



**PDHonline Course M215 (4 PDH)**

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# **HVAC Refresher - Facilities Standard for the Building Services (Part 1)**

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# HVAC Refresher – Facilities Standard for the Building Services (Part 1)

*A. Bhatia*

## Course Content

### Introduction

The main purpose of commercial HVAC (heating, ventilating, and air conditioning) systems is to provide the people inside buildings with "conditioned" air so that they will have a comfortable and safe environment. "Conditioned" air means that air is clean and odor-free, and the temperature, humidity, and movement of the air are within certain comfort ranges. The challenge to system designers and facility executives is achieving goals for improved system performance, efficiency and security without making systems unaffordable or unmanageable. Fortunately there are options that must be explored by designers pursuing high performance buildings.

Here are some basic principles to consider:

- 1) Physically smaller HVAC equipment requires less overall space and improves core efficiencies as compared to larger HVAC equipment.
- 2) Moving less air results in lower fan horsepower, less sound attenuation, smaller equipment and smaller ductwork.
- 3) Quieter HVAC equipment requires less sound attenuation and minimizes special architectural room construction when compared to noisier HVAC equipment.
- 4) Moving less water will result in smaller piping systems, smaller pumps, and lower pump horsepower.
- 5) Smaller pump motors, smaller fan motors, and lower refrigeration horsepower require smaller electrical systems and use less energy.
- 6) Concentrating the major electrical loads such as large motors and refrigeration equipment near the electrical utility service entrance is usually less expensive than locating large electrical loads at a greater distance from the electrical service entrance.
- 7) Using fewer materials and smaller equipment minimizes environmental impact.

This course highlights some important prescriptive/performance requirements for HVAC design and provides some useful rules of thumb information.

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### **HVAC DESIGN CODES & STANDARDS**

- 1) AABC (Associated Air Balance Council)
- 2) ADC (Air Diffusion Council)
  - ADC1062: GRD- Test Code for Grilles, Registers, and Diffusers
  - ADC-01- Directory of ADC Certified Products
- 3) AMCA (Air Moving and Control Association)
  - AMCA 210 - Design and Testing of Fans
  - AMCA 500 - Test Methods for Louvers, Dampers and Shutters

- 4) ANSI (American National Standards Institute)
- 5) ARI (American Refrigeration Institute)
  - ARI 210- Unitary Air Conditioning Equipment
  - ARI 270- Standard for Sound Rating of Outdoor Unitary Equipment
  - ARI 360- Commercial and Industrial Unitary Air Conditioning Equipment
  - ARI 410- Forced Circulation Air-Cooling and Air- Heating Coils
  - ARI 430- Standard for Central Station Air-Handling Unit
  - ARI 650- Standard for Air Outlets and Inlets
- 6) ASHRAE (American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc.) - Fundamentals Handbook, Systems Handbook
  - ASHRAE 14- Methods of Testing for Rating Positive Displacement Condensing Units
  - ASHRAE 15- Safety Code for Mechanical Refrigeration
  - ASHRAE 20- Methods of Testing for Rating Mechanical-Draft Air-Cooled Refrigerant Condensers
  - ASHRAE 34- Number designation and Safety Classification of Refrigerants
  - ASHRAE Standard 55-1992: This standard provides the thermal environmental conditions for human occupancy
  - ASHRAE Standard 62-1999: This standard provides guidelines to the ventilation for acceptable Indoor Air Quality
  - ASHRAE 70- Method of Testing for Rating the Airflow Performance of Outlets and Inlets
  - ASHRAE Standard 90.1-1999: This standard provides guidelines to the energy efficient design of new buildings intended for human occupancy.
  - ASHRAE Standard 105: This standard provides method of measuring and expressing building energy performance.
  - ASHRAE 111- Practices for Measuring, Testing and Balancing a Building Heating, Ventilation, Air Conditioning and Refrigeration Systems
  - ASHRAE Standard 113-1990: This standard provides guidelines to the method for evaluating the air diffusion performance of an air distribution system.
  - ASHRAE Standard 129-1997: This standard describes a test method for evaluating an air distribution system's ability to provide required levels of ventilation air to the building occupants.
- 7) ASTM (American Society of Testing and Materials)
  - ASTM A 53- Standard Specification for Pipe, Steel, Black, and Hot Dipped, Zinc- Coated, Welded and Seamless
  - ASTM A 90-Standard Test Method for Weight of Coating on Zinc Coated (Galvanized) Iron or Steel Articles
  - ASTM A 527- Standard Specification for Steel Sheet, Zinc-Coated (Galvanized) by the Hot-Dip Process, Lock-Forming Quality
  - ASTM A 666- Standard Specification for Austenitic Stainless Steel, Sheet, Strip, Plate, and Flat bar
  - ASTM B 62- Standard Specification for Composition Bronze or Ounce Metal Castings

- ASTM B 88- Standard Specification for Seamless Copper Water Tube
  - ASTM B 280- Standard Specification for Seamless Copper Tube for Air Conditioning and Refrigeration Field Service
  - ASTM C 552- Standard Specification for Cellular Glass Thermal Insulation
  - ASTM E 477- Testing Duct Linear Materials and Prefabricated Silencers for Acoustical and Airflow Performance
- 8) ISO (International Standards Organization)
- ISO 9001- Quality Systems – Model for Quality Assurance in design, development, production, installation, and servicing
  - ISO 9003- Quality Systems – Model for Quality Assurance in final inspection and test
  - ISO 9004- Quality Management and Quality System Elements – Guidelines
- 9) SMACNA (Sheet Metal and Air Conditioning Contractors' National Association, Inc.)
- SMACNA-06- HVAC Duct Construction Standards – Metal and Flexible
- 10) NEMA (National Electrical Manufacturers Association)
- NEMA 250 - Enclosures for Electric Equipment (1000 Volts Maximum)
- 11) NFPA (National Fire Protection Association)
- NFPA 90A - Standard for the Installation of Air Conditioning and Ventilation Systems
- 12) UBC refers to the State adopted International Conference of Building Officials Uniform Building Code, 1994 edition or the year indicated.
- 13) UL (Underwriters Laboratories, Inc.)
- UL 181 -Factory made Air Ducts and Connectors
  - UL 207 -Refrigerant- Containing Components and Accessories, Non-electrical
  - UL 303 - Refrigeration and Air-Conditioning Condensing and Compressor Units
  - UL 465 - Central Cooling Air Conditioners
  - UL 555 - Fire Dampers and Ceiling Dampers
  - UL 900 - UL Standard for Safety Air Filter Units
- 14) Uniform Building Code 1994 (UBC 94)
- 15) Uniform Mechanical Code 1994 (UMC 94)
- 16) Uniform Plumbing code 1944 (UPC 94)
- 17) Uniform Fire Code 1994 (UFC 94)

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### **INTERDISCIPLINE COORDINATION**

HVAC design must be coordinated with all other disciplines, such as, Architectural, Structural, Electrical, Plumbing and Site Planning. The following HVAC related work is usually shown by other disciplines:

- 1) Architectural drawings and specifications show all louvers and attached screens in exterior walls, all flashing for ducts and pipes penetrating roofs and exterior walls, finish and identification, painting walls and ceilings, access panels, chases, furred spaces, door grilles, mechanical equipment rooms and penthouses.

- 2) Structural drawings and specifications show all concrete and structural steel work, including catwalks, concrete housekeeping pads, lintel supports around openings, and platforms for access to HVAC equipment and supports for cooling towers and other large mechanical equipment.
- 3) Electrical drawings and specifications show motor starters and disconnects not furnished as part of HVAC equipment, smoke detectors (duct and/or space mounted), all power wiring to HVAC smoke dampers, motors, heating cable, controls for winterizing piping, day tank and oil piping in the emergency generator room, and muffler exhaust pipe for emergency generator.
- 4) Plumbing provides all domestic water make-up supply and drain outlets, underground oil storage tank(s) and piping for emergency generators.

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### **CLASSIFICATION OF CLIMATES**

Many different systems of climate classification are in use for different purposes. Climatic zones such as tropical, arid, temperate and cool are commonly referred for representing climatic conditions. For the purposes of building design, a simple system based on the nature of the thermal problem in the particular location is often used.

- 1) **Cold** climates, where the main problem is the lack of heat (under heating), or excessive heat dissipation for all or most parts of the year.
- 2) **Temperate** climates, where there is a seasonal variation between under heating and overheating, but neither is very severe.
- 3) **Hot-dry (arid)** climates, where the main problem is overheating, but the air is dry, so the evaporative cooling mechanism of the body is not restricted. There is usually a large diurnal (day - night) temperature variation.
- 4) **Warm-humid** climates, where the overheating is not as great as in hot-dry areas, but it is aggravated by very high humidity's, restricting the evaporation potential. The diurnal temperature variation is small.

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### **SIX CATEGORIES OF CLIMATES**

- 1) **Warm-humid** - 15°N and South of the equator, e.g. Lagos, Mombassa, Colombo, Jakarta etc.
- 2) **Warm-humid Island** - equatorial and trade wind zones, e.g. Caribbean, Philippines and Pacific Islands etc.
- 3) **Hot-dry desert** - 15° to 30° N and South, e.g. Baghdad, Alice Springs, Phoenix etc.
- 4) **Hot-dry maritime desert** - latitudes as 3), coastal large landmass, Kuwait, Karachi etc.
- 5) **Composite Monsoon** - Tropic Cancer/Capricorn, Lahore, Mandalay, New Delhi etc.
- 6) **Tropical uplands** - Tropic Cancer/Capricorn, 900 to 1200 meters above sea level (plateau and mountains), Addis Ababa, Mexico City, Nairobi etc.

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### **HEATING LOAD CALCULATIONS**

- 1) Heat loss occurs from a building envelope whenever the interior temperature exceeds the exterior temperature. The rate at which it occurs is affected primarily by the efficiency of the covering materials (glazing, roof, side walls, doors, window frames and end walls).
- 2) The most common factors affecting heat loss are conduction, temperature difference between indoor/outdoors and infiltration.

- 3) Heat loss estimation shall be made on the worst scenario. Credit for the heat of people present in the building is normally not taken since the building could be unoccupied. Internal heat gain from lighting and appliances is usually neglected.
- 4) If necessary, increase the calculated size of equipment and distribution system(s) by up to 30 percent to compensate for morning recovery due to night setback.

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### **COOLING LOAD CALCULATIONS**

- 1) Cooling Load is the rate at which heat must be extracted from a space to maintain a desired room condition.
- 2) Heat gain occurs from a building envelope whenever the exterior temperature exceeds the interior temperature. The rate at which it occurs is affected primarily by the efficiency of the covering materials (glazing, roof, side walls, doors, window frames and end walls). The other source of heat gain is from the lighting, equipments, occupancy, infiltration and the ventilation air.
- 3) Calculate cooling loads using ASHRAE-based load calculation methodology. Prepare cooling load calculations using the CLTD/SCL/CLF Method, which is based on the cooling load temperature differences (CLTD), the solar cooling load factors (SCL), and the cooling load factors (CLF). Cooling Load Temperature Difference (CLTD): An equivalent temperature difference used for calculating the instantaneous external cooling loads across a wall or roof ( $CLTD = \text{External Cooling Load} / (\text{U-Value} \times \text{Area})$ ). When used for glass, the CLTD calculates only the conduction cooling load. All these methods are described in the ASHRAE Handbook Fundamentals.
- 4) Approved methods for cooling load calculations include:
  - ASHRAE Fundamentals- Load and Energy Calculations
  - ACCA Manual N - Commercial Load Calculation
  - Computer software such as DOE-2, ASEAM or other non-proprietary software based upon ASHRAE or ACCA methods
  - A number of proprietary commercial ASHRAE-based software packages such as TRACE 700, Elite Software, Hevacomp, and software by Carrier Corp etc.
- 5) Factor in extreme conditions such as arid or humid climates. If necessary, increase the calculated size of equipment and distribution system(s) by up to 10 percent to compensate for morning recovery due to night set forward or by up to 10 percent to compensate for unanticipated loads or changes in space usage. Limit the total combined increase above the size calculated of equipment and distribution system(s) to 15 percent total.
- 6) Divide the building into zones. Always estimate the building peak load and individual zones airflow rate. The building peak load is used for sizing the refrigeration equipment and the individual zone loads are helpful in estimating the air-handling unit capacity.

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### **THERMAL ZONING**

Thermal zoning is a method of designing and controlling the HVAC system so that occupied areas can be maintained at a different temperature than unoccupied areas using independent setback thermostats. A zone is defined as a space or group of spaces in a building having similar heating and cooling requirements throughout its occupied area so that comfort conditions may be controlled by a single thermostat. In practice the corner rooms and the perimetric spaces of the building have variations in load as compared to the interior core areas. The following facts may be noted:

- 1) The buildings are usually divided into two major zones.

- **Exterior Zone:** The area inward from the outside wall (usually 12 to 18 feet, if rooms do not line the outside wall). The exterior zone is directly affected by outdoor conditions during summer and winter.
  - **Interior Zone:** The area contained by the exterior zone. The interior zone is only slightly affected by outdoor conditions and usually has a uniform cooling.
- 2) Single-zone models shall be limited to open floor plans with perimeter walls not exceeding 40 feet in length.
  - 3) For large building footprints, assume a minimum of five zones per floor: one zone for each exposure (north, south, east & west) and an interior zone.

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### **OUTDOOR DESIGN CONDITIONS**

- 1) The capacity calculations for the HVAC systems shall be based on the climatic information contained in chapter 26 of 1997 ASHRAE Handbook of fundamentals. Design conditions for the United States appear in Table 1a and 1b; for Canada in Tables 2a and 2b, and the international locations in Tables 3a and 3b.
- 2) The information provided in table 1a, 2a and 3a are for heating design that include:
  - Dry bulb temperatures corresponding to 99.6% and 99% annual cumulative frequency of occurrence,
  - Wind speeds corresponding to 1%, 2.5% and 5% annual cumulative frequency of occurrence,
  - Wind direction most frequently occurring with 99.6% and 0.4% dry-bulb temperatures and
  - Average of annual extreme maximum and minimum dry-bulb temperatures and standard deviations.

*Note that the percentages reflect annual cumulative frequency of occurrence of outdoor weather conditions.*

- 3) The information provided in table 1b, 2b and 3b are for cooling and humidity control that include:
  - Dry bulb temperature corresponding to 0.4%, 1.0% and 2.0% annual cumulative frequency of occurrence and the mean coincident wet-bulb temperature (warm). These conditions appear in sets of dry bulb (DB) temperature and the mean coincident wet bulb (MWB) temperature since both values are needed to determine the sensible and latent (dehumidification) loads in the cooling mode.
  - Wet-bulb temperature corresponding to 0.4%, 1.0% and 2.0% annual cumulative frequency of occurrence and the mean coincident dry-bulb temperature
  - Dew-point temperature corresponding to 0.4%, 1.0% and 2.0% annual cumulative frequency of occurrence and the mean coincident dry-bulb temperature and humidity ratio (calculated for the dew-point temperature at the standard atmospheric pressure at the elevation of the station).
  - Mean daily range (DR) of the dry bulb temperature, which is the mean of the temperature difference between daily maximum and minimum temperatures for the warmest month (highest average dry-bulb temperature). These are used to correct CLTD values.

- 4) Notes on Outdoor Design Conditions

- Winter design conditions shall be based on the 99.6 percent column dry bulb temperature in the *ASHRAE Fundamentals* Volume.
- Summer design conditions for sensible heat load calculations shall be based on the 0.4 percent dry bulb temperature with its mean coincident wet bulb temperature. Design conditions for the summer ventilation load and all dehumidification load calculations shall be based on the 0.4%



dew point with its mean coincident dry bulb temperature. Alternatively for reduced investments and energy efficiency use the 2.5% dry bulb for comfort cooling applications.

- For water-cooled heat rejection equipment (such as cooling tower) use ASHRAE 1% wet bulb and for air-cooled use ASHRAE 1% dry bulb conditions.
- Base the selection of evaporative equipment on the 2.5% wet bulb temperature. For applications where maintaining indoor temperature or humidity conditions is critical, the designer may use the corresponding 1.0% temperatures.
- For the selection of condensers and condensing units that will be subjected to unusually high radiation heat gain, add 3°C (5°F) to the dry bulb temperature specified above.
- Outdoor design temperature for heating to prevent freezing conditions should be based upon the 99% dry bulb temperature.
- Special purpose facilities will require special definition of appropriate interior design conditions.

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### **INDOOR DESIGN CONDITIONS**

Indoor cooling and heating design temperatures should be based upon local energy codes. Design conditions for specific occupancies are specified in ANSI/ASHRAE 55-1992.

For most of the comfort systems, the recommended indoor temperature and relative humidity are:

- **Summer:** 73 to 79°F; The load calculations are usually based at 75°F dry bulb temperatures & 50% relative humidity
- **Winter:** 70 to 72°F dry bulb temperatures, 20 - 30 % relative humidity

These are design conditions and not operating limits. All thermostats shall be adjustable between 60 to 85°F).

#### Notes on Indoor Design Conditions:

- 1) The indoor design temperature for comfort cooling could be taken as 8.3°C (15°F) less than the 2.5% outdoor design temperature, but should not be lower than 23.9°C (75°F) or higher than 25.6°C (78°F).
- 2) The indoor design specific humidity should not exceed the outdoor design specific humidity; otherwise, the indoor design relative humidity will be 50%.
- 3) The indoor design temperature for comfort heating should be 20°C (68°F) in areas with low levels of physical activity and 12.8°C (55°F) in areas with moderate to high levels of physical activity.
- 4) The indoor design temperature for freeze protection should be 4.4°C (40°F).
- 5) Winter humidification should be evaluated for the areas where the outdoor design temperature is -17°C (0 °F) and lower. The winter humidification system, if included, shall be designed to maintain 30% RH.
- 6) Mechanical cooling should be evaluated for the electrical equipment and transformer rooms for locations with the outdoor summer design temperature in excess of 31°C (90 °F); and where the use of a ventilation system would create excessive air movement, in excess of 30 air changes per hour.
- 7) The storage closets, with areas less than 5.6 m<sup>2</sup> (60 sq-feet) need not be cooled or heated.

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### **AIR CONDITIONING LOAD COMPONENTS**

Air conditioning heating & cooling load is generally specified in British thermal unit per hour (Btuh) or tons of refrigeration. One ton of refrigeration is equivalent to heat extraction rate of 12000 Btu per hour. One



Btu equals the amount of heat needed to raise the temperature of one pound of water one degree Fahrenheit. Used for measuring heating and cooling equipment output.

There are two distinctly different components of the air conditioning load; (1) the sensible load (heat gain) and (2) the latent load (water vapor gain).

#### Sensible Loads

There are 3 source categories of sensible heat gain:

- 1) Heat flow from warmer surroundings
- 2) Heat flow from solar radiation
- 3) Heat flow into the space from energy consuming objects within the space; these objects usually include:
  - lighting
  - office appliances
  - cooking or kitchen appliances
  - entertainment appliances
  - occupants within the space
  - any other energy consuming devices

#### Latent Loads

There are 2 source categories of latent heat gain:

- 1) Moisture entering the space from surroundings that are at a higher vapor pressure (higher humidity ratio) than the space.
- 2) Moisture generated within the space from moisture generating objects. These objects usually include:
  - occupants within the space
  - moisture generated by cooking or warming appliances
  - industrial or production machinery which evaporates water

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### ***BUILDING ARCHITECTURAL ASPECTS***

The building architecture has large impact on occupant comfort, HVAC sizing and energy costs. The following guidelines should be noted for energy efficiency:

- 1) **Shape:** The shape of the building has influence on the cooling and heating load. Ideally the building has to have the least aspect ratio (length/width ratio). The lower aspect ratio means the building has the least surface area of the building envelope (least wall area, glazing area and the roof area). Consider for example a 900 square feet area can be built as ~ 34 ft diameter room or can be made as 30 ft x 30 ft square or 60 ft x 15 ft rectangular. For a 10 ft height, the circular room shall have the surface area of 1067 sq-ft; for square the surface area is 1200 sq-ft and for rectangular the surface area is 1500 sq-ft. This indicates that not only the building will use less concrete, brick, wood, glazing or insulation but shall also have lower cooling and heating loss from the building envelope. Therefore the building shall be designed for least aspect ratio where possible.
- 2) **Latitude:**
  - In the lower latitudes there is total overheating, whereas in the higher latitudes overheating only occurs during the summer months.
  - Any breeze in the lower latitude (tropical and arid climates) is beneficial for most of the year whereas in higher latitudes most wind is detrimental and should be screened.

- 3) Orientation:
  - As a general guide, long, narrow buildings facing south with their long axis running east/west will have lower peak cooling loads and may be able to utilize smaller cooling equipment.
  - Buildings facing east or west with their long axis running north/south will have higher peak cooling loads and electricity demand costs, and may require larger cooling equipment.
- 4) Landscaping: Trees planted on the east, west and south sides of a one-or two-story building can effectively reduce summer solar heat gains through windows, which is one of the major contributors to the cooling load on an air conditioning system. External shading with vegetation with natural deciduous trees is very effective at providing shade and cooling by evaporating water through their leaves: during winter they are bare, allowing sunlight to pass through, but during summer they shade the building.
- 5) Day lighting: Day lighting with skylights and other types of architectural glazing features can provide natural lighting creating a pleasant working atmosphere. Day lighting strategies may be particularly effective using skylights in large open areas such as warehouses and manufacturing plants, and in office spaces where the electrical lighting system output can be efficiently varied over a wide range of light levels.
- 6) Shading: Shading devices are designed from knowledge of the sun's azimuth and altitude along with the wall-solar azimuth. Note the following:
  - Tropical regions need both vertical and horizontal shading throughout the year. In higher latitudes, horizontal and vertical shading is only needed during the summer on the south-facing sides of buildings.
  - Beginning at the equator and moving north, the need for solar heating increases while the need for solar shading diminishes.
  - Generally, for the tropical zones as much ventilation as possible is desired. For the arid zone cross ventilation is required, but care has to be taken to filter out high-velocity winds. In the temperate zone, cross ventilation and shielding are both necessary (for summer and winter, respectively). In the cool region, the building should be protected from cold, high-velocity winds, although cross ventilation is still required.
  - In the arid zone, the low level of humidity can be beneficial for evaporative cooling. In the tropical zone the high level of humidity can be very uncomfortable.
- 7) Zoning for transitional spaces: Transitional areas are one that does not require total climate control and natural ventilation may be sufficient. These include lobbies, stairs, utility spaces, circulation, balconies and any other areas where movement takes place.
  - For the tropical and arid zones, the transitional spaces should be located on the north and south sides of the building where the sun's penetration is not as great.
  - In temperate and cool zones the transitional spaces should be located on the south side of the building to maximize solar gain.
- 8) Use of atrium: In the tropical zone the atrium should be located in a way to provide ventilation within the built form. In the arid zone the atrium should be located at the centre of the building for cooling and shading purposes. For the cool and temperate zones the atrium should be at the centre of the building for heat and light.
- 9) Potential of roof/ground floor as useable exterior space: In tropical and arid climates there is a high potential to make use of all external spaces, whereas moving towards the northern latitudes the external spaces have to be covered.
- 10) Vertical cores and structure: The arrangement of primary mass can be used as a factor in climatic design as its position can help to shade or retain heat within the building form.

- For the tropical zone, the cores should be located on the east and west sides of the building form, so as to help shade the building from the low angles of the sun during the major part of the day.
- In arid zone, the cores should also be located on the east and west sides, but with major shading only needed during the summer. Therefore, the cores are located on the east and west sides, but primarily on the south side.
- The arrangement of the primary mass in the temperate zone is on the north face, so as to leave the south face available for solar heat gain during the winter. The cool zone requires the maximum perimeter of the building to be open to the sun for heat penetration. Therefore the primary mass is placed in the centre of the building so as not to block out the sun's rays and to retain heat within the building.

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## **BUILDING ENVELOPE**

The Building Envelope is made up of the elements of a building that enclose conditioned spaces and through which thermal energy may be transferred to or from the exterior. The building envelope uses no energy directly, but its selection strongly affects air-conditioning loads and can reduce the need for electric lighting. The key recommendations include:

### **Glazing U Value**

Fenestration is perhaps the most important part of the envelope design. The fenestration design has a significant impact on solar gains and in combination with interior space planning, determines the day lighting potential.

Glazing material should have a low thermal conductivity (U-value) or high resistance (R-value). Specify low U-factors < 0.40. Even lower values may be desired in extreme heating climates. The units of U-value are: Btu's per hour per square foot per °F (Btu/hr-ft<sup>2</sup>-°F). The following facts may be noted:

- a. U-factors usually range from a high of 1.3 (for a typical aluminum frame single glazed window) to a low of around 0.25 (for a multi-paned, high performance window with low-emissive coatings and insulated frames). The smaller the U-factor, the lesser will be the heat transfer between the inside and outside.
- b. The effects of non-solar heat flow are generally greater on heating needs than on cooling needs because indoor-outdoor temperature differences are greater during the heating season than during the cooling season. Therefore, for the heating season, the key rating parameter is the U-factor. U-factors allow consumers to compare the insulating properties of different windows and skylights. A window with a U-factor of 0.6 will lose twice as much heat under the same conditions as one with a U-factor of 0.3.

### General Recommendations

- c. **Northern Climate Recommendation:** Select windows with a U-factor of 0.35 or less. A low U-factor is useful during cold days when heating is needed. Some double-glazed low-e products have U-factors below 0.30. Some three-layer products have U-factors as low as 0.15.
- d. **Central Climate Recommendation:** Select windows with a U-factor of 0.40 or less. The larger the heating bill, the more important is a low U-factor becomes.
- e. **Southern Climate Recommendation:** Select windows with a U-factor lower than 0.75 and preferably lower than 0.60. A low U-factor is also helpful during hot days when it is important to keep the heat out, but it is less important than SHGC in warm climates.

### **Solar Heat Gain Coefficient (SHGC)**

The Solar Heat Gain Coefficient (SHGC) is the fraction of solar heat that enters the window and becomes heat. *The higher the SHGC the more shall be the solar gain potential through a given window.* The facts and recommendations are as follows:

- a. The SHGC is a ratio between 0 and 1.  $SHGC = 0$  means none of the incident solar gain is transmitted through the window as heat and  $SHGC = 1$  means all of the incident solar energy is transmitted through the window as heat.
- b. A window with a SHGC of 0.6 will admit twice as much solar heat gain as one with a SHGC of 0.3.
- c. Typically, windows with low SHGC values are desirable in buildings with high air-conditioning cooling loads while windows with high SHGC values are desirable in buildings where passive solar heating is needed.
- d. Window solar heat gain coefficients (SHGC) should be selected according to orientation. If south exposures are to admit beneficial solar heat during the heating season, their solar heat gain coefficients should be high. East or West facing windows that get a lot of undesirable sun in mornings and afternoons. Windows in hot climates should have lower SHGC assemblies.

#### General Recommendations

- e. **Northern Climate Recommendation:** To reduce heating, select the highest SHGC you can find (usually 0.30-0.60 for the U-factor ranges required in colder climates) so that winter solar gains can offset a portion of the heating energy need. If cooling is a significant concern, select windows with a SHGC less than 0.55.
- f. **Central Climate Recommendation:** If you have significant air conditioning costs or summer overheating problems, look for SHGC values of 0.40 or less. If you have moderate air conditioning requirements, select windows with a SHGC of 0.55 or less.
- g. **Southern Climate Recommendation:** A low SHGC is the most important window property in warm climates. Select windows with a SHGC less than 0.40.

