James B. Bradford et al. “HVAC Equipment and Systems”
Handbook of Heating, Ventilation, and Air Conditioning
Ed. Jan F. Kreider
Boca Raton, CRC Press LLC. 2001

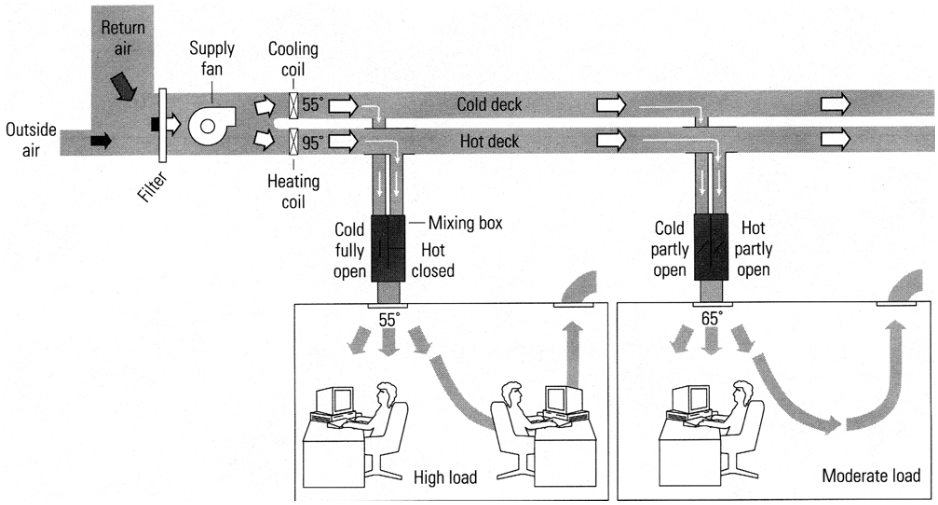


FIGURE 4.3.3 Dual duct system schematic (courtesy of E-source, Boulder, CO).

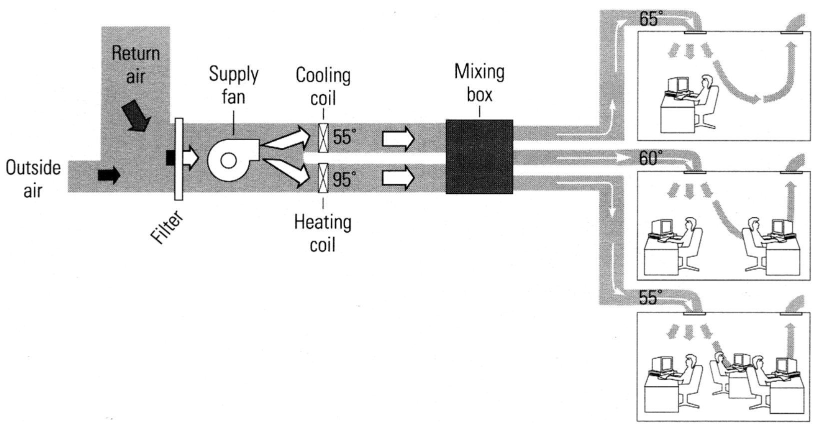


FIGURE 4.3.4 Multizone system schematic (courtesy of E-source, Boulder, CO).

thermostatically controlled zone-terminal reheat occurs to raise the temperature of the cooled air before it enters the zone. Figure 4.3.5 presents an example of a CAV system with a constant supply-air temperature serving several zones. In the figure, the zone with lower internal gains and smaller cooling load requires reheat at the zone terminal box.

VAV systems respond to changing cooling loads by modulating the zone air flow rate instead of the zone air temperature. The flow control of VAV systems is based on maintaining a constant pressure at some point in the main supply air duct. As zone terminal box dampers open and close, the duct pressure changes. The fan flow modulates to maintain the pressure setpoint. Flow variation is achieved by adjusting fan inlet dampers, fan outlet dampers, or the fan motor rpm. Direct fan motor control results in the lowest fan energy use in VAV systems. Figure 4.3.6 presents an example of a variable-air volume system responding to changes in cooling load. In the schematic, the two zones receive air at the same temperature but one has a reduced flow compared to design.

By reducing or eliminating the need for reheat, VAV systems use significantly less energy than CAV systems in meeting the same building loads. VAV systems also tend to be more expensive and more complicated to operate and maintain. Nevertheless, conversion of CAV systems to VAV is a popular energy conservation measure. Energy savings are achieved not only through reductions in fan power but also from reductions in cooling coil and reheat coil loads.

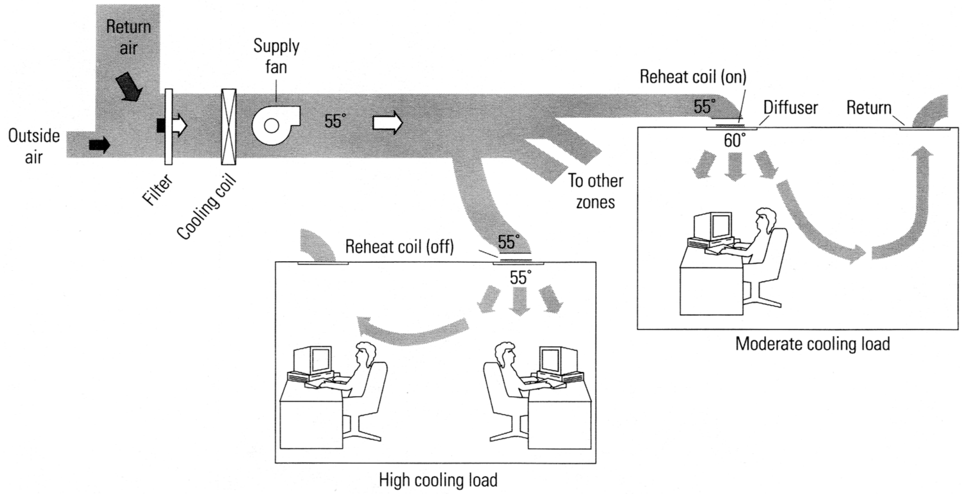


FIGURE 4.3.5 Single duct, multiple zone, constant air volume (CAV) system with reheat (courtesy of E-source, Boulder, CO).

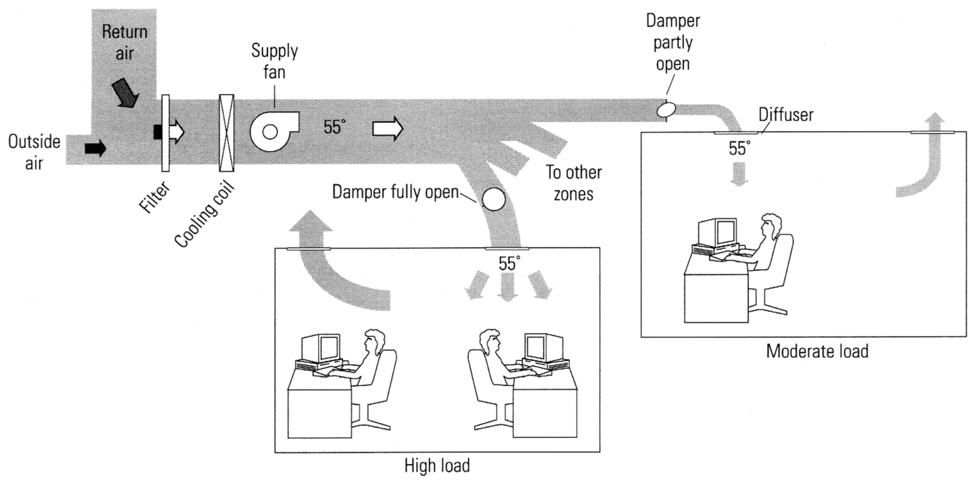


FIGURE 4.3.6 Single duct, multiple zone, variable air volume (VAV) system (courtesy of E-source, Boulder, CO).

System Design Considerations

A properly designed system meets zone loads under both design and off-design conditions. This requires calculation of maximum zone loads and an understanding of the flexibility that various components offer to meet loads in off-design conditions. Methods outlining the calculation of design zone loads are outlined in Chapter 6.1. In addition to zone loads, there are system dependent factors that influence coil loads and equipment size, including

- Supply fan heat gain
- Duct heat transfer
- Duct air leakage
- Component leakage

Details about these topics are described in other sections of this chapter.

Considering the second law of thermodynamics can provide insight for improving distribution system performance through design and operational changes. Second law guidance aids in the detection and avoidance of unnecessary depletion of useful work in a process. Guidelines particularly relevant to HVAC distribution systems include

- Minimize the mixing of streams with different temperatures and pressures.
- Do not discard heat at high temperatures to ambient or to cooling water.
- Do not heat refrigerated streams with hot streams.
- Heat (or refrigeration) is more valuable, the further the temperature is from ambient.
- The larger the mass flow, the larger the opportunity to save (or waste) energy.

Most of these considerations are common sense. Yet, as noted in the description of the CAV system with reheat, they are not adhered to. While there are many criteria other than thermodynamic performance for developing an acceptable design, any opportunities for reducing the depletion of useful work should be recognized when making design decisions.

Air Filtration

Air filtration is used in air handling systems to remove unwanted particulates, smoke, or gases from the air stream. Methods for smoke and gas removal from an air stream are normally reserved for industrial and special processes and are not usually necessary for typical HVAC applications.

There are two main types of particulate air filters used in modern HVAC air systems:

- *Viscous impingement filters* use filter media coated with a viscous substance, such as oil, which acts as an adhesive that catches particles in the flow stream.
- *Dry-type extended surface filters* consist of fibrous bats or blankets of varying thicknesses and density. The bats may be made of various materials such as bonded glass fiber, cellulose, wool felt, or synthetic materials.

Filters are often arranged in a pleated or v-configuration which extends the filter surface and reduces the pressure drop across the media. A common practice is to use inexpensive prefilters upstream of more effective filters to reduce premature loading. This practice extends the filter life and reduces the replacement frequency of the more effective, more expensive final filters.

Filter Testing and Rating

ASHRAE Standard 52.1 (ASHRAE 1992b) provides test procedures for testing and rating filtration devices. Of the many methods used for the testing and rating of air filtration devices, three of the most commonly cited methods include the *arrestance test*, the *dust-spot efficiency test*, and the *DOP penetration test*.

The arrestance test is conducted in a controlled laboratory setting. It consists of releasing a known quantity of material known as *ASHRAE test dust* into the filter to be rated. ASHRAE test dust is comprised of 72% standardized air cleaner test dust, 23% fine powdered carbon, and 5% cotton linters. The percent arrestance, which is the measure of the filter effectiveness, is calculated as follows:

$$\text{Arrestance} = 100 \cdot \left[1 - \frac{\textit{weight gain of filter}}{\textit{weight of dust fed}}\right]$$

Airborne particulates can result in soiled interior building surfaces. The discoloration rate of white filter paper simulates this effect. The dust spot efficiency test uses this approach to measure the effectiveness of the filter in reducing soiling of surfaces. The test measures the changes in light transmitted across the filter to evaluate its effectiveness. Efficiency is calculated from the following equation:

$$DS = 100 \cdot \left[1 - \frac{Q_1 \cdot (T_{20} - T_{21}) \cdot T_{10}}{Q_2 \cdot (T_{10} - T_{11}) \cdot T_{20}}\right]$$

where,

DS = percentage dust spot efficiency
Q_1 = total quantity of air drawn through an upstream target
Q_2 = total quantity of air drawn through a downstream target
T_{10} = initial light transmission of upstream target
T_{11} = final light transmission of upstream target
T_{20} = initial light transmission of downstream target
T_{21} = final light transmission of downstream target

The di-octyl phthalate (DOP) test is reserved for very high efficiency filters, typical of those used in clean rooms. In the test, a smoke cloud of DOP, an oily, high boiling point liquid, is fed into the filter. To determine efficiency, the concentrations of the DOP upstream and downstream of the filter are compared. The percent efficiency, DP, is calculated from the following equation:

$$DP = 100 \cdot \left[1 - \left[\frac{downstream\ concentration}{upstream\ concentration}\right]\right]$$

Application and Performance of Filtration Systems

The required level of filtration varies with the application. The amount of filtration for use in an industrial application may require only the removal of large particles, while the filtration requirement in applications such as clean rooms may be extremely rigorous. Table 4.3.1 outlines the type and effectiveness of filters that should be used for various applications. Note in the table that *A* designates arrestance, *DS*, dust spot efficiency, and *DP*, DOP efficiency percent.

Several types of filters are shown in Figure 4.3.7. From left to right, they include two pleated, disposable filters, a HEPA filter, and a bag filter. The pleated filters are least expensive and least efficient. The HEPA filter is most expensive and most efficient. Pleated filters are appropriate for general HVAC applications. Bag filters are appropriate for most hospital spaces. HEPA filters are appropriate for clean rooms and other aseptic applications.

Filters create air pressure drop in a system. Therefore, they affect air flow and/or fan power draw (kW). Manufacturers provide data regarding the pressure drop associated with both clean filters and dirty filters. When selecting a fan for a particular air handling unit, it is necessary to account for the pressure drop associated with a *fully loaded* (dirty) filter that is nearing the end of its service life or cleaning cycle. Table 4.3.2 gives typical pressure drop values for various filters.

Humidification and Adiabatic Cooling

In some applications, it is desirable or useful to supply humidification to a conditioned space. Some reasons for adding humidification capabilities to the air handling system include

- Space humidity control, where the primary goal is to maintain a humidity setpoint
- Air sensible cooling through adiabatic cooling
- Air cleaning

Air Humidification

Control of space humidity can be important for maintenance of high indoor air quality. Both high and low relative humidity levels can promote the growth of fungus and microbiological organisms. Special process rooms or industrial spaces sometimes require control of humidity to some particular setpoint, which can require humidification control.

Devices used to introduce humidity into a supply air stream include

- *Direct steam injection*, where steam is injected into the supply air stream
- *Pan humidifiers* , where water is evaporated into the air via a heated pan installed in the supply ducting
- *Wetted element*s, where water is applied to an open-textured media in the supply air stream
- *Atomizing devices*, where water is broken into a fine mist via atomizers. One of the more common types of atomizers are nozzles.

TABLE 4.3.1 Performance of Filtration Systems

Application	Prefilter	Filter	Final Filter	Remarks
Warehouses, mechanical rooms		50–80% A 25–30% DS	None	Large particles only; provides coil protection
General offices and laboratories	None	75–90% A 35–60% DS	None	Average housecleaning; provides pollen and some smudge reduction
Conference rooms, cleaner office spaces, specialty rooms	75–80% A 25–40% DS	>98% A 80–85% DS	None	Good housecleaning; no dust settling, significant smudge reduction
Hospitals, R&D, "gray" clean rooms	75–85% A 25–40% DS	>98% A 80–85% DS	95% DP or elect.	High bacteria reduction, effective smudge reduction
Aseptic areas, clean rooms	75–85% A 25–40% DS	>98% A 80–85% DS	99.97% DP (HEPA)	Protection against bacteria, radioactive dust; very clean room

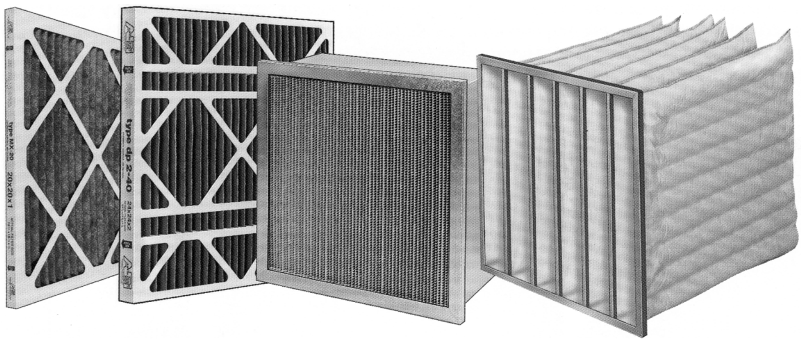


FIGURE 4.3.7 Pleated, HEPA, and bag air filters (courtesy of Airguard Industries).

TABLE 4.3.2 Filter Pressure Drop

Filter Type	Average Efficiency	Rated Face Velocity (fpm)	Clean Filter Pressure Drop (inches water)	Dirty Filter Pressure Drop (inches water)
Flat	85% A	500	0.10–0.20	1.00
Pleated	90% A	500	0.15–0.40	1.00
Bag	90% A	625	0.25–0.40	1.00
HEPA	99.97% DP	250–500	0.65–1.35	2.00

Humidifiers increase the moisture content (thus, latent heat) in the air stream while the stream enthalpy remains essentially constant. The humidification may have a sensible cooling or heating effect on the supply air stream. The change in state of the supply air is dependent on the supply air temperature and humidity, and on the state of the water absorbed by the air.

Evaporative Cooling

A common, but underutilized, application of air humidification is sensible air cooling through evaporative cooling. In an evaporative cooler, water is introduced into the supply air via wetted elements or atomizers (also called "air washers"). The water that is not evaporated by the air is captured in a sump and recirculated through the media or atomizers. Figure 4.3.8 shows a very common evaporative cooler arrangement.

The evaporative, or *adiabatic*, cooling that occurs in an evaporative cooler is the process of evaporating water into the air, thereby causing a sensible heat loss (cooling) of an air stream equal to the air's latent heat gain. Since the sensible loss is equal to the latent gain of the air stream, the process is adiabatic, thus the term *adiabatic cooling*. Since there is very little heat loss or gain in the supply air stream, it maintains a constant enthalpy as the air moves through the cooler.

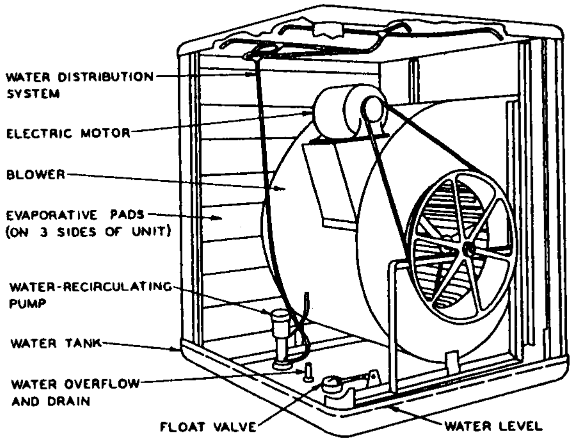


FIGURE 4.3.8 Typical evaporative cooler.

4.3.2 Coils

Coils are a special class of heat exchangers designed to transfer heat to or from an air stream. Coils are used in HVAC systems to provide air heating, preheating, reheating, cooling, and dehumidification. Coils come in various configurations to accommodate the needs of a design engineer in meeting the requirements of particular applications.

Types and Configuration

All HVAC air conditioning coils share the common feature that air is the working fluid on one side of the heat exchanger. The working fluid inside the coil, the primary fluid, may be one of the following.

Liquid — For coils with liquid as the working fluid, heat is transferred from or to the liquid when heating or cooling the air. The liquid working fluid may be water or a water/glycol mixture. Liquid coils are used for heating and cooling applications.

Refrigerant or DX — Direct expansion (DX) coils use refrigerant as the working fluid. They are applicable only for cooling applications. They are named DX systems because the refrigerant is directly expanded as it changes phase from liquid to gas and cools the air stream.

Steam — Steam coils are heating coils that use steam as the working fluid. In a phase change process that is the reverse of a DX coil, steam is condensed in the coil to heat the air stream. The latent heat of vaporization is released as the steam changes phases from gas to liquid.

Combustion gas — In heating coils, the working fluid may be high temperature, fossil fuel combustion gases. Devices that use combustion gases as the working fluid are commonly known as furnaces.

HVAC coils that use liquid, refrigerant, or steam as the working fluid share many components. A cutaway view of a typical coil is shown in Figure 4.3.9. Coil components, as shown in the figure, include fins, tubes, and headers.

The coil *fins* are extended surfaces that increase heat transfer area and improve heat transfer characteristics on the air side of the coil. Fins are usually constructed of aluminum; stainless steel is used for corrosion protection. Other fin materials may be used for special heat transfer properties. The fins are usually press-fitted or brazed onto the primary coil heat transfer surface to assure good contact and high heat transfer rates. Coils are generally specified with a particular number of fins per inch. The number of fins per inch in HVAC applications usually ranges between 6 and 12.

Coil *tubes* carry the working fluid (refrigerant, steam, water, etc.) to or from which heat is transferred to provide the desired air conditioning effect. Tubes are often constructed of copper but may be made of other materials depending on the application. The tubes can be manufactured in various configurations that affect coil heat transfer and pressure drop characteristics. Tube coils can be configured in (1) one

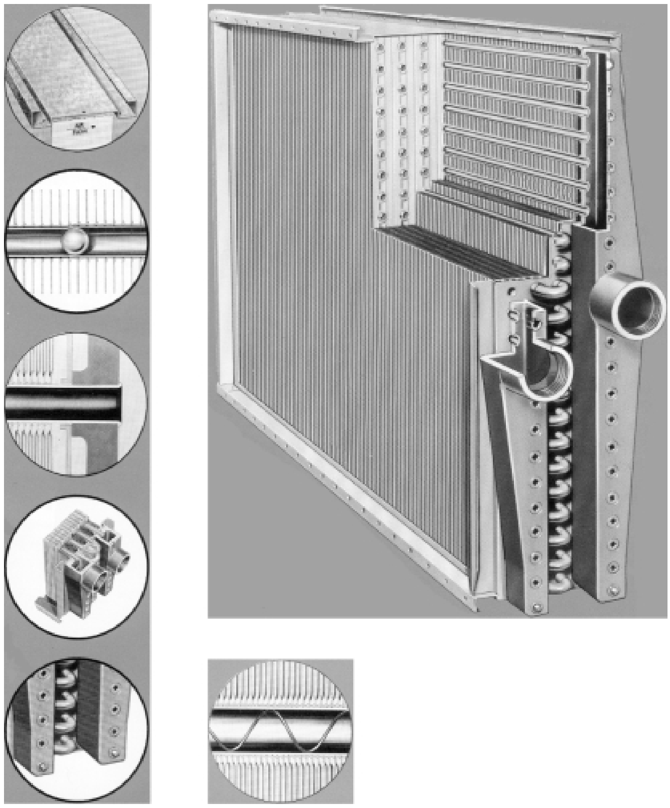


FIGURE 4.3.9 Typical coil configuration (courtesy of the Trane Co.).

or multiple circuits and (2) one or multiple rows. Circuits refer to the number of parallel paths in the coil through which the primary fluid passes. Rows run perpendicular to the depth of the coil. The number of rows equals the number of times a circuit crosses the air stream. In Figure 4.3.9, the primary fluid flows into 22 tubes and each crosses the coil 4 times. Thus, this coil has 22 circuits and 4 rows. However, determining a coil's configuration from external observations is not always reliable.

