thead>
<tr>
<th>Region</th>
<th>Item</th>
<th>Energy Star</th>
</tr>
</thead>
<tbody>
<tr>
<td>North</td>
<td>Windows and Doors</td>
<td>0.35 / -</td>
</tr>
<tr>
<td></td>
<td>U factor / SHGC</td>
<td></td>
</tr>
<tr>
<td>North</td>
<td>Skylights, U factor / SHGC</td>
<td>0.45 / -</td>
</tr>
<tr>
<td>Region</td>
<td>Item</td>
<td>Energy Star</td>
</tr>
<tr>
<td>--------------------------</td>
<td>-------------------------------</td>
<td>--------------</td>
</tr>
<tr>
<td>Central (Heating and</td>
<td>Windows and Doors U factor</td>
<td>0.40 / 0.55</td>
</tr>
<tr>
<td>Cooling)</td>
<td>/ SHGC</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Skylights, U factor / SHGC</td>
<td>0.45 / 0.55</td>
</tr>
<tr>
<td>South (Mostly Cooling)</td>
<td>Windows and Doors U factor</td>
<td>0.75 / 0.40</td>
</tr>
<tr>
<td></td>
<td>/ SHGC</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Skylights, U factor / SHGC</td>
<td>0.75 / 0.40</td>
</tr>
</tbody>
</table>

**LIGHTING WATTAGE ESTIMATION**

<table>
<thead>
<tr>
<th>Location</th>
<th>Rule of thumb (Watts/sq-ft)</th>
</tr>
</thead>
<tbody>
<tr>
<td>General Office Areas</td>
<td>1.5 to 3.0</td>
</tr>
<tr>
<td>Private</td>
<td>2.0 - 5.0</td>
</tr>
<tr>
<td>Conference Rooms</td>
<td>2.0 – 6.0</td>
</tr>
<tr>
<td>Public Places (Banks, Post offices, Courts etc)</td>
<td>2.0 – 5.0</td>
</tr>
<tr>
<td>Precision Manufacturing</td>
<td>3.0 – 10.0</td>
</tr>
<tr>
<td>Computer Rooms/Data Processing Facilities</td>
<td>2.0 – 5.0</td>
</tr>
<tr>
<td>Restaurants</td>
<td>1.5 – 3.0</td>
</tr>
<tr>
<td>Kitchens</td>
<td>1.5 – 2.5</td>
</tr>
<tr>
<td>Pubs, Bars, Clubhouses, Taverns etc</td>
<td>1.5 – 2.0</td>
</tr>
<tr>
<td>Hospital Patient Rooms</td>
<td>1.0 – 2.0</td>
</tr>
<tr>
<td>Hospital General Areas</td>
<td>1.5 – 2.5</td>
</tr>
<tr>
<td>Medical /Dental Centres, Clinics</td>
<td>1.5 – 2.5</td>
</tr>
<tr>
<td>Residences</td>
<td>1.0 – 4.0</td>
</tr>
<tr>
<td>Hotel &amp; Motels (public places and guest rooms)</td>
<td>1.0 – 3.0</td>
</tr>
<tr>
<td>School Classrooms</td>
<td>2.0 – 6.0</td>
</tr>
<tr>
<td>Dining halls, Lunch Rooms, Cafeterias</td>
<td>1.5 – 2.5</td>
</tr>
<tr>
<td>Library, Museums</td>
<td>1.0 – 3.0</td>
</tr>
<tr>
<td>Location</td>
<td>Rule of thumb (Watts/sq-ft)</td>
</tr>
<tr>
<td>----------------------------------</td>
<td>-----------------------------</td>
</tr>
<tr>
<td>Retail, Department &amp; Pharmacist Stores</td>
<td>2.0 – 6.0</td>
</tr>
<tr>
<td>Jewellery Showrooms, Shoes, Boutiques etc</td>
<td>2.0 – 4.0</td>
</tr>
<tr>
<td>Shopping Malls</td>
<td>2.0 – 4.0</td>
</tr>
<tr>
<td>Auditoriums, Theatres</td>
<td>1.0 – 3.0</td>
</tr>
<tr>
<td>Religious Places (Churches)</td>
<td>1.0 – 3.0</td>
</tr>
<tr>
<td>Bowling Alleys</td>
<td>1.0 – 2.5</td>
</tr>
</tbody>
</table>