For buildings where passive solar heating energy is desired, south-facing windows with high SHGC values coupled with low U-factors should be specified.

#### **Visible Transmittance (VT)**

The visible transmittance (VT) is an optical property that indicates the amount of visible light transmitted. While VT theoretically varies between 0 and 1, most values are between 0.3 and 0.8. The higher the VT, the more is the light transmitted. A high VT is desirable to maximize daylight. The following facts may be noted:

- a. A typical clear, single-pane window has a VT of 0.90, meaning it admits 90% of the visible light.
- b. High Glass Visible Transmittance (> 70%) is desired, especially for day lighting applications.
- c. Windows with spectrally selective or low-e coated glazing with low solar heat gain coefficients are effective in hot, sunny climates. Darker glazing tints also provide lower solar heat gain coefficients, but they may yield somewhat decreased visibility.

#### **Light-to-Solar Gain Ratio (LSG)**

The ratio between SHGC and VT is called the light-to-solar gain ratio (LSG.) This provides a gauge of the relative efficiency of different glass types in transmitting daylight while blocking heat gains. The higher the ratio number, the brighter shall be the room without adding excessive amounts of heat.

#### **Shading**

- a. Horizontally oriented adjustable shading devices are appropriate for south-facing windows, while vertically oriented adjustable devices are more effective for shading windows on east and west orientations.
- b. Low-e windows and skylights are the best options for decreasing the transmission of ultraviolet radiation.
- c. Single-pane windows are impractical in heating-dominated climates. In these regions, multiple-pane, low-e, and gas-filled window configurations are advisable.

- d. Reflective window coatings reflect heat away from windows, as well as cutting glare and reducing fading of furniture, drapes, and carpeting inside the house. Note: Do not place reflective coatings on south-facing windows if you want to take advantage of heat gain during the winter.
- e. Double glass is most effective in areas where the conduction component is quite large.
- f. Solar films are more effective in areas of moderate, sunny climates, i.e. Florida.
- g. If exterior or interior shading devices, such as awnings, louvered screens, sunscreens, Venetian blinds, roller shades, or drapes, will be used on windows, lower window solar heat gain coefficients may not be necessary, depending on individual circumstances. Many shading devices can be adjusted to admit more or less solar heat according to the time of day and the season, but windows with lower solar heat gain coefficients require less maintenance.
- h. A properly installed awning can reduce heat gain up to 65% on southern windows and 77% on eastern windows. Also effective are louvers and shutters.
- i. Exterior shading devices are more effective than interior devices in reducing solar heat gain because they block radiation before it passes through a window. Light-color shades are preferable to dark ones because they reflect more and absorb less radiation.

### **Leakage Rating**

An air leakage rating is a measure of the rate of infiltration around a window or skylight in the presence of a strong wind. Air leakage ratings allow consumers to compare the air tightness of different windows and skylights as manufactured.

- a. Select windows with air leakage ratings of 0.2 cubic feet per minute per square foot of window area (CFM/ft<sup>2</sup>) or less. Check the seals between window components for air tightness. To minimize infiltration around installed windows, caulk and weather-strip cracks and joints.
- b. Operable, rather than fixed, windows should be installed in household areas with high moisture production, such as bathrooms, kitchens, and laundry rooms, and in other areas where natural ventilation is desired.

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### ***BUILDING ENVELOPE (Walls and Roof)***

- 1) North-facing is oriented to within 45 degrees of true north, including 45°0'0" east of north (NE), but excluding 45°0'0" west of north (NW).
- 2) Opaque walls must meet a maximum U-value (or minimum insulation R-value) requirement. The R-value of a material is "the [thermal] resistance of a material or building component to the passage of heat in (hr x ft<sup>2</sup> x °F)/Btu." The R-value indicates how well a material prevents heat from flowing through it. R-19 insulation, for example, is only half as effective at slowing heat transfer as R-38 insulation.
- 3) U-value is the inverse of the sum of R-values ( $\Sigma R$ ), which include resistances of surfaces, structural components, and insulation. For wood walls, the maximum U-value should be 0.10, and for metal framed walls the recommended limit is 0.15. Note that the U-value (coefficient of heat transmission) is a measure of the ability of a building envelope to transfer heat; it describes the insulating properties of building envelope components such as walls and roofs. Use building insulation to minimum ASHRAE 90.1 standards. For any type of wall, R-11 insulation will satisfy this requirement of U-factors stated above. An insulation value of R-11 is attainable with about 3½ inches of fiberglass batt insulation or with 2 to 3 inches of foam board insulation. Cellulose or fiberglass blown insulation in a 2x4 framed wall will also meet this requirement.
- 4) Walls with a heat capacity (HC) greater than 7.5 Btu/°F-ft<sup>2</sup> of wall surface area need not to have to be insulated. This includes most, but not all, concrete and concrete block walls.
- 5) Portions of walls shaded by overhangs with a projection factor greater than or equal to 0.2 on north-facing walls or 0.3 on all other orientations need not have to meet the insulation requirement above.

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## **INDOOR LIGHTING**

Each kilowatt hour of lighting requires 3,412 Btu/hr of air cooling. One ton of air conditioning is equivalent to heat extraction rate of 12,000 Btu/hr. Every 100 kilowatts removed during a lighting retrofit cuts cooling load by about 28 tons - enough capacity to cool an 8,000 square foot commercial space. Note the facts below:

- a. A lumen is a measure of the amount of light available from a given light source. A watt is a measure of the power requirement for that light source. The efficacy of a light source is measured by dividing the lumens by the wattage. The more usable light that a light source provides per watt, the greater its energy efficiency.
- b. Use compact fluorescent lamps (CFL); this lasts 10 to 13 times longer than the traditional incandescent light bulb, while consuming 75 percent less energy.
- c. Use fluorescent lamps; this produces approximately five times the lumens per watt of standard incandescent lamps and consumes less energy.
- d. Use T10 lamps; this produces 24% more light than a F40-T12 while consuming 3% more energy.
- e. Use metal halide lamps in place of mercury / sodium vapor lamps
- f. Use high frequency (HF) electronic ballasts in place of conventional magnetic ballasts
- g. Use parabolic luminary's fixtures; this can light at half the watts per square foot normally used in conventional lighting designs.
- h. Use auto controller strategies such as occupancy sensors, motion sensors, photoelectric sensors, dimming devices etc for switching on-off the lighting fixtures.
- i. Design for proper lighting. In general, the light should be brightest on your immediate work area, but should not over-illuminate. Lighting levels should decrease as you move into the general environment of the room.
- j. Consider light troffers for suspended ceilings with ducted supply and plenum return.

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## **EQUIPMENT AND APPLIANCES**

The effect of heat gain from equipment is almost similar to the lighting. Internal heat gain from all appliances—electrical, gas, or steam—should be taken into account. When available, manufacturer-provided heat gain and usage schedules should be utilized to determine the block and peak cooling loads. The heat gain from electrical equipment is based on the data from tables 20 and 21 of chapter 26 of 1985 ASHRAE fundamentals handbook or chapter 28 of 1997 edition.

Typical rate of heat gain from selected office equipment should be based on the latest edition of the ASHRAE Handbook of Fundamentals. Computers, printers, fax machines and copiers consume energy even when these are not in use. With the widespread use of desktop computers, printers and other devices, the commercial establishment can have hundreds of units and the heat gain can just add up. The electrical power requirements for office equipment are available in 2001 ASHRAE Fundamentals, Chapter 29, and Tables 8, 9, & 10.

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## **OCCUPANCY**

Occupants generate both sensible and latent heat components according to activity level. The sensible heat rate increases slightly with higher activity but latent heat increases dramatically because of greater perspiration rates needed to maintain body temperature.

For office spaces, the average density of the *occupiable floor area* of a building is one person per 100 usable square feet. Within areas occupied by workstations, the occupancy load can be as dense as one person per 75 usable square feet in local areas.



Block loads and room loads should be calculated accordingly. Sensible and latent loads per person should be based on the latest edition the *ASHRAE Handbook of Fundamentals*.

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### **VENTILATION –OUTSIDE AIR**

- 1) Base outside air amounts on local code or ASHRAE 62-1999 (20 CFM per person).
- 2) Minimum Outside Air Requirements: While computing the minimum outside air, the following criteria apply:
  - Use 20 CFM per person or 1 air change per hour.
  - Some building codes and utility programs also use 0.35 air changes per hour (ach) as a ventilation target. (To quickly estimate the air flow in CFM needed to meet the 0.35 ach requirement, divide the floor area in square feet by 20).
  - For all air systems, minimum outside air shall be 15% of the supply air volume.
- 3) Perform calculations for each occupancy type to avoid excessive outside air. According to ASHRAE Standard 62-1999, "Where peak occupancy of less than three hours duration occurs, the outdoor air flow rate may be determined on the basis of average occupancy for buildings for the duration of operation of the system, provided the average occupancy used is not less than one-half the maximum." Spaces having intermittent or variable occupancy may then have lower ventilation rates than would be required if the peak occupancy were used to determine the amount of ventilation (e.g., a conference room).
- 4) Apply diversity factors allowed by code to account for non-simultaneous occupancy. The outdoor air quantity shall be increased, if required, to meet the space exhaust.

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### **DESIGN CONSIDERATIONS FOR OPTIMUM VENTILATION AIR**

The outdoor air for ventilation shall be computed in accordance with the ASHRAE 62-1989 standards; however how it is supplied to the occupied spaces would depend upon the type of HVAC system selected.

- 1) **All Air Systems:** With an all air system (example air-handling units), the outside air is mixed with the return air, and the supply air is distributed to various zones. Since the distribution of the supply air is done on the basis of the cooling load requirements, the zones with greater cooling requirements will, naturally, receive more outside air. It is, thus, recognized that the uniform distribution of outdoor air is inherently difficult with an all air system.
- 2) **Terminal Cooling/Heating Systems:** With terminal cooling/heating systems, secondary equipment such as induction units, fan coil units or radiant panels are distributed at various zones. The source of outdoor air is external to the principal cooling and heating equipment and is normally catered for by a 100% fresh air handling unit. This type of arrangement is common in hotel rooms air-conditioning.
- 3) Following is recommended for the ventilation design:
  - Locate intake louvers or fans to make use of prevailing winds
  - Locate fans and intake ventilators for maximum sweeping effect over the working area.
  - If filters are used on gravity intake, size intake ventilator to keep intake losses below 1/8" SP
  - Avoid fans blowing opposite each other, when necessary separate by at least 6 fan diameters.
  - Use Class B insulated fan motors where ambient temperatures are expected to be high for air-over motor conditions.
  - If air moving over motors contains hazardous chemicals or particles, use explosion-proof motors mounted in or out of the airstream, depending on job requirements.



- For explosive atmosphere applications use fans of non-sparking construction. For blast proof building construction in many chemical plants use blast proof valve for intake air.
- Ventilate the building during the coolest parts of the day or night, and seal it up during the hottest part of the day.
- Ventilated attics are about 30°F cooler than unventilated attics. Properly sized and placed louvers and roof vents help prevent moisture buildup and over heating the attic.
- Provisions should be made for the use of 100% unconditioned outdoor air whenever ambient conditions satisfy design and humidity requirements.
- Design the HVAC system with the outdoor air rates required by ASHRAE Standard-62 to maintain indoor air quality. "Build Tight & Ventilate Right".

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### **EXHAUST AIR REQUIREMENTS**

For the space to be maintained under negative pressure exhaust 15 percent more air than the supply. For the space to be maintained under positive pressure, exhaust 15 percent less air than the supply air. In the absence of any specified supply air change/hour, the exhaust air shall be calculated from the following considerations, and the maximum value, thus, derived, shall be used:

- 1) The toilet should be exhaust @ 10 air changes per hour or 2 CFM per sq-ft.
- 2) The mechanical equipment room should be exhausted @ 6 to 12 air changes per hour or in accordance with manufacturer's instructions. The exhaust air requirements shall be coordinated with the equipment specifications. Maintain these spaces under negative pressure.
- 3) The battery rooms shall be exhausted with twin explosion proof fans @ 10 air changes per hour. The hydrogen concentration should not be allowed to exceed 1ppm.
- 4) Individual toilets do not require ducted supply air. Use air transferred from the occupied spaces via door grilles and/or undercuts, to exhaust these areas. Do not transfer more than 150 CFM of air per door undercut.
- 5) Public toilets and congregate baths do require ducted supply air up to 8.5 air changes per hour maximum. The balance air should be drawn from the corridors to maintain negative pressure and to ensure exhaust of 10 air changes per hour.
- 6) The ducted supply air shall be used to ventilate the locker rooms and the public toilets equipped with multiple fixtures. To maintain 15% negative pressure in public toilets, room air from the corridors shall be transferred as make up by door louvers and/or undercuts. All air supplied to the locker rooms (and make up air transferred from the corridors to maintain negative pressure) shall be exhausted outdoors through the toilets/showers adjoining the locker room.
- 7) The designer shall ensure that the make-up air required to maintain negative pressure, and the excess air to maintain positive pressure, are available and taken into account in the air balance calculations.

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### **BUILDING PRESSURIZATION**

To keep dry air flowing through building cavities, systems shall be designed with sequence of operations that assure continuous positive pressure with respect to the outdoor environment until the outdoor temperature falls below 4.5°C (40°F), when the building pressure shall be brought to neutral. These building HVAC systems shall have an active means of measuring and maintaining this positive pressure relationship. The BAS shall alarm when the building pressurization drops below a predetermined low limit. In areas where exhaust systems are used or an indoor air quality contaminant source is located, a negative pressure shall be maintained relative to surrounding spaces.

Calculations shall be provided that show the minimum outside airflow rate required for pressurization. Minimum outside airflow rates shall be adjusted as necessary to assure building pressurization. The recommended values of building pressurization are 0.03 inch-w.g.

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### **INFILTRATION**

- 1) Infiltration is uncontrolled inward air leakage from outside a building, or unconditioned space, including leakage through cracks and interstices, around windows and doors, and through any other exterior or demising partition or pipe or duct penetration. Infiltration rates of not exceed 1/2 to 1/3 air changes per hour is considered OK. Ensure that air tightness meets this parameter.
  - 2) Design air distribution systems to maintain a slightly positive pressure (0.03 inch-w.g.) within the area served in order to reduce or eliminate infiltration unless there is a valid need to maintain a negative pressure in that area.
  - 3) Infiltration of air can be controlled in various ways, many of which are mandatory measures and therefore considered "standard" in new construction. Standard infiltration control measures include weather-stripping, caulking and sealing in and around all exterior joints and openings.
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### **REFRIGERATION CYCLE**

There are two basic refrigeration cycles: one compresses the refrigerant gas and the other absorbs the refrigerant at very low pressures.

#### **Compression Refrigeration**

In the compression refrigeration cycle, HCFC-22 or HFC134a refrigerants remove heat from a medium by changing phases. The refrigerant is compressed in its gaseous phase and passed through a condenser. The refrigerant gas leaving the condenser is then expanded through a valve which lowers its pressure and temperature further until a phase change occurs and a liquid is formed. The liquid refrigerant then passes through an evaporator coil and the water or air flowing through the evaporator gives up its heat to the cold refrigerant making it boil. The refrigeration cycle thus has four paths: evaporator (suction line) to compressor, (hot gas line) to condenser, (liquid line) to metering device, then back to evaporator.

Note the following from energy conservation point of view:

- 1) **Suction pressure:** Lowering the evaporator temperature 10deg from a base of 40°F and 105°F reduces the capacity about 24%, and at the same time increases the compressor HP per ton about 18%.
  - 2) **Discharge Pressure:** Increasing the condensing temperature 15° from a base of 40°F and 105°F reduces the capacity about 13% and increases the compressor HP per ton about 27%. Condensing water temperature for chillers should be no less than 65° F and no greater than 85°F.
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### **ABSORPTION REFRIGERATION**

- 1) In absorption refrigeration machines, the absorption cycle uses a heat-driven concentration difference to move refrigerant vapors (usually water) from the evaporator to the condenser. The high concentration side of the cycle absorbs refrigerant vapors, diluting the absorbent material.
- 2) Lithium bromide is the most common absorbent used in commercial cooling equipment, with water used as the refrigerant. Lithium bromide has a very high affinity for water and is relatively inexpensive and non-toxic. However, it can be highly corrosive and disposal is closely controlled.

#### **Notes on Absorption Refrigeration:**

- 1) Absorption chillers must operate at very low pressures, about 1/100<sup>th</sup> of atmospheric pressure for the water to vaporize at the required -- 40°F.

- 2) Absorption chillers are most economical where waste heat is available. Waste heat can result from incineration of solid wastes, heat recovery engine or gas turbine exhausts, or exhaust from steam turbine drives for refrigeration compressors or electric generators.
- 3) Commercial absorption chillers are available in steam driven or direct fuel fired heat source options.
- 4) Water quality and operating temperatures is very important feature of absorption chillers otherwise problems like crystallization of Lithium Bromide can occur.
- 5) Use a throttling valve to modulate flow to the absorbent generator with chilled water temperature, as well as an automatic steam valve which reduces steam pressure and temperature, for energy efficient part load capacity control.
- 6) Consider using two-stage absorption refrigeration whenever high-pressure steam or high-temperature water is available. These are much more efficient than the single stage machines which use low pressure steam. The life cycle cost analysis will address the economic feasibility of using the higher first-cost, two-stage machine.

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### **COMMON COOLING SYSTEMS**

- 1) **Chilled Water Systems:** In chilled water system the air is cooled with chilled water passing through the tubes of a finned coil (cooling coil). The chilled water system comprises of air cooled (or water cooled) chillers complete with pumping and piping.  
Recommended criteria:
  - Cooling Load 422 kW (120 tons and below): Provide a single chiller with multiple compressors.
  - Cooling Load above 422 kW (above 120 tons): Provide two chillers, each with multiple compressors.
- 2) **DX System:** In DX systems, the air is cooled with refrigerant passing through the tubes of the finned cooling coil. These comprise of a hermetic sealed compressor/s, evaporator (cooling coil fabricated out of copper tubes and aluminum fins), a supply air fan, filter and a condensing unit. DX systems are essentially the factory assembled self-contained units and are also known as local systems. Window air-conditioners, package units and split systems are typical examples of DX systems. The application and unit capacity ranges are as follows:
  - Room air conditioner (capacity range of 0.5 to 3 TR per unit, suitable for an area of not more than 1000 square feet)
  - Packaged unit integral air-cooled condenser (capacity range of 3 to 50 TR, suitable for a maximum an area of 1000 – 10000 square feet)
  - Split system with outdoor air-cooled condenser (capacity range of 0.5 to 50 TR, suitable for an area of 100 – 10000 square feet)
- 3) **Heat Pump:** A heat pump is a device that acts as an air conditioner in the summer and as a heater in the winter. Heat pumps look and function exactly like an air conditioner except it has a reversible cycle. It contains a valve that lets it switch between "air conditioner" and "heater." When the valve is switched one way, the heat pump acts like an air conditioner, and when it is switched the other way it reverses the flow of Freon and acts like a heater.

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### **EQUIPMENT SIZING CRITERIA**

- 1) **Peak Sensible Load:** Select unit(s) with a sensible capacity not less than 100 percent of the calculated peak sensible cooling load. The 1989 version of ASHRAE 90.1 "Energy Standard for Buildings except Low-Rise Residential" called for selecting units with capacities up to 110 percent of the sensible cooling load, but this was deleted from the 1999 version. In most cases the calculated load will fall between standard units sizes, so selecting the closest size which provides at least 100

percent of the sensible load will usually result in a margin of extra capacity. Since the equipment runs at part-load a majority of the time, avoiding oversized equipment will result in lower energy use.

- 2) **Peak Latent Load:** The latent cooling capacity of the unit(s) should equal or exceed the calculated latent-cooling load. An air conditioner with a latent capacity of 25 to 30 percent of the total capacity (a sensible heat ratio of 75 to 70 percent, respectively) can be effective in most areas, although very humid or very dry climates may require respectively higher or lower latent capacity.
- 3) **Safety Margin:** A safety margin of 10% is recommended on the calculated internal heat gain and loss to allow for any future increase in internal heat gain or other load demand. The calculated air quantities for the occupied spaces shall be based on the addition of the safety margin.
- 4) **Air Leakage:** An allowance of 4% shall be made for the leakage of air through the ductwork before it reaches the occupied spaces. The leakage allowance shall be applied to the sum of all individual peak air quantities without any diversity.
- 5) **Static Pressure:** A safety margin of 10% shall be added to the fan total static pressure of the system.

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### ***DIRECT EXPANSION (DX) or UNITARY SYSTEMS***

Depending upon the configuration of the building and available space, the following arrangements shall be evaluated:

- 1) **Individual Air Conditioner:** A type of cooling equipment installed in either walls or windows (with heat-radiating condensers exposed to the outdoor air). These self-contained units are characterized by a lack of pipes or duct work for distributing the cool air; the units condition only air in the room or areas where they are located.
- 2) **Packaged Unit:** A type of heating and/or cooling equipment that is assembled at a factory and installed as a self-contained unit. Packaged units are in contrast to engineer-specified units built up from individual components for use in a given building. This equipment differs from individual air conditioning equipment in that air ducts are used to move the conditioned air to and from the unit. These are available in single or multiple, commercial grade package units with air cooled, or water cooled, refrigeration condensers. The two main categories of packaged air-conditioning units are Rooftop Units and Split Systems.
- 3) **Roof Top Units:** A roof-top packaged unit is an air handling unit (AHU) with its duct, coils, fans, and controls packaged in a weather proof box and are provided with weather proof components to withstand the elements. It is also designed to be supported on a roof with supply and return air ducts located on the bottom of the unit.
- 4) **Remote Units:** Conventional horizontal, or vertical, draw-through indoor, DX air handling units with remote, air cooled, or water cooled, condensing units.
- 5) **Single packaged units** have their duct, coils, fans, and controls packaged in weather proof box and are provided with weather proof components to withstand the elements. They are designed to be located outside and supported on a concrete pad or steel frame from the bottom with supply and return air ducts located horizontally or vertically. Console under window units are also available without duct connections.
- 6) **Split heating, air conditioning and heat pump units** have their ducts, coils, fans, and controls packaged in two boxes. A weather proof exterior box for the compressor, condenser coil, condenser fan and controls is designed to be supported on a concrete pad or steel frame from the bottom. An interior box is provided with supply and return air duct connections, fan, and drip pan. Connections for refrigerant tubing are provided on both boxes. This tubing is routed as needed and insulated to prevent heat loss or gain in the refrigerant and to prevent condensate from forming on the tubing.

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### ***CHILLED WATER SYSTEMS***

Chilled water systems are the engineered systems that are generally supplied as the custom built packaged units or field assembled units. Central systems provide opportunity for economies of scale and results in low capital and operating costs. Refrigeration machines which chill water are often called chillers. Note the facts below:

- 1) A chiller is a mechanical refrigeration device that cools a fluid (usually water). The chilled water is then distributed throughout the building by use of pipes. When large air conditioning is required it is sometimes more feasible to use one large chiller instead of many small air conditioners.
- 2) A "chiller" consists of four major components; a compressor, an evaporator heat exchanger, a condenser heat exchanger and an expansion valve. Compressors are usually of reciprocating, scroll, centrifugal or rotary screw types. The evaporator is usually shell & tube heat exchanger where chilled water would be produced and the condenser can be either air-cooled or water-cooled. Water-cooled chillers use water to transport the heat rejected in the condensers. The water (called "condenser water") is cooled in a cooling tower. "Air-cooled" chillers have condensers which are cooled with ambient air.
- 3) Centrifugal refrigeration machines use centrifugal compressors to compress the refrigerant gas. Centrifugal compressors compress gas as it enters the center of a fluted casting, housing a rotating impeller. The impeller imparts kinetic energy to the gas which turns into potential energy as the gas velocity slows, thus increasing pressure. Compression is a continuous process. One, two, or three stages may be used. Centrifugal compressors are used in large systems.
- 4) Reciprocating or positive displacement refrigeration machines use reciprocating or piston type compressors to compress the refrigerant gas. The gas enters a cylinder through a valve when a piston in the cylinder is going down. The valve closes when the piston starts to go up. As the piston approaches the top of the cylinder the gas is compressed by the decreasing volume. An exhaust valve opens when the piston is near the top of the cylinder allowing the compressed gas to exit. The cycle is then repeated.
- 5) Rotary screw refrigeration machines use rotary helical screw compressors to compress refrigerant gas or increase the pressure of liquid refrigerant. They are positive displacement machines. The twin-screw compressor consists of accurately matched rotors [one lobe (male) and one helix (female)] that mesh when rotating within a close tolerance common housing. One rotor is driven and geared to the other, turning it in a counter-rotating motion.
- 6) Scroll refrigeration machines employ a stationary scroll and a motor driven orbiting scroll to gather refrigerant gas at the circumference and move it through an ever decreasing volume to the center of the scroll where it exists at a higher pressure.
- 7) The screw compressor can operate at any discharge pressure as long as it is within the operating design pressure of the compressor casing. Screw compressors can operate up to 30 compression ratios; reciprocating compressors can operate at compression ratio up to 10. A centrifugal compressor is designed for specific pressure differential (generally 18) and if the suction or the discharge pressures vary, it usually is a hunting or surging condition.
- 8) For air cooled condenser duty, brine chilling or other high head applications, reciprocating and screw liquid chillers are more frequently installed.
- 9) Centrifugal chillers require water-cooled condensers. Air-cooled condenser is not a recommended option for the centrifugal machines operating at low-pressure refrigerants. The screw and reciprocating machines are available in both air-cooled and water-cooled condenser options.
- 10) Where cooling by mechanical refrigeration is justified, the compressor capacity should be provided with two or more compressors rated to provide approximately two thirds of system requirements with any one compressor out of service.
- 11) The centrifugal machine accomplishes the capacity reduction in stages. While this control method is effective down to about 20% of chiller's rated output, it results in decreased operating efficiency. For example a chiller rated at 0.6 kW per ton at full load might require as much as 0.9 kW per ton when lightly loaded.

- 12) Centrifugal Compressor Capacity Control is achieved by variable inlet guide-vane assembly throughout throttling range from full load to 20 percent load.
- 13) The capacity control in reciprocating machine is achieved through 'On-Off' or 'Loading- Unloading' of compressor cylinders.
- 14) Variable frequency drives (VFD) can improve the part-load performance of centrifugal chillers. By reducing the voltage and frequency of the power provided to the HVAC equipment, VFDs can keep part-load operating efficiencies at near-full-load ratings over a wide range of loads.
- 15) The screw compressor offer infinite reduction from 100% to 0% that allows obtaining higher refrigeration capacity from the same machine and affords good part load efficiency.
- 16) The reciprocating compressors stepped capacity control is most efficient at minimum load than the twin-screw compressor or centrifugal compressor.
- 17) The reciprocating compressors are the cheapest followed by centrifugal compressors. Screw compressors are usually more expensive by 10 to 15% than centrifugal at the same operating conditions.
- 18) Centrifugal chillers most commonly use a low-pressure refrigerant such as HCFC-123 or a medium pressure refrigerant such as HFC-134a. The screw and reciprocating machines are positive displacement machines and use higher-pressure refrigerant.
- 19) The use of low-pressure refrigerants to suit the characteristics of the centrifugal compressor implies that the evaporator operates below atmospheric pressure. It has to be ensured that air and moisture do not leak into the system otherwise corrosion may develop which can damage the compressor internals.
- 20) 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
- 21) Compliance:
  - Comply with ASHRAE 15 for chiller design, construction, leak testing, and installation.
  - Comply with ASME Boiler and Pressure Vessel Code: Section VIII, "Pressure Vessels," for constructing and testing evaporator and condenser pressure vessels. Stamp with ASME label.
  - Comply with UL 465.
  - Comply with NFPA 70
  - Performance Testing Requirements in accordance with ARI 550 and ASHRAE 30.
  - Sound-Power-Level Ratings in according with ARI 575.