Coil *headers* distribute the primary fluid into the tube circuits, and they also serve to collect the fluid at the end of the circuits. In Figure 4.3.9, the inlet and outlet coil headers are apparent in the front of the image. Coils are sometimes arranged with face and bypass dampers, which modulate to direct the air across or around the heat transfer surfaces. The dampers control the amount of conditioning the coil imparts on the air stream. Miscellaneous appurtenances for coils also include drain pans for condensate collection and removal, valves for shutoff and control, air removal devices, and flexible piping connectors.

Coil Performance and Selection

Coil heat transfer characteristics are a function of several variables, including number of rows, coil fins per inch, entering air conditions and flow rate, and working fluid entering conditions and flow rate. An air system designer's task is to select a coil that will provide the necessary heat transfer to maintain the supply air setpoint temperature. Evaluating coil performance is complex, involving both heat and mass transfer. Computerized sizing programs are available from coil manufacturers, and simplified sizing methods are also presented in manufacturers' catalogues. Chapter 2.1 summarizes the technical basis for coil performance calculations.

To start the sizing process, the maximum or design load of the zones served by the unit must be determined. Generally for packaged or central systems, design air flow rate is based on the system *cooling* load. This maximum load can be estimated from the following parameters:

- the peak latent and sensible zone loads
- the desired quantity of outside air
- the return air temperature and humidity
- the outside air temperature and humidity

Knowing the above information, the coil selection process can proceed according to the following steps:

1. Specify supply air conditions and determine flow rate.
The design air flow rate is the amount of supply air required to offset the peak zone loads. This flow rate can be determined from the supply air temperature, the return air temperature, and the sum of the zone peak cooling loads. A value of 55°F is often used as the design supply air temperature. The design air flow rate is determined using a sensible heat energy balance as shown below:

$$\dot{m}_{\text{air}} = \frac{Q_{\text{sensible,zones}}}{Cp_{air} \cdot (T_{\text{return air}} - T_{\text{supply air}})}$$

where

Q = peak cooling load (Btu/hr)
T = temperature (°F)
Cp_{air} = heat capacity (for air 0.24 Btu/°F lb)
m = mass flow rate (lb/hr)

In situations where space humidity control is desired, the necessary air humidity leaving the coil can be calculated from the return air humidity ratio and the space latent heat gains.

$$W_{\text{supply air}} = W_{\text{return air}} - \frac{Q_{\text{latent , zones}}}{\dot{m}_{\text{dry air}} \cdot h_{\text{fg, water}}}$$

where

$$\dot{m}_{\text{dry air}} = \frac{\dot{m}_{\text{air}}}{1 + W_{\text{return air}}}$$

and

W = humidity ratio (lbs. of water/lbs of dry air)
Q = peak cooling load (Btu/hr)
h_{fg} = heat of fusion (Btu/lb)
m = mass flow rate (lb/hr)

A good estimate of the supply air humidity ratio can also be calculated by using the total air flow rate (not dry air flow rate) in the first equation above as noted in Chapter 2.2.

The design air flow rate and air-exiting-the-coil conditions are now defined. The supply air temperature and humidity ratio specify the state of the air and pinpoint its location on a psychrometric chart.

2. Calculate the coil entering conditions and loads.
The coil load is generally not equal to the sum of the zone loads. The coil load is a function of several design parameters, including the supply air temperature, air flow rate, percentage of outside air, and outside air temperature and humidity at design conditions. To calculate the coil load, the mixed air

conditions (the temperature and humidity of the air entering the coil) can be easily calculated using mixing equations:

$$T_{\text{mixed air}} = F_{\text{outside air}} \cdot T_{\text{outside air}} + (1 - F_{\text{outside air}}) \cdot T_{\text{return air}}$$

$$W_{\text{mixed air}} = F_{\text{outside air}} \cdot W_{\text{outside air}} + (1 - W_{\text{outside air}}) \cdot T_{\text{return air}}$$

where

F = mass flow rate/supply air mass flow rate

Finally, the design cooling sensible and latent coil loads are calculated as follows:

$$Q_{\text{sensible}} = \dot{m}_{\text{air}} Cp_{\text{air}} (T_{\text{mixed air}} - T_{\text{supply air}})$$

$$Q_{\text{latent}} = \dot{m}_{\text{dry air}} h_{fg} (W_{\text{mixed air}} - W_{\text{supply air}})$$

A very close approximation of latent load can be determined by using the total air flow rate (not the dry air flow rate) in the equation above.

3. Select coil to meet the coil loads at the design conditions.
A large number of parameters affect the ability of a coil to provide the desired supply air conditions given the air flow rate and coil inlet conditions. Some of these parameters include

- Coil face area — The coil face area affects the heat transfer surface area, the air velocity in the coil, and the size of the air handling unit.
- The sensible heat ratio (SHR) — The SHR is the ratio of the coil sensible load to the total load (see Chapter 2.2).
- Coil depth and fin spacing — These parameters affect the coil heat transfer surface area and air velocity in the coil.
- Primary fluid entering temperature and flow rate — The primary fluid conditions and flow are important because they affect the heat capacity of the primary fluid, as well as the heat transfer characteristics inside the coil tubes.
- Thermodynamic limits — The coil effectiveness is the fraction of the theoretical maximum heat that may be transferred to the air stream. This value can not exceed a value of one. For HVAC applications, a typical cooling coil effectiveness value is 0.5.

The goal of the selection process is to select a coil that has adequate capacity to add or remove the necessary heat from the air stream. It also must have a design SHR that matches the desired SHR as closely as possible. Another important consideration is the minimization of life-cycle costs. Energy costs for air-side fans and water-side pumps decrease as pressure drop across the coil decreases. Coil air-side pressure drop is affected by flow rate, face area, and fin spacing. Water-side pressure drop is affected by flow rate, tube size, and number of rows. All of these design parameters impact coil first costs. Thus, the objective is to balance the first cost of the coil with annual energy operating costs.

Computerized selection procedures predict coil performance accurately and easily. Most software permits the performance of the coil to be evaluated in all operating conditions. The same software also facilitates performing life-cycle cost analyses since many different coils that meet all thermal criteria must be evaluated quickly to find the optimal coil. Manual selection procedures are outlined in manufacturers' catalogs. These procedures rely on design performance data presented in tables and charts.

For either computerized or manual methods, the following procedure is generally followed in selecting a water cooling coil.

1. Determine volumetric supply air flow rate from design mass air flow rate (CFM = lbs/hr/4.5).
2. Specify maximum face velocity (typically 300 to 800 fpm).
3. Calculate minimum coil face area.
4. Select an available coil size.
5. Recalculate face velocity based on actual size.
6. Determine the enthalpy of the air entering and exiting the coil under design conditions.
7. Determine total coil load from enthalpies and air flow rate.
8. Specify a water temperature entering the coil (typically 45°F).
9. Specify a water temperature exiting the coil (typically 10–20°F higher than the entering temperature).
10. Calculate required flow rate of water.
11. Assume number of circuits to give 2–5 GPM per feed.
12. From coil performance data, select the lowest number of rows and fin spacing that will meet or exceed total design coil load.
13. Calculate SHR.
14. Depending on SHR value, use appropriate air friction charts to determine air-side pressure drop.
15. Determine total water-side pressure drop across the header and circuit. Circuit pressure drop is dependent on tube size, water velocity, and number of passes.
16. Repeat sizing procedure to select other coils and evaluate each based on minimizing life-cycle costs.

4.3.3 Fans

Fans move air through ducts and system equipment to provide heating, cooling, and ventilation to the building zones. A fan utilizes a power-driven, rotating impeller that creates a pressure differential causing air flow.

Fan Types

Fans are classified by the direction of air flow through the impeller. Two main categories of fans are centrifugal and axial. In centrifugal fans, air flows in a direction radially outward from the shaft. In axial fans, air flows in a direction parallel to the shaft. Within the two fan classifications, there are various subclasses. For either fan type, the energy imparted to the air is mostly in the form of static pressure and partially as velocity pressure. Figure 4.3.10 shows the configuration and fundamental components of both centrifugal and axial flow fans.

Centrifugal Fans

Centrifugal fans produce pressure by changing the magnitude and direction of the velocity of the inlet air. Different types of centrifugal fans are distinguished by their impeller (blade) shape and other features. Operating characteristics of several types of centrifugal fans are presented in Table 4.3.11. The fan types are described below in order of increasing efficiency.

Radial — Because the fan wheel is rugged and simple, radial fans are usually used in industrial systems where the air stream may be contaminated or may be designed to carry solid materials in the air stream.

Forward curved — Forward curved fans generally produce less pressure than backward inclined fans, but are relatively easy to fabricate. They also tend to operate at a lower speed and produce less noise than other centrifugal fan types. Because of these characteristics, forward curved fans are used for low-pressure HVAC applications such as residential systems or packaged air handling systems.

Airfoil — Backward curved fans are of higher efficiency, can produce high static pressures and are *power limiting*. With power-limiting fans the power draw of the fan will climb to a maximum at a maximum efficiency (and particular flow rate) and then decrease as the flow rate increases. Radial and forward curved fans do not exhibit a power-limiting characteristic and, therefore, may overload motors.

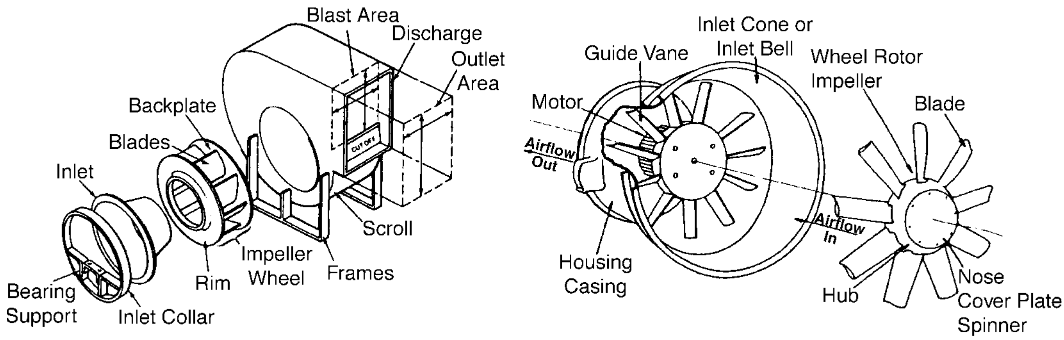


FIGURE 4.3.10 Centrifugal and axial fan components.

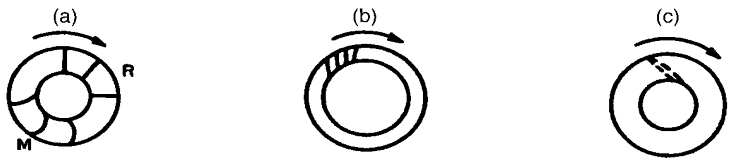


FIGURE 4.3.11 Types of centrifugal fans include: (a) radial, (b) forward curved, and (c) airfoil.

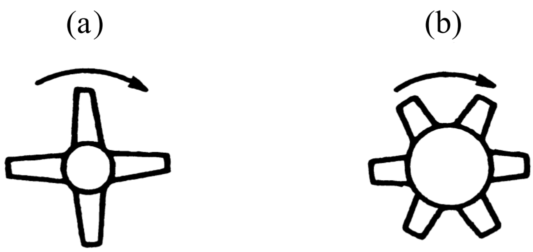


FIGURE 4.3.12 Types of axial fans include: (a) propeller and (b) tubeaxial.

Backward inclined — Backward inclined fans are similar to airfoil fans in both configuration and performance. The primary difference is that unlike airfoil fan blades, backward-inclined blades are simply flat rather than foils. Consequently, their efficiency is slightly lower. Airfoil and backward curved centrifugal fans are most commonly used in commercial HVAC systems due to their efficiency and pressure head capabilities.

Axial Flow Fans

Axial flow fans impart energy to the air by giving it a swirling motion. The different types of axial fans are distinguished by their blade shape, blade pitch, and hub diameter to blade tip diameter ratio (hub ratio). Two types of axial flow fans are presented in Figure 4.3.12. The types of axial flow fans in order of increasing efficiency are described below.

Propeller — Propeller fans generate low pressure differentials and have low efficiency and low cost. They are most often installed unducted for low pressure, high volume applications such as air circulation, spot cooling, or transfer of air from one space to another with very little need for pressure differential. Propeller fans are characterized by an impeller with single thickness blades attached to a small diameter hub/shaft and a simple circular ring housing. Propeller fans essentially have a zero hub ratio (hub diameter/fan diameter), having enough hub only to satisfy the mechanical requirements to drive the fan.

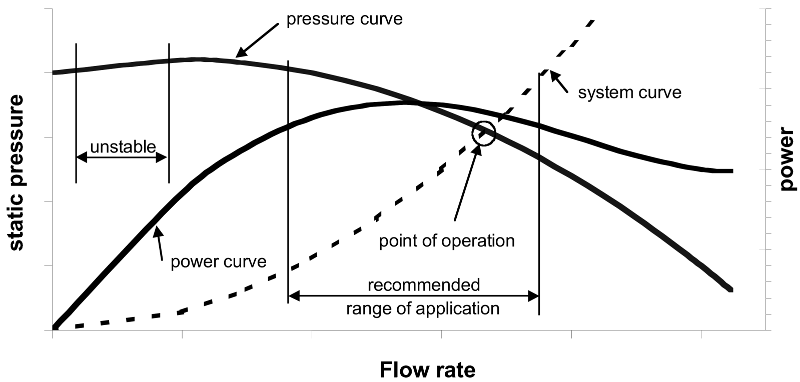


FIGURE 4.3.13 Typical fan performance curve.

Tubeaxial — Tubeaxial fans generally have four to eight airfoil or single thickness blades attached to a larger diameter hub mounted in a cylindrical housing. Tubeaxial fans generally have a hub ratio of 0.25 or greater. Tubeaxial fans are of higher efficiency than propeller fans and are able to generate more pressure.

Vaneaxial — In order to improve the flow vector components and fan efficiency, straightening vanes can be added to the fan housing. Vaneaxial fans are simply tubeaxial units equipped with guide vanes. Vaneaxial fans may be used in general HVAC applications for low, medium, and high pressure systems. They are more compact than centrifugal fans and are advantageous when space restrictions exist.

Fan Arrangement and Classification

Fan specifications include fan class (1, 2, or 3) which reflects the sturdiness of the fan — the higher the class, the higher the fan structural strength, allowing it to operate at higher impeller speeds and in more adverse conditions. Fans are also specified by their arrangement, which indicates the motor location, air discharge orientation, and drive train type (direct drive or pulley drive). Centrifugal fans can be specified as either single width, single inlet (SWSI) or double width, double inlet (DWDI). These designations indicate the width of the impeller wheel and whether the air may enter the fan on only one side or from two sides.

Fan Performance Curves

A typical fan performance curve is shown in Figure 4.3.13. Fan curves are a basic tool used in the design and analysis of performance. As can be seen in the figure, fan curves are presented with air flow rate (CFM) along the horizontal axis and differential pressure (psi or feet of head) along the vertical axis. The relationship between pressure and flow for a particular fan, operating at a particular speed, is presented as a single line. For the particular speed and fan, a given flow and pressure differential operating point will lie on the performance (pressure curve) line. The power curve, also shown, presents the fan power required at a particular flow rate and pressure. Variation of air density or fan speed will change the fan curves in a way that is predictable by fan laws discussed later in this section.

For a particular air distribution system, a *system curve* can be overlaid on a fan curve to predict and visualize how a fan will interact with the system. A system curve is generated by knowing the pressure drop that will occur in a system at a particular flow rate and invoking the second fan law (described below) to generate points at other flow rates. Note in Figure 4.3.13 that two different ranges are shown, the recommended application range and the unstable range. In selecting a fan, it is important for the fan to operate in the recommended application range to ensure good efficiency. If the fan/system combination results in operation in the unstable range, there is a high risk that the flow will exhibit undesirable pulsations.

In lieu of a fan curve, manufacturers very often present their fan performance data in a table. A fan performance table will provide information similar to that found in a fan curve.

Fan Power

A fan rotor adds energy to the air stream in the form of static energy (pressure), kinetic energy (velocity), and heat. The quantity and form (kinetic, static, or heat) of energy added to the air stream is a function of power input, fan efficiency, motor efficiency, and whether the fan motor is situated in the air stream or out of the air stream.

A useful formulation for the calculation of power required to develop a particular flow rate at a particular differential pressure is as follows:

$$Power = C \cdot Flow \cdot \Delta P / (\eta_{\text{fan}} \cdot \eta_{\text{motor}})$$

where

C = system constant
ΔP = air pressure rise
η = efficiency

Fan efficiency is the ratio of the useful work added to the flowing air to the shaft work input; motor efficiency is the ratio of the shaft work to the motor electric power input.

Fan Laws

Fan laws relate performance variables for fans of similar type and geometry. Standardized fan performance curves or tables are generated for a particular fan operating at a particular speed at standard atmospheric conditions. The performance of the fan can be predicted at other speeds, flow rates, pressures, and air densities using fan laws. There are three fundamental fan laws that can be used to predict changes in these variables.

Fan Law Number 1: *In a fixed system, fan flow rate is proportional to fan speed.*

$$CFM_{\text{new}} = CFM_{\text{old}} \cdot \frac{RPM_{\text{new}}}{RPM_{\text{old}}}$$

A fan acts as a constant volume device — the faster the blades operate the higher the volumetric flow rate will be. Since a fan is a constant volume scoop, an important axiom to the first fan law is that changes in density have *no effect* on the volumetric flow rate generated by a fan at a particular speed. However, the mass flow rate increases with density.

Fan Law Number 2: *In a fixed system, the static pressure rise across the fan varies as the square of the fan speed.*

$$SP_{\text{new}} = SP_{\text{old}} \cdot \left[\frac{RPM_{\text{new}}}{RPM_{\text{old}}}\right]^2$$

Since more air mass creates a higher pressure differential across a fan, an important axiom to the second fan law is that the pressure change across a fan is directly proportional to the density of the air. This relationship can be expressed as

$$SP_{\text{new}} = SP_{\text{old}} \cdot \frac{\rho_{\text{new}}}{\rho_{\text{old}}}$$

Fan Law Number 3: *In a fixed system, the power of a fan varies as the cube of the fan speed (RPM).*

$$HP_{\text{new}} = HP_{\text{old}} \cdot \left[\frac{RPM_{\text{new}}}{RPM_{\text{old}}}\right]^3$$

As in the second fan law, an axiom to the third fan law is that fan power varies directly with air density:

$$HP_{\text{new}} = HP_{\text{old}} \cdot \frac{\rho_{\text{new}}}{\rho_{\text{old}}}$$

The fan laws are expressions of the fact that the fan curves of similar fans are homologous. This implies that at the same point of rating, geometrically similar fans have the same efficiency.

The most common use of the fan laws is to determine the performance of a specific fan at altitudes and/or air temperatures other than those at which a particular fan is rated. As altitude or temperature increases, the density of air decreases and the pressure differential and work required will reduce proportionally per fan laws 2 and 3.

System Curve

For a fixed HVAC air distribution system, a given air flow rate requires a specific total pressure in the system. The distribution system components — including dampers, filters, coils, ductwork, and diffusers — represent a resistance that must be overcome. In HVAC applications, the relationship between pressure and flow rate for a system of fixed resistance follows the second fan law.

Using the second fan law, the system curve can be determined from one known operating point (usually design) for a given system. Once determined, the relationship described by the second fan law can be used to calculate other system operating points and plot the system curve. By superimposing the system curve on the fan curve, the operating point of a given fan in a given system can be determined (see Figure 4.3.13). Since fan performance data are generally published for a specific fan speed, the fan laws can be used to generate additional fan curves. The intersection between the system curve and the fan curve identifies the flow and pressure differential at which the fan will operate for a particular system.

System Effect

In the duct design section of this chapter, methods for calculating pressure drop in distribution system equipment are discussed. However, even when these methods are followed, the installed fan performance measured by field tests may differ from the designer's calculated performance. This not uncommon occurrence results because the published fan performance data are based on standardized tests conducted in laboratories using specific fan entry and exit configurations. The fan tests are conducted under ideal conditions where the flow into and out of the fan has no air swirling, and the air stream has a uniform velocity. In actual systems, the inlet or outlet conditions of the fan may be less than ideal in that there may be air swirl or significant velocity gradients at the fan inlet or outlet. These velocity gradients or swirling actions are what cause the system effect to occur. Since testing and system configurations differ, *system effect* factors must be taken into account to properly understand and predict the fan-system operation.