HEAT LOAD FROM OFFICE EQUIPMENT

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

<table>
<thead>
<tr>
<th>Equipment</th>
<th>Continuous</th>
<th>Average</th>
<th>Idle</th>
</tr>
</thead>
<tbody>
<tr>
<td>Computer - 15” Monitor</td>
<td>110</td>
<td>-</td>
<td>20</td>
</tr>
<tr>
<td>- 17” Monitor</td>
<td>125</td>
<td>-</td>
<td>25</td>
</tr>
<tr>
<td>- 19” Monitor</td>
<td>135</td>
<td>-</td>
<td>30</td>
</tr>
<tr>
<td>Laser Printer - Desktop</td>
<td>130</td>
<td>100</td>
<td>10</td>
</tr>
<tr>
<td>- Small Office</td>
<td>320</td>
<td>160</td>
<td>70</td>
</tr>
<tr>
<td>- Large Office</td>
<td>550</td>
<td>275</td>
<td>125</td>
</tr>
<tr>
<td>Fax Machine</td>
<td>30</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Coffee Maker - 10 cup</td>
<td>1050 +</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Microwave Oven - 1 ft³</td>
<td>400</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Refrigerator - 15 ft³</td>
<td>300</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Water Cooler - 3 gal/hr</td>
<td>350</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1/4 hp Motor</td>
<td>270</td>
<td></td>
<td></td>
</tr>
<tr>
<td>5/8 hp Motor</td>
<td>750</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1 hp motor</td>
<td>930</td>
<td></td>
<td></td>
</tr>
<tr>
<td>10 hp Motor</td>
<td>8200</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

(Watts = 746 x hp/\eta_{Motor})

RATE OF HEAT GAIN FROM MISCELLANEOUS APPLIANCES
### Electrical Appliances

<table>
<thead>
<tr>
<th>Manufacturer’s Rating</th>
<th>Recommended Rate of Heat Gain, Btu/h</th>
</tr>
</thead>
<tbody>
<tr>
<td>Watts</td>
<td>Btu/h</td>
</tr>
<tr>
<td>-----------------------</td>
<td>------------------------------------</td>
</tr>
<tr>
<td>Hair dryer</td>
<td>1,580</td>
</tr>
<tr>
<td>Hair dryer</td>
<td>705</td>
</tr>
<tr>
<td>Neon sign, per linear ft of tube</td>
<td>30</td>
</tr>
<tr>
<td>Sterilizer, instrument</td>
<td>1,100</td>
</tr>
</tbody>
</table>

#### Gas-Burning Appliances

| Lab burners | 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**

<table>
<thead>
<tr>
<th>POLES</th>
<th>60 CYCLES</th>
<th>50 CYCLES</th>
</tr>
</thead>
<tbody>
<tr>
<td>2</td>
<td>3600</td>
<td>3000</td>
</tr>
<tr>
<td>4</td>
<td>1800</td>
<td>1500</td>
</tr>
<tr>
<td>6</td>
<td>1200</td>
<td>1000</td>
</tr>
<tr>
<td>8</td>
<td>900</td>
<td>750</td>
</tr>
<tr>
<td>10</td>
<td>720</td>
<td>600</td>
</tr>
</tbody>
</table>

---
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:

\[ H = U \times A \times (CLTD) \]

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:

\[ H = U \times A \times (CLTD) \]

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:

\[ H = A \times (SHGC) \times (SC) \times (CLF) \]

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

**Partitions, Ceilings & Floors**

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

\[ H = U \times A \times (Ta - Tr) \]

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, 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.

\[ H_{sensible} = 1.08 \times CFM \times (T_o - T_c) \]
\[ H_{latent} = 0.68 \times CFM \times \Delta W_{GR} \]
\[ H_{latent} = 4840 \times CFM \times \Delta W_{Lb} \]
\[ H_{total} = 4.5 \times CFM \times (h_o - h_c) \]
\[ H_{total} = H_{sensible} + H_{latent} \]

Where
• $H_{\text{sensible}}$ = Sensible heat gain (Btu/hr)
• $H_{\text{latent}}$ = Latent heat gain (Btu/hr)
• $H_{\text{total}}$ = 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
• $\Delta W_{GR}$ = Humidity Ratio Difference (Gr H₂O/Lb of dry air) = $(W_o - W_c)$
• $\Delta W_{LB}$ = Humidity Ratio Difference (Lb H₂O /Lb of dry air) and = $(W_o - W_c)$
• $W_o$ = Outside humidity ratio, Lb H₂O per Lb (dry air)
• $W_c$ = Humidity ratio of air leaving the cooling coil, Lb H₂O 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.

\[
H_{\text{sensible}} = N \times (H_s) \times (CLF)
\]
\[
H_{\text{latent}} = N \times (H_l)
\]

Where

• $H_{\text{sensible}}$ = Total Sensible heat gain (Btu/hr)
• $H_{\text{latent}}$ = 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.

\[ H = 3.41 \times W \times F_{UT} \times F_{BF} \times (CLF) \]