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### **CHILLER POWER CONSUMPTION**

- 1) Most chillers operate between 40 percent and 70 percent of capacity a majority of the time and rarely operate at full load. Select chillers with a high-integrated part-load value (IPLV) rating so that they operate efficiently under full and part load conditions.
- 2) Consider selecting multiple chillers of different capacities to provide flexibility in meeting varying loads in addition to selecting chillers with high IPLV ratings. It is better to operate chillers near full capacity



and start up additional chillers as needed than it is to operate large chillers at part load most of the time.

- 3) Achieve improved compressor part-load kW/ton ratings by selecting a variable-speed-drive (VSD) on the compressor. The VSD allows the compressor to run at lower speed under part-load conditions.
- 4) Operating chillers to provide higher water supply (CHWS) temperatures increases the efficiency and provides greater cooling capacity (tons) for a given size and constant condenser water supply (CWS) temperature.
- 5) The chilled water temperature differential across the chiller be optimized but shall not be less than 10°F and ideally be selected at 16°F. Consider the impact of piping, pumping cost, cooling coil performance and chilled water piping arrangement in order to optimize the chilled water differential.
- 6) A primary/secondary piping arrangement shall be used to ensure hydronic separation between primary and secondary loops. Primary/Secondary Arrangement: The chilled water flow through the evaporators shall be kept constant. The chilled water flow downstream to the cooling coils, however, shall be variable with automatic, two-way, modulating control valves controlling the flow.
- 7) The energy use of central plant and distribution systems can vary by a factor of two or more based on the system design and operation. For example, an air-cooled chiller could have an energy use of 1 kW/ton or more; whereas, the same sized water-cooled chiller with a cooling tower can have an energy use of less than 0.6 kW/ton.
- 8) The electric, water-cooled, centrifugal and rotary-screw chillers shall be selected in accordance with the ARI Standard 550/590-98 for the lowest full load kW/ton which can be met by at least three manufacturers. Executive Order 13123 and FAR Section 23.704 requires that products be purchased in the upper 25 percent of energy efficiency. Accordingly, DOE recommends following chiller efficiencies, which are based on standard reference conditions, specified in ARI Standard 550/590-98.
- 9) Follow new ARI Standard 550/590-98 on all projects. The two major changes are in regard to the integrated part load value (IPLV), and the fouling factor adjustments used for evaporators in closed loop water systems such as chilled water systems. The IPLV rating conditions and part load weightings are changed to more closely reflect actual operating experience found in the field for a single chiller. The evaporator fouling factor is changed from 0.00025 to 0.0001 based on research sponsored by ASHRAE (American Society of Heating, Refrigerating, and Air- Conditioning Engineers). With ARI certified chillers no field testing of equipment is required for performance and capacity. However, chillers need to be tested in the field for proper and safe operation of the equipment.

#### Chiller Efficiencies for Standard ARI Conditions

Chiller Type	Full Load-kW/Ton	IPLV-kW/Ton
Centrifugal 150-299 Tons	0.59 or Less	0.52 or Less
Centrifugal 300-2000 Tons	0.56 or Less	0.44 or Less
Rotary-Screw 150 Tons or Greater	0.64 or Less	0.49 or Less
Reciprocating	0.85 or Less	



### **AIR-CONDITIONER EFFICIENCY**

When choosing between units with similar prices, capacities and features, energy efficiency should be the deciding factor. Even though an energy efficient unit may be higher priced, it may still be the best buy. High efficiency appliances cost less to operate and can pay back the extra initial cost many times over during their lifetimes.

All room air conditioners bear bright yellow Energy Guide labels which provide information on energy efficiency. Energy Guide labels are mandated by Congress as part of the Energy Policy and Conservation Act of 1975. The label displays an energy efficiency rating (EER) in large black numbers. The higher the rating, the more efficient is the appliance. Units with an EER of 9.0 or above are considered very efficient. To help you compare units, a range for competing air room conditioners of the same cooling capacity is printed on the Energy Guide label below the EER.

The label also provides a cost/use chart to calculate the cost of operating the appliance based on local electricity rates and expected hours of use.

Central air conditioners are rated according to their Seasonal Energy Efficiency Ratio (SEER). Like the EER, the higher the SEER, the more efficient the system. To compare the efficiency of two units with equal cooling capacity, take the difference in SEER's and divide by the large SEER. For example, if system A has a SEER of 6, and system, B has a SEER of 8, system B will provide the same amount of cooling as system A while consuming 25% less energy ( $8 - 6 = 2$ ;  $2/8 = .25$ ). Finally, compare warranties and maintenance agreements when buying an air conditioner. In summary:

- 1) Central air conditioners with output less than 65,000 Btuh are rated according to their Seasonal Energy Efficiency Ratio (SEER), the total cooling output of a central air conditioner in Btu during its normal usage period for cooling divided by the total electric energy input in watt-hours during the same period.
- 2) The SEER takes into account losses such as electricity for fan power and cycling on and off. The SEER is generally higher than the EER, since it takes into account seasonal performance over a 12 month period.
- 3) The SEER is a function of both the condensing unit (compressor) and the type of coil selected to be used with it. Equipment manufacturers typically list efficiencies only of certain compressor-coil combinations. Only the SEER rating determined from the compressor in combination with the largest selling national coil may be used for compliance with the building energy efficiency standards.
- 4) Central air conditioners with outputs of 65,000 Btuh or more are rated using an EER.

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### **REFRIGERANTS**

The refrigerants are synonym with major environmental concern – “Ozone layer depletion”.

The refrigerant shall be fully compatible with all local, state, and federal regulations. The refrigeration equipment selection shall be based on the new EPA approved hydrochlorofluoro-carbon refrigerants, such as, HCFC 123, HFC 134a, and HCFC 22. The latest versions of ASHRAE Standards 15 and 34 shall be followed to ensure its full compliance.

The table below highlights certain facts about most commonly available refrigerants and the refrigerants of the future.

**Summary of Status of Some Refrigerant Groups**

Type	Examples	ODP*	GWP**	Uses	Other Issues
CFC	R12 R502 R11	High	High	Widely used in most applications until 1990.	Now phased out of production
HCFC	R22 R409A R411B	Low	High	Widely used in many applications. Not recommended for use after 1999.	To be phased out of production in 2015. Their use is also regulated increasingly strictly.
NH <sub>3</sub> Ammonia	R717	Zero	Very low	Used in industrial systems since the birth of refrigeration.	Toxic and flammable, reacts with copper.
HFC	R134a R404A R407C R410C R507	Zero	High	Started to be used in place of CFCs from about 1990.	Different compressor oil needed, performance of some HFCs not as good as CFCs. Some reliability problems.
HC E.G. propane, iso-butane	R600a R290 Care 30 Care 50 R1270	Zero	Very low	R290 used in some industrial systems for decades. R600a now used in domestic systems. Care 30 and Care 50 now used in some commercial systems.	Flammable, but are very good refrigerants with few changes needed to a CFC/HCFC system.
CO <sub>2</sub>		Zero	Very low	Widely used before the 1950s but superseded by halocarbons. Now being 'rediscovered' as a primary and secondary refrigerant.	Not yet widespread commercial use as a primary refrigerant, but an interesting prospect. (High operating pressures require special materials and construction.)

\*ODP – Ozone Depleting Potential; \*\*GWP – Global Warming Potential

Select the refrigerant keeping in view the Ozone depletion issue, the long term availability, the operating temperature and pressure conditions, the type of compressor and safety. Consider zeotropic blends that may give advantages like increased evaporator capacity and efficiency. Zeotropic blends are more suited for DX systems, check with the supplier before using them in flooded systems.

## EVAPORATOR

A chiller evaporator has shell and tube design with refrigerant in tubes and water circulated in the shell divided by baffles. Most typical application involves use of chilled leaving temperature of 45°F and entering temperature of 55°F. Higher water supply (CHWS) temperatures increases the efficiency of the chiller and provides greater cooling capacity (tons) for a given size whereas selecting evaporator at higher temperature differential provides benefits on reduced pipe sizes, pumping cost, cooling coil performance.

- 1) Description: Shell-and-tube design with refrigerant in tubes and water circulated in the shell divided by baffles

- Shell material – Carbon steel plate
  - Water tubes – Seamless copper or 90/10copper/nickel alloy tubes expanded into tube sheets; individually replaceable with standard, removable, carbon-steel water boxes.
- 2) Standard working pressures:
    - Refrigerant Working Pressure: 300 psig (2070 kPa).
    - Water Side Working Pressure: 150 psig (1035 kPa).
  - 3) The evaporator can be specified with standard water boxes or marine water boxes. The standard boxes are less expensive than marine water boxes but whereas standard boxes require piping removal for maintenance, marine water boxes allow access to tubes with piping in place.
  - 4) Insulate evaporator, suction lines, and other surfaces where condensation might occur. Typically ½ or ¾ inch flexible electrometric insulation is applied.
  - 5) Pressure Relief: Use frangible-carbon rupture disc option for HCFC-123 (low pressure) machines and resetting type spring loaded relief valve option for HFC-134a (medium-high pressure) machines.

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### **WATER COOLED CONDENSER**

A water cooled condenser has shell and tube design with refrigerant totally enclosed in shell and water in the tubes. Under full load conditions the design condenser cooling water inlet temperature is 85°F and leaving temperature is 95°F.

- 1) **Description:**
  - Shell material – Carbon steel plate
  - Water tubes – Seamless copper or 90/10copper/nickel alloy tubes expanded into tube sheets; individually replaceable; externally finned; with standard, removable, carbon-steel water boxes.
- 2) **Standard working pressures:**
  - Refrigerant Working Pressure: 300 psig (2070 kPa).
  - Water Side Working Pressure: 150 psig (1035 kPa).
- 3) The condenser can be specified with standard water boxes or marine water boxes. The standard boxes are less expensive than marine water boxes but whereas standard boxes require piping removal for maintenance, marine water boxes allow access to tubes with piping in place. A marine water box is installed at the factory and costs about \$3,000-4,000 more for a 1,000 ton chiller with a 2-pass condenser and requires about 18 inches more space lengthwise than a standard water box.
- 4) An alternate online automatic brush cleaning system is more expensive than either of the two (standard or marine water boxes), and requires smooth bore tubes not enhanced tubes with internal grooves or ridges to promote turbulence and augment heat transfer. Smooth bore tubes cost more than enhanced tubes. An automatic brush cleaning system costs about \$20,000-22,000 including smooth bore tubes for a 1,000 ton chiller.

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### **AIR COOLED CONDENSER**

An air cooled condenser has outside air passed over it and the refrigerant in the coil is cooled directly by the air. At full load conditions the design entering air dry bulb temperature is 95°F. Entering condenser refrigerant vapor temperature is 190°F. Leaving condenser refrigerant temperature is 125°F.

- 1) **Description:** Factory assembled and tested; consisting of casing, condenser coils, condenser fans motors, and unit controls.
- 2) **Condensing Coil:** Seamless copper-tube, aluminum-fin coil; factory tested at 425 psig (2930 kPa); Circuit to match compressors

- 3) **Condenser Fans and Drives:** Propeller fans for vertical or horizontal air discharge; directly driven with permanently lubricated ball-bearing motors with integral current- and thermal-overload protection.
- 4) **Low Ambient Control:** Factory-installed damper assembly, fan-speed control, or fan-cycling control.
- 5) **Operating and Safety Controls:** Include condenser fan motor thermal and overload cutouts; 115-V control transformer, if required; magnetic contactors for condenser fan motors and an un-fused factory-mounted and -wired disconnect switch for single external electrical power connection.
- 6) **Exterior Casing:** Manufacturer's standard equipment casing coated with corrosion-resistant exterior finish and with removable doors or panels for service and inspection.
- 7) **Motor Construction:** NEMA MG 1, general purpose, continuous duty, Design B.
- 8) **Verification of Performance:** Rate condensing units according to ARI 210/240 for units smaller than 135000 Btu/hr and according to ARI 340/360 for 135000 Btu/hr units and larger.
- 9) **Testing Requirements:** Factory test sound-power-level ratings according to ARI 270

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### **AIR COOLED V/S WATER COOLED CONDENSERS**

<b>Parameters</b>	<b>Air-cooled Condensers</b>	<b>Water-cooled Condensers</b>
Water loss	Nil	High
Operation Duty & Nominal Capacity	Multiple condensers required above 50 TR cooling capacity	Single condenser combination available for all sizes.
Space Requirements	Low (high above 200 TR)	Low
Energy Efficiency Ratio (EER)	Low	High
Initial Cost	Low	High
Operating Costs	Low	High
Maintenance	Low	High
Seasonal de-rating	High	Low

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### **COOLING TOWERS**

The cooling tower water removes heat of the condenser by principle of evaporative cooling. Water is exposed to outside air to reject heat from the water. This basin water is also treated for corrosion, fouling, and scaling.

- 1) The major component of a cooling tower is a fan to move the air across the fill material. Cooling towers work for cooling higher temperature water or fluids. A typical cooling tower would for example cool 95°F entering water temperature (EWT) down to 85°F leaving water temperature (LWT). Cooling towers are often used in conjunction with water-cooled chillers, but have many stand-alone uses as well.
- 2) The difference of EWT to LWT is termed “range” of cooling tower. This is usually 10°F in HVAC industry.

- 3) The difference in LWT and the ambient wet bulb temperature is termed “approach” of the cooling tower. Cooling towers should be engineered for a 3<sup>0</sup> F - 5<sup>0</sup> F wet bulb temperature approach. Lowering the approach further down implies the cooling tower shall be larger and is economically unviable.
- 4) Select and locate cooling towers to avoid problems with aesthetics, noise, vibrations, air recirculation or drift. Include a noise analysis of the proposed cooling tower relative to adjacent occupancies and consider alternative cooling tower selections, if necessary, to meet noise level of 60 dB(A) at 15 m (50 feet) which may be lowered for critical locations. Consider provisions for security and maintenance lights and receptacles. Provide a permanent service platform and ladders for access to cooling tower basin access doors.
- 5) Cooling towers shall be factory tested by licensed testing agency in accordance with CTI 201-96. The CTI certification is preferable over the field tests.

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### **EVAPORATIVE CONDENSER**

An evaporative condenser is placed directly over a basin, water from the basin is sprayed, and air is forced over the coils resulting in cooling of refrigerant in the coils. At full load conditions the design entering air dry bulb temperature is usually 95°F and entering air wet bulb temperature is 75°F.

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### **WATER TREATMENT**

- 1) Provide water treatment systems for cooling towers for prevention of corrosion, scale, and biological formations. In most cases, a water treatment is required for cooling water systems, closed chilled-water systems, hot water systems, and dual-temperature systems.
- 2) Cooling water is susceptible to scale and corrosion over a period of time. To prevent formation of scale, water is treated prior to using it for coolant purposes. The water treatment methods are classified in three broad categories:
  - Water Treatment (Softening, Dealkalization, Demineralization, Reverse Osmosis)
  - Chemical dosing
- 3) Principle Factors Responsible for Scale Formation
  - Calcium content of water
  - Alkalinity or pH of water
  - Temperature of re-circulation water
  - Higher concentration of solids (TDS)
  - Insufficient bleed off from cooling towers
- 4) The common chemical inhibitors compounds are:
  - Chromates: These are generally used with zinc and poly-phosphates. When chromates are used alone, the level should be maintained otherwise it would promote corrosion. Few countries/states do not permit the use of chromates because of its toxic nature.
  - Poly-Phosphates: These basically extend the solubility limit of Calcium. The main limitations of these are that in case pH increases beyond 7.3, these will be converted into ortho-compound, which will react with calcium and form the sludge.
  - Zinc: These provide film of zinc hydroxide. If pH = 6.5, maintenance is must. If pH > 6.5, zinc will go to the solution. If pH < 6.5, it will precipitate. It will tend to absorb the suspended particulates, which shall lead to higher turbidity.

- Nitrates: These are anodic inhibitors. High level of sulphates and chlorides will affect its performance that leads to high turbidity.
  - Silicates: This cannot be used if pH is high (> 7) and hardness is over 250 ppm.
  - Sodium Meta Phosphate (SHMP): This is effective in a sense that it delays the solubility limit of calcium salts and therefore the scale formation is delayed. The disadvantage however is that it degenerates into ortho-compound and require pH to be controlled with in 6.5 to 7.4. It also acts as food for bio fouling. The latest developments are employing Amino Trimethylene Phosphonate Acid (ATMP) and Hydroxyl Ethylkylene di-Phosphonate Acid (HEDP) that does not have the limitations of SHMP.
- 5) In practice an estimation of saturation index provides a satisfactory solution to estimate whether water is scaling or corrosive tendencies.
  - 6) The provision of di-anodic protection by using polyphosphates and chromium oxide to the extent of 40 ppm and 20 ppm respectively ensures protection to the extent of 1-3 mil per year. Addition of 5/10 ppm of polycrymides benefits the system by keeping  $\text{CaCO}_3$  etc in suspension.
  - 7) To counter the presence of organic matter and algae presence of 6/7 ppm of free chlorine is desired.
  - 8) Several new generic water conditioning and protective chemicals are available today, for less detailed water analysis. This is however a risky and not a recommended option for long run.
  - 9) The plants with high capital investments and high intensity of energy resort to safe process water demineralization and find it economical and worthwhile to ensure highly efficient operations.

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### **AIR HANDLING UNIT**

Air handling unit (AHU) is designed to condition air in rooms or in designated areas in a building called zones. In an AHU, the air flow or the fluid flow in coils can be adjusted for interior environmental control. To compensate for the duct air leakage and any future space internal heat gain, the equipment must be sized in accordance with the following guidelines:

- 1) **Load Calculations:** Heat gain calculations must be done in accordance with the procedure outlined in the latest ASHRAE Handbook of Fundamentals. The calculated supply air shall be the sum of all individual peak room air quantities without any diversity, even for the variable air volume systems.
- 2) **Safety Margin:** A safety factor of 5 percent shall be applied to the calculated room air quantity to allow for any future increase in the room internal load. The adjusted supply air shall be, thus, 5 percent in excess of the calculated supply air.
- 3) **Air leakage:** The air leakage through the supply air distribution ductwork shall be computed on the basis of the method described in the SMACNA Air Duct Leakage Test Manual. The maximum leakage amount shall not exceed 4 % of the adjusted supply air.

### **AHU Components**

The air handling system consists of various components such as fans, ducts, & dampers. Each set of components performs a task critical to the proper operation of the system as well as occupant comfort.

- 1) **Fan selection** – Fans circulates the air and provides the pressure required to push it through ducts, filters and over heating or cooling coils. The fan and motor selection shall be based on the supply air fan capacity and static pressure adjusted, as necessary, for the altitude, temperature, fan inlet and discharge conditions, and the AMCA 201 System Effect Factors. In most cases, vane-axial and backward-curved centrifugal fans are the most efficient AHU fan choice. Consider the rated acoustical properties, space limitations, inlet and outlet conditions, and air quantities/ pressure requirements of the fan before identifying the best fan for the application.
- 2) **Coil and filter selection** – Select AHU coils for low airside and waterside flow resistance, low water flow rates, and operation at warmer chilled water or cooler hot water temperatures. Specify coil control strategies that will minimize water flow and maximize heat transfer. Pay special attention to



the pressure drop of coils and filters. Limit face velocity to 450 - 500 feet per minute (fpm) for VAV systems and 400 - 500 fpm for constant air volume systems.

- 3) **Filter selection** – Filters separate out dust and particulates that could cause discomfort to your building's occupants or possibly damage sensitive equipment. Select prefilters and efficient filters at 250 to 400fpm face velocity. The filter frames are normally fabricated in angular construction (V-shape) to provide increased face area. Over a period of operation, there is decrease in system airflow as the pressure drop across the filter increases. Always size fans for the “dirty” filter condition, which will ensure delivering adequate airflow capacity as the filter becomes loaded.
- 4) **Hot and chilled water piping systems** – Increasing the system pipe diameters and specifying low-friction valves reduces flow resistance through the piping and coils and decreases the system pumping energy.
- 5) **Air distribution systems** – Ducts convey the conditioned air throughout your building, distributing the air to occupants and then returning it to the HVAC system to be conditioned and circulated again. Select air distribution components that offer the lowest pressure drop through the system. Large duct sizes provide low pressure drop and future flexibility if increased airflow is required. Try to minimize fittings such as elbows and transitions, since they have large pressure drops. Dampers control the flow of air through the ducts to the various parts of the building.
- 6) **Mixing Boxes:** Air-handling units shall be provided with mixing boxes where relief air is discharged from the air handling unit. Mixing boxes may also be used on the return side of the unit in lieu of a plenum box. Air flow control dampers shall be mounted within the mixing box or on the ductwork connecting to the mixing box.
- 7) **Control System** - A control system monitors and controls the various parts of this complex equipment, providing heating, cooling, and fresh air to where it is needed while ensuring efficient operation.
- 8) **Compliances:** Central-station air-handling units and components will be designed, fabricated, and installed in compliance with
  - NFPA 90A - Installation of Air Conditioning and Ventilating Systems
  - Electric coils, along with complete central-station air-handling unit, shall be listed and labeled by UL.
  - Central-station air-handling units and their components shall be factory tested according to the applicable portions of ARI 430, "Central-Station Air-Handling Units," and shall be listed and bear the label of the Air-Conditioning and Refrigeration Institute (ARI).
  - Provide motors required as part of air-handling units that are listed and labeled by UL and comply with applicable NEMA 70 standards.
  - Comply with AMCA 301, "Methods for Calculating Fan Sound Ratings from Laboratory Test Data."
  - Establish fan performance ratings for flow rate, pressure, power, air density, rotation, speed and efficiency according to AMCA 210, "Laboratory Methods of Testing Fans for Rating."
- 9) As far as possible, VAV systems should be used with all air systems, except for spaces requiring constant air changes/hours, and/or critical pressure differentials with respect to the adjoining spaces.
- 10) Ideally, air-handling units shall be sized to not exceed 25,000 CFM. Smaller units are encouraged to facilitate flexible zone control, particularly for spaces that involve off-hour or high-load operating conditions. To the extent possible, “plug-n-play” AHU configurations should be considered, facilitating easy future adaptations to space-load changes.
- 11) Psychrometric analysis (complete with chart diagrams) shall be prepared for each air-handling unit application, characterizing full and part load operating conditions.
- 12) Air-handling unit/coil designs shall assure that conditioned space temperatures and humidity levels are within an acceptable range, per programmed requirements, and ASHRAE Standards 55 and 62.



- 13) All air-handling units comprised of coils, fans, filters, etc. shall be of double wall construction. These units shall be factory-fabricated and field-assembled. The larger units could, however, be field-assembled.
- 14) The AHU can be configured as draw through or blow through arrangement. Draw thru arrangement is preferred.
- 15) **Materials:** Formed and reinforced galvanized steel panels, fabricated to allow removal for access to internal parts and components, with joints between sections sealed.
- 16) **Insulation:** Coated, glass-fiber insulation, complying with NFPA 90A, "Installation of Air Conditioning and Ventilating Systems," for insulation.
- 17) Specify ball or roller bearings with inner and outer shaft seals in accordance with ABMA 9 or ABMA 11, L-50 of 200,000 hours.
- 18) Fan motor service factors - 1.15 for poly-phase motors and 1.35 for single phase motor with sufficient torque to accelerate driven loads satisfactorily.
- 19) Motors are typically NEMA MG-1, general purpose, continuous duty, Design B.
- 20) Belt Guards are usually fabricated to OSHA/SMACNA requirements, 0.1046 inch (2.7 mm) thick, 3/4-inch (20-mm) diamond-mesh wire screen welded to steel angle frame or equivalent, prime coated.
- 21) Install fans on open-spring vibration isolators, minimum 1-inch (25-mm) static deflection.
- 22) Always put one "Magnahelec" gauge (0- 1" H<sub>2</sub>O range) on visible side of the air handling unit filter section with 1/4" copper tubing penetrating the filter section on each side of the filters. Air handling unit filter sections shall have doors on both sides of filter section.
- 23) The air handling unit is one of the largest pieces of equipment in air-conditioning system. The dimensions (length x width x height) of AHU are influenced by the airflow capacity of the unit. As a rule of thumb, the face area (width x height) can be determined by dividing the air flow capacity of unit by 500 and length of the unit is estimated by number of modular sections comprising the AHU. Typically include the fan section, coil section, access section, filter section, mixing box and drift eliminator section as a minimum.
- 24) When conceptualizing the AHU footprint in the mechanical room area, keep a space equivalent to the width of the unit on one side for removal and cleaning of filters. Design and construct to facilitate removal and replacement of coil for maintenance and to assure full airflow through coils.
- 25) Where the Air handling units are installed in concealed locations such as those suspended to the ceilings, provide auxiliary drain pans piped through an exterior wall. To the extent practical, access to mechanical rooms housing AHU shall be provided from public corridors so that the on-going functions of other rooms are not disturbed.
- 26) Note the following:
  - 12,000 BTUH = 1 Ton of Cooling
  - 1 MBH = 1,000 BTUH
  - Sensible Cooling/Heating (BTUH) = CFM x 1.08 x Temperature Difference
  - Total Cooling (BTUH) = CFM x 4.5 x Enthalpy Difference
  - Total Cooling = Sensible Cooling + Latent Cooling
  - Sensible Heat is measured in dry bulb temperatures.
  - Latent Heat is measured in wet bulb temperatures and related to change in humidity ratio.

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### **COMBINATION HEATING COOLING SYSTEMS**

Combine heating-cooling systems to avoid duplication of system elements and to reduce costs.

- 1) **All-Air Systems:** In all-air system the piping is distributed to the central air-handler outside the conditioned space. The system relies extensively on ductwork to distribute the cold air while piping is limited. These systems find application on large spaces requiring uniform distribution of air, positive pressurization, near optimum load operation etc. Where outdoor design temperatures are 20°F or below, consider all air systems only in conjunction with double glazing, where sedentary activities are a minimum of 3 feet from the glass, and where proper peripheral air distribution is provided. Uses preheat coils whenever the mixture of return air and ventilation air at outside design temperature is below 35°F.
- 2) **All-Water Systems:** In all-water system only piping is distributed to the multiple terminal units located inside the occupied spaces. The ductwork is eliminated. The system finds application where ceiling heights are low and where retrofitting is desired as a result of space expansion. System yield energy efficiency by shutting off the terminal units if the space is not in use or space conditions are satisfied. Use two-pipe dual-temperature systems for comfort applications where feasible. Four-pipe systems may be used where two-pipe systems are not capable of providing the specified indoor design conditions. Generally, three-pipe systems cannot be justified for comfort applications and should not be used.
- 3) **Air-Water Systems:** In air-water system, piping is distributed to the air-handler unit and as well the terminal units. The system relies extensively on the piping network while the ductwork sizes and runs are minimized. The system finds application where ceiling heights are low and where cross zoning is to be avoided. System yield energy efficiency by shutting off the terminal units if the space is not in use or space conditions are satisfied. Consider combinations of air and water systems such as radiant heating supplemented with single-zone interior air supply for ventilation; hydronic systems at the periphery of a building to offset skin transmission losses only, combined with the use of an air system for space cooling and ventilation loads.