Figure 4.3.14 shows the impact of the system's effect on performance. The system effect causes the system flow coefficient to be higher and the system curve to be steeper than expected. If the system effect is not taken into account during system design, the installed flow rate will be less than the design flow rate. To increase flow, the fan speed and system pressure must increase, which increases power requirements and operating costs as well.

The system effect factor cannot be measured in the field but can be predicted and accounted for using methods outlined in Chapter 32 of the 1997 *ASHRAE Fundamentals Handbook* (ASHRAE 1997) and in AMCA Publication 201 (AMCA 1990). The data account for differences in velocity profiles between fans as tested and fans as installed. The method consists of identifying a flow configuration that most closely

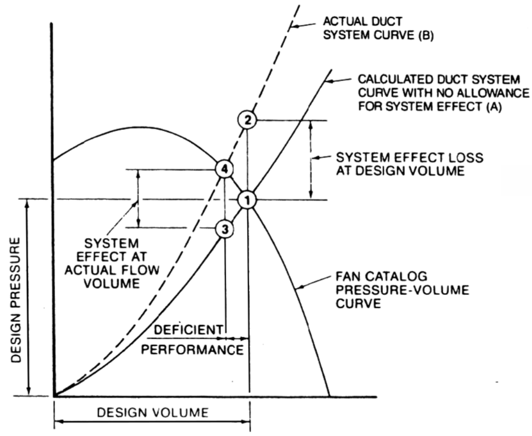


FIGURE 4.3.14 System effect (courtesy of SMACNA, 1990, *HVAC Systems Duct Design*).

matches the particular configuration being considered. Once the flow configuration is identified, the proper system effect curve can be selected and applied to estimate the effect as a pressure loss. An otherwise accurate design can be significantly in error if system effects are not considered.

Fan Flow Modulation

In variable air volume distribution systems, the volume of air delivered by the fan varies as building loads vary. To modulate the flow, the characteristics of either the fan or the system must change. The system performance changes by changing the system resistance, by use of dampers, for example. Fan characteristics may be changed by one of several methods, including a variable-speed drive motor, fan discharge dampers, and fan inlet vane dampers.

To change system resistance, branch duct dampers must accomplish large pressure drops since fan speed remains constant. As the damper resistance increases, the system constant decreases, and the system curve shifts to the left. To achieve low flow rates, extreme pressures may be exerted over branch dampers. This results in unstable flow and noise generation. Thus, most distribution system flow control occurs at the fan.

For fan control with a discharge damper, a damper absorbs excess pressure near the fan while a constant static pressure is maintained in the duct. As the damper closes, the fan curve shifts to the left. It is common to represent fan performance with capacity control dampers as a family of curves, each passing through the origin.

Fan control with inlet-vane dampers has characteristics similar to control with discharge dampers. For this configuration though, the capacity dampers are an integral part of the fan equipment. A properly designed unit can accomplish capacity control with a lower power requirement than discharge dampers.

Changing the fan performance through changing fan speed is the most energy efficient method for accomplishing capacity control because, as illustrated by the fan laws, fan power requirements drop with the cube of the fan flow rate. However, variable speed drive controls are more complex and expensive.

Fan Selection

After the air distribution system has been defined and the system performance curve evaluated, the fan can be selected to meet the system requirements. Fan selection involves choosing the size, type, class, arrangement, and capacity control to accomplish the job most economically.

The most efficient operating area for a fan is usually clearly presented in graphic and tabular presentations of manufacturer data. For variable air volume systems, it is important to know the frequency at which the fan will be operating at different part loads to select the most efficient fan. In general, the less expensive the fan capacity control, the more expensive the operating costs.

4.3.4 Ducts

Ducts are conduits used to carry air from air handling units to or from conditioned or otherwise ventilated spaces. Supply, return, or exhaust air ducts are sized to deliver or remove the amount of air required to meet zone loads or ventilation requirements at the design conditions. Duct sizing and application is influenced by space availability, desired location of room diffusers or returns, and operating versus capital cost tradeoffs. Proper designs must consider allowable noise levels, fire code requirements, duct leakage, operation and maintenance accessibility requirements, and heat conduction rates to or from the duct.

Construction and Codes

Ducts are often constructed of galvanized steel, but other materials such as black carbon steel, stainless steel, aluminum, copper, fiberglass reinforced plastic, and concrete are used in some applications, depending on factors such as corrosion, durability, purpose, price, and pressure requirements. Application considerations prescribe the best material choice. Different duct materials have different properties, such as roughness (which affects air pressure drop), ease of modification, weight, cost, thermal expansion, rigidity, porosity, strength, weldability, and corrosion resistance.

The minimum requirement for duct material strength and thickness is dictated by code and affected by the system air pressure and duct air velocity. Duct friction is a function of surface roughness and velocity. In general, ducts are classified into two velocity regimes — low velocity (below 2500 ft/min) and high velocity (up to 4500 ft/min). Recommended duct friction rates differ between the two regimes. In general, low air velocity applications include constant volume systems and duct sections near spaces where high noise levels are unacceptable. High air velocity systems are used primarily in the main trunks of VAV or industrial systems to reduce duct capital costs and space requirements. Ducts can be either internally or externally insulated to reduce the transfer of both heat and noise.

Industrial Ventilation: A Manual of Recommended Practice (ACGIH 1998) provides an excellent resource outlining the application of ventilation systems in industrial settings. Two publications, *HVAC Systems Duct Design* (SMACNA 1990) and *HVAC Duct Construction Standards* (SMACNA 1995), provide detailed design information on duct construction, installation methods, design, and application for all common systems in use today.

Theory of Air Flow in Ducts

Fluid flow is measured indirectly by a pressure differential measurement. Pressure is lost and flow is hindered by friction arising from the interaction of the fluid with the conduit. This interaction occurs because of both static losses and dynamic losses. Static pressure loss results from the friction of the air on the wall of the duct work. Dynamic losses occur under turbulent flow conditions whenever there is a sudden change in direction or magnitude of the velocity of the air flow.

Bernoulli's equation, which is a specialized form of the first law energy balance, can be applied for the analysis of air flow in ducts. Written in terms of pressure, the equation can be expressed in the following form:

$$SP_1 + VP_1 + HP_1 + \Delta P_{fan} = SP_2 + VP_2 + HP_2 + PL_2$$

States (1) and (2) are points in the ducted air stream, as shown in Figure 4.3.15, where

- SP = static pressure — pressure normal to flow
- VP = velocity pressure — pressure arising from flow velocity that equals $\rho V^2/2g_c$ where g_c is (32.2 lb_m ft/s^2)/lb_f
- HP = potential pressure — pressure arising from columns of air occurring because of a height change (normally a small effect)
- ΔP = total pressure increase across fan (if present)
- PL = pressure loss due to friction

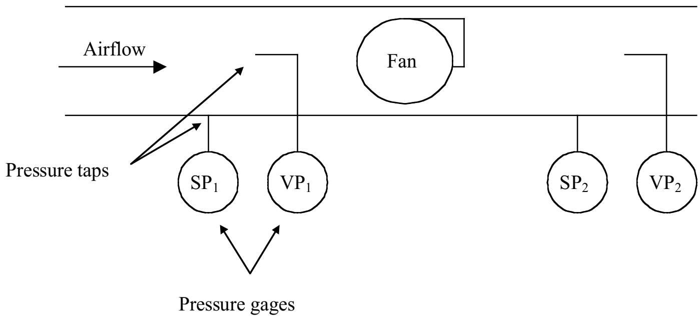


FIGURE 4.3.15 Ducted air flow.

Summing the pressure losses (*PL*) occurring across each duct section and system component (such as dampers, fitting, filters, and coils) determines the pressure loss that must be overcome by the fan. Knowing the pressure loss in the various branches in a complex ducting system allows for the prediction of the fraction of supply air that will follow each branch. Proper application of Bernoulli's equation and friction loss equations allows the designer to size the ducting such that each section will have the proper pressure drop at the design flow rate, resulting in the desired branch flow.

Although duct pressure drop is much lower than component pressure drop, its evaluation is important to minimize the use of dampers to pressure balance the system. The total pressure decrease across a section of ductwork is due to friction and dynamic losses. The theory underlying the frictional and dynamic duct losses is outlined below.

Friction Losses

For fluid flow in conduits, the friction pressure drop (PL or ΔP) can be determined from the Darcy equation

$$\Delta P_f = f\left(\frac{L}{D_h}\right)(VP)$$

where

ΔP_f = pressure loss arising from friction
f = friction factor, a function of Reynolds number and duct internal roughness
L = duct length over which the friction loss occurs
D_h = hydralic cross-sectional diameter
VP = velocity pressure

For circular ducts, the hydraulic diameter is the same as the duct diameter. For other ducts, the hydraulic diameter is equal to

$$D_h = \frac{4A}{P}$$

where

A = the cross-sectional area
P = the perimeter of the ducting

The value of the friction factor is dependent on the flow regime characterized by the Reynolds number. The dimensionless Reynolds number represents the ratio of fluid inertial to viscous forces, specifically

$$\mathrm{Re} = \frac{D_h V}{\nu}$$

where

ν = the kinematic viscosity of the fluid.
V = the fluid velocity

For laminar flow (Re < 2300), the friction factor is only a function of the Reynolds number. For turbulent flow, the friction factor is a function of the Reynolds number and duct surface roughness. The Moody friction factor chart presents the friction factor graphically for both laminar and turbulent flow.

Because use of the Darcy equation can be cumbersome, specialized charts have been developed for determining air pressure loss in galvanized ducts based on the Moody diagram. Figure 4.3.16 presents duct friction loss per unit of duct length as a function of duct volumetric flow rate.

The shaded chart area delimits recommended duct sizes for low and high velocity applications. Duct sizes are specified for circular ducts. Rectangular ducts having equivalent hydraulic characteristics may be substituted for the circular size. Rectangular duct equivalents for circular duct sizes are published in *ASHRAE* (2001) and *SMACNA* (1990). To evaluate friction loss for ducts constructed of materials other than galvanized steel, roughness factors available from SMACNA (1990) can be applied to the pressure losses from Figure 4.3.16.

By invoking the second fan law, the pressure drop for various flow rates can be determined for a fixed duct system by the following relationship:

$$\frac{\Delta P_{\text{new}}}{\Delta P_{\text{old}}} = \frac{V^2_{\text{new}}}{V^2_{\text{old}}} = \frac{CFM^2_{\text{new}}}{CFM^2_{\text{old}}}$$

where

CFM = the air volumetric flow rate
V = air velocity
ΔP = the differential pressure measurement in the duct section of interest.

This relationship is the basis for the system curve equation commonly used in HVAC design.

Dynamic Losses

Dynamic losses occur in fittings when there is a change in flow direction or velocity. Dynamic losses occur across duct fittings, such as elbows, tees, entries, exits, transitions, and junctions. The pressure loss associated with each type of fitting is proportional to the fluid velocity pressure. This relationship can be expressed as

$$\Delta P_d = C \cdot DP$$

where

C = the local loss or dynamic loss coefficient
DP = dynamic pressure ($\rho V^2/2g_c$)

Coefficient values for different fittings have been determined by laboratory testing and are published in several HVAC reference manuals, such as *SMACNA* (1990) and *ASHRAE* (2001). For converging or diverging flow junctions, two loss coefficients are reported — one for the main branch pressure loss and

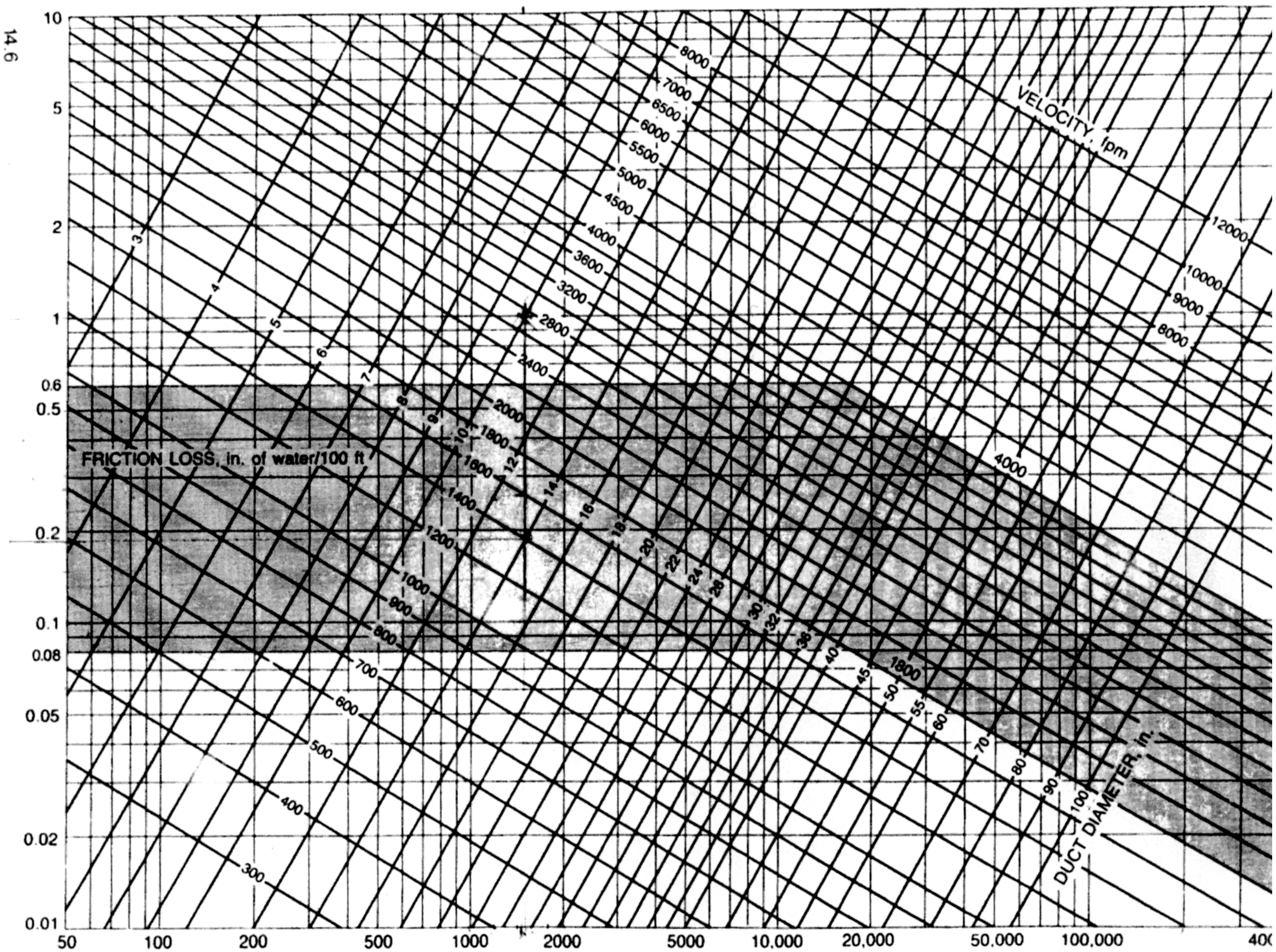


FIGURE 4.3.16 Duct friction loss chart (courtesy of SMACNA, 1990, *HVAC Systems Duct Design*).

the other for the branch duct pressure loss. Since there are many fitting configurations for which there is not an exact match in the literature, the designer needs to use judgement in the estimation of a *C* value for unique or unpublished configurations.

Duct System Components and Considerations

General construction details representing acceptable practice are presented in *HVAC Duct Construction Standards* (SMACNA 1995). While a detailed discussion of construction standards and methods is beyond the scope of this chapter, some of the more important ducting elements are introduced below.

Duct Joints, Seams, Sealing, Hanging, and Reinforcement

Normally, installers are given latitude in the selection of duct seams, hangers, and reinforcements, but the construction must meet applicable codes. Particular construction details needed to meet requirements for static pressure, sealing, materials, duct support, and other provisions are outlined in the *HVAC Duct Construction Standards* (SMACNA 1995).

Turning Vanes

A construction feature that reduces pressure loss in system components and fittings is turning vanes. Used in system components such as elbows and tees, turning vanes reduce the dynamic friction loss by directing the air flow parallel to the duct walls.

Fire Dampers

Fire dampers are required where ducts penetrate a firewall. They must be accessible for checking and resetting, so a duct door is normally installed in the duct adjacent to the fire damper. Fire dampers have a fusible link that melts and separates at a particular temperature, causing the spring loaded damper to slam shut. Similar to fire dampers are smoke and fire dampers that normally include a fusible link, but they may also be actuated by a control system to control smoke contamination in buildings.

Balance Dampers

An important activity that occurs during the construction process is the testing and balancing (TAB) of air systems. TAB professionals make field adjustments to air systems to ensure that proper flow is delivered to or from each zone or space, and that the overall flow of the systems is as specified.

In designing and installing ducting systems, it is important to make adequate provisions for TAB. One of the most important components that must be included to allow TAB is the balance damper. Balance dampers can be of either single or multiple blade construction but generally have a locking mechanism so that the person performing TAB on the system can set and lock the balance dampers in the proper position. Further information on TAB of air systems is available through the American Air Balance Council (AABC) and the National Environmental Balance Bureau (NEBB).

Duct Air Leakage

Duct leakage can significantly increase CFM air flow requirements and energy loss, decreasing overall system efficiency and effectiveness. Duct leakage impacts system performance by increasing fan power requirements, cooling coil loads, and reheat coil loads. The level of impact depends on the leakage rate, fan type, fan control, and system type. Typical leakage levels have been measured by the Florida Solar Energy Center (FSEC) for residential and small commercial buildings. For residential duct systems, leaks are 10–20% of fan flow on each side of the fan. For light commercial buildings, field studies suggest that duct leakage is actually higher than that found in residences — the average leakage in the supply ducts being over 20% of fan flow. Lawrence Berkeley National Laboratory (LBNL) has performed limited measurements of duct leakage in large commercial buildings. The performance data are inconclusive, but it is clear that some fraction of these buildings have significant (i.e., >10%) duct leakage. Even more unclear for this class of buildings is the impact of duct leakage on energy use. Some analyses of this question have been published; however, because the interactions between different factors are so complex, consensus on the appropriate analysis procedure has not been reached. Nevertheless, a conservative evaluation of the cost effectiveness of duct sealing in large commercial buildings warrants sealing ducts in most locations analyzed.

The functional form usually used to describe the relationship between the pressure in a duct and the flow through the leaks in that duct is as follows:

$$CFM_{leaks} = C_{\text{leak}} \Delta P^{n}_{\text{Duct to Space}}$$

where the pressure differential is the difference between the pressure in the ducts and the pressure of the space surrounding the ducts. When testing ducts for leakage, a known measured pressure differential is applied, and the flow required to maintain that pressure differential is determined by using a calibrated fan. By using several data points for Q and ΔP, one can solve for C_{leak} and n. For leaks that look like orifices (e.g., holes), n is 0.5, whereas, for leaks with some length (lap joints between duct sections) n is approximately 0.6 to 0.65.

Duct Conduction and Insulation

Duct heat gains and losses can be significant, particularly for single-story buildings with ducts located outside the conditioned building space. Insulating ducts reduces unwanted heat transfer, condensation on cold surfaces, and noise. Duct heat gains and losses must be determined to size distribution system fans and cooling coils. Methods for estimating duct heat transfer based on entering air temperature, exiting air temperature, and surrounding air temperature are outlined in Chapter 2.1 of this handbook.

The duct heat transfer coefficient is predominantly a function of duct construction, duct insulation level, and duct air flow rate. For example, experimental measurements of duct heat transfer coefficients for a 10-in square metal duct with 2-in $^3/_4$ lb/ft^3 faced fibrous glass range from 0.134 to 0.148 Btu/(hr ft^2 F) for duct velocities ranging from 780 to 3060 feet per minute. For a 10 in round duct with the same insulation and velocity range, the duct heat transfer coefficients range from 0.157 to 0.163 Btu/(hr ft^2 F) (Lauvray, 1978).

Duct System Design

The primary concern of the designer is that the desired amount of air be delivered to the zones. Other important considerations that must be accounted for in the duct system design include

- Available space for installation and access
- Meeting noise criteria
- Air leakage rates to or from the ducting
- Heat gain or loss
- Testing and balancing
- Smoke and fire control
- First costs versus operating costs

Duct Design Methods

Duct design methods are used to determine the size of main and branch duct sections. Several methods have been developed for achieving acceptable designs that balance capital and operating costs. While in many systems, the pressure drop of the duct is only a small fraction of the total system pressure drop, it is important to accurately estimate duct losses so that flow will be properly balanced among zones and rooms served by an AHU.

There are several duct design methods, including (1) equal friction, (2) static regain, (3) velocity reduction, and (4) the T-method. The two most common of the four, equal friction and static regain, are discussed below in detail. In general, the approach of each of the four methods is as follows.

In the *equal friction* method, the goal is to maintain the same pressure gradient throughout the system.

In the *static regain* method, the duct sections are sized so that the friction losses are offset by converting available velocity pressure to static pressure at junctions.

In the *velocity reduction method*, specific duct velocities (which dictate duct size) are specified and designed for various parts of the system.

The *T-method* is an optimization procedure for evaluating the economic trade-off between duct size (capital cost) and fan energy (operating cost).

No matter the methodology used for duct sizing, it is best to create a design that delivers the proper amount of air to each space with little need for throttling through the use of balance dampers. If a balance damper must be significantly closed during TAB, it is often an indication that the ducting has not been designed or constructed carefully.