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

\[ H = 2545 \times \left( \frac{P}{Eff} \right) \times F_{UM} \times F_{LM} \]

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

\[ H = 2545 \times P \times F_{UM} \times F_{LM} \]

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\text{UM} = Motor use factor
• F\text{LM} = Motor load factor
• Note: F\text{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

\[ H = 2545 * P * \left[ \frac{(1.0-Eff)}{Eff} \right] * F_{\text{UM}} * F_{\text{LM}} \]

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\text{UM} = Motor use factor
• F\text{LM} = Motor load factor
• Note: F\text{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:

\[ H = k \times A \times \frac{\Delta T}{t} \]

Where

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

**R-Values/U-Values**

\[ R = \frac{1}{C} = \frac{1}{K \times t} \]

\[ U = \frac{1}{\Sigma R} \]

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)
- \( \Sigma R \) = 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:

\[ H_{\text{sensible}} = A \times U \times (T_i - T_o) \]

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:

\[ H = F \cdot P \cdot (T_i - T_o) \]

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:

\[ H_{\text{sensible}} = V \cdot \rho_{\text{air}} \cdot C_p \cdot (T_i - T_o) \]

Where:

- \( H_{\text{sensible}} \) = Sensible heat loss
- \( V \) = volumetric air flow rate
- \( \rho_{\text{air}} \) is the density of the air
- \( C_p \) = specific heat capacity of air at constant pressure
- \( T_i \) = indoor air temperature
- \( T_o \) = outdoor air temperature
The energy quantity associated with net loss of moisture from the space is latent heat loss which is given by:

\[ H_{\text{latent}} = V \cdot \rho_{\text{air}} \cdot h_{fg} \cdot (W_i - W_o) \]

Where

- \( H_{\text{latent}} \) = Latent heat loss
- \( V \) = volumetric air flow rate
- \( \rho_{\text{air}} \) is the density of the air
- \( W_i \) = humidity ratio of indoor air
- \( W_o \) = humidity ratio of outdoor air
- \( h_{fg} \) = latent heat of evaporation at indoor air temperature

---

**SENSIBLE HEAT FACTOR or RATIO (SHR)**

\[ \text{SHR} = \frac{H_s}{H_t} = \frac{H_s}{H_s + H_L} \]

Where

- \( \text{SHR} \) = 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

\[ Q = 1.08 \times \left( \frac{H_s}{(T_R - T_S)} \right) \]

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 75°F

• $T_S$ = Supply air or leaving air temperature from the cooling coil in °F

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:

<table>
<thead>
<tr>
<th>SHR</th>
<th>LAT ($T_S$)</th>
<th>Room DB ($T_R$)</th>
<th>TD ($T_R - T_S$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>X &lt; .80</td>
<td>54</td>
<td>75</td>
<td>21</td>
</tr>
<tr>
<td>.80 &lt; X &gt; .85</td>
<td>56</td>
<td>75</td>
<td>19</td>
</tr>
<tr>
<td>X &gt; .85</td>
<td>58</td>
<td>75</td>
<td>17</td>
</tr>
</tbody>
</table>

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
- \( \Delta T \) = 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:

\[ C = PCL \times 24 \times \frac{CDD}{\Delta T} \]

Where

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

**FUEL CONSUMPTION BY HEATING UNITS**

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

\[ F = \frac{H}{(e \times FHC)} \]

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:

\[ F = \frac{C}{(SEER \times 1000)} \]

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)
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 = ft³ (cubic feet) and then calculate the Air Volume requirement by multiplying the Room Volume by the Air Change Rate per hour.

\[ ACH = \frac{Cu-ft \times min}{60 \text{ min/hr}} \times \frac{1}{\text{room volume}} \]

\[ CFM = \frac{ACH \times \text{room volume}}{60 \text{ min/hr}} \]

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:

\[ L_v = 4005 \times (R_p)^{\frac{1}{5}} \]

\[ L = 4005 \times A \times (R_p)^{\frac{1}{5}} \]

Where
- **L_v** = 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.

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

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) = \( (C_r - C_s) \times 100 / (C_r - C_o) \)

Where:

\( C_s \) = 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 (Qd), it is necessary to estimate the evaporation rate of the contaminant (qd) according to the following equation:

\[ q_d = \frac{387 \text{ (lbs)}}{(MW)} \times (T) \times (\rho) \]

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
- \( \rho \) = 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:
\[ Q = V \times A \]

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:

\[ Q = V \times [(10X \times 10X) + A] \]

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 rate is predicted by an equation for openings between 0.5 to 2" in width:

\[ Q = 3.7 \times L \times V \times X \]

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 ratios less than 0.15 and flanges greater than three times the slot width, the equation is

\[ Q = 2.6 \times L \times V \times X \]
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:

\[ C_e = \left(\frac{VP}{SPh}\right)^{0.5} = \left[\frac{1}{1+K}\right]^{0.5} \]