The relationship between the three systems with respect to their ability to control the six main HVAC parameters, which are temperature, relative humidity, pressure, air composition, particulates and air velocity.

Parameters	All-Air Systems	Air-Water Systems	All-Water Systems
Temperature	Good using modulation of chilled water through coils in air handling units	Good, it uses central air-handling unit and the terminal unit coils for temperature control. The control is achieved through modulation of chilled water through air-handling coil and usually using "on-off" solenoid valve at the terminal units.	Moderate, using modulation or 'on-off' of valve to the terminal units. Air control is also possible using 2 or 3 speed fan motor.
Relative Humidity	Moderate using coils for dehumidification and humidifiers in air handling units. Since the system relies on the chilled water for dehumidification, the system also simultaneously uses reheat for temperature control, which is waste of	Good, it relies primarily on the dehumidification through the chilled water coil of air-handling unit that processes 100% outside air. The secondary control is at the terminal units, but in practice the primary air control is	Poor using chilled water at low temperature at the terminal units. Necessary plumbing arrangements need to be considered for taking condensate drainage to the discharge receptacles.

Parameters	All-Air Systems	Air-Water Systems	All-Water Systems
	which is waste of energy.	primary air control is good enough as high humidity is primary a result of outside air.	
Pressurization	Moderate by permitting requisite amount of outside air	Good by use of 100% outside air handling unit which can easily be controlled	Poor as terminal units use 100% re-circulation
Air composition	Good by modulating the outside air and return air dampers	Good by allowing the right amount of ventilation at the first place.	Poor as the system uses 100% re-circulation units and rely on window/wall openings or natural infiltration for ventilation
Particulates	Good by allowing pre-filtration and high efficiency filtration in the air handling units	Good by allowing pre-filtration and high efficiency filtration in the air handling units	Moderate by allowing filtration at the terminal units. The filtration efficiency is however low due to static pressure concerns. Maintenance in occupied spaces is another concern.
Air velocity	Good by varying the fan speed and dampers, swirl diffusers with VAV boxes	Moderate by constant supply of fresh air. The throw of terminal units is a concern to cover wide areas.	Poor control as the terminal units have a limited through and are 100% re-circulation units

### HEATING & COOLING COILS

- 1) **Coil Construction:** Individual finned tube coils should generally be between six and eight rows with at least 12 fins per inch to ensure that the coils can be effectively and efficiently cleaned. Dehumidifying coils shall be selected for no more than negligible water droplet carryover beyond the drain pan at design conditions. All hot water heating and chilled water cooling coils shall be copper tube and copper finned materials.
- 2) Equipment and other obstructions in the air stream shall be located sufficiently downstream of the coil so that it will not come in contact with the water droplet carryover. Cooling coils shall be selected at or below 2.5 m/s face velocity (500 fpm) to minimize moisture carryover. Heating coils shall be selected at or below 3.8 m/s face velocity (750 fpm).
- 3) **Coil-Performance Tests:** according to ARI 410, "Forced-Circulation Air-Cooling and Air-Heating Coils."
- 4) **Face area of cooling coil in Sq-ft** = Air handler capacity in CFM / 500 FPM

- 5) The larger face area of the coil reduces the air velocity and pressure drop across them.
- 6) For higher latent load (moisture removal requirements), use deeper cooling coil (> 4 row deep). Deeper the coil, higher shall be the pressure drop.
- 7) Higher the fins per linear inch (FPI), lesser shall be the by-pass factor and higher shall be the surface contact. The air pressure drop though will be high. The recommended FPI should not exceeding 12.
- 8) The approximate pressure drop across cooling coil designed for 500 FPM:
  - 4 Row, 8 FPI coil has approx. 0.50 in-WG
  - 6 Row, 8 FPI coil has approx. 0.75 in-WG
  - 8 Row, 8 FPI coil has approx. 1.0 in-WG
- 9) The approximate pressure drop across heating coil designed for 700 FPM:
  - 1 Row, 8 FPI coil has approx. 0.15 in-WG
  - 2 Row, 8 FPI coil has approx. 0.30 in-WG
- 12) Determine the optimum supply and return water temperature differential by life cycle cost analysis. Dehumidification is normally accomplished in the cooling coil. This may be a factor in determining required cooling water temperature. Also, designing for warm chilled water temperatures increases the number of hours of potential “free” cooling using a waterside economizer.
- 13) Depending on sensible heat ratio characteristics, effective moisture control may require cooling coil air discharge dew point temperatures as low as 50°F. As required, provide face-by-pass or heat recovery features to re-heat cooling coil discharge temperatures for acceptable space entry. Provide a direct form of re-heat and/or humidification only if space conditions require tight environmental control, or if recurring day-long periods of unacceptable humidity levels would otherwise result. The use of re-heat is highly energy inefficient and is prohibited by various states/codes.
- 14) For direct expansion coils, refrigerant distributor must be designed for low pressure drop, for down feed with solder connections, and with maximum of 12 circuits for each distributor. Make sure that the length of each circuit is same to allow for same pressure drop.
- 15) Drip pans are placed under cooling coils to collect condensate resulting from cooling building room air and dropping its dew point temperature. Drain lines from air handlers shall be plastic (PVC) pipe or galvanized iron. Clean-outs shall be provided at proper locations and need be sloped correctly to assure good drainage.
- 16) The external drain of each air handling unit coil section shall have a deep seal trap and be extended to an open sight drain. Intermediate drain trough shall be provided for each coil bank more than one coil high.
- 17) **Piping:** Piping to coils shall be offset and shall have shut-off valves and flanges or unions to permit removal of the coil from the side of the air handling unit.
- 18) All constant air volume AHU chilled water coils shall be provided with three way, two position chilled water valves in order to maintain a wet coil so as to dehumidify at all times. Variable air volume AHU chilled water coils shall be provided with two-way, modulating chilled water valves in addition to a three way, two position chilled water valve installed at the end of the loop to prevent pump dead heading.

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### **AIR HANDLING UNIT FANS**

There are two main types of fans, centrifugal and axial.

- ✓ Centrifugal fans discharge air perpendicular to the axis of the impeller rotation. As a general rule, centrifugal fans are preferred for higher pressure ducted systems.

- ✓ An axial fan discharges air parallel to the axis of the impeller rotation. As a general rule, axial fans are preferred for high volume, low pressure, and non-ducted systems.

1) **Fan Types**

Centrifugal	Axial	Special Designs
Forward Curved (FC)	Propeller	Roof ventilators
Radial Fans (RF)	Tubeaxial	Inline Centrifugal
Backward Inclined (BI)	Vaneaxial	Plug / Plenum fans
Backward Curved (BC)		Blowers
Aerofoil Bladed (AF)		

2) **Fan Laws:**

- Law #1: CFM varies directly with RPM
- Law #2: Static pressure varies with the square of the RPM
- Law #3: Horsepower varies with the cube of the RPM

3) **Fan Selection Criteria**

Before selecting a fan, the following information is needed:

- Air volume required – CFM
- System resistance – SP
- Air density (Altitude and Temperature)
- Type of service
- Environment type
- Materials/vapors to be exhausted
- Operation temperature
- Space limitations
- Fan type
- Drive type (Direct or Belt)
- Noise criteria
- Number of fans
- Discharge
- Rotation
- Motor position
- Expected fan life in years

4) **Centrifugal Fan Pressure Classes:**

- Class 1: Fans up to 4.5" pressure
- Class 2: Fans from 4.5" to 7"-8" pressure, heavier gauges & larger shafts than Class 1

- Class 3: Fans above 7"-8" pressure, heavier gauges & larger shafts than Class 2

#### 5) Fan Performance Tables and Curves

Performance tables provide a simple method of fan selection. The performance curve also is a valuable tool when evaluating fan performance in the field. Fan performance tables and curves are based on standard air density of 0.075 lb/ft<sup>3</sup>. When altitude and temperature differ significantly from standard conditions (sea level and 70° F) performance modification factors must be taken into account to ensure proper performance.

#### 6) Fan Testing

Fans are tested in accordance with strict requirements of AMCA (Air Moving and Conditioning Association) Standard 210. This standard specifies in detail the procedures and setups to be used in testing the various types of fans. Fans are tested under these standard test conditions so that all fans are rated on an equal basis. Thus, fans of different manufacturers and of different types can be rated and compared using the same basis of testing and obtaining performance data.

When fan performance is measured in field conditions, the difference between the ideal laboratory condition and the actual field installation must be considered. Consideration must also be given to fan inlet and discharge connections as they will dramatically affect fan performance in the field. If possible, readings must be taken in straight runs of ductwork in order to ensure validity. If this cannot be accomplished, motor amperage and fan RPM should be used along with performance curves to estimate fan performance.

#### 7) General

- All fans shall bear the AMCA seal and performance shall be based on tests made in accordance with AMCA Standard 210. Fans should be selected on the basis of required horsepower as well as sound power level ratings at full load and at part load conditions.
- Fan motors shall be sized so they do not run at overload anywhere on their operating curve. Fan operating characteristics must be checked for the entire range of flow conditions, particularly for forward curved fans. Fan drives shall be selected for a 1.5 service factor and fan shafts should be selected to operate below the first critical speed.
- From power consumption standpoint, "Variable Speed Drives" (VSD) are the most efficient followed by blade pitch control method. Inlet vanes save some power, while discharge dampers throttling at the fan save little. From a first-cost standpoint, dampers are the least costly. Inlet vanes and blade pitch control follow, with variable speed motors being the most expensive. Centrifugal double width double-inlet forward curved and airfoil fans are preferable for VAV systems.
- Fan power is proportional to flow cubed. Cutting fan speed by about 50%, which reduces fan energy use to 1/8th.
- A fan operating at a higher elevation or temperature will move the same volume of air as it will at standard conditions; however it will generate less total pressure and will require less horsepower. Note that the volume of air is not affected by variations in air density. In other words, if a fan will move 3,000CFM at 70°F it will also move 3,000CFM at 250°F. Since 250°F air weighs only 34% of 70°F air, the fan will require less BHP but also create less pressure than specified. Therefore, when selecting a fan to operate at a non-standard density using standard air density tables and curves, corrections must be made to the parameters affected by air density. These parameters are static pressure and brake horsepower.
- Fan installation should minimize system effect. For example use a straight duct length on the discharge of both inline and centrifugal fans. In order to achieve a uniform velocity profile, a 100% effective duct length must be used. To calculate the 100% effective length, use 2.5 duct diameters for 2500 FPM (or less). Add one duct diameter for each additional 1000 FPM.

## **AIR DISTRIBUTION SCHEMES**

Air should travel from the furnace/air conditioner throughout the system at an average velocity of about 700 FPM. With an air velocity of about 700FPM, air conditioner are designed to move 400CFM per ton and furnaces 12CFM/1000Btu

Vertical Throw - the maximum distances below the ceiling of the air pattern at terminal velocities of 100, 75, and 50 FPM.

For comfort cooling or heating, typically 1 or 1.5 CFM of supply air is required per sq-ft of the floor area.

You might have one or more of the following systems:

- 1) Constant Air Volume Systems
- 2) Reheat System
- 3) Dual Duct System
- 4) Multi-Zone System
- 5) Variable Air Volume Systems

### **Constant Volume (CV) Systems**

- 1) A classification of HVAC equipment for which the air flow rate is constant. The main system air supply fan operates only at a single speed, thus the delivered air flow rate is constant. This system operation is in contrast to the Variable Air Volume (VAV) system operation, which allows variation in the supply air flow.
- 2) When supplied with temperature reset, this type of system becomes a constant volume/variable temperature system (CVVT). The efficiency is increased somewhat but humidity problems can result.
- 3) When the size of the building permits, serve zones with similar solar exposure and loads, and in reasonable proximity from the same air handling unit. For example, serve south/west perimeter offices from one unit, north/east perimeter offices from a second unit, and central spaces from a third unit.
- 4) Avoid inefficient terminal or zone reheat for constant volume systems.

### **Reheat Systems**

In a reheat system, a constant amount of air is cooled and dehumidified. Reheating Coils are part of some air-conditioning systems used primarily to raise the temperature of circulated air after it is over cooled to remove moisture. Reheat systems are a very inefficient and some codes do not permit the use of reheat.

### **Dual Duct (DD) Systems**

Often found in buildings constructed during the 1960's and 1970's, dual duct systems are a relatively effective, yet extremely inefficient method of conditioning air.

DD systems consist of two independent systems, one warm, one cool, which circulate air in parallel sets of ducts through all sections of the building. Hot and cold air are mixed in local mixing boxes in each zone or room and then fed into that area. Depending on the temperature needs of the area, the mixture of hot and cold air is adjusted until the desired temperature is reached.

Unfortunately, with a dual duct system, you must pay to cool, heat and circulate this volume of air, which is typically much larger than the actual volume required by your building.

### **Multi-Zone Systems**

Multi-Zone systems are similar to dual duct systems in that two streams of hot and cold air are mixed to produce a desired temperature. In contrast to dual duct systems, multi-zone systems mix the air in large mixing boxes near the fans instead of individual boxes located at each area or room. This conditioned air is then fed to each zone, with each zone, depending on its load, receiving air at a different temperature. Like dual duct systems, multi-zone systems are quite inefficient as well



### Variable Air Volume (VAV) Systems

- 1) Variable air volume systems usually referred to as "VAV" supply varying quantities of conditioned (heated or cooled) air to different parts of a building according to the heating and cooling needs of those specific areas. VAV systems can substantially reduce fan and cooling energy.
- 2) For blowers over 1 hp, use fans with adjustable-speed drive (ASD) or variable pitch fan blades for the highest efficiency. For blowers under 1 hp, select electronically commutated motors (ECM).
- 3) Multi-speed motors are less efficient and less effective than adjustable-speed drives or variable pitch fan blades.
- 4) VAV w/o Reheat Characteristics:
  - Controls space temperature by varying the quantity of supply air.
  - Supply temperature is constant.
  - Energy savings is due to reduced supply air quantities and therefore reduced horsepower.
- 8) VAV w/Reheat integrates heating at the VAV terminal unit to offset heating load, limit maximum humidity, provide reasonable air movement, and provide ventilation air.
- 9) VAV operates by taking temperature signal from the thermostats. The modulation of VAV damper varies the pressure in the ductwork which is sensed by the duct static pressure sensors. This in turn provides control signal to the adjustable speed drive of the air handling unit fan. The duct static pressure sensors should be placed at  $2/3^{\text{rd}}$  of the way to the furthest VAV box.
- 10) Use VAV boxes with linear-response dampers, direct digital (DDC) control and adjustable minimum openings.
- 11) Minimum CFM for VAV Boxes:
  - 20% of design flow: Perimeter Spaces
  - 0% of design flow: Interior Spaces.

When interior spaces are occupied or lights are on, the VAV terminal unit will maintain a minimum flow to offset the heat gain. Therefore, the only time a VAV terminal unit serving an interior space will be closed is when the space is unoccupied and lights are off.

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### **TERMINAL UNITS TYPES & CHARACTERISTICS**

Terminal units are typically located at the end of a conditioned air supply duct and may or may not include a fan for control of interior building air.

- 1) **Terminal Unit Types:** There are two types; Pressure independent terminal units and pressure dependent terminal units
  - Pressure Independent Terminal Units: Terminal unit airflow is independent of pressure upstream of box. Recommend using pressure independent terminal units.
  - Pressure Dependant Terminal Units: Terminal unit airflow is dependant on pressure upstream of box.
- 2) **Parallel Fan Powered Terminal Units:**
  - Primary air is modulated in response to cooling demand and fan is energized at a predetermined reduced primary airflow.
  - Fan is located outside the primary airstream to allow intermittent fan operation.
- 3) **Series Fan Powered Terminal Units:**
  - A constant volume fan mixes primary air with air from the ceiling plenum.

- Fan is located within the primary airstream and runs continuously.
- 4) Induction Terminal Units:
    - Reduces cooling capacity by reducing primary air and inducing room or ceiling plenum air.
    - Incorporates reduced supply air quantity energy savings of VAV system and air volume to space is constant to reduce effect of stagnant air.
  - 5) Constant Volume Reheat (CVR) Terminal Units:
    - CVR terminal units provide zone/space control for areas of unequal loading, simultaneous cooling/heating, and close tolerance of temperature control.
    - Conditioned air is delivered to each terminal unit at a fixed temperature then reheated to control space temperature.
    - Energy inefficient system.
  - 6) Constant Volume Bypass Terminal Units:
    - Variation of CVR system. Constant volume primary air system with VAV secondary system.
    - Supply air to space varied by dumping air to return air plenum.
  - 7) Dual Duct Terminal Units:
    - Constant volume of supply air is delivered to the space.
    - Space temperature is maintained by mixing varying amounts of hot and cold air.
    - Energy inefficient system
  - 8) VAV Dual Duct Terminal Units:
    - Variable volume of supply air is delivered to space.
    - Space temperature is maintained by supplying either hot or cold air in varying amounts and limiting the amount of hot and cold air mixing.
    - More energy efficient the standard dual duct systems.
  - 9) Single Zone Systems:
    - Supply unit serves single temperature zone and varies supply air temperature to control space temperature.

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### **CENTRIFUGAL PUMPS**

- 1) The 'Affinity Laws' equations relating rotodynamic pump performance parameters of flow, head and power absorbed are:
  - Capacity varies directly as the speed or impeller diameter ( $\text{GPM} \propto \text{rpm} \propto D$ )
  - Head varies as the square of speed or impeller diameter ( $\text{GPM} \propto \text{rpm}^2 \propto D^2$ )
  - BHP varies as the cube of the speed or impeller diameter ( $\text{BHP} \propto \text{rpm}^3 \propto D^3$ )

If you double the speed of a pump you will get twice the capacity, four times the head and it will take eight times the horsepower to do it.
- 2) The effect of varying speed with a centrifugal pump is to vary both head and flow. Variation of speed with a positive displacement pump will vary only the flow rate.
- 3) Check the equation: Cooling load (BTUH) = Flow (GPM) x 500 x (Ti – To), where Ti & To is the inlet and outlet temperature across heat exchanger (chiller or condenser). As the temperature range

across the heat exchanger (chiller or condenser) is increased, the flow rate is decreased for the same capacity. Smaller flow means smaller pipes & pumps; this equates to capital savings.

- 4) Check the equation:  $\text{Pump BHP} = \text{Flow (GPM)} \times \text{Head (ft)} \times \text{Specific Gravity} / (3960 \times \text{Pump Efficiency})$ . In making a choice of pumping systems, engineers put emphasis on the efficiency. While this is OK to have the maximum possible efficiency, one must carefully look at other part of the formula; viz. the pump flow and the head. These two parameters are often neglected. The true energy savings can be found from the correct sizing of parameters (TDH in ft x flow in GPM). For example, a pump efficiency of 70% will correlate to a system efficiency of 35%, if the pump is selected and operated at twice required pump head. If the flow requirements were similarly specified, system-pumping efficiency would decrease to the order of 17.5%.
- 5) As a generality, the larger the pump, the higher is the efficiency. While it is true that the large pumps offer higher efficiency, don't be misguided by this generic statement. It will almost always be true that a smaller pump matched to the system will operate at lowest cost-even though its efficiency as a pump is lower.
- 6) The pumps best efficiency point (B.E.P.) is between 80% and 85% of the shut off head. At this point there is little to no radial thrust on the impeller. Also the "power in" is closest to the "power out".
- 7) Power requirements for the over-headed pump will be to the order of 2.5 times that otherwise needed; if the actual needed pump power draw were 10 HP, the 100% over headed pump would draw 25 BHP. Wastage of 15 HP or \$ 1750 per year would occur at \$ 0.015 per KWH.
- 8) Chilled water pumping is most energy efficient for evaporator selection at 15°F to 17°F temperature differential. Condenser water pumping should be at a 10°F to 12°F temperature differential.
- 9) End suction pumps are preferred for chilled water and cooling water supply.
- 10) Pumps shall be of a centrifugal type and shall generally be selected to operate at 1750 RPM. Both partial load and full load must fall on the pump curve. The number of primary chilled water and condenser water pumps shall correspond to the number of chillers and a separate pump shall be designed for each condenser water circuit.
- 11) For smaller pumps up to 20kW use monoblock pumps and with capacity ratings > 20kW use horizontal split casing pumps.
- 12) Higher flow rates are better served by double suction pump in which axial forces tend to balance one another.
- 13) Pumps piped in series must have the same capacity (impeller width and speed). Series pumping is most effective when the system head pressure curve is steep. When the head pressure is not a constraint, parallel pumping is preferred.
- 14) Pumps piped in parallel must have the same head (impeller diameter and speed). Parallel pumping provides high degree of redundancy and standby capacity. When one pump is out of operation, the other pump continues to pump water through the system.
- 15) Primary/secondary (P/S) piping (pumping) is an efficient arrangement for chilled water distribution for large facilities and has established itself as the default piping technique for multi-load hydronic systems. Its ability to "keep the peace" among several simultaneously operating circulators is what makes it the preferred choice over other types of distribution systems.
- 16) The key to all primary-secondary applications is the use of a common pipe, which interconnects the primary and secondary circuits. The length of this pipe should be kept very short in order to keep the pressure drop very low and the supply and return tees to the secondary circuit should be a maximum of four pipe diameters apart. By keeping the pressure drop very low, water that is flowing in the primary loop will not flow into the secondary circuit until its circulator turns on.
- 17) Variable volume pumping systems should be considered for all secondary piping systems with pump horsepower greater than 10 kW (15 HP). The specified pump motors shall not overload throughout

the entire range of the pump curve. Each pump system shall have a standby capability for chilled, hot water, and condenser water pumps.

- 18) Variable-pumping systems for chilled and hot water supply to AHU must use 2-way valve to maximize the energy saving benefits.
- 19) Pumps specifically the chilled water pumps shall be specified with mechanical seals. Do not use a mechanical seal when packing is more than adequate for the intended service.
- 20) All pumps shall be serviceable without removing the volute from piping connections.
- 21) The mass of the pump concrete foundation must be 5 times the mass of the pump, base plate, and other equipment that is being supported, or vibration will occur.
- 22) Up to 500 horsepower (375 KW), the foundation must be 3 inches (76 mm.) wider than the base plate all around. Above 500 horsepower (375 KW) the foundation should be a minimum of 6 inches (150 mm.) wider.

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## **FILTRATION**

- 1) Air filtration shall be provided in every air handling system. Air-handling units shall have a disposable pre-filter and a final filter. The filter media shall be rated in accordance with *ASHRAE Standard 52*. Pre-filters shall be 30 percent to 35 percent efficient. Final filters shall be filters with 85 percent efficiency capable of filtering down to 3.0 microns per *ASHRAE 52*. Filter racks shall be designed to minimize the bypass of air around the filter media with a maximum bypass leakage of 0.5 percent.
- 2) Filters shall be sized at 2.5 m/s (500 FPM) maximum face velocity.
- 3) Filter media shall be fabricated so that fibrous shedding does not exceed levels prescribed by *ASHRAE 52*. The filter housing and all air-handling components downstream shall not be internally lined with fibrous insulation. Double-wall construction or externally insulated sheet metal housing is acceptable.
- 4) The filter change-out pressure drop, not the initial clean filter rating, must be used in determining fan pressure requirements. Differential pressure gauges and sensors shall be placed across each filter bank to allow quick and accurate assessment of filter dust loading as reflected by air-pressure loss through the filter and sensors shall be connected to building automation system.
- 5) Determine number of filters required from a manufacturer's data (usually 2 CFM for each sq- in of filter area maximum)
- 6) **Pre-filters** shall be permanent metal frames with 2" replaceable, disposable pads. Final filters may be the bag type on large built-up systems.
- 7) **Permanent Filters:** Cleanable, 2 inches thick, with aluminum-or stainless-steel frames, clean airflow resistance of 0.10 inch-WG at face velocity of 500 fpm and ASHRAE 52.1 filter-arrestance efficiency of 60 to 80 percent.
- 8) **Disposable Filters:** 2-inch thick, viscous-coated fibers encased in fiberboard cell with perforated-metal media support, clean airflow resistance of 0.10"-WG at face velocity of 300 fpm and ASHRAE 52.1 filter-arrestance efficiency of 70 to 82 percent.
- 9) Pressure drop is proportional to velocity squared. Doubling the surface area of a filter (example by increasing duct diameter, or pleating the filter) allows a 50% reduction in velocity while maintaining equivalent flow and thereby reduces pressure drop to 1/4th.
- 10) Typical Pressure Drops
  - 30% ASHRAE, 2" Pleated Filters - 3" P.D. Clean and 0.6" P.D. Dirty
  - 60% ASHRAE, Rigid Final Filters - 0.5" P.D. Clean and 1.2" P.D. Dirty
  - 95% ASHRAE, Rigid Final Filters - 0.65" P.D. Clean and 1.4" P.D. Dirty

## 11) Filter Comparison

Filter Type	ASHRAE Arrestance Efficiency	ASHRAE Atmospheric Dust Spot Efficiency	Initial Pressure Drop (IN.WG)	Final Pressure Drop (IN.WG)
Permanent	60-80%	8-12%	0.07	.5
Fiberglass Pad	70-85%	15-20%	0.17	.5
Polyester Pad	82-90%	15-20%	0.20	.5
2" Throw Away	70-85%	15-20%	0.17	.5
2" Pleated Media	88-92%	25-30%	0.25	.5-.8
60% Cartridge	97%	60-65%	0.3	1.0
80% Cartridge	98%	80-85%	0.4	1.0
90% Cartridge	99%	90-95%	0.5	1.0
HEPA	100%	99.97%	1.0	2.0

- 12) Filters are normally installed in air handling units, ahead of the coils, and in a position to filter both recirculation and outside air. Install at 45 - 60° to horizontal, never horizontal
- 13) For clean room applications, high efficiency particulate air (HEPA) filters should be used. HEPA filters are replaceable extended-media dry-type having a minimum particle collective efficiency of 99.97 to 99.997% for a 0.3 micron particle, and a maximum clean filter pressure drop of 2.54 cm (1") water gauge when tested at rated air flow capacity. 0.3 micron is 1/75,000 of an inch or 1/300, the diameter of the human hair. These are tested by pressurized housing to a minimum of 3.0 inch-w.g or to design operating pressure, whichever is higher. Test for air leaks in according to ASME N510 pressure-decay method.
- 14) Activated-Carbon filters are recommended in the areas subjected to volatile organic compounds (VOC's). Capacity: 8.8 lb (4.0 kg) of activated carbon per 2000 CFM of airflow.
- 15) Compliance:
  - Comply with NFPA 90A and NFPA 90B.
  - Comply with provisions of ASHRAE 52.1 for method of testing and rating air-filter units.
  - Comply with ARI 850.
  - NFPA 70 compliance for the electronic air cleaners and electrical devices and accessories

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**HUMIDIFIERS**

- 1) Humidification makeup water requirements:
    - 1 lb. water = 7000 grains
    - Steam or Electric Humidifiers: 5.6 GPM per 1000 KW Input or 5.6 GPM per 3413 MBH
    - Evaporative and Spray Coil Humidifiers: 5.0 GPM per 1000 Lbs/Hr
  - 2) Humidification can be accomplished by direct injection of steam into the air stream, vaporizing water from a pan by heating it, passing air through a moist porous pad, or by spraying water from nozzle into the air stream.
  - 3) Any pan-type humidifiers installed in special purpose areas shall use distilled water, if available.
  - 4) Avoid spray coil systems.
  - 5) It is best practice to place humidifiers at least 10 feet from any duct transformation, elbow, fitting or outlet. Consideration must be given to length of vapor trail and air handling unit and ductwork design must provide sufficient length to prevent vapor trail from coming in contact with items downstream of humidifier before vapor has had time to completely evaporate.
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## DESICCANT DEHUMIDIFICATION

As new building codes require more outside air to be circulated into buildings, the amount of humidity that an air-conditioning system must remove has increased substantially. Conventional cooling systems are consuming more and more energy removing moisture, often overcooling the air to below its dewpoint just to get the water out.