Equal Friction

The most common manual method for duct design is the equal friction method, in which all duct sections are sized to maintain an equal pressure gradient in the system (thus the term *equal friction*). The method is normally applied to low pressure/low velocity systems. The gradient commonly chosen is 0.1 in water gauge of pressure drop per 100 ft duct length. With this method, branch ducts of different lengths will not be balanced. Achieving a balanced system to ensure appropriate zone flow rates relies on the proper adjustment of the balance dampers in the zone boxes.

Static Regain

The static regain method is the most commonly used computerized method for duct sizing. When precisely applied, the static regain method results in a system that is self-balancing. The method is based on the requirement that the static pressure losses occurring in a duct section are regained by a decrease in velocity pressure in the following section. Velocity pressure decreases through a decrease in velocity. A decrease in velocity occurs through an increase in duct size. This design approach results in each duct section having nearly the same static pressure, thereby creating a system that is self balancing.

Design Software

While the equal friction method can be completed by hand calculations, the static regain method requires a computerized analysis. Both methods are available in computerized form from major HVAC equipment manufacturers. Although the static regain method is complicated and requires several calculation iterations, in computerized form it is as straightforward as the equal friction method.

4.3.5 Terminal Units

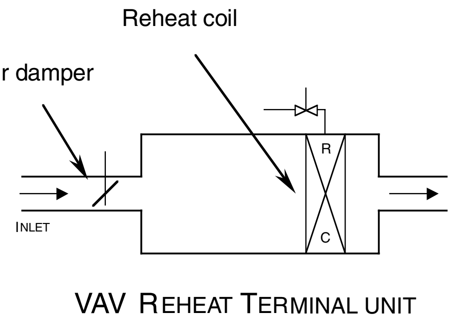
Terminal units (also known as terminal "boxes") receive conditioned air from a central air handling unit and vary the volume and/or temperature of the air delivered to the conditioned space to maintain the zone setpoint. A single terminal box serves one thermal zone. Each box may distribute air to several zone supply diffusers. Air distribution systems can be distinguished by their terminal type, and the basic control of a central air handling system is dependent on the terminal type associated with the unit. Terminal unit types are described below.

Constant Volume Reheat

Constant volume reheat terminal units are simply reheat coils located in the supply ducts near the zones served. A constant volume of supply air from the central air handling unit flows through the reheat coil and into the zone. If the supply air is too cold to maintain the zone setpoint temperature, the reheat coil, which usually uses electricity or hot water as its heat source, will modulate to reheat the supply air, thereby maintaining the zone setpoint. Because constant volume reheat systems often operate in a mode where the air is cooled by a vapor compression cycle, only to be reheated at the terminal units, they tend to use a large amount of energy when compared to other system types.

Variable Volume without Reheat

Variable volume (VAV) terminal units without reheat are common. The terminals consist of a modulating damper that changes the quantity of air into the space in response to a zone thermostat to maintain the zone



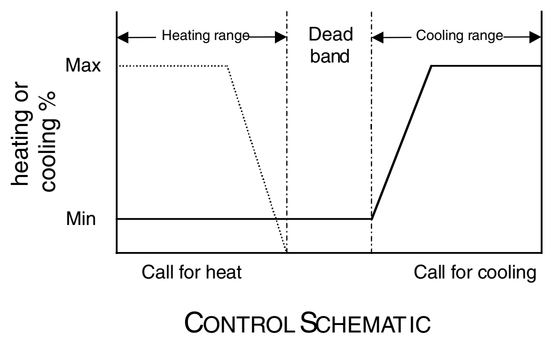


FIGURE 4.3.17 Variable air volume terminal unit with reheat.

setpoint temperature. Often, variable volume terminal units will have a minimum damper position or volume to assure that adequate air ventilation and circulation occurs in the space. In such units, overcooling can occur since no provision is made for the tempering of the supply air when zone cooling loads are small.

Variable Volume Reheat

Variable volume (VAV) terminal units with reheat have an air damper that modulates the quantity of supply air from the central air handler in response to the zone thermostat. Additionally, a reheat coil is provided to temper the supply air when there is no zone cooling load, or if heat is required in the space. Sometimes, the variable volume box and the reheat coil are separated in a zone. In this case, the system is composed of standard variable volume terminal units (without reheat) and zone baseboard heating coils. The coils add heat to the room during low cooling loads or when the zone requires heating.

Figure 4.3.17 shows a schematic drawing of a VAV terminal unit with a reheat coil. In most cases, the units are controlled as shown on the right side of the figure. Specifically, when there is a call for heat, as shown on the x-axis, the VAV damper will be closed to its minimum position, and the reheat coil will be in full heating mode. As the call for heat decreases, the controller will begin to reduce the output to the heating coil until the heating coil is fully off, while the primary air damper remains at its minimum position. On a call for cooling, the heating coil will remain off, and the primary air damper will modulate open until the box goes to maximum cooling.

Dual Duct Constant Volume

Dual duct constant volume terminal units are served with warm and cool primary air from the central air handling unit (Figure 4.3.18). The terminal unit blends these two air streams to produce the necessary supply air temperature to maintain zone setpoint. The terminal unit contains a damper, or set of dampers, that varies the percentage of cool and warm air, but not total volume, supplied to the zone to maintain the zone setpoint temperature. Like their constant volume reheat terminal unit counterparts, these units can use a great deal of energy because the supply air streams are heated in the hot deck and cooled in the cold deck, only to be blended to some intermediate temperature to satisfy the zone loads.

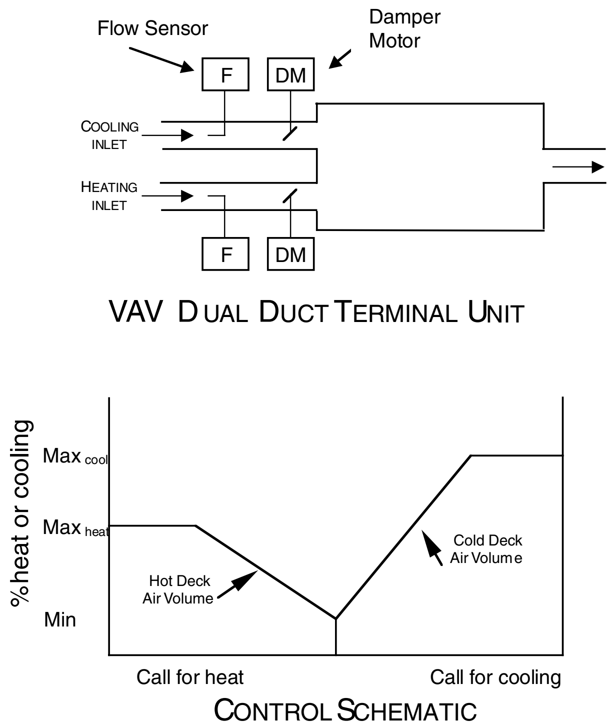


FIGURE 4.3.18 Dual duct terminal unit.

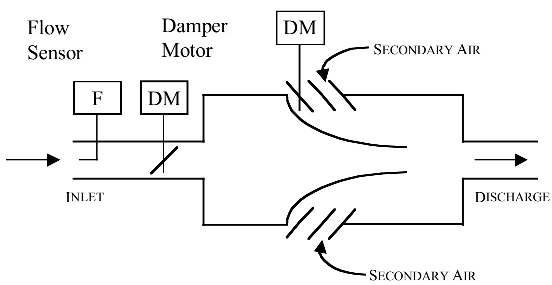


FIGURE 4.3.19 Induction terminal unit.

Dual Duct Variable Volume

Like the dual duct constant volume units, dual duct variable volume terminals are served by both warm and cool air. Unlike the constant volume units, the variable volume units vary both the percentage and the total volume of air into the space to maintain the zone temperature setpoint.

Other Terminal Unit Variations

Two primary terminal units types, single duct and dual duct, are outlined above. Variations and enhancements on these terminal units are described below.

1. Induction terminal units — A VAV induction system has a terminal unit that blends primary air from the central unit with recirculated (secondary) return air from the zone. The higher volume of blended air is then introduced to the space via the supply diffusers. The increased air flow is advantageous to increase the mixing of air in the space.

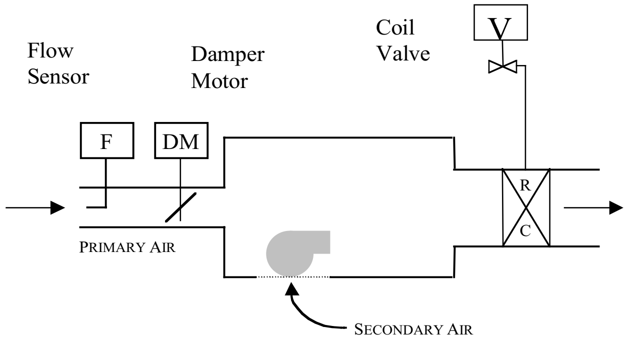


FIGURE 4.3.20 Parallel fan-powered terminal unit.

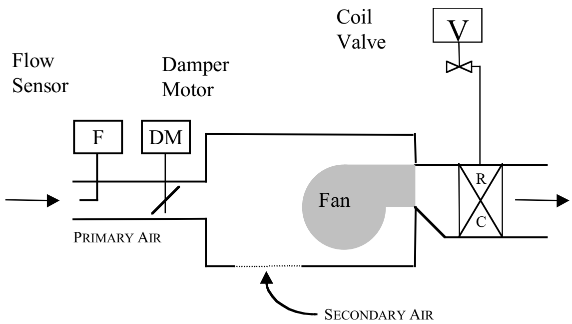


FIGURE 4.3.21 Series fan-powered terminal unit.

Induction boxes, in general, require a higher supply duct static pressure than other terminal unit types in order to induce the secondary air flow and operate properly. For this reason, operating costs using induction VAV boxes are usually higher. Figure 4.3.19 illustrates an induction terminal unit.

2. Fan-powered terminal units — Secondary air may also be introduced to the terminal boxes with a *fan-powered induction* terminal unit. In these units, the fan draws air from the plenum or zone. The induction fan may be installed in a parallel or series arrangement (Figures 4.3.20 and 4.3.21) with the primary air flow. In series, it operates continuously and provides a constant air flow to the zone. In parallel, it operates intermittently to induce the secondary air flow as needed to meet heating demand.

Because fan-powered terminal units have a local fan, the pressure requirements are lower than other terminal types, which can reduce the load on the main air handling unit fan. However, since each of the terminal unit fans require energy, the overall energy use of a system with fan-powered terminals is often greater than systems that have terminals not fan-powered.

Induction units and fan-powered boxes are often used in VAV systems since they allow higher zone air flow rates with a negligible increase in coil loads. A higher flow rate into the zone ensures higher air velocities and greater occupant comfort.

3. Bypass terminal units — Smaller central air handling units (generally below 20 tons of cooling capacity) used in VAV applications usually do not include provisions for the modulation of the primary air volume. In these situations, a bypass terminal unit (Figure 4.3.22) is sometimes used. Bypass units receive a constant volume of air from the central air handling unit but supply a variable air volume to the zone as needed to maintain setpoint. The remainder of the primary air is diverted, via a modulating damper, to a return air plenum or duct.

These units can be more efficient than constant volume units since the need to reheat this air is avoided. Because the supply air flow is constant, this is not a true VAV system; hence supply air temperature reset is

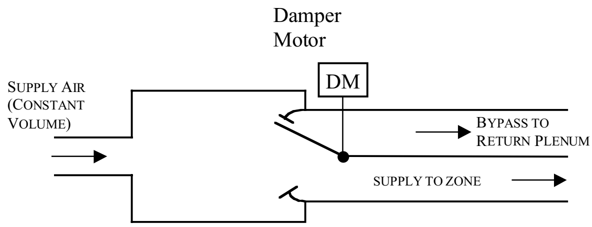


FIGURE 4.3.22 Bypass terminal unit.

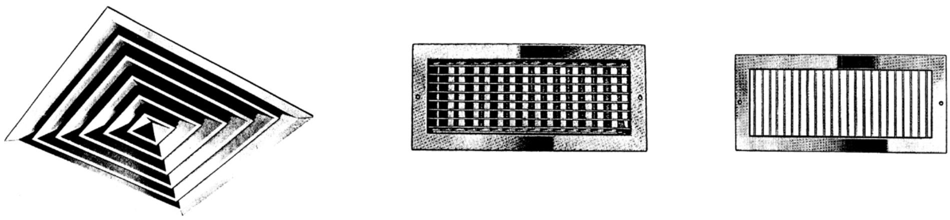


FIGURE 4.3.23 A square ceiling-mounted diffuser, a register, and a grille (courtesy of Carnes Company, Inc.).

often used to prevent a high percentage of the supply air from bypassing the space and reducing air circulation and occupant comfort.

4.3.6 Diffusers

This section describes the equipment and design fundamentals for supply-air outlets and return-air and exhaust-air inlets of air distribution systems. Just before entering or exiting a conditioned building space, conditioned air passes through a *diffuser*, *register*, or *grille*. These devices are not interchangeable since each varies in the way it transitions air to or from the conditioned space. Diffusers are designed to entrain room air with high velocity supply air to induce room air circulation. Registers have slotted or perforated openings and are equipped with a damper to provide direction or volume control of supply air. Grilles are registers without dampers and do not provide any mixing or flow control. Generally, supply air is pushed through diffusers and return air is pulled through grilles. A typical example of each type of device is presented in Figure 4.3.23.

Supply Air Diffusers

Diffusers are installed at supply-air outlets and come in a variety of configurations, including square, circular, and slot. Generally, square and circular ceiling- mounted diffusers consist of a series of concentric rings or louvers. These diffusers discharge air radially in all directions. A slot diffuser is an elongated outlet with an aspect ratio of 25:1 or greater and a maximum height of approximately 3 in (SMACNA, ASHRAE 1988). Figure 4.3.24 shows a slot diffuser in a ceiling mount.

Diffusers are important for maintaining a safe and comfortable indoor environment. They promote proper mixing of the supply air with the room air. Generally, conditioned air is supplied to the outlet at velocities higher and temperatures lower than those acceptable in the occupied zone. Supply-air outlet diffusers slow and temper the supply air by entraining room air into the primary air stream. The *entrainment* of secondary air results in a surface or *Coanda effect*, illustrated in Figure 4.3.25.

The surface effect is caused by the primary air stream moving adjacent to a ceiling or wall, creating a low pressure area adjacent to the surface, and causing the air stream to flow parallel to the surface throughout the length of throw. This effect inhibits the horizontal drop of the cold, primary air stream. Diffusers which result in larger areas of surface spread tend to have a larger surface effect. The surface effect permits the temperature differentials between the primary and secondary air to be large while still maintaining occupant comfort.

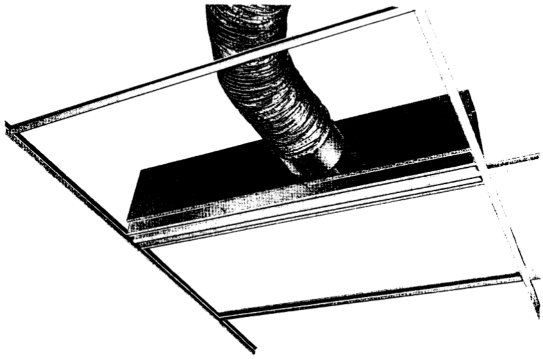


FIGURE 4.3.24 Ceiling-mounted slot diffuser (courtesy of Carnes Company, Inc.).

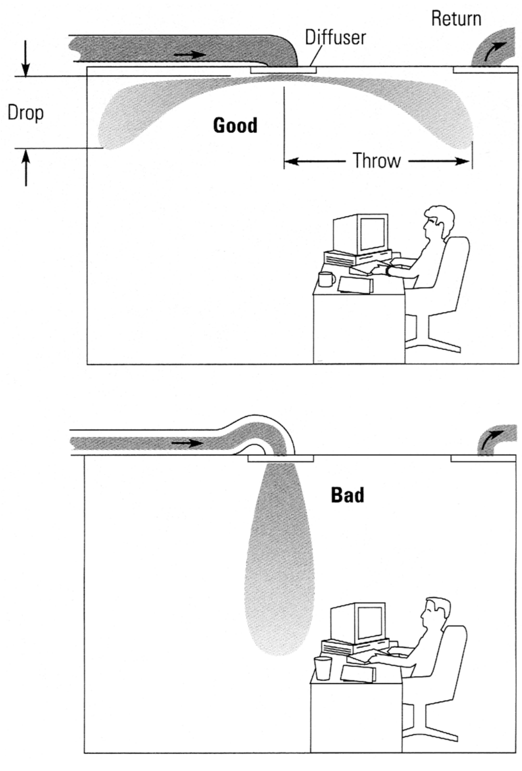


FIGURE 4.3.25 Coanda effect of supply air diffuser (courtesy of E-source, Boulder, CO).

Outlet Fixture Design Procedure

The general procedure for selecting and locating supply-air outlet devices can be summarized as follows.

1. Calculate zone supply air flow rate from design load calculations.
2. Select type and quantity of room outlets by evaluating outlet air flow rate, outlet flow pattern, and building structural characteristics.
3. Locate outlets to provide uniform room temperature through use of a uniform distribution pattern with modifications due to high gain or loss parts of a zone, e.g., windows.
4. Select proper outlet size from manufacturer's literature, based on outlet air flow, discharge velocity and throw, distribution pattern, pressure loss, and sound level requirements.

TABLE 4.3.3 Sidewall, Sill Throw, and Total Pressure for a Slot Diffuser

CFM/FT	Neck Size	2.0 in	2.5 in	3.0 in	4.0 in	5.0 in	6.0 in	8.0 in	10.0 in	12.0 in
100	Total Pressure (iwg)	.19	.081	.063	.032	.020	.013			
	Sidewall Throw (ft.)	20–27	18–26	17–24	15–23	13–22	11–19			
	Sill Throw (ft.)	18–22	16–20	15–18	12–15	9–13	7–10			
200	Total Pressure (iwg)			.25	.13	.080	.052	.026	.017	.012
	Sidewall Throw (ft.)			30–40	27–38	25–36	23–33	20–29	16–25	14–21
	Sill Throw (ft.)			25–32	23–28	20–24	17–22	15–19	13–17	11–14

TABLE 4.3.4 Characteristic Room Length Definitions

Diffuser Type	Characteristic Length (L)
High sidewall grille	Distance to wall perpendicular to jet
Circular ceiling diffuser	Distance to closest wall or intersecting jet
Ceiling slot diffuser	
Perforated, louvered ceiling diffuser	
Sill grille	Length of room in direction of jet flow
Light troffer diffuser	Distance to midplane between outlets plus distance from ceiling to occupied zone

Proper selection and spacing of supply-air outlet devices will reduce the occurrence of excess fluctuation of conditions and maintain comfort within the room. The physiological effect of temperature, humidity, and air motion on the human body is measured in terms of the *effective draft temperature*. The percent of locations in a room where the effective draft temperature is within the comfort range is defined as the air diffusion performance index (ADPI). A higher ADPI means more desirable conditions are achieved. To simplify diffuser specification and completing step 4 above, guidelines have been developed for selecting supply-air outlets based on maximizing ADPI in the building zone.

An understanding of several terms is required to apply the guidelines. The diffuser *throw* is the maximum distance from the outlet device to a point in the air stream where the velocity in the stream cross-section equals a specified terminal velocity. *Sidewall throw* is the horizontal distance from the diffuser. *Sill throw* is the horizontal plus vertical distance from the diffuser mounted in the sill or floor. The *drop* is the vertical distance from the diffuser.

For most devices the *terminal velocity*, V_t, for which the throw is defined is specified as 50 fpm. For ceiling slot-diffusers the value has been set at 100 fpm. The throw distance for a particular terminal velocity is denoted as T_v. The subscript refers to the terminal velocity for which it is defined, such as T_{50}.

The throw of the device is impacted by the flow rate through the device and its neck area. The pressure drop of the device is also impacted by these two parameters. Table 4.3.3 presents sample catalog data for a ceiling-mounted slot diffuser similar to the one depicted in Figure 4.3.24. The neck dimensions are listed as diameter inches. Flow rate is presented in terms of active diffuser length. The total pressure is the static and velocity pressure drop through the device. Sidewall and sill throw minimum and maximum values are based on T_{100} and T_{50}, respectively.

In selecting and spacing diffusers, the throw is compared to the *characteristic room length*, L, the distance from the outlet device in the principal horizontal direction of the air flow to the nearest boundary wall or intersecting air jet. Definitions of characteristic room length for several diffuser types are listed in Table 4.3.4.

Guidelines

Table 4.3.5 recommendations for T_v/L values for room loads ranging from 20–80 Btu/h ft^2 for several types of supply-air outlets. Recommended values are given for meeting a maximum ADPI. Also, a range of T_v/L values is listed that meet a minimum ADPI. To use the selection guidelines, the room dimensions, load, and air volume requirements must be known. With this information available, proceed as follows.

TABLE 4.3.5 ADPI Selection Guide

Terminal Device	Room Load, Btu/(h · ft²)	T_{50}/L for Max. ADPI	Maximum ADPI	For ADPI Greater Than	Range of $T_{0.25}/L$
High	80	1.8	68	—	—
sidewall	60	1.8	72	70	1.5–2.2
grilles	40	1.6	78	70	1.2–2.3
	20	1.5	85	80	1.0–1.9
Circular	80	0.8	76	70	0.7–1.3
ceiling	60	0.8	83	80	0.7–1.2
diffusers	40	0.8	88	80	0.5–1.5
	20	0.8	93	90	0.7–1.3
Sill grille	80	1.7	61	60	1.5–1.7
straight	60	1.7	72	70	1.4–1.7
vanes	40	1.3	86	80	1.2–1.8
	20	0.9	95	90	0.8–1.3
Sill grille	80	0.7	94	90	0.8–1.5
spread	60	0.7	94	80	0.6–1.7
vanes	40	0.7	94	—	—
	20	0.7	94	—	—
Ceiling	80	0.3*	85	80	0.3–0.7
slot diffusers	60	0.3*	88	80	0.3–0.8
(for T_{100}/L)	40	0.3*	91	80	0.3–1.1
	20	0.3*	92	80	0.3–1.5
Light	60	2.5	86	80	<3.8
troffer	40	1.0	92	90	<3.0
diffusers	20	1.0	95	90	<4.5
Perforated and	11–51	2.0	96	90	1.4–2.7
louvered ceiling				80	1.0–3.4
diffusers					

Source: ASHRAE (2001).
* The column value is actual T_{100}/L.