Where

- \( K \) = Loss factor
- \( VP \) = Velocity pressure in duct
- \( SPh \) = Absolute static pressure about 5 duct diameters down the duct from the hood

**DUCTWORK EQUATIONS**

**Velocity in Duct**

Velocity in duct can be expressed as

\[ V = \frac{Q}{A} = 144 \times \frac{Q}{a \times b} \]

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

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

\[
P = \frac{\left(\frac{\pi \times DS}{12}\right) + \left(\frac{DL - DS}{12}\right)}{2}
\]

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

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)
• \( Deq \) = 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**

\[ BHP = Q \times SP / (6356 \times \text{Fan Eff.}) \]

Where

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

---

**Fan Motor Horsepower**

\[ \text{Motor HP} = BHP / \text{Motor Eff} \]

Where

- \( BHP \) = Break Horsepower
- \( \text{MotorEff} \) = Motor drive efficiency usually 80-95%

---

**Tip Speed**

\[ Ts = 3.14 \times D \times N \]

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
- \( \rho_2 \) = Density at duty point 2, lb/ft\(^3\)
- \( \rho_1 \) = Density at duty point 1, lb/ft\(^3\)

Effect of Fan Speed

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

\[
Q_2 = Q_1 \left( \frac{N_2}{N_1} \right)
\]

2) Fan pressure varies with square of fan speed ratio

\[
P_2 = P_1 \left( \frac{N_2}{N_1} \right)^2
\]

3) Fan power varies with cube of fan speed ratio

\[
H_2 = H_1 \left( \frac{N_2}{N_1} \right)^3
\]

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:
\[ L = 2C + 1.57 \times (D+d) + \frac{(D-d)^2}{4C} \]

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* = \( GPM \times TDH \times SG \) / (3960 \times Efficiency)

*Pump BHP* = \( GPM \times PSI \times SG \) / (1713 \times 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 \times 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* = \( \frac{BHP}{Motor\ Eff} \)

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: \( SG = \frac{P_F}{P_W} \)

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

\( PSI = \text{Head (feet)} \times \text{Specific Gravity} / 2.31 \)

**Velocity Head**

\( VH = \frac{V^2}{2g} \)

Where

- \( VH = \text{Velocity head in ft} \)
- \( V = \text{Velocity in ft/s} \)
- \( g = \text{Acceleration due to gravity (32.17 ft/s}^2\)
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 \( \frac{p}{y} \) and velocity head \( \frac{v^2}{2g} \). 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).

\[
h + \frac{p}{y} + \frac{v^2}{2g} = E = \text{Constant}
\]

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

\[
\text{NPSHA} = \text{HA} \pm \text{HS} - \text{HF} - \text{HVP}
\]

Where

- \( \text{NPSHA} \): Net Positive Suction Available at Pump expressed in feet of fluid
- \( \text{NPSHR} \): Net Positive Suction Required at Pump (Feet)
- \( \text{HA} \): 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)
- \( \text{HS} \): 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.
- \( \text{HF} \): 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;

\[ Ns = \left( \frac{N_r \times Q}{H} \right)^{\frac{1}{2}} \]

Where

- \( Ns \) = Specific speed
- \( Q \) = Flow in US gallons per minute (GPM)
- \( N_r \) = 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 than the one with specific speed of 1000.

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**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:

\[ Q = \frac{H}{(8.01 \times \rho \times c \times \Delta T)} \]

Where:

- \( Q \) = Water volume flow rate (GPM)
• $H$ = Heat load (Btu/hr)
• $\Delta T$ = Intended temperature drop ($^\circ$F)
• $\rho$ = Fluid's density at the average system temperature (lb/ft$^3$)
• $c$ = the fluid's specific heat at the average system temperature (Btu/lb/$^\circ$F)
• $8.01$ = a constant

In small to medium size hydronic systems, the product of $(8.01 \times \rho \times c)$ 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)
• $\Delta T$ = temperature difference ($^\circ$F)

---

**Evaporator Flow Rate**

The evaporator water flow rate can be expressed as

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

Where

• $Q_e$ = Evaporator water flow rate (GPM)
• $H_{\text{tons}}$ = Air conditioning cooling load (tons)
• $\Delta T$ = Temperature differential between inlet and outlet ($^\circ$F)