Recently, new desiccant-based cooling systems have emerged on the market. Desiccants are materials that absorb moisture from their surrounding space. Most people know them as small, usually white, packets found in the packaging of electronics and dried foods. Desiccant materials used in building applications can be regenerated. In other words, the moisture is driven out of them by the application of heat, whereupon they can be reused to absorb more water from the air. The heat is generally derived from gas, steam, or waste heat from the building.

Both the ASHRAE Handbook Fundamentals and Handbook Applications describe several applications where desiccant dehumidification should be considered. The best applications for desiccant cooling systems are buildings with large dehumidification loads, long hours, and those in warm and humid climates—for example, hospitals, swimming pools, or supermarket fresh produce areas. The cost of gas or waste heat used for regeneration is typically much lower than the cost of electricity used for conventional dehumidification.

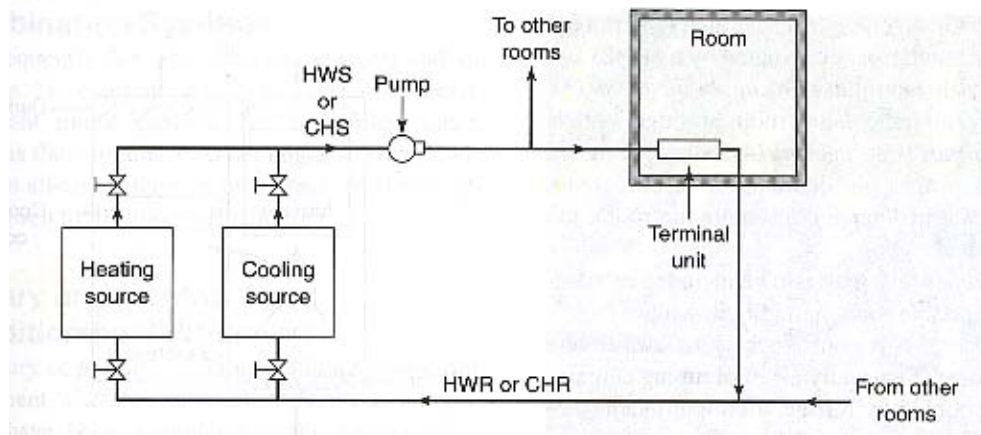
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## PIPING SYSTEMS

Piping systems supply steam, hot water and chilled water for heating and cooling applications. These systems also provide make-up water to cooling towers, boilers and return steam condensate to boilers. Condensate from cooling coils is taken to a drain by pipe systems. The hydronic piping in HVAC applications may be classified as two pipe, three pipe or four pipe systems.

### Two Pipe Systems

Both heating and cooling of a space can be accomplished using the same piping by connecting a hot water boiler and water chiller in parallel (see figure below), and using each when needed.



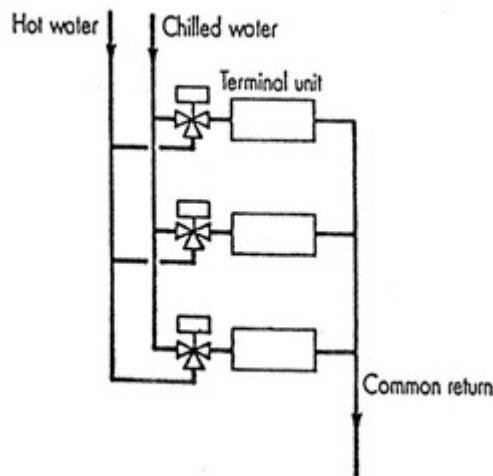
Two pipe systems consist of a network of insulated pipes; one pipe supplies chilled water and the second pipe returns it to the chiller. The secondary water is cold in summer and intermediate seasons and warm in winter. With water-changeover, chilled water is circulated during the cooling season and hot water during the heating season. The problems occur during the mid-seasons where cooling can be required part of the time and heating part of the time and no heating or cooling the rest of the time.

Note that the water flow rate required for heating is much low than the chilled water flow. The piping and pumping is sized for the maximum flow of chilled water. Using the same infrastructure of piping shall result in very low velocities during heating. To overcome this if not all, 50% of pumps may need to be

operative. Energy is wasted in terms of pumping cost. It is better to use 4-pipe system and lower flow rates with smaller pumps.

### Three- Pipe System

Three pipe systems have separate chilled and hot water supplies with a common return. These systems are rarely used because they consume more energy due to excessive mixing of the chilled and hot water in the common return.

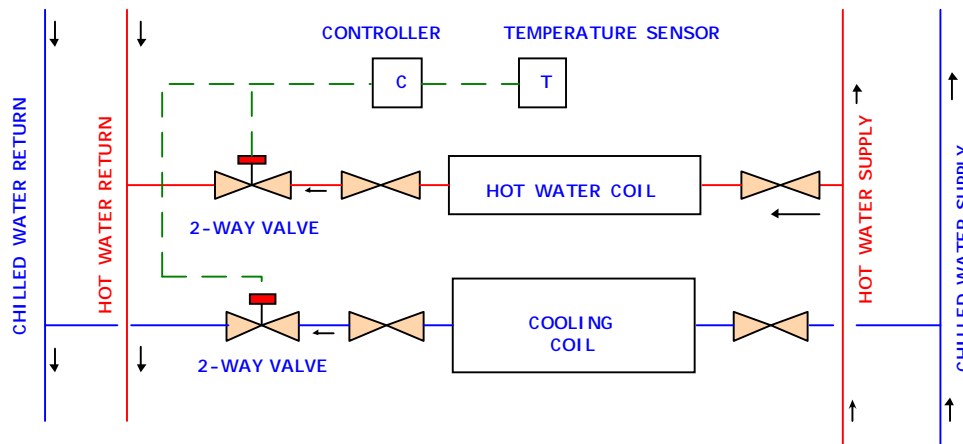


### Four-Pipe System

Four-pipe systems provide two independent water systems - one dedicated to chilled water and one to hot water i.e. four-pipe systems have a cold water supply, cold water return, warm water supply and warm water return.

The system is further categorized as the independent load system and common load systems.

- 1) Independent load systems have two separate water coils, one served by hot water, the other by cold water. The systems make use of 2-way on-off valves.
- 2) Common load systems can have a single coil in the air handler but still supplied independently with 4-pipe system. The systems make use of 3-way diverting valves.





All piping required for HVAC systems shall be sized based on the criteria listed below:

- 1) Pipe sizing shall be based on "Cameron Hydraulic Data"; with C=100 for open systems (Example: Cooling Tower) and C=150 for closed systems (Example: Chilled and Hot Water Systems).
- 2) For closed systems, the maximum friction loss shall be 1200 mm (4 ft) of water per 30 m (100 ft) of pipe with maximum velocity of 1.2 m/s (4 fps) for systems in occupied areas, and up to 2.4 m/s (8 fps) for mains and large branches.
- 3) For open systems, the maximum friction loss shall be 1200 mm (4 ft) of water per 30 m (100 ft) of pipe and a maximum velocity of 2.4 to 3.0 m/s (8 to 10 fps). The minimum pipe size shall be 20-mm (3/4-inch).
- 4) Square law: Pressure drop is proportional to velocity squared. Doubling the diameter of pipe reduces velocity to 1/4th and also pressure drop to 1/4th.
- 5) Chilled water and condenser water piping shall preferably be schedule 40 black steel pipe with welded joints and dielectric fittings at points of connections to dissimilar metals. Victualic joint coupling is not recommended. Above ground chilled water piping runouts to the individual AHU of sizes 2 inches and smaller may be Type "L" copper not to exceed 20 feet in length. Black steel pipe installed underground shall be coated with a bitumastic material.
- 6) Piping systems must be hydrostatically tested for leaks before being put into service. All hot and chilled water piping shall be tested at 1½ times the operation pressure at the operating temperature for a period of 24 hours.
- 7) Hot and chilled water piping shall include flow switches, pressure switches and gages, thermometers and thermo-wells, solenoids, and relief valves for system control.
- 8) In hydronic systems an air separator should be incorporated to remove air from the water and an expansion tank is provided to account for changing density in the water.
- 9) Install flanges in piping 2½" and larger adjacent to each valve, and at the final connection to each piece of equipment.
- 10) Install drains at low points in mains, risers, and branch lines consisting of a tee fitting, ¾" ball valve, and short ¾" threaded nipple and cap.
- 11) Install dielectric fittings to connect piping materials of dissimilar metals in dry or wet piping systems (water, steam).
- 12) Seal pipe penetrations through exterior walls using sleeves and mechanical sleeve seals. Pipe sleeves smaller than 6" shall be steel; pipe sleeves 6" and larger shall be sheet metal.
- 13) Install strainers on the supply side of control valve, pressure reducing or regulating valve, and solenoid valve. Provide Y-Type strainers with 125 PSI working pressure rating for low pressure applications, and 250 PSI pressure rating for high pressure application. Strainer shall be Type 304 stainless steel, with 3/64" perforations at 233 per square inch.
- 14) Steel Pipe Joints (2" and Smaller): Thread pipe with tapered pipe threads in accordance with ANSI B2.1. Pipe larger than 2": Weld pipe joints (except for exterior water service pipe) in accordance with ASME Code for Pressure Piping, B31.
- 15) The mid-span deflection due to dead weight loading should be limited to no more than 1/8 inch for lines that are required to drain. For lines that are not required to drain, deflection should be limited to ½ inch.
- 16) For the condenser and chilled water pipe sizes 10 inches and larger, the pipe support shall be engineered and all supports/hanger sizes and locations shall be shown on drawings.
- 17) Buried Pipe
  - The trench for installation of buried piping should be of sufficient width and depth to provide necessary bedding and cover, depending on traffic volume to facilitate joining, trapping, and future maintenance concerns.

- As applicable, analysis of buried piping should consider soil, surface, internal pressure, thermal growth, soil settlement and seismic loads.
- Underground piping should be buried beneath the frost line. If conditions do not allow this, heat tracing and insulation should be used.
- Pipe guides and anchors should be provided to keep pipes in accurate alignment; direct the expansion movement; and prevent buckling, swaying, and undue strain. Spider-type supports should be provided inside the encasement piping to permit lead detection.

#### 18) Steam and Condensate Systems

- Steam lines should slope 1/8 inch per foot in the direction of steam flow. Lines with lesser slopes should have provisions for slower warm-up to allow condensate time to flow to the traps. Condensate removal provisions should also be placed at shorter intervals to reduce accumulation of condensate.
- Each low point should have a steam trap and free blow with drainage provisions to a lower elevation. If condensate from steam traps discharges into a common header, a check valve is typically located downstream of each trap. The maximum backpressure that is possible in the header is normally designed to not impede the flow of condensate.
- Drip legs should include a steam trap and blowdown drains. For steam lines less than 3 inches, the drip should be the same diameter as the run pipe and the blowdown line and valve should be at least 1/2 inch.
- Blowdown (free blow) lines are normally sized to accommodate all condensation developed during steam line warm up.
- Provisions should be made to drain condensate from the upstream side of isolation valves. Small bypass lines can be installed around pressure reduction stations and around larger isolation valves. Condensate drainage from each drip leg is typically adequate for drainage of all condensate from the full-open warm-up bypass valve. Bypass valves are normally sized to not exceed the flow capacity of the downstream drip legs to handle the condensate collection during the initial startup of the steam system.
- Steam header control valves and pressure regulators should be provided with upstream strainers. Pressure regulators should have upstream drip legs and manual bypass lines.
- Steam traps should provide adequate capacity to accommodate condensation loads during warm-up as well as during normal operation and to compensate for line size, length, and insulation type and thickness. Traps for end user equipment (e.g., heating coils) normally accommodate full condensate load from the equipment warm-up in order to be considered automatic draining.
- Hot water and steam distribution pipe shall incorporate electrically actuated solenoids for stopping fluid flow and relief valves to prevent over pressure of vessels and systems.

#### 19) Piping at Pumps

- Piping configurations at pumps require special considerations to ensure the safe and effective operation of the pump. There should be at least 10 diameters of pipe between the suction of the pump and the first elbow. This is especially critical in double ended pump designs as the turbulent inlet flow can cause shaft thrusting, and subsequent bearing problems.
- Proper piping layout permits both the suction and the discharge pipes to be supported independently of the pump so that little load is transmitted to the pump casing.
- Suction piping should be at least one size larger than the suction flange at the pump.
- Use eccentric reducers rather than concentric reducers at the pump suction. Concentric reducers will trap air. Be sure the eccentric reducer is not installed up side down. The top of the reducer should go straight into the suction flange.

- Startup strainers necessary near pumps should have adequate space for installation.
- 20) **Expansion Joints:** Metallic or rubber bellows-type expansion joints should be used in the design of piping systems in the following instances:
- to absorb the thermally induced dimensional changes in the piping system;
  - to minimize the stresses and moments in the system;
  - to minimize the loads imposed on equipment nozzles;
  - to reduce the recurring problem of rotating equipment misalignment due to nozzle loads, particularly at pumps;
  - where space is inadequate for a conventional flexible piping arrangement;
  - where a conventional piping arrangement would result in excessive pressure drop in the system;
  - where the economics favor the expansion joint over a conventional piping arrangement;
  - to compensate for differential expansion of a pipe within a pipe or at flued heads
- 21) **Freeze Protection**
- In cold climates external piping should be provided with freeze protection. Freeze protection typically consists of either steam or electric heat tracing. Economic considerations dictate which method is selected.
  - Avoid routing piping along the inside of exterior building walls or un-insulated siding in cold climates. Where possible, route piping near radiant heat sources to prevent freezing.
  - Review the location of steam relief valve vent stacks to ensure that the condensing vapor does not cause ice to form on nearby equipment, walkways, or roads.
  - Consider local area frost penetration when designing underground piping.
- 22) **Tubing:**
- Carbon Steel Tubing shall meet ASTM A179
  - Stainless Steel Tubing shall meet ASTM A269
  - Alloy 400 Tubing shall meet ASTM B165
  - Seamless Copper Tubing, Types K, L, & M shall meet ASTM B88

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## VALVES

Valves should be located in systems to provide proper isolation for maintenance tasks, normal operations, and consideration should also be given to provide future expansion of the system without major system outages.

- 1) Gate valves are unsuitable for throttling and should be used in the full open or closed position.
- 2) Globe valves should be used primarily for throttling service only unless system flow reverses and the globe valve serves as a stop valve.
- 3) Substituting a globe valve for a gate valve in a piping system is similar to adding another 100 feet (31 meters) of piping to the system. On the discharge side of the pump this will cause the pump to run off of its economical point with a resultant shaft bending. On the suction side of the pump it will probably cause "cavitation".
- 4) Butterfly valves should be used for stop valves or for throttling purposes in water systems.
- 5) Ball valves should be used for bubble-tight stop valves in relatively clean fluid services.

- 6) Valve stems, T Branches and elbows should be installed perpendicular to the pump shaft, not at right angles.
- 7) Valve access is best accomplished in the natural routing of pipe from point to point, avoiding the use of vertical loops and pockets. Control valves should be located adjacent to walkways wherever possible. (Access platforms should be provided where valves are otherwise inaccessible.)
- 8) A minimum of 4 inches of knuckle clearance is recommended around hand wheels.
- 9) Chain-wheel operators can be used as long as they do not present a hazard to operating personnel.
- 10) Control valves should be located near the operating equipment to be observed while on local manual control.
- 11) Control valve manifolds should have a minimum of at least three diameters of straight pipe both upstream and downstream of the control valve.
- 12) Concentric reducers are recommended to make size reductions at the control valve.
- 13) The use of underground chilled or condenser water shutoff valves in valve boxes is prohibited. Install the valves inside the building mechanical rooms in an accessible location and properly identified.
- 14) Valves for the chilled water system shall be as follows:
  - Bronze body ball valves with stainless steel balls and stems for isolating and throttling duty in sizes up to 2-1/2 inches.
  - Bubble tight, ductile iron, lug body butterfly valves with EPDM seats and silicon bronze discs for isolating and throttling duty in sizes 3 inches and larger.
  - Ductile iron body plug valves for precision balancing applications.
  - Required at each piece of equipment, branch tap, main riser and at other locations to allow system isolation and balancing.

## HVAC DUCTWORK

Duct systems should be designed in accordance with the general rules outlined in the latest ASHRAE Guide and Data Books, SMACNA Manuals and Design Guide Section of the Associated Air Balance Council Manual. Note the following facts:

- 1) Supply duct system, with total external static pressure 2 inches and larger, shall be designed for a maximum duct velocity of 2500 fpm for duct mains and a maximum static pressure of 0.25 inch of water gage per 100 ft. Static pressure loss and regain shall be considered in calculating the duct sizes. Size supply branch ducts for a maximum duct velocity of 1500 fpm.
- 2) All other duct systems such as return and exhaust, including branch ducts, shall be designed for a maximum velocity of 1500 fpm for the duct mains and a maximum static pressure of 0.10 inch of water gage per 100 ft, with the minimum duct area of 48 sq in that is 8 in x 6 in size.
- 3) Primary air ductwork (fan connections, risers, main distribution ducts) shall be medium pressure classification as a minimum. Secondary air ductwork (run-outs/branches from main to terminal boxes and distribution devices) shall be low pressure classification as a minimum.
- 5) Table below provides pressure classification and maximum air velocities for all ductwork.

DUCTWORK CLASSIFICATION		
Static Pressure	Air Velocity	Duct Class
1 in-WG	< 2000 FPM	Low Pressure

2 in-WG	< 2000 FPM	Low Pressure
3 in-WG	< 2500 FPM	Medium Pressure
4 in-WG	> 2000 FPM	Medium Pressure
6 in-WG	> 2000 FPM	Medium Pressure
10 in-WG	> 2000 FPM	High Pressure

- 6) Supply, return and exhaust air ducts shall be designed and constructed to allow no more than 3 percent leakage of total airflow in systems up to 3"-WG.
- 7) Systems above 3"-WG through 10"-WG ducts shall be designed and constructed to limit leakage to 0.5 percent of the total air flow.
- 8) Duct sizes should generally be kept as large as possible – a small difference in diameter makes a huge difference to the amount of restriction, as the back pressure from friction losses is inversely proportional to the power of 4. This means that halving the size of the duct increase the pressure lose by 32 times! As a rule of thumb the pressure drop as a function of velocity range is depicted below:

Duct Pressure Classification	Velocity Range	Pressure Drop
Low Pressure	1500 – 1800 FPM	0.10 – 0.15 in-WG
Medium Pressure	2000 – 2500 FPM	0.20 – 0.25 in-WG
High Pressure	2500 – 3500 FPM	0.40 – 0.45 in-WG

- 9) Designing air distribution systems to avoid excessive duct lengths/fittings, high air velocities, and pressure drop can have a major impact on energy. For a given airflow rate, doubling the duct diameter reduces duct pressure drop by 75 percent. Even an increase of one standard duct size can generate significant energy savings.
- 10) Round, smooth galvanized-steel ducts are recommended for maximum air-carrying capacity with minimum pressure loss. Round ducts require 27 percent less metal per unit of air-handling capacity than rectangular ducts and have lower installation costs, which may result in substantial capital cost savings. They are also quieter during operation and easier to fabricate and seal than rectangular ducts.
- 11) When rectangular ducts must be used due to space limitations, keep the width-to-height ratio close to 1:1 for lower pressure drop. Avoid using duct with an aspect ratio greater than 3 to 1, if possible. Aspect ratio of duct is the width by height ratio. Note the following advantages of low aspect ratio:
  - As the aspect ratio increases, the friction loss increases.
  - They contain the largest volume per unit of wetted perimeter length.
  - They tend to have the lowest weight per contained volume. This is somewhat a function of the specific construction technique used.
  - They will tend to be structurally more rigid, especially round ducts.
  - Due to the structural rigidity, they will produce less noise, especially drumming.

- 16) Using larger ducts often requires not only more sheet metal but also thicker or heavier gauge material. Check and compare tabular data below for different duct dimensions suitable for airflow of 10000 CFM at a nominal 0.2 in-WG per 100 feet length.

Duct Size - inches			Aspect Ratio	Cross Sectional Area - sq.ft.	Perimeter - ft.	Ratio of Cross Sectional Area to Perimeter	Gauge	Pounds of Sheetmetal per lineal foot of duct	Velocity - fpm
Height	Width	Diameter							
N/A	N/A	29.0	N/A	4.59	7.59	0.60	24	9.10	2,180
26.5	26.5	N/A	1.0	4.88	8.83	0.55	26	8.00	2,051
18.0	41.0	N/A	2.3	5.13	9.83	0.52	24	11.37	1,951
14.0	56.0	N/A	4.0	5.44	11.67	0.47	24	13.49	1,837
12.0	70.0	N/A	5.8	5.83	13.67	0.43	24	15.80	1,714

Round duct weight information based on spiral construction.

Another example: a case where two parallel 60"x36" ducts with total cross-section area of 30 ft<sup>2</sup> have lower installation cost than one duct of 120"x30" with cross-section area of 25 ft<sup>2</sup> because the metal gauge for the single duct is 18 and the gauge for the two smaller ducts is 20.

- 17) Ensure that duct design incorporates all features necessary to accommodate testing, adjusting, and balancing (TAB). For example, provide adequate length of duct, both upstream and downstream of fans and coils.
- 18) An intelligent placement of the air handler (in middle) and routing the ductwork can minimize the pressure losses and air velocities.  
These unique components not only require special consideration in calculating their pressure loss, they also greatly influence the design of the duct system.
- 19) Frictional pressure losses are inversely proportional to duct diameter raised to the fifth power. Thus, simply choosing a duct diameter that is one standard size larger than initially calculated can produce a large reduction in energy use.
- 20) Ducts should always be at least 6 inches in the smallest dimension to allow for the use of standard tools during installation and repair.
- 21) Ducts, except for runs to outlets, should be sized in 2 inches increments, e.g., 10," 12," and 14" diameters. Some ducts may appear slightly oversized using this approach, but avoiding extra transition fittings between different sizes of duct will reduced installation costs and performance will be nearly the same. Recommendations for round duct diameters are:
  - 1 inch increments from 6" to 10" diameter
  - 2 inches increments from 10" to 40" diameter
  - 4 inches increments above 40" diameter
- 22) Branch ducts should be at least 2 inches smaller than the trunk ducts to allow use of standard 1 inch flanges for connections to branch ducts.
- 23) The final duct connection to an outlet in a suspended ceiling grid should be reinforced flexible duct in lengths of 6 feet or shorter and installed without any kinks. This length allows easy offsets to make final connections even if the ceiling grid turns out to be quite different than anticipated.
- 24) Duct taps should be 2 inches smaller than main duct to properly construct and seal duct.
- 25) Duct size should be 2 inches wider than diffusers, registers, and grilles.
- 26) All rectangular duct taps should be made with shoe (45 degree) collar fittings.
- 27) Recommended Sheet Metal Thickness for Ducts



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

28) Ductwork Materials:

- Galvanized Steel: HVAC Applications; Most Common; Galvanized steel sheets meeting ASTM A90, A525, and A527, Lock Forming Quality
- Carbon Steel: Breechings, Flues and Stacks; Carbon steel meeting ASTM A569 for stacks and breechings 24" and larger; Galvanized sheet steel meeting ASTM A527 with ANSI/ASTM A525 G90 zinc coating for stacks and breechings less than 24 inches.
- Aluminum: Moisture Laden Air Streams; Aluminum base alloy sheets meeting ASTM B209, Lock Forming Quality.
- Stainless Steel: Kitchen Hood and Fume Hood Exhaust; Stacks and Breechings (Prefabricated); Type 304, 304L, 316, or 316L stainless steel sheets meeting ASTM A167:
  - 304 and 316: Non-welded applications.
  - 304L and 316L: Welded applications.
- Concrete: Underground Ducts, Air Shafts; Reinforced concrete pipe meeting ASTM C76, Class IV.
- Copper: Ornamental.
- Polyvinyl Steel and Stainless Steel (PVS and PVSS): Chemical Exhaust

29) Duct Location Recommendations:

- Plan schematic ductwork layout. Where possible, locate air handling units centrally over zones to minimize duct sizes and lengths for supply, return, exhaust and outside air ducting.
- If a central return air system is planned, locate the unit(s) near the returns. Consider sound issues before locating a unit directly above an air terminal.
- For rooftop units with ductwork located within the building, specify down-flow configuration to reduce ductwork turns, heat loss and air leakage.
- Locate ductwork within conditioned space whenever possible. Heat gains and losses are minimized and go directly to the conditioned space.
- Plan locations to minimize duct lengths, turns and fittings
- Ducts outside the building incur the greatest thermal losses and energy penalties, especially when exposed to solar gain. Factors affecting losses include air leakage, heat conduction, solar

radiation, and solar reflection effects. Ducts in unconditioned spaces and outside the building should be well insulated. Locate away from sources of heat and in shaded areas when possible.

- Locate units for easy access and serviceability. Provide manufacturer recommended clearance for service access and proper airflow.
- With the exception of ceiling return plenums and under-floor air delivery, building cavities should not be used for air distribution.
- When chases, furred spaces or other cavities (except ceiling return plenums) are used for air pathways, provide sealed ductwork within the cavity to convey the air.
- Unconditioned attic spaces containing ducts should be well ventilated.
- Building cavities used as ducts, such as enclosed support platforms, mechanical rooms, ceiling spaces, wall and cavities and chases, are usually less airtight than standard ducts and plenums.

30) Flexible Ducting:

- Minimize flexible duct use, which can increase pressure drop by 63 percent compared with galvanized-steel ducts. Limit use to diffuser connections with a maximum length of 10 feet each.
- If flexi-duct must be used, ensure it is fully extended and cut off any excess. If there is any excess left in the run, it is likely to spring back, which reduces the inside diameter and hence increases the restriction. A 20% excess left in the duct run of flexiduct increases the air restriction by 3-7 times!
- Flexible air duct shall be supported at 5'-0" intervals and on both sides of wall sleeves with 2" wide hanger bands with rolled edges.
- The maximum length of flexible air duct shall not exceed 10'-0" unless authorized by the Designer.