1. Make a preliminary selection of outlet type, number, and location.
2. Determine room characteristic length, L.
3. Select the recommended T_v/L ratio from the table.
4. Calculate the throw distance, T_v.
5. Select the appropriate supply outlet size from manufacturer's literature.
6. Verify that selection meets other design criteria for noise and pressure drop.

Return and Exhaust Air Grilles

Grilles are located at return-air or exhaust-air inlets and consist of a framed set of vertical or horizontal vanes. The vanes control the air flow in the vertical or horizontal plane and may be fixed or adjustable. They return air to the central system or exhaust air to the outside. They ensure proper, unrestricted air flow in the space and maintenance of building pressure. Return-air inlets may be connected to a duct or to another space (often the plenum above the conditioned space). Exhaust-air inlets remove air directly from the building and are always ducted.

In general, the same type of equipment used for outlets can be used for inlets. However flow outlets require accessory devices to deflect, equalize, and turn the air stream from the duct approach to produce a uniform air flow into the outlet device. For the return air, these accessories are not required, but return-air branch ducts should be equipped with volume dampers to balance the air flow. In some applications, exhaust intakes are designed and positioned to remove contaminants or heat directly from the source. Some examples of such applications include laboratory fume hoods and kitchen exhaust canopies.

The major concern with return and exhaust inlet devices is that there be a sufficient number of inlets to maintain inlet velocities within the recommended range. In general, return- and exhaust-air diffusers do not affect the air patterns in the space. However, do not mount return-air inlets close to supply-air outlets or flow short-circuiting will occur.

The location of the return and exhaust inlets can help increase HVAC efficiency. For HVAC systems predominately operating in the cooling mode, improved performance is achieved when heat is removed at the source rather than having it distributed through the space. Due to the nature of some loads, such as solar, it may be difficult to remove them at the source. For lighting loads, mounting return-air inlets near ceiling-mounted fixtures keeps the heat from dispersing into the space.

Design Procedure

The general procedure for selecting and locating inlet devices is as follows.

1. Calculate room return- and exhaust-air flow rates from design load calculations.
2. Select type and quantity of room inlets by evaluating inlet air flow rate, inlet velocity, pressure loss, and sound level.
3. Locate inlets to enhance room air circulation and removal of undesirable loads and contaminants.
4. Select proper inlet size from manufacturer's literature based on inlet airflow, inlet velocity, pressure drop, and sound level.

4.3.7 Air Handling System Control

Air handling systems are sized to meet the design peak capacity. Since the design capacity is needed during only a very small percentage of the year, it is necessary to reduce system capacity to meet the particular setpoints when operating at conditions less than full system capacity. Control systems as applied to air handling systems function to reduce system capacity and meet off-peak loads.

Typically, mechanical designers provide a sequence of operations in the design documents that outline how the system is supposed to function. The sequence of operation is used as a basis for the selection and application of control systems to air handling systems. Once the systems are installed and commissioned, the duty of system operation and maintenance falls upon the building operator.

This section lists the types of control systems normally found in air handling units and discusses air handler sequences of operation and the control philosophies normally used in controlling commonly used air system components and arrangements.

Control System Types

There are three basic types of control systems:

- *Pneumatic control* — systems that use air pressure for sensing system states, such as temperature, and for controlling and actuating devices such as valves and dampers.
- *Electric* — systems that use electric volts or current for system sensing and actuation.
- *Digital/electronic* — systems that use volts or current for system sensing and actuation and digital systems for system programming, sensing, and actuation.

Regardless of the specific type of control system used for a particular system, the system is designed to meet the requirements of the sequence of operations.

Control of the Major Components of Air Handling Systems

Following is an outline of the philosophies used in control of air handling system components.

The fans in CAV systems may operate *intermittently* or *continuously*. With intermittent fan operation or cycling, the system fans run only as needed to meet zone loads. Intermittent operation is commonly found in residential buildings and small commercial buildings. In continuous fan operation, air is

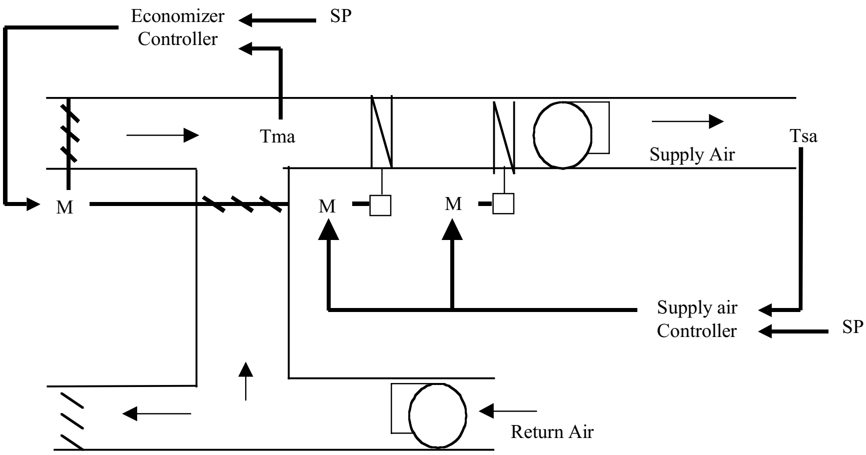


FIGURE 4.3.26 AHU control schematic.

continuously provided to the zones during occupied hours. This type of operation improves indoor air quality since outdoor ventilation air is constantly supplied. Continuous fan operation is ubiquitous in commercial buildings and may be mandatory by code to meet ventilation and other requirements.

Air handling systems used for comfort control are often shut off during unoccupied periods to conserve energy and to reduce system wear due to long run hours. Start/stop control of such systems can be accomplished through the use of time clocks. Clocks can be of the 24-hour, 7-day type, or, in the case of digital or other electronic control systems, be programmed to start or stop a system at any time during an entire year.

When a system is commanded into stop mode, it is always important to protect against potential system damage, such as the freezing of coils. To avoid such situations, outside air dampers will normally be commanded to their fully closed positions, and other steps will be taken to guard against freezing, such as the opening of hot water coil valves when sensed temperatures in critical areas are too close to freezing.

Control of Ventilation and Economizers

Airside economizers, so called to differentiate them from other types of economizers such as waterside economizers, normally consist of a set of dampers that work together to blend return and outside air to maintain a mixed or supply air, or space setpoint. Figure 4.3.26 shows a typical airside economizer control setup.

In this system the controller provides an output signal to modulate the return and outside air dampers to maintain the mixed air temperature at setpoint. Often, an economizer will be designed so that it maintains a supply-air setpoint. Use of the mixed-air control point, however, can be advantageous since it can reduce system hunting and stability problems due to the mass of the heating and cooling coils.

The use of temperature as the controlled variable results in a *dry-bulb economizer.* Another commonly used economizer scheme is the *enthalpy economizer* which is used to maximize the amount of free cooling by using the air stream (return or outside air) that has the lowest enthalpy. Enthalpy economizers can be difficult to control and maintain since they rely on accurate determination of air humidity, a notoriously problematic state to measure.

Ventilation Control

The control of ventilation rates is important for maintaining acceptable indoor air quality while maintaining efficient operation. The maintenance of the minimum ventilation rate is often accomplished through the use of a ventilation damper that is fully open when the system is on, and is fully closed when the system is off. Such a minimum outside air damper is particularly effective in a CAV system, where the system can be balanced during system startup to ensure that a specified quantity of outside air is always introduced into the supply air stream.

In VAV systems, where the flow of supply air is not constant, maintenance of a specified quantity of outside air is more difficult. Several methods to maintain constant ventilation rates while the supply air rate modulates have been used, with varying degrees of success. Some of the methods include

- Supply air/return air fan tracking, where the return-air fan is controlled to move slightly less air than the supply fan, thereby, at least in theory, ensuring a certain quantity of ventilation air.
- Use of mixed air, return air, and outside air temperature sensors to calculate the fraction of outside air entering the supply fan based on first law mixing equations.
- Use of 100% outside air fans (injection fans) to positively deliver a constant quantity of ventilation air into the mixing box. This method works well at the cost of added capital equipment expenditures.
- Use of differential pressure sensors, situated across a fixed orifice. In this configuration, a negative pressure with respect to the outside is maintained in the mixing box, and the outside air damper is modulated to maintain a constant pressure across the fixed orifice.
- Use of CO_2 monitors in the conditioned space or return air ducting to maintain the CO_2 concentrations at acceptable levels.

Control of Air Handling Unit Fans

In CAV systems, fans operate at a constant speed to provide a constant volume of air to and from the conditioned space. In this case, the supply and return fans are balanced during system commissioning to maintain the constant flow rates. The only control, then, of the fans is start/stop control.

In VAV systems, the supply and usually return fans are modulated using variable inlet vanes or variable speed drives to vary the air flow to the required rate. In nearly all cases, the supply fan is modulated to maintain a duct static pressure setpoint at some location in the supply air ducting. As the associated VAV terminal boxes modulate open to provide more cooling to the spaces, the pressure in the supply air ducting will decrease. In response to the decreased duct pressure, the control system will increase the air flow into the ducting to maintain the pressure setpoint. Conversely, as the associated VAV boxes close to reduce cooling to the spaces, the duct pressure will tend to increase, causing the control system to reduce the air flow rate to maintain the static pressure setpoint.

Control of Main Air Handling Unit Heating and Cooling Coils

The coils in the air handling units are modulated to maintain temperature and, in some cases, humidity setpoints. For air handlers that serve multiple spaces, the most common approach is to modulate the coil capacity to maintain a supply-air temperature setpoint. The controller modulates a control valve in the case of steam, hot water, and chilled water cooling coils. The controller modulates or stages the source in systems with vapor-compression air conditioning, a furnace, or electric resistance heating coils. Figure 4.3.26 shows a control schematic for the maintenance of a supply-air setpoint in a central air handling unit with hot water heating coils and chilled water cooling coils. As shown in the figure, the controller receives a signal of the supply air temperature, compares it to the setpoint value, and sends the appropriate signal to adjust the coil valve position.

Control of Terminal Units

VAV terminal units modulate the supply air volume, the supply air temperature, or both, delivered to the conditioned space to maintain the space setpoint temperature. Two basic types of local control loops are commonly used on VAV terminal units. One type results in *pressure dependent* operation, and the other results in *pressure independent* operation.

Terminal units where the space thermostat controls the primary damper directly, as shown in Figure 4.3.27, are pressure dependent. In the figure, T_z is the actual zone temperature, *setpoint* is the desired zone temperature, *C* is the local controller, and *DM* is the air damper motor. In this arrangement, the controller output directly operates the air damper actuator. Pressure dependent boxes are so called because the flow rate of the primary air through the terminal depends on the static air pressure at the inlet of the unit and the primary air damper position.

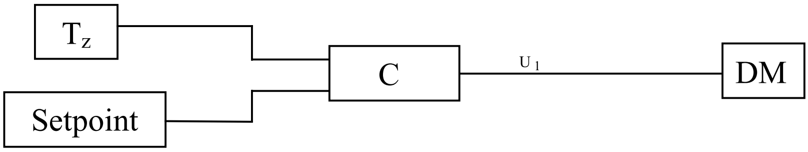


FIGURE 4.3.27 Pressure dependent terminal box local control system arrangement.

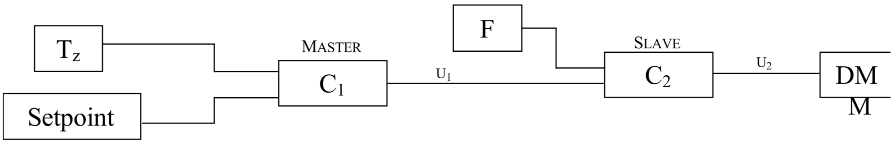


FIGURE 4.3.28 Pressure independent terminal unit control system arrangement.

Pressure dependent VAV terminal units are usually less expensive than pressure independent terminal units because they have simpler control systems and monitoring equipment.

To achieve pressure independence, the control system is arranged in a master/slave configuration, as shown in Figure 4.3.28, where:

T_z is the zone temperature
Setpoint is the desired zone temperature
C_1 is the master controller that resets the flow rate setpoint
F is the measured air flow of the primary air stream
C_2 is the slave controller modulating to maintain the setpoint flow rate from C_1
DM is the primary air damper position

The master-slave control system arrangement results in pressure independence because the flow rate into the zone is directly controlled with the slave controller, so it is not a function of the pressure at the inlet of the terminal unit. Pressure independent terminal units are more stable under varying pressure conditions but are more expensive to purchase and maintain because of the need for a flow measurement device (F) and a second controller.

Sequences of Operation for Air Handing Systems

Using the fundamental control schemes outlined above, a sequence of control operations that governs the control of the entire air handling system can be developed. The sequence of operation describes the method of control, control points, control circuits, control sequence, control features, and operation. Specific procedures for the following operations are generally included:

- Start-up
- Fan-speed control
- Supply- and mixed-air temperature control
- Room-air temperature control
- Equipment safety interlocks

The start-up describes the manual and automatic methods to start the system fans and the sequence of control contacts used to power other system components. Fan speed is actively controlled in VAV systems. Thus, the sequence of operation for VAV systems includes a section on fan speed control. Equipment safety interlocks include safety switches, limit switches, and smoke detection. A sequence of operation for a number of different HVAC systems is presented in Appendix G of *Air-Conditioning Systems Design Manual* (ASHRAE 1993).

4.3.8 Secondary Air System Design

The previous sections of this chapter outline the various components found in typical air systems. Selection of each component and integrating it into a working system that satisfies all design requirements and parameters is the designer's task. Following is a step-by-step process that can be used in system design.

1. Define system requirements and design conditions.
For general HVAC systems, the requirements include the following items:

- Inside air temperature and humidity based on comfort requirements as outlined in ASHRAE Standard 55 — *Thermal Environmental Conditions for Human Occupancy* (ASHRAE 1992a).
- Design values for outside air temperature and humidity.
- Design sensible and latent loads for each conditioned zone as determined from the building load analysis.
- Ventilation requirements as outlined in ASHRAE Standard 62.1 – *Ventilation for Acceptable Indoor Air Quality* (ASHRAE 1989). Ventilation requirements are usually driven by ventilation standards for the maintenance of indoor air quality but may be based on pressurization requirements, exhaust makeup air requirements, or other needs.
- Noise criteria.
- Space and system configurations requirements.

2. Select a supply air temperature and calculate the zone design flow rates.
A value of 55°F is typical (lower temperatures usually result in more efficient system operation and lower energy costs).

3. Select and arrange diffusers and return air grilles in each space.
Select number, spacing, and layout to provide adequate flow, ventilation, and air movement. In VAV systems, be sure to consider both design conditions and low flow rates when selecting the diffusers.

4. Select terminal units.
If designing a system that serves multiple zones, terminal units should be selected so they supply the design amount of air at or near their full open positions. Each unit should allow for adequate turn-down for off-peak conditions. To avoid circulation and ventilation problems, avoid sizing VAV terminal units to supply less than 0.6 CFM/ft^2 in the minimum damper position.

5. Layout and size duct work.
Start duct layout from the terminal units to the diffusers, then from the air handling unit(s) to the terminal units. Be sure to consider duct air leakage, heat loss, and noise in sizing the ducts. Provide adequate provisions for test and balance. Place fire and smoke dampers where appropriate.

6. Size the main air handling unit and components.

- Select heating coils, cooling coils, humidification devices. The coils must be adequate to offset the load arising from the space, the outside ventilation air, and the gains or losses in the supply and return air paths.
- Size mixing box and dampers. Include return-air dampers, mixed-air dampers, outside-air dampers. Dampers must be sized to provide good control of the air streams without causing too much pressure drop.
- Select and arrange filtration systems for the application.
- Select the fans to offset the pressure losses in each of the system components.

Because of duct air leakage, the amount of supply air leaving the air handler will be less than the amount of air introduced to the zones. This should be accounted for in sizing the air handling unit components or measures should be taken to seal all duct circuits.

7. Provide for mounting and maintenance.
Be mindful of weight and clearances. Be sure to provide access for component replacement, maintenance, and cleaning.

8. Prepare a sequence of operations.

9. Generate plans and specifications.

4.3.9 Air System Commissioning and Operation

After an air secondary system is designed and installed, it is particularly important that the system be commissioned to ensure its effective operation. Commissioning involves identifying building system equipment, control, and operational problems and fixing them so that the building performs according to the design intent. To ensure efficient system operation and long life, it is also important that the system operators and service technicians be trained to have a strong fundamental understanding of the system they are charged with operating.

Commissioning is discussed in Chapter 7.1.

Definition of Terms

Airside economizer: An air system control option that maximizes the use of outdoor air for cooling. The economizer consists of dampers, temperature and humidity sensors, actuators, and controls.

Coanda effect: The diversion of an air stream from its normal flow path due to its attachment to an adjacent surface (such as a ceiling or wall). The effect results from a low pressure region between the fluid and the surface. In supply air streams, the effect prevents cold air from dropping in a narrow column into the space.

Constant air volume system: CAV systems supply a constant flow rate of air when the fans are on. Single duct or dual duct and single zone or multiple zone systems may be constant volume systems. Single duct CAV systems commonly include terminal reheat.

Diffuser: An air distribution system outlet comprised of deflecting members designed to discharge air in various directions and promote mixing of primary air with secondary air.

Direct expansion (DX) coils: The evaporator coil in a refrigeration compression cycle. The refrigerant, the primary working fluid, absorbs heat from the secondary fluid and changes phase from liquid to gas.

Drop: The vertical distance between the supply air outlet and the lower edge of a horizontally projected air stream at the end of its throw.

Dual duct system: One type of a category of systems that supplies heating and cooling in separate ducts to multiple zones. In this type, the warm and cool air streams are mixed close to the zone served as compared to multizone systems that mix the two air streams close to the central supply fan.

Effectiveness: A measure of heat exchanger efficiency that equals the ratio of the actual amount of heat transferred to the maximum heat transfer possible between the fluid streams. The theoretical maximum is dependent on the fluids' entering state (i.e., temperature) and heat capacity.

Entrainment: The capture of surrounding air (secondary air) by the supply air stream (primary air) as it is discharged from a zone diffuser.

Evaporative cooling: The adiabatic exchange of heat between an air stream and a wetted surface or water spray. Sensitive cooling of the air stream occurs as it becomes saturated and approaches the wet-bulb temperature. It is an effective cooling method for dry climates.

Grille: A louvered or perforated device for air passage which can be located on the ceiling, wall, or floor.

Multizone system: One type of a category of systems that supplies heating and cooling in separate ducts to multiple zones. In this type, the warm and cool air streams are mixed close to the central supply fan as compared to dual duct systems that mix the two air streams close to the zone served. This system type is poorly named since many systems serve multizones although they are not necessarily *multizone* systems.

Pressure dependent terminal boxes: A type of VAV terminal box where the space thermostat controls the box damper position directly. For a given damper position, the actual flow rate through the box is system pressure dependent.

Pressure independent terminal boxes: A type of VAV terminal box where the space thermostat and a supply air flow sensor control the box damper position. The flow sensor compensates for changes in system pressure making the box pressure independent.

Register: A grille equipped with a movable damper to control the direction of flow and/or volume of flow.

Sensitive heat ratio (SHR): The ratio of sensible cooling to total cooling in an air cooler. The SHR establishes a line of constant slope on a psychrometric chart for a graphical solution to moist air cooling analysis.

System effect: An increase in system pressure drop resulting from the ducting configuration entering and exiting the air-handling unit fan. Abrupt duct transitions can result in unintentional air twirling that negatively impacts fan performance.

System curve: A graphic presentation showing the relationship between air flow rate and pressure drop for a particular system.

Throw: Upon leaving a supply air outlet, the distance the maximum velocity air stream travels before being reduced to a specified terminal velocity, usually defined as 50 or 100 feet per minute.

Terminal velocity: The maximum velocity of an air stream leaving a supply air outlet at the end of its throw.

Variable air volume systems: VAV systems modulate the flow of air supplied to the zones. Single duct or multiple duct multiple zone systems may be VAV systems.

Zone: A building thermal space that has comfort conditions controlled by a single thermostat.

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Additional Information

Information regarding the design requirements of HVAC systems can be found in the following resources. For comfort, refer to ASHRAE Standard 55 — *Thermal Environmental Conditions for Human Occupancy* (ASHRAE 1992a). For design values of outdoor design conditions for cities in the U.S. and worldwide, refer to values published in *ASHRAE Fundamentals*, Chapter 24 (ASHRAE 1997). Outdoor air ventilation requirements are outlined in ASHRAE Standard 62.1 — *Ventilation for Acceptable Indoor Air Quality* (ASHRAE 1989). For a sample sequence of control operations written for several different types of HVAC systems, refer to Appendix G of *Air-Conditioning Systems Design Manual* (ASHRAE 1993).

Industrial Ventilation: A Manual of Recommended Practice (ACGIH 1998) provides an excellent resource outlining the application of ventilation systems in industrial settings. *HVAC Systems Duct Design* (SMACNA 1990) and *HVAC Duct Construction Standards* (SMACNA 1995) provide detailed design information on duct construction, installation methods, design, and application for all common systems in use today.