---

**Condenser Flow Rate**

The condenser water flow rate can be expressed as

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

Where

• $Q_c$ = Condenser water flow rate (GPM)
• $H_{\text{tons}}$ = Air conditioning cooling load (tons)
• $\Delta T$ = Temperature differential between inlet and outlet ($^\circ$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

\[ Q_{\text{Cond}} = Q_{\text{air}} \times \frac{\Delta W_{\text{lb}}}{(SpV \times 8.33)} \]

\[ Q_{\text{Cond}} = Q_{\text{air}} \times \frac{\Delta W_{\text{GR}}}{(SpV \times 8.33 \times 7000)} \]

Where

- \( Q_{\text{Cond}} \) = Air Conditioning condensate generated (GPM)
- \( Q_{\text{air}} \) = 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)
- \( \Delta W_{\text{lb}} \) = Specific Humidity diff. between inlet and outlet of air stream across coil (lb-H₂O per lb of dry air)
- \( \Delta W_{\text{GR}} \) = Specific Humidity diff. between inlet and outlet of air stream across coil (Gr. H₂O 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:

\[ Q = \frac{H}{(C_p \times \rho \times \Delta T)} \]

Where

- \( Q \) = Volumetric flow rate (GPM)
- \( H \) = Heat flow rate (Btu/hr)
- \( C_p \) = Specific heat capacity (Btu/lb-°F)
- \( \rho \) = Density (lb/ft³)
- \( \Delta T \) = Temperature difference (°F)

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

\[ Q = \frac{H \times (7.48 \text{ gal/ft}^3)}{((1 \text{ Btu/lb-}°\text{F}) (62.34 \text{ lb/ft}^3) (60 \text{ min/h}) \Delta T)} \]

Or

\[ Q = \frac{h}{(500 \times \Delta T)} \]

Where

- \( Q \) = Water flow rate (GPM)
- \( H \) = Heat flow rate (Btu/hr)
• $\Delta T =$ Temperature difference (°F) (usually 20°F)

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:

$$m = \frac{h}{((1.2 \text{ Btu/lb}^{-1}\text{°F}) \times \Delta T)}$$

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**

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

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:

<table>
<thead>
<tr>
<th>Material</th>
<th>Hazen-Williams Coefficient - C -</th>
</tr>
</thead>
<tbody>
<tr>
<td>Brass</td>
<td>130 - 140</td>
</tr>
<tr>
<td>Cast-Iron - new unlined (CIP)</td>
<td>130</td>
</tr>
<tr>
<td>Cast-Iron 10 years old</td>
<td>107 - 113</td>
</tr>
<tr>
<td>Cast-Iron 20 years old</td>
<td>89 - 100</td>
</tr>
<tr>
<td>Cast-Iron 30 years old</td>
<td>75 - 90</td>
</tr>
<tr>
<td>Concrete</td>
<td>120 - 140</td>
</tr>
<tr>
<td>Copper</td>
<td>130 - 140</td>
</tr>
<tr>
<td>Ductile Iron Pipe (DIP)</td>
<td>120</td>
</tr>
<tr>
<td>Galvanized iron</td>
<td>120</td>
</tr>
</tbody>
</table>
Darcy-Weisbach equation and Head Loss

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

The friction factor or coefficient - $\lambda$ - 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:

$$H_{loss} = \lambda \left(\frac{L}{d_h}\right) \left[\frac{v^2}{2g}\right]$$

Where

- $H_{loss}$ = head loss (ft)
- $\lambda$ = friction coefficient
- $L$ = length of duct or pipe (ft)
- $g$ = acceleration of gravity (32.2 ft/s²)
- $d_h$ = 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:

$$Re = \frac{D \times v \times \rho}{\mu}$$

Where

- $D$ = characteristic length (For a pipe or duct the characteristic length is the pipe or duct diameter, in m)
- $v$ = velocity (m/s)
- $\rho$ = density (kg/m³)
- $\mu$ = dynamic (absolute) viscosity (Ns/m²)
The table below shows Reynolds Number for one gallon of water flowing through pipes of different dimensions:

<table>
<thead>
<tr>
<th>Pipe Size (inches)</th>
<th>1</th>
<th>1 ½</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>6</th>
<th>8</th>
<th>10</th>
<th>12</th>
<th>18</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reynolds number with one (1) gal/min</td>
<td>3800</td>
<td>2500</td>
<td>1900</td>
<td>1270</td>
<td>950</td>
<td>630</td>
<td>475</td>
<td>380</td>
<td>320</td>
<td>210</td>
</tr>
</tbody>
</table>

**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:

\[ d_h = \frac{4 \times A}{P} \]

Where:

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

**Friction Coefficient (\( \lambda \))** 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:

\[ \lambda = \frac{64}{Re} \]

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.

\[ \text{Approach} = \text{LWT} – \text{WBT} \]

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

\[ \text{Range} = \text{EWT} – \text{LWT} \]

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**

\[ Q = \frac{H}{(\text{Range} \times 500)} \]