31) Duct Layout Considerations:

- Plan duct layout to minimize duct lengths, turns and fittings while providing good air distribution.
- If the duct size on the fan is reduced (e.g. fitting a 150mm duct on a 200mm fan) and in particular if the intake is reduced as well, then less air gets to the tips of the fan blade, which is what does the work. Efficiency is reduced to around 60% or less, just by doing that.
- At the very minimum, the duct size should be at least as large as the size of the fan, i.e. a 150mm fan uses 150mm ducting. This is more important for axial fans, where it is the blade tips that do all the work.
- If the duct is smaller than the fan inlet, or is not in line, then effectively the fan is unable to get any air to the fan tips, it is only the inner section doing the work.
- If the exhaust duct is smaller than the fan outlet, the air is likely to suffer from turbulence, and hence is less efficient.
- Install radius or section elbows at > 45-degree turns.
- Install turning vanes in supply ducts that turn immediately below a roof penetration. Use vanes with airfoil shape to reduce pressure drop in corners.
- Correct vane settings are critical. Improper vane settings can result in pressure drops as high as 200 percent of normal.
- Ceiling return plenums, when constructed for reasonable air-tightness can reduce construction costs, pressure losses and fan energy.
- In conditioned rooms without return registers, provide transfer grilles from the room to general return air area. An alternative is to ensure that undercut doors are provided with sufficient free area for face velocities under the door of 100-300 fpm.

- Avoid opposed-blade dampers. These dampers cannot be relied upon to take more than  $\frac{1}{4}$  -  $\frac{1}{2}$  closures without noise. 12a For air balancing, provide accessible volume dampers near the branch take-off.
- Avoid designs which require airflow to turn immediately before/after a supply/return outlet.
- Installing inlet vanes at return outlets and extractors at supply outlets can reduce pressure losses and correct airflow in sharp turns.
- Use smooth WYE branch fittings.
- The first elbow in the ductwork leaving the unit should be no closer than 2 feet from the unit, to minimize resistance and noise.
- A shorter total effective duct length (straight run length plus fitting equivalent length) will result in lower pressure losses and lower operating pressure.
- Select low pressure drop fittings and accessories. Well designed aerodynamic fittings are more expensive but their use is justified because the lower pressure losses result in more economical operation.
- Ensure that there is adequate static pressure for proper operation of air side devices such as flow regulators, mixing boxes, distribution boxes and air terminals.

32) Ductwork Testing:

- -3"-WG and Lower: 1.5 x Pressure Rating
- -2" to +2" WG: Generally not tested
- +3" WG and Higher: 1.5 x Pressure Rating

33) Maximum ductwork hanger spacing:

- Horizontal: 8 feet maximum
- Vertical: 16 feet and at each floor

34) Perform frequent inspections of the duct system during construction, while ducts are accessible, to ensure adherence with correct installation and sealing procedures.

35) Velocity in feet per minute (FPM) = Airflow in cubic feet per minute (CFM) / Sq-ft of duct cross-section

36) Total Pressure = Static Pressure + Velocity Pressure

37) Velocity Pressure =  $(\text{FPM}/4005)^2$

- 4005 FPM = 1 inch-w.g velocity pressure
- 2000 FPM = 0.25 inch-w.g velocity pressure
- 1000 FPM = 0.067 inch-w.g velocity pressure
- 500 FPM = 0.016 inch-w.g velocity pressure
- 100 FPM = 0.006 inch-w.g velocity pressure

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**AIR PLENUM**

- 1) Ceiling spaces should be used as return air plenums only when the thermal and air barrier are located at the roof or if there is conditioned space above.
- 2) T-bar ceilings used as return air plenums should be constructed carefully to minimize leaks between the plenum and the outdoors or unconditioned spaces.

- 3) The barrier between the plenum and conditioned space need not be airtight, with the exception of large leaks that may cause short circuiting of room supply and return air.
- 4) To determine the width required for an extended plenum, count the number of round pipes it will serve, multiply this number by 2, and then add 2 — the total is the required duct width in inches. For 8 inches round pipes, multiply by 3, then add 2. If the desired size is not available, choose the next larger size.

For example

A duct serving four 6" round pipes will have a width of  $4 \times 2 = 8 + 2 = 10$ .

A duct serving five 8" round pipes should have a width of  $5 \times 3 = 15 + 2 = 17$ . Since rectangular duct is not available in a 17" width, use the next larger standard size, which is 18".

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### **DUCT INSULATION**

- 1) Insulate ducts in accordance with the minimum requirements of local codes or ASHRAE 90.1-19992a, whichever are more stringent.
- 2) Insulation exposed to weather should be suitable for outdoor service and provided with a weather-protective covering. Cellular foam insulation should be provided with a weather-protective covering or painted with a coating that is water retardant and provides shielding from solar degradation of the material.
- 3) In cooling load-dominated areas, ducts located on roofs should be covered with a reflective coating.
- 4) Avoid compressing insulation against building or hangers.

Read more about insulation specs in subsequent paragraphs.

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### **DUCT SEALING**

- 1) Seal all ducts, including ducts in conditioned space, in accordance with the minimum requirements of SMACNA and the guidelines listed in Tables 6.2.4.3 A and B of ASHRAE 90.1-1999.
- 2) If the duct leakage is 20% of the total air flow, the efficiency of the cooling system can drop by 50%. Heating efficiency is similarly affected. Duct leakage also lowers the heating and cooling capacity, and can lessen equipment life.
- 3) All ducts should be sealed for SMACNA Seal Class B minimum
- 4) SMACNA-HVAC Ductwork Leakage Classes
  - Seal Class A: 2–5% Total System Leakage (All Transverse joints, longitudinal seams, and duct penetrations).
  - Seal Class B: 3–10% Total System Leakage (All Transverse joints and longitudinal seams)
  - Seal Class C: 5–20% Total System Leakage (All Transverse joints)
- 5) Seal openings greater than 1/16" wide at duct, plenum and cabinet joints and where equipment such as pipes, conduits, damper levers and sensors penetrate ducts.
- 6) Approved sealants are flexible gaskets, fiber-reinforced mastic or mastic used with mesh tape. Sealants should be UL 181 listed, water based, non-toxic and water resistant with high solids content. Sealants used outdoors shall be rated for outdoor use and resistant to weather and solar degradation.
- 7) Duct systems sealed with mastics generally exhibit lower leakage. If tape is used, methods and materials should comply with the ADC - Flexible Duct Performance & Installation Standards.
- 8) Oil base caulking and glazing compounds should not be used.

- 9) Apply sealants according to the manufacturer's instructions with proper surface preparation and in correct temperature conditions.
- 10) Ensure that the ductwork connections to the air handling units or curb are sealed airtight and that the seals can withstand vibration. Provide airtight seals at duct connections to air outlets, duct access panels and equipment cabinet panels. These are commonly overlooked sites for leaks.
- 11) Do not apply sealant in spiral duct lock seams. This can result in poor seam closure and less satisfactory control.

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### **BASIC KITCHEN DESIGN EXHAUST REQUIREMENTS**

- 1) Compile total equipment list and layout from kitchen designer
- 2) Calculate heat gain including heat rejection from freezers/refrigerators (reach-in & walk-in)
- 3) Calculate the ventilation requirement and add the exhaust requirements for the hoods. Provide 12 to 15 air changes per hour or as calculated by ventilation, whichever is greater.
- 4) Verify overhang of hood over deepest equipment item (6" min. required, 12" + is desirable)
- 5) Listed hoods are not penalized for additional CFM by greater overhang
- 6) Identify source of replacement air
- 7) Include 20% transfer air
- 8) Extend hood beyond cook surface 0.4 x distance between hood and cooking surface

Note the specific requirements below:

- 9) Exhaust/Makeup Air:
  - Duct velocity should be between 1500–2500 FPM
  - Supply shall be equal to exhaust
  - Dishwasher exhaust: 600 – 1200 CFM
  - Hood velocities shall not be less than 50 FPM over face area between hood and cooking surface
  - Exhaust over cooking areas: 150-400 CFM/LF
  - Exhaust rate for wall type hood - 80 CFM/ft<sup>2</sup>
  - Exhaust rate for island type hood - 125 CFM/ft<sup>2</sup>
- 11) Duct Sheet Metal Gauge:
  - 16 gauge Galvanized Steel
  - 18 gauge 304 Stainless Steel
- 12) Duct Slope:
  - Lengths 75 and less: ¼" per foot toward hood
  - Lengths greater than 75 feet: 1" per foot toward hood
  - Extend hood beyond cook surface 0.4 x distances between hood and cooking surface
- 13) Hood Exhaust: Canopy Hoods (attached to wall)
  - 200 CFM/Sq-ft over char broilers
  - 100 CFM/ Sq-ft over high temperature appliances
  - 75 CFM/ Sq-ft over medium temperature appliances

- 50 CFM/ Sq-ft over low temperature appliances

14) **Hood Exhaust: Canopy Hoods (exposed all sides)**

- 300 CFM/ Sq-ft over char broilers
- 150 CFM/ Sq-ft over high temperature appliances
- 100 CFM/ Sq-ft. over medium temperature appliances
- 75 CFM/ Sq-ft over low temperature appliances

15) **Non-Canopy:** 300 CFM/Lineal Ft. of cooking surface.

16) **Hood Construction:** 22 gauge Galvanized Steel or 22 gauge 304 Stainless Steel.

17) **1991 NFPA 96 Exhaust/Makeup Air Requirements**

- 1500 FPM minimum Duct Velocity
- Ducts shall not pass through fire walls or partitions
- Ducts shall lead directly as possible to the outside
- Ducts shall not be connected with other ventilating or exhaust systems
- Ducts shall terminate a minimum of 40" above roof surface, 10 feet from outside air intakes and property lines, and 3 feet above any air intake within 10 feet.

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**LABORATORY FUME HOODS**

- 1) Minimum face velocities for laboratory hoods shall be 100 FPM measured at an 18" sash opening and have label identifying what fan serves it, (hood-location, room, lab, etc).
- 2) Exhaust fans shall be located on the roof and have label identifying what it serves.
- 3) Exhaust stacks shall be at least 10 feet above the surface of the roof, discharged vertically, with a minimum velocity of 2500 fpm.
- 4) Rain protection should be provided by use of a "no loss" stack as described in the Industrial Ventilation (latest edition).
- 5) The opportunity for entrainment of exhaust contaminants into buildings should be minimized by locating building air intakes as far away as practical from exhaust discharges and vehicle exhaust sources.
- 6) All hoods shall be provided with and air monitoring device to alarm at low flow.

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**PROCESS EXHAUST SYSTEMS**

- 1) Collect fumes and heat as near the source of generation as possible.
- 2) Make all runs of ducts as short and direct as possible.
- 3) When turns are required in the duct system use long radius elbows to keep the resistance to a minimum (preferably 2 duct diameters).
- 4) After calculating duct resistance, select the fan having reserve capacity beyond the static pressure determined. The connection to a fume hood or other piece of equipment will generally require between 1.0 and 3.0 inches WC negative pressure.
- 5) Use same rationale regarding intake ventilators and motors as in General Ventilation guidelines above.
- 6) Install the exhaust fan at a location to eliminate any recirculation into other parts of the plant.



- 7) When hoods are used, they should be sufficient to collect all contaminating fumes or particles created by the process.
- 8) Ductwork material must be selected to suit the material or chemical being exhausted—carbon steel, 304 or 316 stainless steel, fiberglass reinforced plastic (FRP), and polyvinyl chloride (PVC) are some examples.
- 9) Sprinklers are generally required in FRP and PVC ductwork systems in all sizes larger than 8 inch in diameter.
- 10) Process exhaust systems should be provided with a blast gate or butterfly damper at each tap for a hood or equipment, at each lateral, and at each submain. At all fans, large laterals, and submains, a tight shutoff style butterfly damper should be provided for balancing and positive shutoff in addition to the blast gate. Blast gates should be specified with a wiper gasket, of EPDM or other suitable material, to provide as tight a seal as possible for blast gates; otherwise blast gates tend to experience high leakage rates.
- 11) Process exhaust ductwork should be sloped a minimum of 1/8 inch per foot with a drain provided at the low point. The drain should be piped to the appropriate waste system.
- 12) Process exhaust systems are required, in most cases, to undergo a treatment process—scrubbing, abatement, burning, or filtering.
- 13) Duct sizing must be based on capture velocities and entrainment velocities of the material or chemical being exhausted. For most chemical or fume exhaust systems, the mains, risers, submains, and large laterals should be sized for 2,000 to 3,000 feet per minute and small laterals and branches should be sized for 1,500 to 2,500 feet per minute. Discharge stacks should be sized for 3,000 to 4,000 feet per minute discharge velocity and should terminate a minimum of 8 feet above the roof and a minimum of 10 feet from any openings or intakes.

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## AIR OUTLETS

A supply outlet is positioned to mix conditioned air with room air and is responsible for most of the air movement within a room.

- **Diffuser:** Circular, square, or rectangular air distribution outlet, generally located in the ceiling and comprised of deflecting members discharging supply air in various directions and planes and arranged to promote mixing of primary air with secondary room air.
- **Grille:** A louvered or perforated covering for an opening in an air passage, which can be located in a sidewall, ceiling, or floor.
- **Register:** A combination grille and damper assembly over an air opening.

Occupant comfort requires that supply register locations be carefully selected for each room. In selecting outlets, keep in mind that cool air tends to drop away from the outlet, and warm air tends to rise to the ceiling. Following key point should be noted:

- 1) Install diffusers, registers, and grilles according to NFPA 90A, "Standard for the Installation of Air-Conditioning and Ventilating Systems"
- 2) Test performance according to ASHRAE 70, "Method of Testing for Rating the Performance of Air Outlets and Inlets."
- 3) The normal air velocity used for comfortable air distribution is 50 fpm while the acceptable range is from 25 to 75 fpm.
- 4) In general the velocity criteria for outlet devices is as follows:
  - Diffuser: Neck Velocity 1000 – 1200 FPM (NC 35-40)
  - Spot cooling, air curtain or laminar flow perforated plenum: 150-200 FPM

- Plaque, induction/slots, neck velocity: 1000- 1200 FPM
- 5) In cold climates, perimeter floor outlets that blanket portions of the exterior wall (usually windows) with supply air are generally preferred.
  - 6) In hot climates, ceiling diffusers or high wall outlets that discharge air parallel to the ceiling are preferred.
  - 7) In moderate climates, outlet location is less critical. Outlet locations near interior walls can significantly reduce duct lengths (decreasing costs), thermal losses (if ducts are located outside the conditioned space), and blower requirements.

### Supply Air Outlet Types

Type	Characteristics	Applications
Fixed blade grille	Single set of vertical or horizontal blades	Long perimeter grilles installations
Adjustable single deflection blade grille	One set of vertical or horizontal adjustable blades	Sidewall installation where single plane air deflection is required
Stamped plate grilles	Stamped from single sheet of metal with square, round or ornamental design openings	No adjustment of air deflection possible. Use for architectural design purposes only
Variable area grille	Similar to adjustable double deflection blade grilles with means to effectively vary the discharge area	Use with variable volume system to minimize variation of throw with variable supply air volume
Curved blade grilles	Curved blades to provide horizontal air pattern	Ceiling installation, High sidewall installation, Perimeter installation
Perpendicular flow slot diffuser	Generally 25 to 1 dimensional aspect ratio for maximum height of 3 inches	High sidewall installation. Perimeter installation in sills, curbs and floors
Parallel-flow slot diffuser	Generally 25 to 1 dimensional aspect ratio with maximum height of 3 inches	Ceiling installation
Air light fixture slot diffuser	Use in conjunction with recessed fluorescent light fixtures with fixed or adjustable air discharge patterns	Ceiling installation- ordered to match light fixture
Multi-passage round ceiling	Series of flaring rings or louvers forming series	Install in centre of area served

Type	Characteristics	Applications
diffuser Multi-passage square and rectangular ceiling diffuser	of concentric air passages	served
Adjustable pattern round ceiling diffuser  Adjustable pattern square or rectangular ceiling diffuser	Series of flaring rings or louvers forming series of concentric air passages. Air discharge pattern adjustable from horizontal or vertical or down blow pattern	Install for control of diffuser discharge pattern or where specific requirement to direct airflow pattern either horizontal or vertical.
Multi-pattern square and rectangular ceiling diffuser	Special louvers discharge air in one or more directions	Install in centre of area served or adjacent to partitions. Set pattern according to flow requirements
Half round diffuser	Matches round diffuser	Install in ceiling adjacent to partition of sidewall
Supply and return concentric diffuser	Combination diffuser with return grille in centre of diffuser	Install in centre of area served
Light fixture air diffuser combination	Combination diffuser light fixture	Ceiling installation combined with light fixture pattern
Perforated face diffuser	Perforated face plate with or without deflection device to obtain a horizontal discharge pattern	Install in centre or area served or control discharge pattern when installed off centre of area served
Variable area diffuser	Parallel or concentric passages or perforated face with means to vary discharge area	Use with variable volume system to minimize variation to throw with variable supply air volume
Air distribution ceilings	Ceiling system provided with round holes or slots	Use with ceiling supply plenum- particularly suited to large zones of uniform room temperature
Linear grille	Linear slot width ½ to 1 inch continuous length with adjustable airflow blades	Coiling and perimeter with air deflection adjustable form 1-way horizontal to vertical to 2-way horizontal

Type	Characteristics	Applications
Egg crate grille	Fixed square grid	Ceiling or sidewall (no pattern adjustment)

#### Locating and Selecting Returns and Exhaust Devices

- 1) Air motion is not significantly affected by the location of return and exhaust inlets. Also, the location of return and exhaust inlets will not compensate for ineffective supply air distribution.
- 2) The location of the return register has only a secondary effect on room air motion. However, returns can help defeat stratification and improve mixing of room air if they are placed high when cooling is the dominant space-conditioning need and low when heating is dominant.
- 3) Keep return air inlets away from supply outlet. A return air inlet that is located directly in the primary airstream of the supply outlet will short circuit the supply air back into the return without mixing with room air.
- 4) Return air inlets are generally located so that the room air returned is the greatest temperature difference from that being supplied to the room.

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#### **OUTDOOR AIR INTAKE**

The location of outdoor air intakes is critical and the basic guidelines include the following:

- 1) Locate outside air intake upstream of prevailing wind and away from sources of pollution and odors such as parking lots, roads, smoking areas, exhaust outlets, plumbing vents, debris, dust or fine particles.
- 2) Maximize distance between intakes and cooling towers, plumbing vents, loading docks, traffic, etc.
- 3) Special care must be taken to assure that exhausts from hoods, emergency generators, etc., will not be pulled into the building through make-up or fresh air intakes.
- 4) Maintain a minimum distance of 10 meters (30 feet) between intakes and exhausts, more if possible.
- 5) Locate intakes and exhausts on different building faces.
- 6) Do not add the air exhausted from the separate room or local exhaust to the return air or transfer it to any other areas.

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#### **LOUVERS**

- 1) Normal exhaust louver sizing based on 700 FPM through free area of louver
- 2) Normal intake louver sizing based on 500 FPM through free area of louver
- 3) Free Area Range:
  - **Metal:** 40–70% of Gross Area
  - **Wood:** 20–25% of Gross Area
- 4) Pressure Loss across louver is in range 0.01–0.10 inch- w.g
- 5) Transfer Ducts: 0.03–0.05 inch-w.g /100 ft: 1,000 FPM Maximum
- 6) Pressure Loss across transfer grilles: 0.03–0.05 inch-w.g
- 7) Pressure Loss across gravity relief air shafts: 0.03–0.05 inch- w.g @ 1,000 FPM Maximum

**DAMPERS**

- 1) Dampers serve to modulate the flow of air through the ducts to the various parts of the building, reducing or increasing the flow of air depending upon conditions.
  - 2) Opposed blade dampers should be used for balancing, mixing, and modulating control applications.
  - 3) Parallel blade dampers should be used for 2 positioning applications (open/closed) such as air handling unit equipped with emergency smoke exhaust arrangement.
  - 4) Typical pressure loss through dampers is 0.15 inch-w.g @ 2000 FPM, full open position
  - 5) Specify dampers with low leakage rate according to AMCA 500, "Test Methods for Louvers, Dampers and Shutters," Ideally the leakage rate shall not exceed 2 percent of air quantity at 2000-fpm (10-m/s) face velocity through damper and 4-inch-w.g (1000-Pa) pressure differential.
    - Standard dampers leak at a rate of 10 15 CFM/sq-ft at 1inch-w.g differential
    - Ultra low leakage dampers leakage rate is not greater than 6 CFM/sq-ft at 1 inch-w.g differential
    - Maximum leakage rate in low leakage dampers is less than 10 CFM/sq-ft @ 4inch-w.g differential maximum
  - 6) Size dampers at a flow rate of approximately 1200 to 1500 CFM/Sq-ft rather than on duct size.
- 

**FIRE DAMPERS**

The HVAC systems shall be designed to meet the requirements of the National Fire Protection Association codes, NFPA 45, 72E, 90A, 96, 99, and 101.

- 1) Fire dampers in HVAC applications are generally interlocking blade or expanding curtain type
- 2) Fusible link fire dampers are dampers which close when a fusible link melts and permits the damper to close.
- 3) Type A: Return air and transfer air; frame and damper is located in the airstream
- 4) Type B: Low pressure supply air; frame and damper is totally recessed out of the airstream
- 5) Type C: High pressure supply air; frame and damper is totally recessed out of the airstream
- 6) All fire dampers are required to have sleeves per NFPA.

6) Fire Damper Requirements:

1988 UBC Code:

- 1-hour Construction: 1 Hr rated fire dampers are required.
- 2, 3, and 4-hour Construction: 2 and 3 Hr rated fire dampers are required.

1991 NFPA 90A:

- 1-hour Construction: Dampers are not required.
- 2, 3, and 4-hour Construction: 2 and 3 Hr. dampers are required.

U.L. 555:

U.L. 555 requires fire dampers to bear an affixed label stating whether the damper is static or dynamic rated. Static fire dampers are tested without air movement in the ductwork whereas dynamic fire dampers are tested with air movement in the ductwork.

- Dynamic rated fire dampers must be U.L. tested and show airflow and maximum static pressure against which the damper will operate (fully close). Fire dampers are tested to 4 inch static pressure for "no duct" applications and 8 inches static pressure or "in duct" applications.

- Static rated fire dampers have not been U.L. tested against airflow and may not close under medium to high airflow conditions that may be encountered in HVAC systems which do not shut down in event of fire (i.e., smoke control systems).
- Recommend using dynamically rated fire dampers in all applications.
- Provide access doors to access all fire dampers, smoke dampers, smoke detectors, volume dampers, motor operated dampers, humidifiers, coils (steam, hot water, chilled water, electric), and other items located in ductwork which require service and/or inspection.

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### **SMOKE DAMPER**

Smoke management systems use smoke dampers to inhibit the passage of smoke from one zone to another. The subdivision of the building spaces into smoke zones shall conform to NFPA 101. All smoke partitions shall be shown on the HVAC floor plans exactly as depicted on the architectural floor plans. By appropriate opening and closing of smoke dampers in HVAC ducts, building zones free of fire can be pressurized with air while the zone with the fire can have smoke pulled from it.

#### 1991 NFPA 90A:

Smoke dampers shall be installed in systems over 15,000 CFM in the supply and return.

#### *Exceptions:*

- When AHU is located on the floor it serves and only serves that floor.
- When the AHU is located on the roof and only serves the floor immediately below it.
- Smoke dampers shall be installed at or adjacent (2 feet maximum distance from barrier) to the point where air ducts pass through required smoke barriers.

#### Smoke Detectors Requirements

- Air distribution systems with capacity greater than 2,000 CFM shall be equipped with smoke detector downstream of any filters and ahead of any branch connections in supply air system.
- At each story prior to the connection to a common return and prior to any recirculation or fresh air inlet connection in return systems over 15,000 CFM capacity and serving more than 1 story.
- Activation shall shut down fan, except smoke control equipment shall switch to smoke control mode.
- All duct-mounted smoke detectors, furnished by the electrical discipline, shall be shown on the HVAC drawings. The duct-mounted smoke detectors, if and when furnished in conjunction with the duct mounted smoke dampers, shall always be installed on the upstream side of the dampers.
- For a fully sprinkler building with quick response sprinklers, duct mounted smoke detectors or smoke dampers are not required in supply, return, and exhaust air ducts crossing the smoke barriers. This is a trade off permitted by the Life Safety Code NFPA 101.

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### **SMOKE EXHAUST SYSTEMS**

- 1) An engineered smoke control system shall be designed based on the requirements of NFPA 92B, 1995 edition and Uniform Building Code (UBC-1994 edition) Sections 402 and 905 requirements.
- 2) The smoke removal system shall be designed for manual as well automatic actuation. In the automatic mode, either a sprinkler flow switch, or a grid of the beam type of smoke detectors should energize the smoke removal system. The sprinklers are usually installed at 55 feet or below.
- 3) The make-up air system for the smoke removal, if required, will vary with the layout of the protected zone (such as atrium) and the surrounding spaces. Normally, upon activation of the smoke removal



system, the dedicated air-handling unit serving the zone, for its normal cooling and heating requirements, should be designed to go to 100 percent outdoor air mode. The additional make-up air, to comply with the code, should be brought in by a dedicated make-up air unit from the surrounding spaces. This unit should operate during the smoke removal mode only. Depending upon the severity of weather, 40° F or below, the make-up air should be heated to 55°F temperature. For milder climate, the make-up air can be admitted to the zone directly from outdoor, with or without ductwork, depending upon the physical configuration of the zone and the surrounding spaces. The use of the building air from the other air-handling units serving the occupied spaces shall be avoided to prevent any possibility of contamination.

### **CLEAN ROOM DESIGN**

The industry differentiates between the cleanliness of rooms by referring to class numbers. Federal Standard 209E, "Airborne Particulate Cleanliness Classes in Clean Rooms and Clean Zones", September 11, 1992, categorize clean rooms in six general classes, depending on the particle count (particles per cubic foot) and size in microns ( $\mu\text{m}$ ). The first three classes allow no particles exceeding 0.5 microns ( $\mu\text{m}$ ), and the last three allowing some particles up to 5.0 microns.

Clean Room Class	Class Limits "not to exceed" particles per cu ft for particle sizes shown				
	0.1 $\mu\text{m}$	0.2 $\mu\text{m}$	0.3 $\mu\text{m}$	0.5 $\mu\text{m}$	5 $\mu\text{m}$
<b>1</b>	<b>35.0</b>	<b>7.50</b>	<b>3.0</b>	<b>1.0</b>	<b>--</b>
<b>10</b>	<b>350</b>	<b>75.0</b>	<b>30.0</b>	<b>10.0</b>	<b>--</b>
<b>100</b>	<b>--</b>	<b>750</b>	<b>300</b>	<b>100</b>	<b>--</b>
<b>1000</b>	<b>--</b>	<b>--</b>	<b>--</b>	<b>1000</b>	<b>7.0</b>
<b>10000</b>	<b>--</b>	<b>--</b>	<b>--</b>	<b>10000</b>	<b>70.0</b>
<b>100000</b>	<b>--</b>	<b>--</b>	<b>--</b>	<b>100000</b>	<b>700</b>

Interpreting the table above, a class 100,000 clean room limits the concentration of airborne particles equal to or greater than 0.5 microns to 1 00,000 particles in a cubic foot of air.