4.4 Electrical Systems

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This chapter outlines some general concepts related to electric systems for HVAC applications in buildings. First, a review of basic characteristics of an electric system operating under alternating current is provided. Then, electric equipment commonly used in HVAC systems is described. Finally, design procedures for electrical distribution systems specific to motors are illustrated step-by-step with specific examples. Throughout this chapter, several measures of improving the energy efficiency of electrical systems are provided. Moreover, simplified calculation methods are presented to evaluate of the cost effectiveness of the proposed energy efficiency measures.

In most buildings and industrial facilities, electric systems consume a significant part of the total energy use. Table 4.4.1 compares electricity consumption in three sectors (residential, commercial, and industrial) for both the U.S. and France, which is representative of most western European countries. It is clear that in the U.S. electric energy is used more significantly in commercial and residential buildings than in the industrial facilities where fossil fuels (such as coal, oil, and natural gas) are typically used.

For residential buildings, lighting and heating, ventilating, and air conditioning (HVAC) each account for approximately 20% of total U.S. electricity use. Refrigerators represent another important energy end-use in the residential sector with about 16% of electricity. For the commercial sector as a whole, lighting accounts for over 40% while HVAC accounts for only 11% of the total electricity use. However, for commercial buildings with space conditioning, HVAC is one of the major electricity end-uses and can be more energy intensive than lighting. Moreover, computers and other office equipment (such as printers, copiers, and facsimile machines) are becoming an important electric energy end-use in office buildings.

To ensure that all electric equipment operate safely, it is important to design a reliable distribution system. In the U.S., the National Electric Code (NEC) provides specific requirements for a safe design of electrical installations. For buildings, a typical electrical installation includes the following equipment:

- A unit substation with a transformer to step down the voltage
- A set of lighting panelboards and motor control centers that house circuit breakers, fuses, disconnect switches, and overload loads
- A set of wiring distribution systems including feeders and branch circuits consisting of electrical conductors and conduits

To properly design an electric installation, it is important to first estimate the load associated with all the electric utilization equipment, including lighting fixtures, appliances, and motors.

TABLE 4.4.1 Percentage Share of Electricity in Total Energy Use in Three Sectors for U.S.[a] and France[b]

Sector	U.S.	France
Residential buildings	61%	52%
Commercial buildings	52%	68%
Industrial facilities	12%	52%

[a] *Source:* Office of Technology Assessment (1995).
[b] *Source*: Electricité de France (1997).

4.4.1 Review of Basics

Alternating Current Systems

For a linear electrical system subject to an alternating current (AC), the time variation of the voltage and current can be represented as a sine function:

$$v(t) = V_m \cos\omega t \tag{4.4.1}$$

$$i(t) = I_m \cos(\omega t - \phi) \tag{4.4.2}$$

where

V_m and I_m are the maximum instantaneous values of voltage and current, respectively. These maximum values are related to the effective or root mean square (rms) values as follow:

$$V_m = \sqrt{2} * V_{rms} = 1.41 * V_{rms}$$

$$I_m = \sqrt{2} * I_{rms} = 1.41 * I_{rms}$$

In the U.S., the values of V_{rms} are typically 120 V for residential buildings or plug-load in the commercial buildings, 277 V for lighting systems in commercial buildings, and 480 V for motor loads in commercial and industrial buildings. Higher voltages can be used for certain industrial applications.

ω is the angular frequency of the alternating current and is related to the frequency f as follows:

$$\omega = 2\pi f$$

In the U.S., the frequency f is 60 Hz, that is 60 pulsations in one second. In other countries, the frequency of the alternating current is $f = 50$ Hz.

ϕ is the phase lag between the current and the voltage. In this case, the electrical system is a resistance (such as incandescent lamp), the phase lag is zero, and the current is on phase with the voltage. If the electrical system consists of a capacitance load (such as a capacitor or a synchronous motor), the phase lag is negative and the current is in advance relative to the voltage. Finally, when the electrical system is dominated by an inductive load (such as a fluorescent fixture or an induction motor), the phase lag is positive and the current lags the voltage.

Figure 4.4.1 illustrates the time variation of the voltage for a typical electric system. The concept of root mean square (also called effective value) for the voltage, V_{rms}, is also indicated in Figure 4.4.1. It should be noted that the cycle for the voltage waveform repeats itself every 1/60 s (since the frequency is 60 Hz).

The instantaneous power, $p(t)$, consumed by the electrical system operated on one-phase AC power supply can be calculated using Ohm's law:

$$p(t) = v(t) \cdot i(t) = V_m I_m \cos\omega t \cdot \cos(\omega t - \phi) \tag{4.4.3}$$

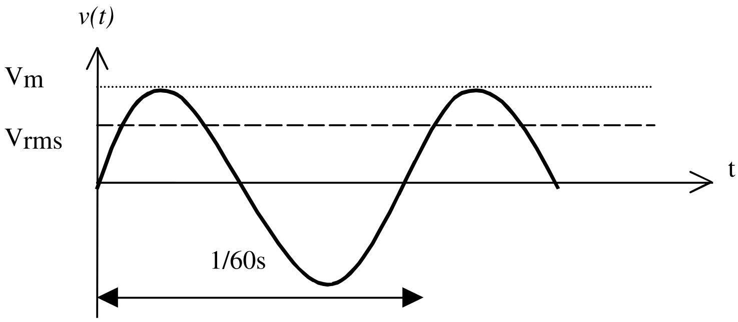

FIGURE 4.4.1 Illustration of the voltage waveform and the concept of V_{rms}.

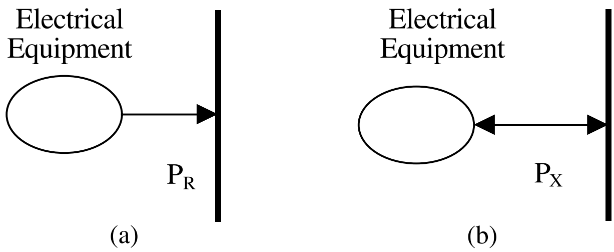

FIGURE 4.4.2 Illustration of the direction of electricity flow for (a) real power, and (b) reactive power.

The above equation can be rearranged using some basic trigonometry and the definition of the rms values for voltage and current:

$$p(t) = V_{rms} \cdot I_{rms}(\cos\phi \cdot (1 + \cos 2\omega t) + \sin\phi \cdot \sin 2\omega t) \tag{4.4.4}$$

Two types of power can be introduced as a function of the phase lag angle ϕ: the real power P_R and the reactive power P_X as defined below:

$$P_R = V_{rms} \cdot I_{rms}\cos\phi \tag{4.4.5}$$

$$P_X = V_{rms} \cdot I_{rms}\sin\phi \tag{4.4.6}$$

Note that both types of power are constant and are not a function of time. To help understand the meaning of each power, it is useful to note that the average of the instantaneous power consumed by the electrical system over one period is equal to P_R:

$$\bar{P} = \frac{1}{T}\int_0^T p(t)dt = P_R \tag{4.4.7}$$

Therefore, P_R is the actual power consumed by the electrical system over its operation period (which consists typically of a large number of periods T). P_R is typically called real power and is measured in kW. Meanwhile, P_X is the power required to produce a magnetic field to operate the electrical system (such as induction motors) and is stored and then released; this power is typically called reactive power and is measured in kVAR. A schematic is provided in Figure 4.4.2 to help illustrate the meaning of each type of power.

While the user of the electrical system actually consumes only the real power, the utility or the electricity provider has to make available to the user both the real and reactive power. The algebraic sum of P_R and

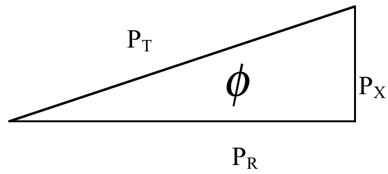

FIGURE 4.4.3 Power triangle for an electrical system.

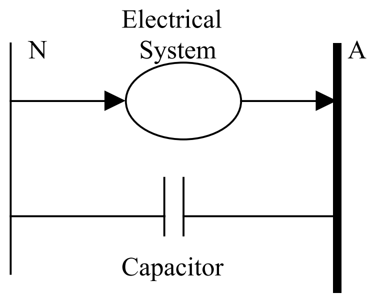

FIGURE 4.4.4 The addition of a capacitor can improve the power factor of an electrical system.

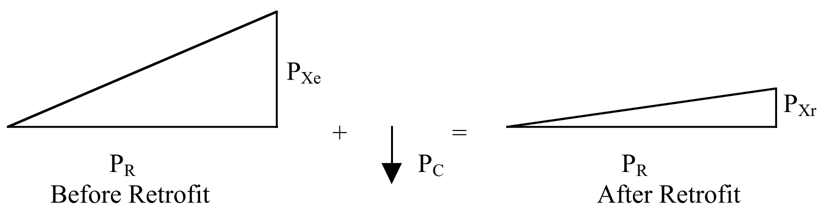

FIGURE 4.4.5 Effect of adding capacitors on the power triangle of the electrical system.

P_X constitutes the total power, P_T. Therefore, the utility has to know, in addition to the real power needed by the customer, the magnitude of the reactive power, and thus the total power.

As mentioned earlier, for a resistive electrical system, the phase lag is zero, and, thus, the reactive power is also zero (see Equation 4.4.6). Unfortunately, for commercial buildings and industrial facilities, the electrical systems are not often resistive and the reactive power can be significant. In fact, the higher the phase lag angle ϕ, the more important the reactive power P_X. To illustrate the importance of the reactive power relative to the real power P_R and the total power P_T consumed by the electrical system, a power triangle is used to represent the power flow, as shown in Figure 4.4.3.

Power Factor Improvement

As mentioned in the previous section, the reactive power must be supplied by the utility even though it is not actually registered by the power meter (as real power used). The magnitude of this reactive power increases as the power factor decreases. To account for the loss of energy due to the reactive power, most utilities have established rate structures that penalize any user who has a low power factor. Therefore, a significant savings in utility costs can be achieved by improving the power factor. As illustrated in Figure 4.4.4, this power factor improvement can be obtained by adding a set of capacitors to the entire electrical system. The size of these capacitors, P_C, is typically measured in kVAR (the same unit as the reactive power) and can be determined as indicated in Figure 4.4.5 using the power triangle analysis:

$$P_C = P_{Xe} - P_{Xr} = P_R \cdot (\tan\phi_e - \tan\phi_r) \tag{4.4.8}$$

where,

P_{Xe} and P_{Xr} are reactive power, respectively, before retrofit (existing conditions) and after retrofit (retrofitted conditions).
P_C is the reactive power of the capacitor to be added.
ϕ_e and ϕ_r are phase lag angle, respectively, before retrofit (existing conditions) and after retrofit (retrofitted conditions).

Using the values pf_e of power factor before and after Pf_r the retrofit, the size of the capacitors can be determined:

$$P_C = P_R \cdot [\tan(\cos^{-1} pf_e) - \tan(\cos^{-1} pf_r) \tag{4.4.9}$$

The calculations of the cost savings due to power factor improvement depend on the utility rate structure. In most rate structures, one of three options, summarized below, is used to assess the penalty for low power factor. Basic calculation procedures are typically needed to estimate the annual cost savings in the utility bills:

- *Modified billing demand:* In this case, the demand charges are increased in proportion to a fraction by which the power factor is less than a threshold value. The size for the capacitors should be selected so the system power factor reaches at least the defined threshold value.
- *Reactive power charges:* In this rate, charges for reactive power demand are included as part of the utility bills. In this option, the size of the capacitors should ideally be determined to eliminate this reactive power (i.e., so that the power factor is unity).
- *Total power charges:* This rate is similar to the rate described above but the charges are set for the building/facility total power. Again, the capacitors should be sized so the power factor is equal to unity.

The calculations of the cost savings due to power factor improvement are illustrated in Example 4.4.1.

Example 4.4.1

Problem: Consider a building with a total real power demand of 500 KW with a power factor of $pf_e = 0.70$. Determine the required size of a set of capacitors to be installed in parallel with the building service entrance so that the power factor becomes at least $pf_r = 0.90$.

Solution

The size in kVAR of the capacitor is determined using Equation 4.4.9:

$$P_C = 500 \cdot [\tan(\cos^{-1} 0.70) - \tan(\cos^{-1} 0.90) = 268kVAR$$

Thus a capacitor rated at 275 kVAR can be selected to ensure a power factor of the building electrical system of 0.90.

4.4.2 Electrical Motors

In the U.S., there were 125 million operating motors in the range of 1 to 120 hp in 1991. These motors consumed approximately 55% of the electricity generated in the U.S. (Andreas, 1992). In large industrial facilities, motors can account for as much as 90% of the total electrical energy use. In commercial buildings, motors can account for more than 50% of the building electrical load.

Motors convert electrical energy to mechanical energy and are typically used to drive machines. The driven machines serve several purposes in the building, including moving air (supply and exhaust fans), moving liquids (pumps), moving objects or people (conveyors, elevators), compressing gases (air compressors, refrigerators), and producing materials (production equipment). To select the type of motor to be used for a particular application, several factors have to be considered, including

- The form of electrical energy that can be delivered to the motor: direct current (DC) or alternating current (AC), single or three phase.
- The requirements of the driven machine, such as motor speed and load cycles.
- The environment in which the motor is to operate: normal (where a motor with an open-type ventilated enclosure can be used), hostile (where a totally enclosed motor must be used to prevent outdoor air from infiltrating the motor), or hazardous (where a motor with an explosion-proof enclosure must be used to prevent fires and explosions).

The basic operation and the general characteristics of AC motors are discussed in the following sections. In addition, simple measures are described to improve the energy efficiency of existing motors.

Overview of Electrical Motors

There are basically two types of electric motors used in buildings and industrial facilities: (1) induction motors and (2) synchronous motors. Induction motors are the more common type, accounting for about 90% of the existing motor horsepower. Both types use a motionless stator and a spinning rotor to convert electrical energy into mechanical power. The operation of both types of motor is relatively simple and is briefly described below.

Alternating current is applied to the stator, which produces a rotating magnetic field in the stator. A magnetic field is also created in the rotor. This magnetic field causes the rotor to spin in trying to align with the rotating stator magnetic field. The rotation of the magnetic field of the stator has an angular speed that is a function of both the number of poles, N_P, and the frequency, f, of the AC current, as expressed in Equation 4.4.10:

$$\omega_{mag} = \frac{4\pi \cdot f}{N_P} \tag{4.4.10}$$

The above expression is especially useful when we discuss the use of variable frequency drives for motors with variable loads.

One main difference between the two motor types (synchronous and induction) is how the rotor field is produced. In an induction motor, the rotating stator magnetic field induces a current, thus a magnetic field, in the rotor windings which are typically of the squirrel-cage type. Because its magnetic field is induced, the rotor cannot rotate with the same speed as the stator field (if the rotor spins with the same speed as the stator magnetic field, no current can be induced in the rotor since the stator magnetic field remains unchanged relative to the rotor). The difference between the rotor speed and the stator magnetic field rotation is called the *slip factor.*

In a synchronous motor, the rotor field is produced by application of direct current through the rotor windings. Therefore, the rotor spins at the same speed as the rotating magnetic field of the stator, and, thus, the rotor and the stator magnetic field are synchronous in their speed.

Because of their construction characteristics, the induction motor is basically an inductive load and thus has a lagging power factor, while the synchronous motor can be set so it has a leading power factor (i.e., acts like a capacitor). Therefore, it is important to remember that a synchronous motor can be installed both to provide mechanical power and to improve the power factor of a set of induction motors. This option may be more cost-effective than just adding a bank of capacitors.

Three parameters are important to characterize an electric motor during full-load operation:

- The mechanical power output of the motor, P_M. This power can be expressed in kW or horsepower (hp). The mechanical power is generally the most important parameter in selecting a motor.
- The conversion efficiency of the motor, η_M. This efficiency expresses the mechanical power as a fraction of the real electric power consumed by the motor. Due to various losses (such as friction,

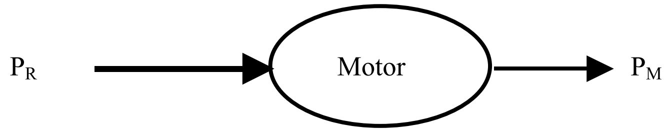

FIGURE 4.4.6 Definition of the efficiency of a motor.

core losses due to the alternating of the magnetic field, and resistive losses through the windings), the motor efficiency is always less than 100%. Typical motor efficiencies range from 75 to 95% depending on the size of the motor.

- The power factor of the motor, pf_M. As indicated earlier in this chapter, the power factor allows the estimation of the reactive power needed by the motor.

Using the schematic diagram of Figure 4.4.6, the real power used by the motor can be calculated as follows:

$$P_R = \frac{1}{\eta_M} \cdot P_M \tag{4.4.11}$$

Therefore, the total power and the reactive power needed to operate the motor are, respectively,

$$P_T = \frac{P_R}{pf_M} = \frac{1}{pf_M \cdot \eta_M} \cdot P_M \tag{4.4.12}$$

$$P_X = P_R \tan\phi = \frac{1}{\eta_M} \cdot P_M \cdot \tan(\cos^{-1} pf_M) \tag{4.4.13}$$

Energy Efficient Motors

General Description

Based on their efficiency, motors can be classified into two categories: (1) standard efficiency motors, and (2) high or premium efficiency (i.e., energy efficient) motors. The energy efficient motors are 2 to 10 percentage points more efficient than standard efficiency motors, depending on the size. Table 4.4.2 summarizes the average efficiencies for both standard and energy efficient motors that are currently available commercially. The improved efficiency for the high or premium motors are mainly due to better design with use of better materials to reduce losses. However, this efficiency improvement comes with a higher price of about 10 to 30% more than standard efficiency motors. These higher prices are partially the reason that only one fifth of the motors sold in the U.S. are energy efficient.

However, the installation of premium efficiency motors is becoming a common method of improving the overall energy efficiency of buildings. The potential for energy savings from premium efficiency motor retrofits is significant. In the U.S. alone, it was estimated that replacing the 125 million operating motors (in the range of 1 to 120 hp) with premium efficiency models would save approximately 60 Thw of energy per year (Nadel et al., 1991).

To determine the cost effectiveness of motor retrofits, there are several tools available, including the MotorMaster developed by the Washington State Energy Office (WSEO, 1992). These tools have the advantage of providing large databases for cost and performance information for various motor types and sizes.

Adjustable Speed Drives (ASDs)

With more emphasis on energy efficiency, an increasing number of designers and engineers are recommending the use of variable speed motors for various HVAC systems. Indeed, the use of adjustable speed

TABLE 4.4.2 Typical Motor Efficiencies

Motor mechanical power output kW (hp)	Average nominal efficiency for standard efficiency motor	Average nominal efficiency for premium efficiency motor
0.75 (1.0)	0.730	0.830
1.12 (1.5)	0.750	0.830
1.50 (2.0)	0.770	0.830
2.25 (3.0)	0.800	0.865
3.73 (5.0)	0.820	0.876
5.60 (7.5)	0.840	0.885
7.46 (10)	0.850	0.896
11.20 (15)	0.860	0.910
14.92 (20)	0.875	0.916
18.65 (25)	0.880	0.926
22.38 (30)	0.885	0.928
29.84 (40)	0.895	0.930
37.30 (50)	0.900	0.932
44.76 (60)	0.905	0.933
55.95 (75)	0.910	0.935
74.60 (100)	0.915	0.940
93.25 (125)	0.920	0.942
111.9 (150)	0.925	0.946
149.2 (200)	0.930	0.953

Source: Adapted from Hoshide, 1994.

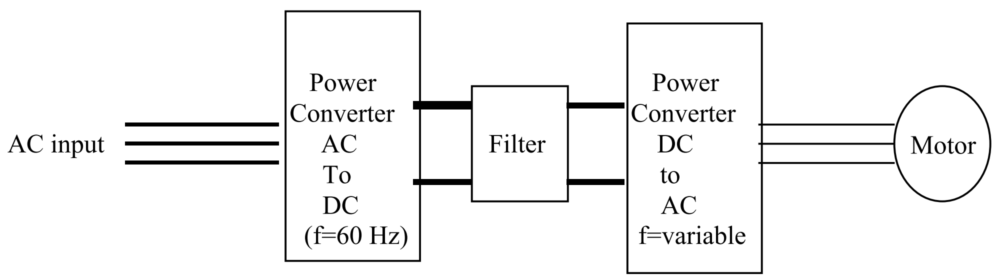

FIGURE 4.4.7 Basic concept of ASD power inverter.

drives (ASDs) is now becoming common, especially for supply and return fans in variable air volume (VAV) systems and for hot and chilled water pumps in central heating and cooling plants.

Electronic ASDs convert the fixed-frequency AC power supply (50 or 60 Hz) first to a DC supply and then to a variable frequency AC power supply, as illustrated in Figure 4.4.7. Therefore, the ASDs can change the speed of AC motors with no moving parts presenting high reliability and low maintenance requirements.

In order to achieve the energy savings potential for any HVAC application, the engineer needs to know the actual efficiency of the motors. For ASD applications, it is important to separate the losses between the drive and the motor to achieve an optimized HVAC system with the lowest operating cost. Specifically, the engineer would need to know the loss distribution, including the iron losses, copper losses, friction and windage losses, and the distribution of losses between stator and rotor. To determine these losses and thus the motor efficiency, accurate measurements are needed. Unfortunately, existent power measurement instruments are more suitable for sinusoidal rather than distorted waveforms (which are typical for ASD applications).