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

\[ MR = \text{water lost through evaporation (ER) + bleed (BR) + drift (DR)} \]

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

\[ ER = Q \times \text{Range} / 1,000 \]

The drift loss is roughly 0.2 to 0.5%

\[ DR = 0.002\% \times Q \]

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:

\[ MR = \text{COC} \times BR \] …… (eq.2)

And that:

\[ MR = ER \times \left(\frac{\text{COC}}{(\text{COC} - 1)}\right) \] …… (eq.3)

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

\[ BR = ER / (\text{COC} - 1) \]

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:

\[ \text{Efficiency} = \frac{(\text{EWT} - \text{LWT}) \times 100}{(\text{EWT} - \text{WBT})} \]

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
\[ C_v \text{ max} = \frac{\dot{Q}_{gpm}}{\sqrt{\Delta P_{\text{max}} / S_g}} \]

Where

1) \( S_g \) is the specific gravity of the fluid, water = 1
2) \( \Delta P \) is in psi
3) \( C_v \) = 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 \( C_v \) and then flow rate at various pressure drops. *The rated \( C_v \) is established with the valve fully open. As the valve partially closes to some intermediate position the \( C_v \) 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 \( \Delta P_{\text{max}} \). When the valve is completely open, the pressure drop across the valve is at its lowest point and is referred to \( \Delta P_{\text{min}} \). The ratio \( (N) \Delta P_{\text{min}} / \Delta P_{\text{max}} \) 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:

\[
LMTD = \frac{(TD_2 - TD_1)}{\ln \left( \frac{TD_2}{TD_1} \right)}
\]

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:

\[
AMTD = \frac{(T_{P1} + T_{P2})}{2} - \frac{(T_{S1} + T_{S2})}{2}
\]

Where

- AMTD = Arithmetic Mean Temperature Difference °F
• \( T_{P1} \) = primary inlet temperature °F
• \( T_{P2} \) = primary outlet temperature °F
• \( T_{S1} \) = secondary inlet temperature °F
• \( T_{S2} \) = secondary outlet temperature °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 \( T_{S1} = T_{S2} \) °F

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

---

**HUMIDIFICATION**

\[
GR_{reqd} = \left( \frac{W_{Gr}}{SpV} \right)_{Room \ air} - \left( \frac{W_{Gr}}{SpV} \right)_{Supply \ air}
\]

\[
Lb_{reqd} = \left( \frac{W_{Lb}}{SpV} \right)_{Room \ air} - \left( \frac{W_{Lb}}{SpV} \right)_{Supply \ air}
\]

\[
Q_{steam} = \frac{(Q_{air} \times GR_{reqd} \times 60)}{7000}
\]

\[
Q_{steam} = Q_{air} \times Lb_{reqd} \times 60
\]

Where

• \( GR_{reqd} \) = Grains of Moisture Required (Gr H\textsubscript{2}O per Cu-ft of air)
• \( Lb_{reqd} \) = Pounds of Moisture Required (Lb H\textsubscript{2}O 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)
• \( \Delta W_{lb} \) = Specific Humidity (lb-H\textsubscript{2}O per lb of dry air)
• \( \Delta W_{Gr} \) = Specific Humidity (Gr. H\textsubscript{2}O per lb of dry air)

---

**Humidifier Sensible Heat Gain**

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

Where

Page 55 of 71
• 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:

\[ MRC = \left( Q_{\text{air}} \times \left( \frac{60 \text{ min/hr}}{Vs} \right) \times (\text{GPP}_{\text{in}} - \text{GPP}_{\text{out}}) \right) / 7000 \text{ grains/lb} \]

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

The difference in \((\text{GPP}_{\text{in}} - \text{GPP}_{\text{out}})\) 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

\[ V_{et} = 2 \times \left\{ V_w \times \left[ \left( \frac{v_1}{v_0} \right) - 1 \right] - 3 \times a \times (T_1 - T_0) \right\} \]

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

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

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

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

1) \( V_e \) = Necessary expansion tank volume (Gallons)

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

3) \( \nu_0 \) = Specific volume of water at initial (cold) temperature \( T_0 \) in \( \text{ft}^3/\text{lb} \), refer 1989 ASHRAE Fundamentals, Chapter 2, Table 25 or Part 27, Properties of Air and Water

4) \( \nu_1 \) = 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 \(^\circ\)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 \(^\circ\)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 \((\alpha_{\text{STEEL}} = 6.5 \times 10^{-6} \text{ and } \alpha_{\text{COPPER}} = 9.5 \times 10^{-6})\)

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.

\[
1TR = 12,000 \text{ Btu/hr}
\]
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.

\[ \text{KW/ton} = \frac{P_c}{H_r} \]