The 4 important air-conditioning design considerations for clean room system design are:

1. Supplying airflow in sufficient volume and cleanliness to support the cleanliness rating of the room.
2. Introducing air in a manner to prevent stagnant areas where particles could accumulate.
3. Conditioning air to meet clean-room temperature, humidity and filtration requirements.
4. Ensuring enough conditioned makeup air to maintain the specified positive pressurization.

#### Important Regulatory and Guideline Information

- The Institute of Environmental Sciences (IES): Consideration for Clean room Design, IES - RP - CC012.1
- Testing Clean Rooms (IES-RP-CC-006-84-T), outlines performance tests procedures. IES-CC-011-85T for Glossary of terms and definitions related to contamination control.
- IES - RP - CC - 006: Testing Clean rooms
- IES - RP - CC007: Testing ULPA Filters
- Fed Std. 209E: Prepared by the Institute for Environmental Sciences, under the authority of the General Services Administration of the Federal Government offers specific guidelines in terms of non-viable particulate levels.

- Chapter 32 of ASHRAE Guide and data book on Systems and Application, 1997 provides information on Clean Spaces.
- ISO / TC 209: Clean room and Associated Controlled Environments
- JIS - B - 9920: Measuring Methods for Airborne Particles in Clean rooms and Evaluating Methods for Air Cleanliness of Clean rooms; Japanese Standards Association.
- NEBB, Procedural Standards for Certified Testing of Clean rooms (refer part III section 4 for details)

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### **NATURAL VENTILATION GUIDELINES**

Several general guidelines should be followed when designing for natural ventilation:

- 1) In hot, humid climates, maximize air velocities in the occupied zones for bodily cooling. In hot, arid climates, maximize air flow throughout the building for structural cooling, particularly at night when temperatures are low.
- 2) Take advantage of topography, landscaping, and surrounding buildings to redirect airflow and give maximum exposure to breezes. Use vegetation to funnel breezes and avoid wind dams that reduce the driving pressure differential around the building. Site objects should not obstruct inlet openings.
- 3) The stack effect requires vertical distances between openings to take advantage of the effect; the greater the vertical distance, the greater the ventilation.
- 4) Openings with areas much larger than calculated are sometimes desirable when anticipating increased occupancy or very hot weather.
- 5) Horizontal windows are generally better than square or vertical windows. They produce more airflow over a wider range of wind directions and are most beneficial in locations where prevailing wind patterns shift.
- 6) Window openings should be accessible to and operable by occupants. Vertical air shafts or open staircases can be used to increase and take advantage of stack effects. However, enclosed staircases intended for evacuation during a fire should not be used for ventilation.

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### **INSULATION (General)**

- 1) **Materials:**
  - Calcium Silicate Temperature Range: 0 – +1200°F
  - Fiberglass Temperature Range: -20 – +1000°F
  - Mineral Wool Temperature Range: +200—+1900°F
  - Urethane, Styrene, Beadboard Temperature Range: -350—+250°F
  - Cellular Glass Temperature Range: -450—+850°F
  - Ceramic Fiber Temperature Range: 0—+3000°F
  - Flexible Tubing and Sheets Temperature Range: -40—+250°F
- 2) Insulation, adhesives, mastics, sealants, and coverings shall have a flame spread rating of 25 or less and a smoke developed rating of 50 or less as determined by an independent testing laboratory in accordance with NFPA 255 and UL 728 as required by ASHRAE 90A and 90B. Coatings and adhesives applied in the field shall be non-flammable in the wet state.
- 3) Hangers on chilled water and other cold piping systems should be installed on the outside of the insulation to prevent hangers from sweating.
- 4) Insulation Thickness is given in the following table:

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**COLD PIPE INSULATION**

- 1) Where the piping fluid and ambient temperature differential is 40°F (4.4°C); insulate the following cold HVAC piping systems:
  - HVAC chilled water supply and return piping
  - HVAC make-up water piping
  - Air conditioner condensate drains piping
- 2) Insulate each cold piping system with one of the following types and thickness of insulation:
  - Fiberglass: 1" thick for pipe sizes up to and including 4" or 1-1/2" thick for pipe sizes over 4"
  - Cellular Glass: 1-1/2" thick for pipe sizes up to and including 4" or 2" thick for pipe sizes over 4"
  - Flexible Unicellular: 3/4" thick for pipe sizes up to 1-1/2" (largest size permitted)

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**HOT PIPING INSULATION (LOW PRESSURE)**

- 1) Insulate the following hot low pressure HVAC piping systems (steam piping up to 15 psi, water piping up to 250°F (121°C)).
  - HVAC hot water supply and return piping
  - Low pressure steam and condensate piping
  - Condenser water supply and return piping
  - Heated fuel piping
  - Hot gas refrigerant piping
- 2) Insulate each piping system specified above with one of the following types and thickness of insulation:
  - Fiberglass: 1" thick for pipe sizes up to and including 1" or 1-1/2" thick for pipe sizes 1-1/4" through 4", or 2" thick for pipe over 5"
  - Calcium Silicate or Cellular Glass: 1-1/2" thick for pipe sizes up to and including 1" or 2-1/2" thick for pipe sizes 1-1/4" through 4", or 3" thick for pipe sizes over 5"

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**HOT PIPING INSULATION (MEDIUM PRESSURE)**

- 1) Insulate the following hot medium pressure HVAC piping (steam piping from 16 to 75 psi, water piping 251 to 305°F (122 to 152°C)):
    - HVAC hot water supply and return piping
    - Medium pressure steam and condensate piping
  - 2) Insulate each piping system specified above with one of the following types and thickness of insulation:
    - Fiberglass: 1-1/2" thick for pipe sizes up to and including 1", or 2" thick for pipe sizes 1-1/4" through 2", or 2-1/2" thick for pipe sizes 2-1/2" through 4", or 3" thick for pipe sizes over 5".
    - Calcium Silicate or Cellular Glass: 2-1/2" thick for pipe sizes up to and including 1", or 3" thick for pipe size 1-1/4" through 2", or 4" thick for pipe sizes 2-1/2" through 4", or 4-1/2" thick for pipe sizes over 5".
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**HOT PIPING INSULATION (HIGH PRESSURE)**

- 1) Insulate the following hot high pressure HVAC piping (steam piping from 76 to 200 psi, water piping 306 to 450°(152 to 232°C):
  - High temperature hot water supply and return piping
  - High pressure steam and condensate piping
- 2) Insulate each piping system specified above with one of the following types and thickness of insulation:
  - **Fiberglass:** 1-1/2" thick for pipe sizes up to and including 1", or 2" thick for pipe sizes 1-1/4" through 2", or 2-1/2" thick for pipe sizes 2-1/2" through 4", or 3-1/2" thick for pipe sizes over 5". Provide Class 13 for superheated steam over 450 deg. F (232 deg. C).
  - **Calcium Silicate or Cellular Glass:** 2-1/2" thick for pipe sizes up to and including 1", or 3" thick for pipe size 1-1/4" through 2", or 4" thick for pipe sizes 2-1/2" through 4", or 5" thick for pipe sizes over 5".

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**COLD DUCTWORK INSULATION**

- 1) Insulate cold ductwork system with one of the following types and thickness of insulation:
  - **Rigid Fiberglass:** 1-1/2" thick, increase thickness to 2" in machine, fan and equipment rooms.
  - **Flexible Fiberglass:** 1-1/2" thick, application limited to concealed locations.
  - **Cellular Glass:** 2-1/2" thick.
  - **Flexible Unicellular:** 1" thick.
- 2) **Insulation Omitted:** Do not insulate fibrous glass ductwork, or lined ductwork.

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**HOT DUCTWORK**

Insulate hot ductwork system such as kitchen range and hood exhaust ductwork with one of the following types and thickness of insulation:

- **Rigid Fiberglass:** 2" thick.
- **Flexible Fiberglass:** 2" thick, application limited to concealed locations.
- **Calcium Silicate:** 3" thick. Use for range and hood exhaust ductwork, in addition to other applications where indicated.

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**COLD EQUIPMENT INSULATION**

- Refrigeration equipment, including chillers, tanks and pumps
- Drip pans under chilled equipment
- Cold water storage tanks
- Cold and chilled water pumps
- Pneumatic water tanks
- Roof drain bodies

Insulate each item of equipment specified above with one of the following types and thickness of insulation:

- **Fiberglass:** 2" thick for surfaces above 35°F (2°C), or 3" thick for surfaces 35°F (2°C) and lower
- **Cellular Glass:** 3" thick for surfaces above 35°F (2°C), or 4-1/2" thick for surfaces 35°F (2°C) and lower
- **Flexible Unicellular:** 1" thick

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### **HOT EQUIPMENT INSULATION**

- Boilers (not pre-insulated at factory).
- Hot water storage tanks
- Water heaters
- Heat exchangers
- Condensate receivers
- Hot water expansion tanks
- Hot water pumps
- Condensate pumps
- Fuel oil heaters
- Flash tanks

Insulate each item of equipment specified above with one of the following types and thickness of insulation:

- **Fiberglass:** 2" thick, except 3" thick for low-pressure boilers and steam-jacketed heat exchangers
- **Calcium Silicate:** 3" thick, except 4-1/2" thick for low-pressure boilers and steam-jacketed heat exchangers
- **Flexible Unicellular:** 1" thick. Do not use for equipment operating above 180° F (82°C)

Breaching between heating equipment outlet and stack or chimney connection, except for double wall or factory insulated breaching shall be insulated with one of the following types and thickness of insulation:

- **Fiberglass:** 3-1/2" thick
- **Calcium Silicate:** 5" thick

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### **BUILDING ENVELOPE INSULATION**

Weatherization measures--such as insulating, weather-stripping, and caulking--help seal and protect the building against the summer heat in addition to keeping out the winter cold. The attic is a good place to start insulating because it is a major source of heat gain. Table below shows the insulating values of common types of insulation used:

Type	Form	R-value per inch
Polyurethane	Expanding Foam	6
Expanded Polystyrene	Rigid board	4 - 5
Polyisocyanurate, faced	Rigid board	7

faced		
Fiberglass	Batts or Loose	3.5
Cellulose	Loose or wet blown	3.5
Mineral Wool	Loose	2.5
Rock Wool	Loose	2.5

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### **ROOF CURBS & SEALING AROUND ROOFTOP UNITS**

- 1) Top of curb 8" above roof - common to use 12" high roof curb
- 2) Three Types:
  - Straight Sides
  - Canted
  - Raised Cant
- 3) OD should be 2"-3" less than ID of equipment sitting on curb to allow for flashing.
- 4) Seal rooftop units and curbs according to the manufacturer's instructions. Correctly install manufacturer-supplied gasket during unit placement. This is often overlooked during installation.
- 5) Provide a solid connection and airtight seal between the ductwork and the unit or curb. Manufacturers often recommend a 3-inch flexible connector from the unit to the ductwork to minimize vibration transfer. This connector and connection should be air- and watertight.
- 6) Seal around pipe and conduit penetrations with sealant rated for outdoor use.

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### **SOUND ATTENUATION**

- 1) Sound power (generating from the fan) and sound pressure (received by the ear) are measured in dB (decibels): Ten times logarithm of the ratio of a given power to a reference power. In sound the current reference power is 10-12 watts.
- 2) When specifying sound criteria for HVAC equipment, refer to sound power level, not sound pressure level.
- 3) When comparing sound power levels, remember the lowest and highest octave bands are only accurate to about +/-4 dB.
- 4) When trying to calculate the additive effect of two sound sources, use the approximation as below (note that the logarithms cannot be added directly).
- 5) With two sound sources of equal dBs, the equivalent dB is the original plus 3 dB.  
(Ex. 60 dB + 60 dB = 63 dB)
- 6) With two sound sources of unequal dBs, if difference is 10 dB or more, use higher dB.  
(Ex. 70 dB + 60 dB = 70 dB)
- 7) Sones - a single number method to predict loudness levels, usually for small fans.
- 8) Lower frequencies are the most difficult to attenuate.
  - 2 x sound pressure (single source) = +3 dB(sound pressure level)
  - 2 x distance from sound source = -6dB (sound pressure level)

- +10 dB(sound pressure level)= 2 x original loudness perception
- 9) Centrifugal fan noise maximizes in 3rd octave band (250 Hz).
  - 10) Sound Attenuators Types
    - Rectangular: 3, 5, 7, and 10 foot lengths
    - Round: 2 or 3 times the diameter
  - 11) Sound Attenuator Location: Where necessary, select passive silencers with minimal flow restriction.
    - Centrifugal and Axial Fans Discharge: 1 duct diameter from discharge for every 1,000 FPM
    - Centrifugal and Axial Fans Intake: 0.75 duct diameters from intake for ever 1,000 FPM
    - Elbows: 3 duct diameters up and down stream
    - Terminal Boxes: 1 duct diameter down stream
    - Mechanical Equipment Rooms: Install in or close to mechanical equipment room wall opening
  - 12) The inside surface should be perforated to allow the sound to penetrate the absorbent material. Another simple but effective silencer can be made by forming 6-8 feet of flexible duct into a smooth "S" shape.
  - 13) Avoid the use of acoustical duct liners. The fibrous, coated or foam surfaces increase pressure drop by 50-250 percent. In addition, the fibers may enter the air stream and/or provide a medium for microbial growth.
  - 14) To control noise, size airside components for maximum velocities for controlling noise generation listed in Appendix 3 of ACCA Manual Q.5b Avoid locating air terminals in close proximity to air-handling units. Provide at least one change in direction between air terminals and air-handling units.

### **EQUIPMENT VIBRATION ISOLATION**

**System Natural Frequency:** The natural frequency of a system is the frequency at which the system prefers to vibrate. It can be calculated by the following equation:

$$F_n = 188 (1/d)^{1/2} \text{ (cycles per minute)}$$

The static deflection corresponding to this natural frequency can be calculated by the following equation:

$$d = (188/F_n)^2 \text{ (inches)}$$

By adding vibration isolation, the transmission of vibration can be minimized. A common rule of thumb for selection of vibration isolation is as follows:

Equipment RPM	Static Deflection of Isolation	
	Critical Installation	Non-critical Installation
1200+	1.0 in	0.5 in
600+	1.0 in	1.0 in
400+	2.0 in	1.0 in
300+	3.0 in	2.0 in

- Critical installations are upper floor or roof mounted equipment.
- Non-critical installations are grade level or basement floor.
- Always use total weight of equipment when selecting isolation.
- Always consider weight distribution of equipment in selection.



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### **VIBRATION ISOLATION -SEISMIC**

Earthquake-resistive design for HVAC equipment shall be provided where Seismic Zone Factor, "Z" value is 0.10 or greater, and for piping and ductwork where "Z" value is 0.20 or greater. Earthquake-resistive design shall comply with the requirements of the Uniform Building Code (UBC).

HVAC equipment, ductwork and piping shall be braced in accordance with the most current edition of Seismic Restraint Manual Guidelines for Mechanical Systems (SMACNA) and National Uniform Seismic Installation Guidelines (NUSIG). Seismic restraints are required for the following items when suspended by individual hangers greater than 12" in length from supporting structure:

- 1) **Ductwork:** Ducts greater than 6 sq. ft. rectangular or 28" round to be braced 30' transverse and 60' longitudinal.
- 2) **Piping:** Gas or fuel pipe 1" diameter and larger to be braced 20' transverse and 40' longitudinal.
- 3) **Piping:** In boiler and mechanical rooms with piping 1-1/4" ID and larger and in other areas with piping 2-1/2" ID or greater to be braced 40' transverse and 80' longitudinal.

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### **BEARINGS**

All rotating equipment use bearings. Bearing life is determined in accordance with methods prescribed in ISO 281/1-1989 or the Anti Friction Bearing Manufacturers Association (AFBMA) Standards 9 and 11, modified to follow the ISO standard. The life of a rolling element bearing is defined as the number of operating hours at a given load and speed the bearing is capable of enduring before the first signs of failure start to occur. Since seemingly identical bearings under identical operating conditions will fail at different times, life is specified in both hours and the statistical probability that a certain percentage of bearings can be expected to fail within that time period.

For example, a manufacturer specifies that the bearings supplied in a fan or compressor has a minimum life of L-10 in excess of 40,000 hours at maximum catalogued operating speed. We can interpret this specification to mean that a minimum of 90% of the bearings in this application can be expected to have a life of at least 40,000 hours or longer. To say it another way, we should expect less than 10% of the bearings in this application to fail within 40,000 hours.

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### **HEATING SYSTEMS**

Where the outdoor winter design temperature is 40°F and below, a thermostatically controlled, forced air, heating terminal device(s) shall be provided for the building entrances, vestibules, attic spaces, and exterior stairs. In addition, the vestibules shall be pressurized by supplying conditioned air at the rate of 2.0 CFM per square foot and allowing it to exfiltrate outside by not providing any ducted exhaust or return air connections. In general following rules of thumb may be noted for direct fired heating systems:

- 1) Heating Load = 24 BTUH / sq- ft for insulated building
- 2) Heating Load = 40 BTUH / sq-ft for non-insulated building
- 3) Combustion Air = 4 CFM outside air / 1,000 BTUH Input
- 4) 1 PSI (Gas Pressure) = 28 inch- w.g = 2.04 in- Hg (Mercury)

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### **HEATING EQUIPMENT CHARACTERISTICS**

**Furnace:** A type of space-heating equipment with an enclosed chamber where fuel is burned or electrical resistance is used to heat air directly without steam or hot water. The heated air is then distributed throughout a building, typically by air ducts.

There are various types of heat systems available, each having its own advantages and disadvantages. Table below provide the characteristics of various heating equipment:

Equipment Type	Typical Available Output Ranges	Advantages	Disadvantages
Gas Furnaces	40000 to 150000 Btu/hr	<ul style="list-style-type: none"><li>• Burner relative quite</li><li>• Warm comfortable heat delivered to spaces</li><li>• Clean burning , minimal service</li><li>• Available in up flow, down flow and horizontal configurations</li></ul>	<ul style="list-style-type: none"><li>• Use limited to availability of fuel</li><li>• Flue required</li><li>• Lower output unit (15000 to 30000 Btu/hr) compatible with actual requirements not readily available</li><li>• Seasonal efficiency reduced by over sizing</li></ul>
Oil Furnaces	72800 to 168000 Btu/hr (0.65 to 1.5 GPH nozzle size)	<ul style="list-style-type: none"><li>• Favorable alternative to electricity in colder climate</li><li>• Warm comfortable heat delivered to spaces</li><li>• Available in up flow, down flow and horizontal configurations</li></ul>	<ul style="list-style-type: none"><li>• Burner relatively noisy</li><li>• Flue and storage tank required</li><li>• Lower output unit (15000 to 30000 Btu/hr) compatible with actual requirements not readily available</li><li>• Seasonal efficiency reduced by over sizing</li><li>• Occasional service required to assure clean combustion</li></ul>
Electric Furnaces	17065 to 136520 Btu/hr (5 kW to 40 kW)	<ul style="list-style-type: none"><li>• Warm comfortable heat delivered to spaces</li><li>• No flue required</li><li>• Small space required</li><li>• Same unit adaptable to up flow, down flow and horizontal configurations</li><li>• Many sizes available in small increments (5 kW) allow close sizes to actual loads</li></ul>	<ul style="list-style-type: none"><li>• High operating costs in most areas</li></ul>

Equipment Type	Typical Available Output Ranges	Advantages	Disadvantages
		<ul style="list-style-type: none"> <li>Minimum service</li> </ul>	
Heat Pump (air to air)	18000 to 60000 Btu/hr (1 ½ to 5 tons)	<ul style="list-style-type: none"> <li>No flue required</li> <li>Small space required similar to electric furnace</li> <li>Same unit often adaptable to up flow, down flow and horizontal configurations</li> <li>Economical operation compared to electric resistance furnace</li> <li>Use same equipment for both heating and cooling</li> </ul>	<ul style="list-style-type: none"> <li>Low air temperature at registers</li> <li>Higher air volume requires more critical duct design</li> <li>Economical heating output limited to nominal tonnage of air-conditioner size</li> <li>Least efficient when coldest outdoors</li> <li>Less efficient for cooling than available high efficiency cooling only, condenser-coil combinations</li> <li>Servicing more sophisticated than furnace</li> </ul>
Heat Pump (water to air)	18000 to 60000 Btu/hr (1 ½ to 5 tons)	<ul style="list-style-type: none"> <li>No flue required</li> <li>Small space required similar to electric furnace</li> <li>Same unit often adaptable to up flow, down flow and horizontal configurations</li> <li>Extremely efficient in both heating and cooling modes</li> </ul>	<ul style="list-style-type: none"> <li>Requires dependable ground water source within economical operating temperature range</li> <li>Requires means of returning water to ground or other drainage</li> <li>Potential freeze up problems in cooler climates</li> <li>Installed costs higher dependent on the cost of obtaining ground water</li> </ul>
Coal & Wood Furnace	80000 to 200000 Btu/hr	<ul style="list-style-type: none"> <li>Provide abundance of warm heat utilizing economical fuels</li> <li>Ideal for retrofit in poorly insulated older homes</li> </ul>	<ul style="list-style-type: none"> <li>Economical use limited to availability of wood or coal</li> <li>Requires large space for coal, wood storage and larger furnace</li> <li>Requires frequent daily firing, coal handling and</li> </ul>

Equipment Type	Typical Available Output Ranges	Advantages	Disadvantages
		<ul style="list-style-type: none"> <li>Coal furnace can burn coal, wood and even trash</li> <li>Operation not dependent on electric supply, hence not affected by outages</li> </ul>	firing, coal handling and ash removal is messy and inconvenient <ul style="list-style-type: none"> <li>Difficult to accurately control temperature</li> <li>Tends to overheat space during moderate weather</li> </ul>

### **FUEL COMPARISON**

Degree day method is a simplest method for computing fuel requirements. It presupposes that the energy required maintaining comfort is a function of a single parameter: the outdoor dry-bulb temperature. One degree-day corresponds to a difference of 1° F between the outside mean temperature and 65°F (if the mean is less than 65°F; the balance point temperature). Monthly heating degree days are defined as the number of degrees the average monthly temperatures are below the balance point temperature multiplied by the number days in the month. Annual degree days are an accumulation of the monthly degree days. For example if the 24-hour mean temperature is 57°F on a given day, then the degree days are 65-57=8°F.

The degree-day concept has traditionally been used to determine the coldness of a climate. For example, Pittsburgh, Columbus, Ohio, and Denver, Colorado, have comparable annual degree days (about 6000 DD/year). It can be expected that the same structure in all these locations would have about the same heating bill. Move the building to Great Falls, MT (7800 DD/year), it would have a higher heating bill; but in Albuquerque, NM, (4400 DD/year), it would have a relatively lower heating cost.

From the above data, we can make an educated guess about the annual heat loss. To determine the annual heat loss, divide the energy loss rate by the design temperature difference and then multiply it by 24 hours per day and the number of annual degree days (from the weather files of the location). For example, a house with a design heating load of 30,000 Btu/hr in Pittsburgh (outdoor design temperature of 4°F) will use:

$$[30,000 \text{ Btu/hr} \times 24 \text{ hr/day} / (65-4) (\text{°F})] \times 6000 \text{ DD/yr} = 71 \text{ million Btu/yr}$$

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### **HOT WATER HEATING SYSTEM**

- 1) As far as possible, direct gas fired heating or hydronic hot water heating system shall be used to heat the space.
- 2) If the hot water is produced from the steam, the incoming steam pressure shall be reduced to 207 kPa (30 PSIG) by installing a suitable PRV station.
- 3) If the use of the building steam from the central boiler plant is not feasible due to elaborate and cumbersome steam/condensate piping distribution layout, or due to insufficient boiler capacity, dedicated hot water boilers should be used. The electric resistance heating shall not be used unless it is proven cost effective.
- 4) Hot Water Heating Selection Criteria

The following criteria shall apply for sizing and selecting hot water heating systems:

- **Pumping Capacity 3.7 kW (5.0 HP or less):** If the pumping capacity is equal to or less than 3.7 kW (5.0 HP), two converters and pumps of 100% capacity each shall be provided. One converter and pump shall act as standby.
- **Pumping Capacity Larger than 3.7 kW (5.0 HP):** If the pumping capacity is larger than 3.7 kW (5.0 HP), two converters and pumps of 50% capacity each shall be provided. No standby equipment is required.
- **System Configuration:** As far as possible, provide a common heating system for all heating terminal units and for the unit mounted preheat coils for which there is no danger of coil freeze up. With a common heating system for perimeter heating and interior reheat applications; care must be taken to ensure that the hot water temperature, reset schedule, and the flow rates can satisfy perimeter and interior duties with by selected parameters. Two independent heating systems, one for interior reheat and another for perimeter heating shall be considered only if such an arrangement is proved cost effective.
- **System Controls:** The maximum hot water temperature, to meet the peak heating demand, shall not exceed 80 °C (180 °F). The hot water temperature shall be on an inverse reset schedule with the outdoor temperature and the minimum hot water temperature shall not be less than 58 °C (140 °F).
- **Pump Control:** The pump(s) shall be allowed to ride on the pump curve during part load conditions. Care shall be taken to ensure that at least 15% water flow remains in circulation and water flows up to the end of each water branch (or a riser). With two pumps of 50% in operation, staging of the pumps shall be based on the actual flow and/or outdoor temperature.

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## **HEAT ELECTRIC**

- 1) Electricity is inherently less efficient than gas as a heating energy source because it must account for losses associated with generation from depletable fossil fuels and transmission to the building site.
- 2) Electric resistance baseboard heaters are assumed to have a line efficiency of 1.00 and an equivalent HSPF of 3.413.
- 3) Electric radiant heating panels may be assigned a slightly higher efficiency of 1.04 (equivalent HSPF of 3.550) owing to their mode of delivering sensible heat if the following criteria are met:
  - The radiant heating is installed on the interior surface of the building envelope.
  - The radiant surface is not likely to be behind furnishings (i.e. it is installed high on a wall or on a ceiling).
  - The radiant surface is not designed with fins or covers.
  - The radiant surface does not use a fan to deliver its heat.
- 4) Do not use electric resistance heating except where the existing heating system is inadequate to heat the addition, or where natural gas is not available within a reasonable distance or where it is unreasonable from an engineering or economic position to extend the existing heating system to the new area.
- 5) **Electric-Resistance Coils** for heating should be finned-tube construction with 80 percent nickel, 20 percent chromium elements, mounted in copper-plated steel tube and surrounded by magnesium-oxide powder; spirally wound with copper-plated steel fins brazed to tubes; mounted in aluminized-or galvanized steel frame.
- 6) **Control Panel:** NEMA 250, Type 1 enclosure, with thermal cutouts, primary and secondary controls, back-up contactors, circuit fusing, airflow switch, and fused control transformer. Include integral primary automatic and secondary manual reset thermal-protection devices and airflow switches.
- 7) Consider infrared radiant heating for high-bay areas or where spot heating is required.

- 1 KW = 1000 Watts
- 1 KW = 3,413 BTUH

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### **OIL BURNERS**

- 1) The selection of oil burners will depend on the grade of the oil being burned, the size of installation, and the need for modulating control.
- 2) For light oil, atomizing will be accomplished using oil pressure, air, or steam atomizing burners.
- 3) For heavy oil, atomizing will be accomplished using air or steam atomizing burners.