The use of low-cost solid state power devices and integrated circuits to control speed have made most of the commercially available ASDs draw power with extremely high harmonic content. Some investigators (Filipski and Arseneau 1990; Domijan et al. 1995; Czarkowski and Domijan, 1997) have studied the behavior of commercially available power measurement instruments subjected to voltage and current waveforms typical of ASD motor connections. In particular, three drive technologies used in HVAC industry were investigated by Czarkowski and Domijan (1997) as part of ASHRAE project 770-RP (Domijan et al., 1996), namely PWM induction, switched reluctance, and brushless DC drives. The main finding of their investigation is that existing power instruments fail to accurately measure power losses (and thus motor efficiency) due to the high harmonic content in the voltage spectra, especially for brushless DC and switched reluctance motors which represent a substantial portion of the HVAC market.

Typically, the motor losses were measured indirectly by monitoring the power input and the power output then by taking the difference. This traditional approach requires an extremely high accuracy for the power measurements to achieve "reasonable" estimation of motor losses, especially for premium efficiency motors (with low losses). However, a new approach to measure the ASD motor losses *directly* has been proposed (Fuchs and Fei, 1996).

Energy Savings Calculations

There are three methods to calculate the energy savings due to energy efficient motor replacement. These three methods are outlined below.

Method 1: ***Simplified Method***

This method has been and is still used by most energy engineers to determine the energy and cost savings incurred by motor replacement. Inherent to this method, two assumptions are made: (1) the motor is fully loaded, and (2) the change in motor speed is neglected.

The electric power savings due to the motor replacement is computed as follows:

$$\Delta P_R = P_M \cdot \left(\frac{1}{\eta_e} - \frac{1}{\eta_r}\right) \tag{4.4.14}$$

Where,

P_M is the mechanical power output of the motor.
η_e is the design (i.e., full-load) efficiency of the existing motor (e.g., before retrofit).
η_r is the design (i.e., full-load) efficiency of the energy-efficient motor (e.g., after retrofit).

The electric energy savings incurred from the motor replacement is thus

$$\Delta kwh = \Delta P_R \cdot N_h \cdot LF_M \tag{4.4.15}$$

where,

N_h is the number of hours per year during which the motor is operating.
LF_M is the load factor of the motor's operation during one year.

Method 2: ***Mechanical Power Rating Method***

In this method, the electrical peak demand of the existing motor is assumed to be proportional to its average mechanical power output:

$$P_{R,e} = \frac{P_M}{\eta_{op,e}} \cdot LF_{M,e} \cdot PDF_{M,e} \tag{4.4.16}$$

where,

$\eta_{op,e}$ is the motor efficiency at the operating average part-load conditions. To obtain this value, the efficiency curve for the motor can be used. If the efficiency curve for the specific existing motor is not available, a generic curve can be used.

$LF_{M,e}$ is the load factor of the existing motor and is the ratio between the average operating load of the existing motor and its rated mechanical power. In most applications, the motor is oversized and operates at less than its capacity.

$PDF_{M,e}$ is the peak demand factor and represents the fraction of typical motor load that occurs at the time of the building peak demand. In most applications, $PDF_{M,e}$ can be assumed to be unity since the motors often contribute to the total peak demand of the building.

Since the mechanical load does not change after installing an energy efficient motor, it is possible to consider a smaller motor with a capacity $P_{M,r}$ if the existing motor is oversized with a rating of $P_{M,e}$. In this case, the smaller energy efficient motor can operate at a higher load factor than the existing motor. The new load factor, LF_r, of the energy efficient motor can be calculated as follows:

$$LF_r = LF_e \cdot \frac{P_{M,r}}{P_{M,e}} \tag{4.4.17}$$

Moreover, the energy efficient motors often operate at a higher speed than the standard motors they replace since they have lower internal losses. This higher speed actually has a negative impact since it reduces the effective efficiency of the energy efficient motor by a factor called the slip penalty. The slip penalty factor, $SLIP_P$, is defined as shown in Equation 4.4.18:

$$SLIP_P = \left(\frac{\omega_{M,r}}{\omega_{M,e}}\right)^3 \tag{4.4.18}$$

where,

$\omega_{M,e}$ is the rotation speed of the existing motor

$\omega_{M,r}$ is the rotation speed of the energy-efficient motor

Using an equation similar to Equation 4.4.16, the peak electrical demand for the retrofitted motor (e.g., energy efficient motor) can be determined:

$$P_{R,r} = \frac{P_{M,r}}{\eta_{op,r}} \cdot LF_{M,r} \cdot PDF_{M,r} \cdot SLIP_P \tag{4.4.19}$$

The electrical power savings due to the motor replacement can thus be estimated:

$$\Delta P_R = P_{R,e} - P_{R,r} \tag{4.4.20}$$

The electric energy savings can be therefore calculated using Equation 4.4.15.

Method 3: *Field Measurement Method*

In this method, the motor electrical power demand is measured directly on-site. Typically, current, I_M, voltage, V_M, and power factor, pf_M, readings are recorded for the existing motor to be retrofitted. For three-phase motors (which are common in industrial facilities and in most HVAC systems for commercial buildings), the electrical power used by the existing motor can be either directly measured or calculated from current, voltage, and power factor readings as follows:

$$P_{R,E} = \sqrt{3} \cdot V_M \cdot I_M \cdot pf_M \tag{4.4.21}$$

The load factor of the existing motor can be estimated by taking the ratio of the measured current over the nameplate full-load current, I_{FL}, as expressed in Equation 4.4.22:

$$LF_{M,E} = \frac{I_M}{I_{FL}} \tag{4.4.22}$$

TABLE 4.4.3 Level of Participation in Lighting Conservation Programs by U.S. Commercial Buildings

Lighting Retrofit	Percent Participation in Number of Buildings	Percent Participation in Floor Area of Spaces
Energy efficient lamps and ballasts	31	49
Specular reflectors	18	32
Time clock	10	23
Manual dimmer switches	10	23
Natural lighting control sensors	7	13
Occupancy sensors	5	11

Source: From EIA, 1995

A study by Biesemeyer and Jowett (1996) has indicated that Equation 4.4.20 more accurately estimates the motor load ratio than an approach based on the ratio of the motor speeds (i.e., measured speed over nominally rated speed) used by BPA (1990) and Lobodovsky (1994). It should be noted that Equation 4.4.20 is recommended for load ratios that are above 50% since, for these load ratios, a typical motor draws electrical current proportional to the imposed load.

The methodology for the calculation of the electrical power and energy savings is the same as described for the Mechanical Power Rating Method using Equations 4.4.17 through 4.4.20.

4.4.3 Lighting Systems

Lighting accounts for a significant portion of the energy use in commercial buildings. For instance, in office buildings, 30 to 50% of electricity consumption is used to provide lighting. In addition, heat generated by lighting contributes to the thermal load to be removed by the cooling equipment. Typically, energy retrofits of lighting equipment are very cost effective with payback periods of less than two years in most applications. In the U.S., lighting system conservation features were the most often installed measures to reduce energy costs in commercial buildings, as shown in Table 4.4.3. The data for Table 4.4.3 are based on the results of a survey (EIA, 1995) to determine the level of participation of commercial buildings in a variety of specific types of conservation programs and energy technologies.

To better understand the retrofit measures that need to be considered in order to improve the energy efficiency of lighting systems, a simple estimation of the total electrical energy use due to lighting can be provided by Equation 4.4.23:

$$Kwh_{Lit} = \sum_{j=1}^{J} N_{Lum,j} \cdot WR_{Lum,j} \cdot N_{h,j} \tag{4.4.23}$$

where,

$N_{Lum,j}$ is the number of lighting luminaires of type j in the building to be retrofitted. Recall that a luminaire consists of the complete set of a ballast, electric wiring, housing, and lamps.
$WR_{Lum,j}$ is the wattage rating for each luminaire of type j. In this rating, the energy use due to both the lamps and ballast should be accounted for.
$N_{h,j}$ is the number of hours per year when the luminaires of type j are operating.
J is the number of luminaire types in the building.

It is clear from Equation 4.4.23 that there are three options to reduce the energy use due to lighting, as briefly discussed below:

(a) Reduce the wattage rating for the luminaires including both the lighting sources (e.g., lamps) and the power transforming devices (e.g., ballasts), which would therefore decrease the term $WR_{Lum,j}$ in Equation 4.4.23. In the last decade, technological advances, such as compact fluorescent lamps and electronic ballasts, have increased the energy efficiency of lighting systems.

(b) Reduce the time of use of the lighting systems through lighting controls, which would therefore decrease the term $N_{h,j}$ in Equation 4.4.23. Automatic controls have been developed to decrease the use of a lighting system, so illumination is provided only during times when it is actually needed. Among energy efficient lighting controls are occupancy sensing systems and light dimming controls through the use of daylighting.

(c) Reduce the number of luminaires, which would therefore decrease the term $N_{Lum,j}$ in Equation 4.4.23. This goal can be achieved only in cases where delamping is possible due to over-illumination.

In this section, only measures related to the general actions described in items (a) and (b) are discussed. To estimate the energy savings due to any retrofit measure for the lighting system, Equation 4.4.23 can be used. The energy use due to lighting has to be calculated before and after the retrofit, and the difference between the two estimated energy uses represents the energy savings. Throughout the section, examples of lighting retrofit are presented.

Energy Efficient Lighting Systems

Improvements in the energy efficiency of lighting systems have provided several opportunities to reduce electrical energy use in buildings. This section discusses the energy savings calculations for the following technologies:

- High efficiency fluorescent lamps
- Compact fluorescent lamps
- Compact halogen lamps
- Electronic ballasts

First a brief description is provided for the factors that an auditor should consider to achieve and maintain an acceptable quality and level of comfort for the lighting system. Second, the design and the operation concepts are summarized for each available lighting technology. Then, the energy savings that can be expected from retrofitting existing lighting systems using any of the new technologies are estimated and discussed.

Typically, three factors determine the proper level of light for a particular space. These factors include age of the occupants, speed and accuracy requirements, and background contrast (depending on the task being performed). It is a common misconception to consider that overlighting a space provides higher visual quality. Indeed, it has been shown that overlighting can actually reduce the illuminance quality and the visual comfort level within a space, in addition to wasting energy. Therefore, it is important, when upgrading a lighting system, to determine and maintain the adequate illuminance level as recommended by the appropriate authorities. Table 4.4.3 summarizes the lighting levels recommended for various activities and applications in selected countries, including the U.S., based on the most recent illuminance standards.

High Efficiency Fluorescent Lamps

Fluorescent lamps are the most commonly used lighting systems in commercial buildings. In the U.S., fluorescent lamps illuminate 71% of the commercial space. Their relatively high efficacy, diffuse light distribution, and long operating life are the main reasons for their popularity.

A fluorescent lamp consists generally of a glass tube with a pair of electrodes at each end. The tube is filled at very low pressure with a mixture of inert gases (primarily argon) and with liquid mercury. When the lamp is turned on, an electric arc is established between the electrodes. The mercury vaporizes and radiates in the ultraviolet spectrum. This ultraviolet radiation excites a phosphorous coating on the inner surface of the tube which emits visible light. High efficiency fluorescent lamps use a krypton-argon mixture which increases the efficacy output by 10 to 20% from a typical efficacy of 70 lumens/watt to about 80 lumens/watt. Improvements in phosphorous coating can further increase the efficacy to 100 lumens/watt.

TABLE 4.4.4 Recommended Lighting Levels for Various Applications in Selected Countries (in Lux Maintained on Horizontal Surfaces)

Application	France AEF (92&93)	Germany DIN5035 (90)	Japan JIS (89)	U.S./Canada IESNA (93)
Offices				
General	425	500	300–750	200–500
Reading Tasks	425	500	300–750	200–500
Drafting (detailed)	850	750	750–1500	1000–2000
Classrooms				
General	325	300–500	200–750	200–500
Chalkboards	425	300–500	300–1500	500–1000
Retail Stores				
General	100–1000	300	150–750	200–500
Tasks/Till Areas	425	500	750–1000	200–500
Hospitals				
Common Areas	100	100–300	—	—
Patient Rooms	50–100	1000	150–300	100–200
Manufacturing				
Fine Knitting	850	750	750–1500	1000–2000
Electronics	625–1750	100–1500	1500–300	1000–2000

The handling and the disposal of fluorescent lamps is highly controversial due to the fact that mercury inside the lamps can be toxic and hazardous to the environment. A new technology is being tested to replace the mercury with sulphur to generate the radiation that excites the phosphorous coating of the fluorescent lamps. The sulphur lamps are not hazardous and would present an environmental advantage to the mercury-containing fluorescent lamps.

The fluorescent lamps come in various shapes, diameters, lengths, and ratings. A common labeling method used for fluorescent lamps is

$$F.S.W.C - T.D.$$

where

F stands for fluorescent lamp.
S refers to the style of the lamp. If the glass tube is circular, then the letter C is used. If the tube is straight, no letter is provided.
W is the nominal wattage rating of the lamp (such as 4, 5, 8, 12, 15, 30, 32, 34, 40, etc.)
C indicates the color of the light emitted by the lamp: W for white, CW for cool white, BL for black light
T refers to tubular bulb.
D indicates the diameter of the tube in eighths of one inch (1/8 in = 3.15 mm) and is, for instance 12 (D = 1.5 in = 38 mm) for the older and less energy efficient lamps and 8 (D = 1.0 in = 31.5 mm) for more recent and energy efficient lamps.

Thus, F40CW-12 designates a fluorescent lamp that has a straight tube, uses 40W electric power, has a cool white color, and is tubular with 38 mm (1.5 in) diameter.

Among the most common retrofit in lighting systems is the upgrade of the conventional 40W T12 fluorescent lamps to more energy efficient lamps such 32W T8 lamps. For a lighting retrofit, it is recommended that a series of tests be conducted to determine the characteristics of the existing lighting system. For instance, it is important to determine the illuminance level at various locations within the space especially in working areas such as on benches and/or desks.

Compact Fluorescent Lamps

These lamps are miniaturized fluorescent lamps with small diameters and shorter lengths. The compact lamps are less efficient than full-sized fluorescent lamps with only 35 to 55 lumens/Watt. However, they are more energy efficient and have longer lives than incandescent lamps. Currently, compact fluorescent lamps are being heavily promoted as an energy saving alternative to incandescent lamps, even though they may have some drawbacks. In addition to their high cost, compact fluorescent lamps are cooler and thus provide less pleasing contrast than incandescent lamps.

Compact Halogen Lamps

Compact halogen lamps are adapted for use as direct replacements for standard incandescent lamps. Halogen lamps are more energy efficient, produce whiter light, and last longer than incandescent lamps. Indeed, incandescent lamps convert typically only 15% of their electrical energy input into visible light — 75% is emitted as infrared radiation, and 10% is used by the filament as it burns. In halogen lamps, the filament is encased inside a quartz tube which is contained in a glass bulb. A selective coating on the exterior surface of the quartz tube allows visible radiation to pass through but reflects the infrared radiation back to the filament. This recycled infrared radiation permits the filament to maintain its operating temperatures with 30% less electrical power input.

Halogen lamps can be dimmed and present no power quality or compatibility concerns as can be the case for the compact fluorescent lamps.

Electronic Ballasts

Ballasts are integral parts of fluorescent luminaires since they provide the voltage level required to start the electric arc and regulate the intensity of the arc. Before the development of electronic ballasts in the early 1980s, only magnetic or "core and coil" ballasts were used to operate fluorescent lamps. While the frequency of the electrical current is kept at 60 Hz (in countries other than the U.S., the frequency is set at 50 Hz) by the magnetic ballasts, electronic ballasts use solid state technology to produce high frequency (20–60 MHz) current. The use of high frequency current increases the energy efficiency of the fluorescent luminaires since the light is cycling more quickly and appears brighter. When used with high efficiency lamps (T8 for instance), electronic ballasts can achieve 95 lumens/Watts as opposed to 70 lumens/Watts for conventional magnetic ballasts. It should be mentioned however that efficient magnetic ballasts can achieve the same lumen/Watt ratios as electronic ballasts.

Other advantages that electronic ballasts have relative to their magnetic counterparts include

- *Higher power factor.* The power factor of electronic ballasts is typically in the 0.90 to 0.98 range. Meanwhile, conventional magnetic ballasts have a low power factor (less than 0.80) unless a capacitor is added, as discussed in Section 4.4.2.
- *Less flicker.* Since magnetic ballasts operate at 60 Hz current, they cycle the electric arc about 120 times per second. As a result, flicker may be perceptible, especially if the lamp is old, during normal operation or when the lamp is dimmed to less than 50% capacity. However, electronic ballasts cycle the electric arc several thousand times per second and flicker problems are avoided, even when the lamps are dimmed to as low as 5% of capacity.
- *Less noise.* Magnetic ballasts use electric coils and generate an audible hum which can increase with age. Such noise is eliminated by the solid state components of the electronic ballasts.

Lighting Controls

As illustrated by Equation 4.4.23, energy savings can be achieved by not operating the lighting system at full capacity when illumination becomes unnecessary. The control of the lighting system can be achieved by several means, including manual on/off and dimming switches, occupancy sensing systems, and automatic dimming systems using daylighting controls.

While energy savings can be achieved by manual switching and manual dimming, the results are typically unpredictable since they depend on occupant behavior. Scheduled lighting controls provide a more efficient approach to energy savings but can also be affected by the frequent adjustments from occupants. Only automatic light switching and dimming systems can respond in real-time to changes in occupancy and climatic changes. Some of the automatic controls available for lighting systems are briefly discussed next.

TABLE 4.4.5 Energy Savings Potential with Occupancy Sensor Retrofits

Space Application	Range of Energy Savings
Offices (Private)	25–50%
Offices (Open Space)	20–25%
Rest Rooms	30–75%
Conference Rooms	45–65%
Corridors	30–40%
Storage Areas	45–65%
Warehouses	50–75%

Occupancy Sensors

Occupancy sensors save energy by automatically turning off the lights in spaces that are not occupied. Generally, occupancy sensors are suitable for most lighting control applications and should be considered for lighting retrofits. It is important to properly specify and install the occupancy sensors to provide reliable lighting during periods of occupancy. Indeed, most failed occupancy sensor installations result from inadequate product selection and improper placement. In particular, the auditor should select the proper motion sensing technology used in occupancy sensors. Two types of motion sensing technologies are currently available in the market:

Infrared sensors — register the infrared radiation emitted by various surfaces in the space, including the human body. When the controller connected to the infrared sensors receives a sustained change in the thermal signature of the environment (as is the case when an occupant moves), it turns the lights on. The lights are kept on until the recorded changes in temperature are not significant. The infrared sensors operate adequately only if they are in direct line-of-sight with the occupants and thus must be used in smaller enclosed spaces with regular shapes and without partitions.

Ultrasound sensors — operate on a sonar principle like submarine and airport radars. A device emits a high frequency sound (25–40 KHz) beyond the hearing range of humans. This sound is reflected by the surfaces inside a space (including furniture and occupants) and is sensed by a receiver. When people move inside the space, the pattern of sound waves changes. The lights remain on until no movement is detected for a preset period of time (e.g., 5 minutes). Unlike infrared radiation, sound waves are not easily blocked by obstacles such as wall partitions. However, the ultrasound sensors may not operate properly in large spaces which tend to produce weak echoes.

Based on a study by EPRI, Table 4.4.5 shows typical energy savings expected from occupancy sensor retrofits. Significant energy savings can be achieved in spaces where occupancy is intermittent, such as conference rooms, rest rooms, storage areas, and warehouses.

Light Dimming Systems

Dimming controls allow the variation of the intensity of lighting system output based on natural light level, manual adjustments, and occupancy. A smooth and uninterrupted decrease in the light output is defined as a continuous dimming as opposed to stepped dimming in which the lamp output is decreased in stages by preset amounts.

Computer software, such as RADIANCE (LBL, 1991), can accurately estimate the energy savings from dimming systems that use natural light controls (e.g., daylighting). With such computer tools, an engineer can predict the percentage of time when natural light is sufficient to meet all lighting needs.

Example 4.4.3 provides a simple calculation procedure to estimate the energy savings from a lighting retrofit project.

Example 4.4.3

Problem: Consider a building with a total of 500 luminaires of four 40W lamps/luminaire. Determine the energy saving after replacing those with two 32W high efficacy lamps/luminaire. This building is operated 8 hours/day, 5 days/week, 50 weeks/year.

Solution

The energy saving in kWh is

$$\Delta kWh = 500 \cdot (4 * 40 - 4 * 32) \cdot 8 \,.\, 5 \,.\, 50 \cdot \frac{1}{1000} = 32{,}000\ kWh/yr$$

Thus, the energy saving is 116,800 kWh/year.

4.4.4 Electrical Distribution Systems

All electrical systems have to be designed in order to provide electrical energy as safely and reliably as economically possible. Figure 4.4.8 shows a typical one-line diagram of an electrical system for a building. The main distribution panel includes the switchgear-breakers, to distribute the electric power, and the unit substation, to step down the voltage. The unit substation consists typically of a high voltage disconnect switch, a transformer, and a set of low voltage breakers. The circuit breakers for lighting and plug-connected loads are housed in lighting panelboards while the protective devices for motors are assembled typically into motor control centers (MCCs). Specifically, an MCC consists of: overload relays, to prevent the current from the motor from exceeding dangerous levels, and fuse disconnect switches or breakers to protect the motor from short circuit currents.

An important part of any electrical system is the electrical wiring that connects all the system components. Three types of connecting wires can be identified:

- Service entrance conductors are those electrical wires that deliver electricity from the supply system to the facility. For large facilities, electricity is typically supplied by an electric utility at a relatively high voltage (13.8 kV) requiring a transformer (part of a unit substation) to step down the voltage to the utilization level.
- Feeders are the conductors that deliver electricity from the service entrance equipment location to the branch-circuits. Two types of feeders are generally distinguished: the main feeders that originate at the service entrance (or main distribution panel) and the subfeeders that originate at distribution centers (lighting panelboards or motor control centers).
- Branch circuits are the conductors that deliver electricity to the utilization equipment from the point of the final over-current device.