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:

\[ \text{COP} = \frac{H_u}{H_a} \]

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.

\[ EER = \frac{H_c}{P_a} \]

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 = \((\text{Btu Input} - \text{Btu Stack Loss}) / \text{Btu Input}\) \times 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:

\[
\text{Decibel} = 10 \log \left( \frac{P}{P_{\text{ref}}} \right)
\]

Where

- \(P\) = signal power (W)
- \(P_{\text{ref}}\) = 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.

\[
L_{\text{wt}} = 10 \log \left( \frac{n \cdot N}{N_0} \right) = 10 \log \left( \frac{N}{N_0} \right) + 10 \log (n)
\]

\[
= L_{\text{ws}} + 10 \log (n)
\]

Where

- \(L_{\text{wt}}\) = Total sound power level (dB)
- \(L_{\text{ws}}\) = Sound power level from each single source (dB)
- \(N\) = sound power (W)
- \(N_0 = 10^{-12} \) - 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)).

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**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:

\[ L_{wt} = 10 \log \left( \left( \frac{N_1 + N_2 + \ldots + N_n}{N_o} \right) \right) \]

---

**STEAM & CONDENSATE EQUATIONS**

Some common steam and condensate equations can be expressed as

**Steam Heating**

\[ m_s = \frac{H}{960} \]

Where

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

---

**Steam Heating Liquid Flow**

\[ m_s = Q_L \times 500 \times S_{GL} \times C_{PL} \times \Delta T_L / L_s \]

Where

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

---

**Steam Heating Air or Gas Flow**

\[ m_s = Q_G \times 60 \times p_G \times C_{PG} \times \Delta T_G / L_s \]
Steam Pipe Sizing Equations

\[ \Delta P = \left[ \frac{0.01306 \times W^2 \times (1 + \frac{3.6}{ID})}{3600 \times D \times ID^5} \right] \]
\[ W = 60 \times \left[ \frac{P \times D \times ID^5}{0.01306 \times (1 + \frac{3.6}{ID})} \right]^{1/2} \]
\[ W = 0.41667 \times V \times A_{\text{inches}} \times D = 60 \times V \times A_{\text{feet}} \times D \]
\[ V = \frac{2.4 \times W}{A_{\text{inches}}} \times D = \frac{W}{60 \times A_{\text{feet}} \times D} \]

Where

- \( \Delta P \) = 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)
- \( A_{\text{inches}} \) = Actual Cross Sectional Area of Pipe (Sq-inches)
- \( A_{\text{feet}} \) = Actual Cross Sectional Area of Pipe (Sq-ft)

Steam Condensate Pipe Sizing Equations

\[ FS = \frac{H_{SS} - H_{SC}}{H_{LC}} \times 100 \]
\[ W_{CR} = \frac{FS}{100} \times W \]

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.

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**RELIEF VALVE SIZING**

Liquid System Relief Valves and Spring Style Relief Valves:

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

Liquid System Relief Valves and Pilot Operated Relief Valves:

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

Steam System Relief Valves:

\[
A = \frac{W}{(51.5 \times K \times P \times K_{SH} \times K_N \times K_B)}
\]

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

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

Gas and Vapor System Relief Valves (SCFM):

\[
A = \frac{(SCFM \times (TGZ)^{\frac{1}{2}})}{(1.175 \times C \times K \times P \times K_B)}
\]

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)

• $\Delta P =$ Differential Pressure (Psig); $\Delta P =$ 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*

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

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**Steel Pipe Equations**
\[ A = 0.785 \times ID^2 \]
\[ WP = 10.6802 \times T \times (OD - T) \]
\[ WW = 0.3405 \times ID^2 \]
\[ OSA = 0.2618 \times OD \]
\[ ISA = 0.2618 \times ID \]
\[ AM = 0.785 \times (OD^2 - ID^2) \]

Where

- \( A \) = Cross-Sectional Area (Sq-inches)
- \( WP \) = Weight of Pipe per Foot (Lbs)
- \( WW \) = 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)
- \( AM \) = 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. \( C_v \) 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 (\( K_v \)) - is also in common use, but express the capacity in SI-units.