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### **GAS BOILERS**

- 1) Consider substituting High Efficiency Boilers (Gas Fueled with efficiency >85%) for conventional boilers.
- 2) Sizes are currently available up to 150,000 Btu/H. These are workable in temperate or cold climates for medium-sized office buildings with requirements for backups.

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### **FUEL- FIRED SPACE HEATING EQUIPMENT EFFICIENCY (AFUE)**

The AFUE or Annual Fuel Utilization Efficiency of a space heater is the "measure of the percentage of heat from the combustion of gas or oil which is transferred to the space being heated during a year, as determined using the applicable test method in the Appliance Efficiency Regulations."

AFUE is similar to the thermal efficiency in that it does not include energy consumption from standby wattage and fans.

The *Appliance Efficiency Regulations* require that AFUE of all new central furnaces manufactured on or after January 1, 1992 be at least 78 percent for equipment with output capacity less than 225,000 Btu/hr. Central furnaces with outputs greater than or equal to 225,000 Btu/hr are rated according to their Steady State (or Thermal) Efficiency. Other space heating equipment manufactured on or after January 1, 1994 has the following AFUE requirements:

Appliance	Capacity	AFUE
Furnace	Up to 300000 Btuh	78%
Gas Steam Boilers	Up to 300000 Btuh	75%
Other Boilers	Up to 300000 Btuh	80%

As with cooling equipment, higher efficiency equipment will satisfy space heating requirements using less fuel.

- 1) Typically, 81-83% AFUE furnaces carry little cost increase over those meeting federal minimum efficiency standards (currently 78%).
- 2) Higher efficiency gas furnaces or boilers (greater than 90% AFUE) are usually direct vent, sealed combustion units and, because the flue gases are cooler, PVC pipe can often be used for venting,

eliminating the need for a chimney. Through-the-wall venting may offset the higher cost of the equipment.

- 3) Keep in mind that sealed combustion equipment not only offers an energy efficiency benefit, but may be advantageous with respect to indoor environmental quality as well.
- 4) The sealed combustion chamber eliminates any possibility of back drafting or spillage of combustion gases into the home.
- 5) In many cases, it is advantageous to invest in high AFUE heating equipment in climates that have a significant heating load or high fuel costs. Energy efficiency investment dollars may be better spent elsewhere in cooling-dominated climates.

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## **HEAT PUMPS**

Heat Pump is a type of heating and/or cooling equipment that draws heat into a building from outside and, during the cooling season, ejects heat from the building to the outside.

Heat pumps are vapor-compression refrigeration systems whose indoor/outdoor coils are used reversibly as condensers or evaporators, depending on the need for heating or cooling.

Different categories of heat pumps include Single-Package, Split-System, Packaged Terminal Heat Pumps, and Water Loop Heat Pumps. The use of the closed loop, water cooled, incremental, heat pump system shall be evaluated for mild climate locations and for the buildings with relatively large interior area. The heat shall be the primary source of heating with steam to hot water, or electric resistance heat, as back up.

### **Heat Pump Efficiency (Cooling Mode)**

- 1) Heat pumps are often the cost-effective solution for space conditioning equipment in climates where outdoor temperatures are moderate and there is a need for both heating and cooling.
- 2) High efficiency equipment uses less energy to accomplish the same task. The intent of this guideline is to reduce the electrical energy necessary to cool your space.
  - SEER 11-12\* - (9%-17% improvement above SEER 10 air conditioner)
  - SEER 13-14 - (23%-29% improvement above SEER 10 air conditioner)
  - SEER 15-18 - (33% -44% improvement above SEER 10 air conditioner)
  - SEER 19+ - (47%+ improvement above SEER 10 air conditioner)
- 3) Select equipment that carries the ENERGY STAR label, or check manufacturer's literature for information on SEER. For cooling-dominated climates, it often makes sense to invest energy efficiency dollars in high SEER equipment. For climates with little cooling load, investments in efficiency may be better spent elsewhere.
- 4) As of January 2006, the federal minimum efficiency for air conditioning units will be SEER 13.

### **Heat Pump Efficiency (Heating Mode)**

- 1) To reduce the amount of nonrenewable energy used to meet the space heating requirements of a space, following guidelines need to be adopted.
  - 7.2-7.9 HSPF (6%-16% increase in efficiency)
  - 8.0-8.9 HSPF (18%-31% increase in efficiency)
  - 9.0-10.5 HSPF (32%-54%% increase in efficiency)
  - >10.5 HSPF (>54% increase in efficiency)
- 2) Select equipment that carries the ENERGY STAR label, or check manufacturer's literature for HSPF data.



- 3) The current federal minimum Heating Season Performance Factor (HSPF), the standard measure of heat pump efficiency in the heating mode, is 6.8.
- 4) Air-to-air heat pumps are not recommended if winter temperatures often drop below 35° F.
- 5) If heating is the predominant load, consider a heat pump with an HSPF of 8.0 or higher.

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### **HEATING & COOLING UNIT EFFICIENCY RATINGS**

- 1) Gas (fossil fuel) heating equipment is rated according to its Annual Fuel Utilization Efficiency (AFUE).
- 2) The heating efficiency of electric heat pumps with less than 65,000 Btuh cooling capacity is rated according to Heating Seasonal Performance Factor (HSPF).
- 3) The heating efficiency of heat pumps with cooling capacity of 65,000 Btuh or more is rated according to Coefficient of Performance (COP).
- 4) Electric resistance heating is rated according to its HSPF.
- 5) All electric cooling with less than 65,000 Btuh output capacities is rated according to the Seasonal Energy Efficiency Ratio (SEER).
- 6) Electric cooling with an output capacity of 65,000 Btuh or more is rated according to its Energy Efficiency Ratio (EER).
- 7) Heat pump cooling is rated according to its SEER or EER, and heat pump heating by the HSPF or COP.

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### **ELECTRICITY**

- 1) 1 HP (motor) = 0.746 KW (operating energy)
- 2)  $5 \text{ HP} \times 0.746 \text{ KW/HP} \times 3413 \text{ BTUH/KW} = 12,700 \text{ BTUH} = 1 \text{ Ton of Cooling}$
- 3) Watts = Volts x Amps
- 4) Efficiency =  $746 \times \text{Output Horsepower (HP)} / \text{Input Watts}$
- 5) KW (1 Phase) =  $\text{Volts} \times \text{Amps} \times \text{Power Factor} / 1000$
- 6) KW (3 Phase) =  $\text{Volts} \times \text{Amps} \times 1.732 \times \text{Power Factor} / 1000$
- 7) HP (1 Phase) =  $\text{Volts} \times \text{Amps} \times \text{Efficiency} \times \text{Power Factor} / 746$
- 8) HP (3 Phase) =  $\text{Volts} \times \text{Amps} \times 1.732 \times \text{Efficiency} \times \text{Power Factor} / 746$

### **MOTORS**

All motors shall have premium efficiency as per *ASHRAE 90.1*. 1/2 HP and larger shall be polyphase. Motors smaller than 1/2 HP shall be single phase. For motors operated with variable speed drives, provide insulation cooling characteristics as per NEC and NFPA .

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### **ARRANGEMENT OF MECHANICAL SPACES**

- 1) **Minimum Space Requirements:** A minimum of 4 percent of the typical floor's gross floor area shall be provided on each floor for air-handling equipment. A minimum of 1 percent of the building's gross area shall be provided for the central heating and cooling plant (location to be agreed upon during preparation of concept submission).

Space requirements of mechanical and electrical equipment rooms shall be based upon the layout of required equipment drawn to scale within each room.

- 2) **Service Access:** Space shall be provided around all HVAC system equipment as recommended by the manufacturer and in compliance with local code requirements for routine maintenance. Access doors or panels should be provided in ventilation equipment, ductwork and plenums as required for in-site inspection and cleaning.

Equipment access doors or panels should be readily operable and sized to allow full access. Large central equipment shall be situated to facilitate its replacement.

The HVAC design engineer should be cognizant of the necessity to provide for the replacement of major equipment over the life of the building and should insure that provisions are made to remove and replace, without damage to the structure, the largest and heaviest component that cannot be further broken down.

In addition, adequate methods of access shall be included for items such as: chillers, boilers, heat exchangers, cooling towers, reheat coils, VAV boxes, pumps, hot water heaters and all devices that have maintenance service requirements.

Access to major equipment and working platform surfaces for employees shall be in accordance with OSHA Regulations.

- 3) **Drawings:** The floor plans and at least two cross sections shall be drawn at 1:50 (1/4") scale to show clearances and access spaces. The designer shall not rely on the construction shop drawings to be assured of the adequate maintenance space. The tube pull space shall be clearly shown on the drawings. Overhead hoist beams shall be provided, as required, to facilitate the removal of the condenser and evaporator heads. Piping shall have the necessary offsets and mechanical couplings or flanges to permit removal of heads and tubes.
- 4) **Vertical Clearances:** Main mechanical equipment rooms generally shall have clear ceiling heights of not less than 3.6 m (12 feet). Catwalks shall be provided for all equipment that cannot be maintained from floor level. Where maintenance requires the lifting of heavy parts [45 kg (100 pounds) or more], hoists and hatchways shall be installed.
- 5) **Horizontal Clearances:** Mechanical rooms shall be configured with clear circulation aisles and adequate access to all equipment. The arrangement shall consider the future removal and replacement of all equipment.

The mechanical rooms shall have adequate doorways or areaways and staging areas to permit the replacement and removal of equipment without the need to demolish walls or relocate other equipment. Sufficient space areas (noted by outlining manufacturer's recommendations) for maintenance and removal of coils, filters, motors, and similar devices shall be provided. Chillers shall be placed to permit pulling of tubes from all units. The clearance shall equal the length of the tubes plus 600 mm (2 feet).

Air-handling units require a minimum clearance of 750 mm (2 feet 6 inches) on all sides, except the side where filters and coils are accessed. The clearance on that side should equal the length of the coils plus 600 mm (2 feet).

- 6) **Roof-Mounted Equipment:** No mechanical equipment except for cooling towers, air-cooled chillers, evaporative condensers, and exhaust fans shall be permitted on the roof of the building. Access to roof-mounted equipment shall be by stairs, not by ship's ladders.
- 7) **Housekeeping Pads:** Housekeeping pads shall be at least 150 mm (6 inches) wider on all sides than the equipment they support and shall be 150 mm (6 inches) thick.
- 8) **Mechanical Rooms:** Mechanical equipment rooms must be designed in accordance with the requirements of *ASHRAE Standard 15: Safety Code for Mechanical Refrigeration*.

All mechanical rooms must be mechanically ventilated to maintain room space conditions as indicated in *ASHRAE 62*, *ASHRAE 15*. Water lines shall not be located above motor control centers or disconnect switches and shall comply with requirements of *NEC*.

Mechanical rooms shall have floor drains in proximity to the equipment they serve to reduce water streaks or drain lines extending into aisles.

Every mechanical & electrical room shall have sufficient lighting and duplex convenience outlets to enable maintenance to plug in drop cord trouble lights, operate small tools, drills, etc. Lighting shall not be obstructed by duct-work and/or piping. Column and wall mounted lighting shall be included as necessary.

- 9) **Chiller Equipment Rooms:** All rooms for refrigerant units shall be constructed and equipped to comply with *ASHRAE Standard 15: Safety Code for Mechanical Refrigeration*. Chiller staging controls shall be capable of DDC communication to the central building Energy Management System.
- 10) **Electrical Equipment Rooms:** No water lines are permitted in electrical rooms, except for fire sprinkler piping. Sprinkler piping lines must not be located directly above any electrical equipment.
- 11) **Communications Closets:** Communications closets must be cooled in accordance with the requirements of EIA/TIA Standard 569. Closets which house critical communications components shall be provided with dedicated air-conditioning systems that shall be connected to the emergency power distribution system.
- 12) **Elevator Machine Rooms:** A dedicated heating and/or cooling system must be provided to maintain room mechanical conditions required by equipment specifications.
- 13) **Emergency Generator Rooms:** The environmental systems shall meet the requirements of NFPA Standard 110: *Emergency and Standby Power Systems* and meet the combustion air requirements of the equipment. Rooms must be ventilated sufficiently to remove heat gain from equipment operation. The air supply and exhaust shall be located so air does not short circuit. Generator exhaust should be carried up to roof level in a flue or exhausted by way of compliance with the generator manufacturer's installation guidelines. Horizontal exhaust through the building wall should be avoided.
- 14) **UPS Battery Rooms:** Battery rooms must be equipped with eye wash, emergency showers and floor drains. The battery room must be ventilated / exhausted directly to the outdoors at a rate calculated to be in compliance with code requirements and manufacturer's recommendations, and the exhaust system must be connected to the emergency power distribution system.

Fans shall be spark-resistant, explosion proof, with motor outside the air stream, ductwork to be negative pressure system of corrosion-resistant material, with exhaust directly to outdoors in a dedicated system. Acoustical enclosures shall be provided to maintain a maximum NC level of 35. Coordinate with electrical design specifications to include HVAC support equipment in UPS extended servicing agreements.

- 15) **24-Hour Spaces:** All areas designated as requiring 24-hour operations shall be provided with a dedicated and independent HVAC system. All spaces handling BAS computer processing of Fire Alarm Monitor and Control Systems, Security Monitor and Control Systems and/or energy monitoring and control systems shall be provided with dedicated HVAC systems to maintain temperature, humidity and ventilation requirements at all times.

Twenty-four hour systems shall have dedicated chiller(s), cooling tower(s) boiler(s), and associated pumping systems. However, central system(s) can be used to provide chilled water and hot water during the normal operating hours, or as a backup for the 24-hour system(s).

Twenty-four hour systems with a capacity of up to 50 tons should be configured with an air-cooled chiller. In the event the building's 24-hour operation load, including the dedicated perimeter ventilation system, exceeds 50 tons, the cooling systems may be combined with a central system of which a dedicated central chilled water supply loop shall be provided along with 24-hour chiller.

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### **EQUIPMENT SELECTION FOR RIGHT ACCESS**

Equipment shall be located to be accessible for installation, operation and repair. Mechanical spaces shall be of suitable size to permit inspection and access for maintenance, and to provide space for future equipment when required.

- 1) **Physical Attributes:**

- To a great extent space availability will dictate the type of chiller to be installed. The area and space requirements for chiller package must be examined. If space is tight, the rotary chiller might be the best solution.
- Decision for air-cooled v/s water-cooled chiller package should be evaluated on life cycle analysis. In addition to availability of quality water concerns, it should be noted that space requirement for the cooling tower is roughly 25% the area needed by the air-cooled system on large installation.
- If the space is adjacent to an area where noise is a major concern, reciprocating chillers are not the best option. Even the high-pitched whine of centrifugal compressor can be a serious problem. Rotary chiller is a best choice.

2) Outdoor Equipment Location:

- For ease of maintenance of the units as well as maintenance of the roof, air cooled condensers, cooling towers and allied equipment should be mounted at ground level on concrete slabs appropriately screened with attractive fencing or plantings. In the event that this is impractical and for justifiable reasons, units must be roof-mounted, provide architectural screening for the equipment and provide service walkways on the roof to and surrounding the units.
- Provide a permanent means of access to the units from the interior of the building without use of portable ladders.
- Locate units away from sources of debris such as leaves or grease, which can inhibit unit performance and increase maintenance costs.
- Ensure that there is adequate drainage at the equipment location to avoid water pooling.
- Do not locate equipment where it is vulnerable to falling or drifting snow or ice. Provide a rack or platform to elevate and support equipment per manufacturer's instructions when recommended by the manufacturer for protection from snow.
- If possible, locate split-system outdoor units in shaded areas or provide screening to shade the unit from solar heat gain. Avoid excessive distance between split-system condenser and evaporator components.
- Level units to avoid water accumulation in the drain pans of rooftop units.

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### **MECHANICAL INSTALLATION**

- 1) Terminal boxes shall be located so that space and access is provided for service and filter change. If fan powered VAC boxes are to be used in the ceiling, they should be mounted above entrance doors and access panel at ceiling height.
- 2) Fan coil units should not be installed above ceiling, and concealed fan coil units in general shall be avoided. Access shall be provided for service of the unit and for filter changes. If ceiling mounting is the only alternative, provide recess mounted units. Floor mounted fan coil units shall have sloped tops to eliminate books, papers, flowers, etc., being placed on top of units.
- 3) Conceal all pipe installations in walls, pipe chases, utility spaces, above ceilings, below grade or floors, unless indicated otherwise.
- 4) Install piping free of sags or bends and with ample space between piping to permit proper insulation applications.
- 5) Install exposed piping at right angles or parallel to building walls. Diagonal runs are not permitted, unless expressly indicated on the Drawings.
- 6) Install piping tight to slabs, beams, joists, columns, walls, and other permanent elements of the building. Provide space to permit insulation applications, with 1" clearance outside the insulation. Allow sufficient space above removable ceiling panels to allow for panel removal.

- 7) Locate groups of pipes parallel to each other, spaced to permit applying full insulation and servicing of valves.
- 8) Install unions adjacent to each valve and at the final connection to each piece of equipment and plumbing fixture having 2" and smaller connections, and elsewhere as indicated.

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### **HVAC CONTROLS**

Another aspect of HVAC efficiency involves controlling the operation of equipment. It starts with adequate zoning of the building to allow closer control of space conditions and to reduce the need to supply conditioned air to all of the building when only portions are occupied. The modern HVAC system today uses direct digital control (DDC) systems use sensors, electric actuators, and microprocessors which provide a marked upgrade in system functionality over that attainable with pneumatic control.

- 1) DDC systems provide operators the ability to remotely monitor existing conditions, change set points, diagnose, and sometimes fix problems from a workstation or laptop PC.
- 2) A single operator could monitor many buildings with DDC control, in addition to performing other operating and maintenance (O & M) tasks.
- 3) Energy management and reporting functions improve operational efficiency and cost savings. Operational problems can be identified early and fixed before they become larger and more expensive.
- 4) DDC systems involve vendor-specific application software and a communications network. Each software application provides varying degrees of flexibility and a unique software approach to DDC. The systems are often incompatible.
- 5) System Controls: The chillers shall be equipped with the factory furnished safety controls and flow switches which shall be interfaced with the operating controls to meet the intent of the system design. The operating controls shall be selected by the designer to suit the system capacity and configuration. The following guidelines are offered:
  - Single Chiller Installation: With a single chiller, the capacity control shall be generally based on monitoring the return chilled water temperature and/or maintaining variable fixed leaving chilled water temperature.
  - Two Chillers Installation: With two chillers, primary/secondary pumping arrangement shall be used with variable speed drives for the secondary pumps, if the pumps are 5.5 kW (7.5 HP) and larger. For smaller secondary pumps, the capacity reduction of the chillers shall be affected without the use of the variable speed drives for the secondary pumps.
- 6) Smoke management systems: When HVAC control systems interface with fire systems, signals from the fire system are used to take action to minimize the fire and/or provide safe shut down of equipment. Automatic and manual control can be used in HVAC control systems for fire safety and smoke management. Air distribution systems have manually operated devices that stop the operation of supply, return and/or exhaust air, and fans in an emergency. Automatic shutdown capability automatically, according to a predetermined fire management plan, closes dampers and shuts down fans when detectors located in the supply ducts, return ducts, or the building fire protection system detect a fire.
- 7) Controls should be installed for energy-efficient HVAC system operation in accordance with ASHRAE 90.1-1999. This includes:
  - Programmable Thermostatic Controls
  - Off-Hour Controls
  - Heat Pump Auxiliary Heat
  - Humidification and Dehumidification

- Freeze Protection
- Ventilation Controls for High-Occupancy Areas
- Economizers
- Simultaneous Heating and Cooling Limitation
- Fan Power Limitation
- Variable Air Volume (VAV) Fan Control
- Condenser Fan Speed Control

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### **HVAC BALANCING**

Testing and balancing of air and water systems shall be performed in complete accordance with AABC National Standards for Total System Balance, latest edition, as published by the Associated Air Balance Council. A representative of Physical Plant and Designer shall be present during testing.

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### **EQUIPMENT IDENTIFICATION-COLOR CODES**

Piping systems in mechanical rooms should be completely painted with the applicable colors listed below and have appropriate self-sticking or strap-on identifications and arrows indicating direction of flow. Piping and ducts in chases above ceiling, etc. should be color banded and have stencil markings at appropriate intervals. On straight runs of piping, markings should be no further than 30 feet apart; and stencil identifications, color bands, and direction arrows should be near each valve, pressure reducing valve, heat exchanger, etc. Where pipe passes through walls or floors, marking should be near the penetration on both sides. Markings should be at each directional change of all piping systems. Mechanical room pipe color and the color of bands are to be as follows:

The following color coding is stated in terms of OSHA Safety Color Designations:

- 1) Condensate Return (COND) - Corrugate Brown
- 2) Steam - High Pressure (25-75 psi) – (HPS#\_\_) - Safety Orange
- 3) Steam - Low Pressure (0-25 psi) – (LPS#\_\_) - Safety Orange
- 4) Chilled Water Supply (CWS) - Dark Blue
- 5) Chilled Water Return (CWR) - Dark Blue
- 6) Domestic Cold Water (DOM CW) – Safety Green
- 7) Distilled Water (DSTW) - Safety Green
- 8) Condenser Water to Cooling Tower (CTW) - Light Gray
- 9) Cooling Water to Condensers (CCW) - Light Gray
- 10) Domestic Hot Water (DHW) - Dark Yellow (Gold)
- 11) Domestic Hot Water Recirculation (DHWR) - Dark Yellow (Gold)
- 12) Hot Water Supply (HWS) - Safety Orange
- 13) Gas (GAS) - Safety Yellow
- 14) Air (AIR)- Safety Green
- 15) Nitrogen (Nitrogen) - Safety Yellow
- 16) Helium (Helium) - Safety Yellow
- 17) Hydrogen (Hydrogen) - Safety Yellow



- 18) Vacuum (VAC)- Safety Green
- 19) Chemical (labeled)- Safety Blue
- 20) Fire Protection (labeled) - Safety Red
- 21) Sprinkler (SPKR) - Safety Red
- 22) Fire Line (FIRE) - Safety Red
- 23) Drains, Vents,- Supports, Hangers - Black

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## HVAC CONTRACT DRAWINGS

The contract drawings shall include those listed below. For uniformity, drawings shall be arranged in the order listed.

- 1) General Notes, Abbreviations and Symbols
- 2) Standard Equipment Schedules; include schedules for existing air handling units, fans, pumps, etc., that will require alteration or rebalancing
- 3) Standard Details and other necessary details
- 4) Flow Diagrams for Air Supply, Return and Exhaust for each HVAC system
- 5) Temperature Control Diagrams and Sequence of Operation for all HVAC Systems, including "Sequence of Operation" written on the drawings alongside the control diagrams
- 6) Flow and Control Diagrams for Chilled Water and Hot Water Systems
- 7) Flow diagrams shall show entire system on a single drawing
- 8) Flow and Control Diagrams for Refrigeration Systems
- 9) Flow and Control Diagrams for Steam and Condensate Piping Systems
- 10) Riser Diagrams for chilled water, hot water, drain, steam and condensate and supply air, return air and exhaust air systems where applicable. Required flow diagrams may eliminate the need for riser diagrams
- 11) Demolition of existing HVAC works, if applicable. Minor demolition may be shown on new construction drawings. Extensive demolition requires drawings for demolition only
- 12) Floor Plans 1:100 (1/8" = 1'-0") for Equipment, Piping and Ductwork
- 13) Floor Plans and Sections 1:50 (1/4" = 1'-0") for Mechanical Rooms
- 14) Floor Plans 1:50 (1/4" = 1'-0") for Mechanical Chases at each floor showing ducts, dampers, piping and plumbing
- 15) Sections shall be shown, as required, to clarify installation, especially thru areas of possible conflict. Show all the equipment including plumbing and electrical
- 16) Room numbers and names shall be shown on HVAC plans at every review stage including schematic submissions. Where there is insufficient room on HVAC floor plans to show room names, room numbers only may be shown on the floor plan with the room numbers and names tabulated on the drawing.

### Ductwork Drawings

- 1) All ductwork, without any exceptions, shall be shown in double line. The minimum duct size shall be 200 mm x 150 mm (8 inch x 6 inch).
- 2) Complete flow diagrams of the supply return and exhaust air systems shall be shown on the drawings. Flow diagrams shall show the Cu M/Min (CFM) required in all mains and major branches (such as zone/floor) and the size of each main and major branch. Flow diagrams shall show and



identify all air handling units, fans, and other major components in the air system. These diagrams are to facilitate checking and air balancing.

- 3) Manual air volume balancing devices shall be provided in supply return and exhaust mains, branch mains and terminal branches. Ceiling access panels are to be installed, where required, for access to balancing devices. Location of balancing devices shall be shown on the contract drawings.
- 4) Dampers in room diffusers and registers shall be used only for minor balancing requiring a maximum pressure drop of approximately 25 Pa (0.10 inch of water gage). Registers and/or diffusers shall not be located on main ducts or main branches. They shall be located on individual branch ducts with opposed blade balancing dampers in the branch to reduce room noise transmission.
- 5) Air quantities on plans shall be "rounded off" to the nearest increment of 0.30 Cu M/Min (10 CFM).
- 6) Smoke detectors for air conditioning systems are specified in the Electrical Specifications, but the locations at air handling units shall be shown on the "H" drawing control diagrams and floor plans. Coordinate diffuser location and blow direction with space detector locations shown on the Electrical Drawings to avoid false smoke alarms caused by air discharge.
- 7) Provide fire dampers and smoke dampers in accordance with requirements of relevant codes and design criteria. Provide a schedule for smoke dampers showing sizes, pressure drops, and compliance with the maximum velocity limit. Show duct transitions on drawings.

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## SAFETY REQUIREMENTS

Safety requirements shall be followed in accordance with newest ANSI/ASHRAE Standard 15-2001. ASHRAE 15 addresses general requirements, refrigerant storage restrictions, design and construction procedures, conformance to National electrical code, periodic testing of safety equipment for a refrigerating machinery room. The focus of this design alert is on safety of personnel operating and maintaining the equipment. Few of the key revised requirements are:

- 1) Refrigerant leak detector to activate alarms and mechanical ventilation system
  - 2) The alarms shall be visual and audible inside the machinery room and outside each entrance to the machinery room. The alarms shall be of manual reset type with the reset located inside the machinery room.
  - 3) Machinery room shall be vented to the outdoors, utilizing mechanical ventilation. The air supply and exhaust ducts for the machinery room shall serve no other area.
  - 4) No flame producing device or continuously hot surface over 800°F is permitted as a permanent installation in the machinery room.
  - 5) Clear headroom shall not be less than 7.25 ft. below equipment situated over passageway.
  - 6) The machinery room shall have a door that opens directly to the outside or through a vestibule equipped with self-closing tight fitting doors.
  - 7) Walls, floor, and ceiling separating the machinery room from other occupied areas of the building shall be of at least one-hour rating.
  - 8) Exterior openings, if present, shall not be under any fire escape or any open stairway.
  - 9) Detectors, alarms and mechanical ventilating systems shall be periodically tested in accordance with manufacturer's specifications and the requirements of the local authorities.
  - 10) Emergency shutdown procedures, including precautions to be observed in case of a breakdown or leak, shall be displayed on a conspicuous card located as near as possible to the refrigerant compressor.
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