Specific formulas used to estimate \( C_v \) for different fluids is indicated below:

**Flow Coefficient - \( C_v \) for Liquids**

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

**Flow expressed by volume**

\[ C_v = Q \times \left( \frac{SG}{\Delta p} \right)^{1/2} \]

Where

- \( Q \) = water flow (US gallons per minute)
• $SG = \text{specific gravity (1 for water)}$

• $\Delta p = \text{pressure drop (psia)}$

**Flow is expressed by weight**

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

*Where*

• $w = \text{water flow (lb/h)}$

• $SG = \text{specific gravity (1 for water)}$

• $\Delta p = \text{pressure drop (psia)}$

---

**Flow Coefficient - $C_v$ - 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:

$$C_v = \frac{m}{1.61 \, p_i}$$

*Where*

• $m = \text{steam flow (lb/h)}$

• $p_i = \text{inlet steam absolute pressure (psia)}$

• $p_o = \text{outlet steam absolute pressure (psia)}$

---

**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:

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

---

**Flow Coefficient - $C_v$ - 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:

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

*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)

pi = inlet gas absolute pressure (psia)

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

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

Where

\[ \Delta p = (p_i - p_o) \]

\[ p_o = \text{outlet gas absolute pressure (psia)} \]

---

**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

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**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) - m³/s
   - m³/h = 3600 x m³/s
   - CFM = 0.59 x m³/h

3) Area (A) – m²
   - 1 m² = 1550 in² = 10.764 ft² = 1.1968 yd² = 3.861x10⁻⁷ mile²

4) Density – kg/m³
   - 1,000 kg/m³ = 62.43 Lbs/Cu-foot = 8.33 Lbs/Gal. = 0.1337 Cu-foot/Gal
   - Standard air (ρ) = 1.2 kg /m³

5) Energy - kJ
   - 1 kWh = 3.6x10⁶ J = 859.9 kcal = 2.656x10⁶ foot lb = 3.412x10³ Btu
   - 1 kJ = 1 kNm = 1kWs = 10³ J = 0.947813 Btu = 737.6 foot lb = 0.23884 kcal

6) Energy per unit mass
   - 1 kJ/kg = 0.4299 Btu/ lbm = 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 (m³ per min of air flow) x ΔT (temperature difference in deg C)
   - Latent heat: HL (Btu/hr) = 0.68 x CFM (cu-ft per min of air flow) x ΔW (Humidity Ratio Difference (Grains of H₂O/Lb of dry air)
   - Latent heat: HL (KJ/hr) = 177734.8 x CMM (m³ per min of air flow) x ΔW (Humidity Ratio Difference (Kg H₂O/ Kg of dry air)
   - Total heat: HT (Btu/hr) = 4.5 x CFM (cu-ft per min of air flow) x Δh (Enthalpy Difference Btu/Lb of dry air)
• Total heat: \( H_T (\text{KJ/hr}) = 72.09 \times \text{CMM (m}^3\text{ per min of air flow)} \times \Delta h \) (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 = 3.6x10^6 J/h

• 1 kcal/hr = 1.163x10^{-3} kW = 3.969 Btu/h = 1.582x10^{-3} hp = 1.560x10^{-3} 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 = 2.043x10^4 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 = 3.73x10^4 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 \circ F = 5.678 W/m^2 K = 4.882 kcal/h m^2 \circ C

• 1 W/m^2 K = 0.85984 kcal/h m^2 \circ C = 0.1761 Btu/ foot^2 h \circ F

• 1 kcal/h m^2 \circ C = 1.163 W/m^2 K = 0.205 Btu/ foot^2 h \circ F

11) Length

• 1 m (meter) = 3.2808 foot = 39.37 in = 1.0936 yd = 6.214x10^{-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 = 2.778x10^{-4} kg/s = 3.67x10^{-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/m² = 0.3170 Btu/(h foot²) = 0.85984 kcal/(h m²)

16) Pressure

• 1 atm = 101.325 kN/m² = 101.325 kPa = 1.013 bar = 14.696 psia (lb/in²) = 0 psig = 29.92 in-Hg = 760 mm mercury at 62 °F (16.7 °C) = 760 torr = 33.95 Foot H₂O = 407.2 In-wg (Water Gauge) = 2116.8 Lbs/Sq-ft

17) Specific Energy

• 1 J/kg = 0.43 Btu/lb = 2.39x10⁻⁴ kcal/kg
• 1 kcal/kg = 1.80 Btu/lb = 4,187 J/kg

18) Specific Heat Capacity

• 1 J/ (kg K) = 2.389x10⁻⁴ kcal/ (kg °C) = 2.389x10⁻⁴ Btu/ (lb °F)
• 1 kcal/ (kg °C) = 4,186.8 J/ (kg K) = 1 Btu/ (lb °F)

19) Specific Volume

• 1 m³/kg = 16.02 foot³/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.) = 3.785x10⁻³ m³ = 3.785 dm³ (litre) = 0.13368 foot³ = 4.951x10⁻³ yd³ = 0.8327 Imp. Gal (UK) = 4 Quarts = 8 Pints

25) Volume Flow

• 1 m³/h = 0.5886 foot³/min (CFM) = 3.667 Imp. Gal (UK)/min = 4.403 gal (US)/min
• 1 GPM = 3.785 LPM (litres per min)