Transformers

The transformer is the device that allows change to the voltage level of an alternating current. In particular, it is common to use transformers at generating stations to increase the transmission voltages to high levels (13,800 volt) and near or inside buildings to reduce the distribution voltages to low levels for utilization (480 or 208 volt).

A typical transformer consists of two windings: primary and secondary. The primary winding is connected to the power source, while the secondary winding is connected to the load. Between the primary and the secondary windings, there is no electrical connection. Instead, the electric energy is transferred through inductance within the core, which is generally made of laminated steel. Therefore, transformers operate only on alternating current.

There are basically two types of transformers: liquid-filled and dry-type. In liquid-filled transformers, the liquid acts as a coolant and as insulation dielectric. Dry-type transformers are constructed so that the core and coils are open to allow for cooling by the free movement of air. In some cases, fans may be installed to increase the cooling effect. The dry-type transformers are widely used because of their lighter weight and simpler installation, compared to liquid-filled transformers.

A schematic diagram for a single-phase transformer is provided in Figure 4.4.9. A three-phase transformer can be constructed from a set of three single-phase transformers electrically connected so that

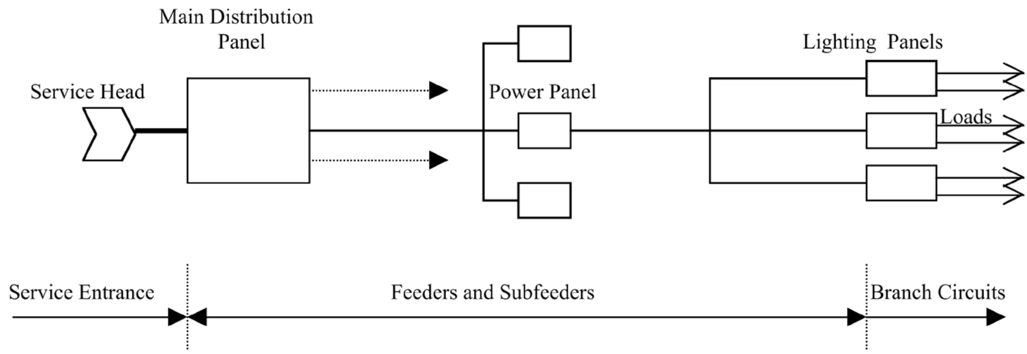

FIGURE 4.4.8 A schematic one-line diagram for a basic electrical distribution system within a building.

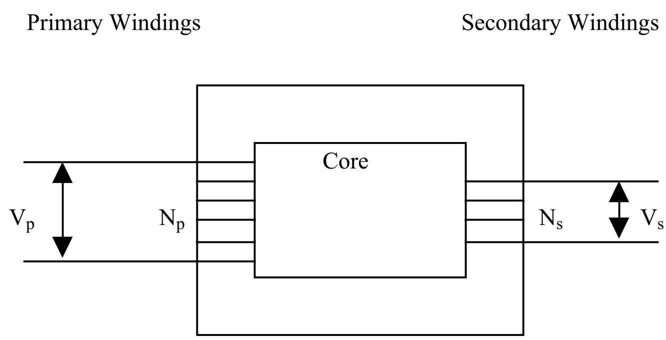

FIGURE 4.4.9 Simplified model for a single-phase transformer.

the primary and the secondary windings can be either wye or delta configurations. For buildings, delta-connected primary and wye-connected secondary is the most common arrangement for transformers. It can be shown that the primary and the secondary voltages, V_p and V_s, are directly proportional to the respective number of turns, N_p and N_s, in the windings:

$$\frac{V_p}{V_s} = \frac{N_p}{N_s} = a \tag{4.4.24}$$

where a is the turns ratio of the transformer. As indicated by Equation 4.4.24, the turns ratio can be determined directly from the voltages without knowing the actual number of turns on the transformer windings.

Transformers are rated by their volt-ampere capacity of the secondary windings. For large transformers, the power output in kilo volt-ampere, or kVA, rating is generally used as expressed in Equation 4.4.25:

$$kVA = \frac{\sqrt{3} \cdot V_s \cdot I_s}{1000} \tag{4.4.25}$$

where V_s and I_s are respectively the rated line-to-line voltage and the rated line current of the secondary.

Transformers are typically very efficient with energy losses (in the core and windings) representing only 1–2% of the transformer capacity. It may be cost effective to invest in more efficient transformers, especially if they are used continuously at their rated capacity, as illustrated in Example 4.4.4.

Example 4.4.4

Problem

Determine the cost effectiveness of selecting a unit with an efficiency of 99.95% rather than 99.90% for a 500-kVA rated transformer. Assume the following:

- The cost of electricity is $0.10/kWh.
- The installed costs of 99.0% and 99.5% efficient transformers are respectively $7000 and $9500.
- The average power factor of the load is 0.90.
- The no-load losses are the same for both transformers.

For the analysis, consider two cases for the length of time during which the transformer is used at its rated capacity:

(a) 10 hours/day and 250 days/year
(b) 16 hours/day and 300 days/year

Solution

To determine the cost effectiveness of installing an energy efficient transformer, a simplified economic analysis based on the payback period is used. The savings in energy losses in kWh for the high efficiency transformer can be calculated as follows:

$$kWh_{saved} = N_h \cdot kVA \cdot pf \cdot \left(\frac{1}{\eta_{std}} - \frac{1}{\eta_{eff}}\right)$$

where

N_h is the total number of hours (per year) during which the transformer is operating at full-load [for case (a) $N_h = 10*250 = 2500$ hrs/yr; for case (b) $N_s = 16*300 = 4800$ hrs/yr].
kVA is the rated transformer power output [500 kVA].
pf is the annual average power factor of the load [pf = 0.90].
η_{std} and η_{eff} are the efficiency of the standard transformer and the efficient transformer [0.990 and 0.995], respectively.

The energy savings and the payback period for each case are presented below.

For case (a) $N_h = 2500$ *hrs/yr*

The energy savings in kWh is calculated as follows:

$$kWh_{saved} = 2500 * 500 * 0.90 * \left(\frac{1}{0.990} - \frac{1}{0.995}\right) = 5710\ kWh/yr$$

Therefore, the payback period, *PB*, for investing on the efficient transformer is

$$PB = \frac{\$9000 - \$7000}{5710 kWh * \$0.07/kWh} = 5.0\ years$$

For case (b) $N_h = 4800$ *hrs/yr*

The energy savings in kWh is calculated as follows:

$$kWh_{saved} = 4800 * 500 * 0.90 * \left(\frac{1}{0.990} - \frac{1}{0.995}\right) = 10{,}964\ kWh/yr$$

Thus, the payback period, *PB*, for investing on the efficient transformer is

$$PB = \frac{\$9000 - \$7000}{10{,}964(10964)kWh * \$0.07/kWh} = 2.6 \ years$$

It is clear that it can be cost-effective to consider investing in a more energy efficient transformer, especially when the load is supplied during longer periods of time. It should be noted that additional energy savings can be expected during no-load conditions for the energy efficient transformer.

Electrical Wires

The term *electrical wire* is generic and refers typically to both a conductor and a cable. The conductors are copper or aluminum wires that actually carry electrical current. Cable refers generally to the complete wire assembly, including the conductor, insulation, and any shielding and/or protective covering. A cable can have more than one conductor, each with it own insulation.

The size of an electrical conductor represents its cross-sectional area. In the U.S., two methods are used to indicate the size of a conductor: the American wire gauge (AWG) for small sizes and thousand of circular mils (MCM). For the AWG method, the available sizes are from number 18 to number 4/0 — the higher the number the smaller the conductor size. For buildings, the smallest size of copper conductor that can be used is number 14 which is rated for a maximum loading of 15 amperes. AWG size designation became inadequate soon after its implementation in the early 1900s due to the ever-increasing electrical load in buildings. For larger conductors, the cross-sectional area is measured in circular mils. A circular mil corresponds to the area of a circle that has a diameter of 1 mil or 1/1000th of an inch. For instance, a conductor with a diameter of .5 in (500 mils) has a circular mil area of 250,000 which is designated by 250 MCM.

To determine the correct size of conductors to be used for feeders and branch circuits in buildings, three criteria generally need to be considered:

- *The rating of the continuous current under normal operating conditions.* The National Electric Code (NEC, 1996) refers to the continuous current rating as the ampacity of the conductor. The main parameters that affect the ampacity of a conductor include the physical characteristics of the wire, such as its cross-sectional area (or size) and its material, and the conditions under which the wire operates, such as the ambient temperature and the number of conductors installed in the same cable. Table 4.4.5 indicates the ampacity rating of copper and aluminum conductors with various sizes. Various derating correction and factors may need to be applied to the ampacity of the conductor to select its size.
- *The rating of short circuit current under fault conditions.* Indeed, high short circuit currents can impose significant thermal or magnetic stresses not only on the conductor but also on all the components of the electrical system. The conductor has to withstand the relatively high short circuit current since the protective device requires some finite time before detecting and interrupting the fault current.
- *The maximum allowable voltage drop across the length of the conductor.* Most electrical utilization equipment is sensitive to the voltage applied to it. It is therefore important to reduce the voltage drop that occurs across the feeders and the branch circuits. The NEC recommends a maximum voltage drop of 3% for any one feeder or branch circuit, with a maximum voltage drop from the service entrance to the utilization outlet of 5%.

For more details, the reader is referred to section 220 of the NEC that covers the design calculations of both feeders and branch circuits.

Two conductor materials are commonly used for building electrical systems: copper and aluminum. Because of its highly desirable electrical and mechanical properties, copper is the preferred material used for conductors of insulated cables. Aluminum has some undesirable properties and its use is restricted. Indeed, an oxide film, which is not a good conductor, can develop on the surface of aluminum and can

TABLE 4.4.5 Ampacity of Selected Insulated Conductors Used in Buildings

Conductor Size (AWG or MCM)	THW (Copper)	THHN (Copper)	THW (Aluminum)	THHN (Aluminum)
18	—	14	—	—
16	—	18	—	—
14	20	25	—	—
12	25	30	20	25
10	35	40	30	35
8	50	55	40	45
6	65	75	50	60
4	85	95	65	75
3	100	110	75	85
2	115	130	90	100
1	130	150	100	115
1/0	150	170	120	135
2/0	175	195	135	150
3/0	200	225	155	175
4/0	230	260	180	205
250	255	290	205	230
300	285	320	230	255
350	310	350	250	280
400	335	380	270	305
500	380	430	310	350
600	420	475	340	385
700	460	520	375	420
750	475	535	385	435
800	490	555	395	450
900	520	585	425	480
1000	545	615	445	500
1250	590	665	485	545
1500	625	705	520	585
1750	650	735	545	615
2000	665	750	560	630

Source: Adapted from NEC Table 310-16.

cause poor electrical contact, especially at the wire connections. It should be noted that aluminum can be considered in cases when cost and weight are important criteria for the selection of conductors. However, it is highly recommended, even in these cases, to use copper conductors for the connections and the equipment terminals to eliminate poor electrical contact.

To protect the conductor, several types of insulation materials are used. The cable (which is the assembly that inludes the conductor, insulation, and any other covering) is identified by letter designations depending on the type of insulation material and the conditions of use. In buildings, the following letter designations are used.

- *For the insulation material type:* A (asbestos), MI (mineral insulation), R (rubber), SA (silicone asbestos), T (thermoplastic), V (varnished cambric), and X (cross-linked synthetic polymer).
- *For the conditions of use:* H (heat up to 75°C), HH (heat up to 90°C), UF (suitable for underground), W (moisture resistant).

Thus, the letter designation THW refers to a cable that has a thermoplastic insulation rated for maximum operating temperature of 75°C and suitable for use in dry as well as wet locations.

Moreover, some types of electrical cables have outer coverings that provide mechanical/corrosion protection, such as lead sheath (L), nylon jacket (N), armored cable (AC), metal-clad cable (MC), and nonmetallic sheath cable (NM).

For a full description of all types of insulated conductors, their letter designations, and their uses, the reader is referred to the NEC, article 310 and table 310-13.

In general, electrical cables are housed inside conduits for additional protection and safety. The types of conduits commonly used in buildings are listed below:

- Rigid metal conduit (RMC) can be of either steel or aluminum and has the thickest wall of all types of conduits. Rigid metal conduit is used in hazardous locations such as areas of high exposure to chemicals.
- Intermediate metal conduits (IMC) has a thinner wall than the rigid metal conduit but can be used in the same applications.
- Electrical metallic tubing (EMT) is a metal conduit but with a very thin wall. The NEC restricts the use of EMT to locations where it is not subjected to severe physical damage during installation or after installation.
- Electrical nonmetallic conduit (ENC) is made of nonmetallic material such as fiber or rigid PVC (polyvinyl chloride). Generally, rigid nonmetallic conduit cannot be used where subject to physical damage.
- Electrical nonmetallic tubing (ENT) is a pliable corrugated conduit that can be bent by hand. Electrical nonmetallic tubing can be concealed within walls, floors, and ceilings.
- Flexible conduit can be readily flexed and thus is not affected by vibration. Therefore, a common application of the flexible conduit is for the final connection to motors or recessed lighting fixtures.

It should be noted that the number of electrical conductors that can be installed in any one conduit is restricted to avoid any damage of cables (especially when the cables are pulled through the conduit). The NEC restricts the percentage fill to 40% for three or more conductors. The percentage fill is defined as the fraction of the total cross-sectional area of the conductors — including the insulation — over the cross-sectional area of the inside of the conduit.

When selecting the size of the conductor, the operating costs and not only the initial costs should be considered. As illustrated in Example 4.4.5, the cost of energy encourages the installation of larger conductors than are required by the NEC, especially for the smaller sized conductors (i.e., numbers 14, 12, 10, and 8). Unfortunately, most designers do not consider the operating costs in their design for several reasons, including the uncertainties in electricity prices.

Example 4.4.5

Problem

Determine if it is worthwhile to install number 10 (AWG) copper conductor instead of number 12 (AWG) on a 400 ft branch circuit that feeds a load of 16 amperes. Assume that

- The load is used 10 hours/day and 250 days/year.
- The cost of electricity is \$0.10/kWh.
- The installed costs of No. 12 and No. 10 conductors are, respectively, \$60.00 and \$90.00 per 1000 ft cable.

Solution

In addition to the electric energy used to meet the load, which is independent of the conductor size, there is an energy loss in the form of heat generated by the flow of current, *I*, through the resistance of the conductor, *R*. The heat loss in Watts can be calculated as follows:

$$Watts = R \cdot I^2$$

Using the information by the NEC (Table 8), the resistance of both conductors No. 12 and 10 can be determined to be, respectively, 0.193 ohm and 0.121 ohm per 100 ft. Thus, the heat loss for the 400-ft branch circuit if No. 12 conductor is used can be estimated as follows:

$$Watts_{12} = 0.193 * 400/100 \cdot (16)^2 = 197.6\ W$$

Similarly, the heat loss for the 400-ft branch circuit when No. 10 conductor is used is found to be

$$Watts_{10} = 0.121 * 400/100 \cdot (16)^2 = 123.9\ W$$

The annual cost of copper losses for both cases can be easily calculated:

$$Cost_{12} = 197.6\ W * 250\ days/yr * 10\ hrs/Day * 1\ kW/1000\ W * \$0.10/kWh = \$49.4/yr$$

$$Cost_{10} = 123.9\ W * 250\ days/yr * 10\ hrs/Day * 1\ kW/1000\ W * \$0.10/kWh = \$31.0/yr$$

Therefore, if No. 10 is used instead of No. 12, the payback period, *PB*, for the higher initial cost for the branch circuit conductor is

$$PB = \frac{(\$90/1000\,ft - \$60/1000\,ft) * 400\,ft}{(\$49.4 - \$31.0)} = 0.68\ yr = 8\ months$$

The savings in energy consumption through the use of larger conductors can thus be cost-effective. Moreover, it should be noted that the larger size conductors reduce the voltage drop across the branch circuit which permits the connected electrical utilization equipment to operate more efficiently. However, the applicable code has to be carefully consulted to determine if a larger size conduit is required when larger size conductors are used.

Branch Circuits

In general, branch circuits originate at the panelboards and/or motor control centers to serve lighting fixtures, general use receptacles, specific purpose equipment, and motors. In commercial buildings, branch circuits for lighting and receptacles are common, and their design requirements are detailed in Articles 210 and 220 of the NEC. The design requirements for motor branch circuits are generally more involved and are considered in article 430 of the NEC.

Branch Circuits for Lighting

Branch circuits for fluorescent lighting and for smaller wattage medium-based incandescent lamps (up to 300 watts) are restricted to 15 or 20 amperes. Fixed lighting units with heavy-duty lampholders can be connected to circuits rated up to 50 amperes when installed in other-than dwelling units. In general, the lighting used to illuminate areas such as offices and schools is considered to be a continuous load. Since, the NEC restricts the maximum loading on a circuit supplying a continuous load to 80% of the circuit rating, the maximum loading on a 20-ampere lighting circuit is 16 amperes.

Branch Circuits for Receptacles

The NEC defines a receptacle as a contact device installed at the outlet for the connection of a single attachment plug. The minimum load for an outlet is set by the ampere rating of the appliance served by the outlet. However, the majority of receptacles are installed for general purpose use. Therefore, the exact loads of receptacles are generally unknown. To compute the loads on a receptacle branch circuit, a minimum loading of 180 volt-amperes should be allowed for each general use receptacle outlet, regardless of whether a single, duplex, or triplex receptacle is installed. Thus, the maximum number of general use receptacles allowed on a 15- and 20-ampere branch circuit is 10 and 13, respectively (assuming a power supply source of 120 volt).

Branch Circuit for Motors

Electric motors have unique starting and running characteristics. Therefore, the branch circuits for motors have to be designed with special considerations. In particular, the starting currents of motors can be as high as six times that of their rated full-load running currents. To avoid the motor from shutting down, protective devices have to be properly designed to account for the transient starting current that can last up to 15 seconds. Moreover, the protective devices have to be able to react accurately to any

TABLE 4.4.6 Maximum Size of Overcurrent Protection Device as Required by the NEC

Wire size	Copper wire	Aluminum or copper-clad aluminum wire
No. 14 AWG	15 amperes	No. 14 AWG aluminum is not permitted
No. 12 AWG	20 amperes	15 amperes
No. 10 AWG	30 amperes	25 amperes

overloads and to protect the motor from being damaged. Thus, the branch circuit of motors includes the following components:

- Protection device for short circuit and ground fault protection
- Conductors to supply electric power to the motor
- Motor controller for overload protection
- Disconnection means to safely isolate the motor from the power source supply

Protective Devices

One of the main requirements in the design of electrical systems is to minimize power outages and damage in cases of fault conditions. Protective devices provide the means to isolate the faulted segment of the electrical system as quickly and safely as possible. Specifically, a protective device has two major functions: the detection of the fault condition and disconnection of the faulted section from the remainder of the electrical system. Some protective systems combine both functions, such as fuses, while other types separate the two actions, such as high voltage circuit breakers. Article 240 of the NEC covers the over-current protective devices.

Abnormal or fault conditions can occur on an electrical system for several reasons, including the following:

- Overloads occur when electrical equipment draws excessive current demands. Fault currents are considered overload currents when they are up to 600% of the full-load capacity of the electrical system.
- Short circuits result in considerably large flows of current in excess of 600% of the full-load current rating. Typically, short circuits are due to electrical failures, such as breakdown in the conductor insulation (arcing fault) or an accidental connection of two phases (bolted fault).
- Single phasing on three-phase systems such as motors.
- Over-voltages and transient surges that occur when the electrical system is subject to lightning.

Protective devices are characterized and rated using the following parameters:

- Maximum continuous voltage that can be applied to the electrical system without causing the conductor insulation to fail.
- Maximum continuous current that can flow in the electrical system without resulting in overheating
- Interrupting current defined as the maximum current up to which the protective device can safely operate to disconnect the electrical system.
- Short-time ratings, including the momentary current (the maximum current that the protective device can withstand without failure) and the specified time current (the current that the protective device can withstand for a specified time — typically 0.5 seconds — without failure).

In general, the size of the protective device should be less than the ampacity of the conductor being protected. Table 4.4.6 provides the maximum size of the overcurrent protective device required by the NEC depending on the conductor size of the branch circuit or feeder.

Two types of devices are commonly used to protect electrical systems in buildings: fuses and circuit breakers.

Fuses

The NEC defines the fuse as an over-current protective device with a circuit opening and fusible part that is heated and severed by the passage of current through it. Currently, there are several types of fuses suitable for various applications. The basic construction of all fuses has remained essentially the same over the years. However, for current-limiting fuses, the fusible element is made of silver, packed in a quartz filler, and hermetically sealed inside a ceramic case. For motor applications with high starting current, dual-element time-delay fuses are used to prevent the protective device from tripping each time the motor is operated.

The main advantages of fuses compared to other types of protective devices are

- Low initial cost
- Little maintenance since fuses are simple to construct
- Generally compact and require little space to be installed
- High current interrupting capabilities
- Inherently fail-safe devices since when fuses fail, they automatically open the circuit

However, fuses also present several disadvantages, including

- Can cause single phasing in three phase systems
- Are not flexible since the time response of the fuses are fixed and not adjustable
- Must be replaced after each operation

Circuit Breakers

The NEC defines the circuit breaker as a device designed to open and close a circuit by non automatic means and to open the circuit automatically on a predetermined matter without injury to itself when properly applied within its rating. Circuit breakers are available with various voltage and continuous current ratings, as well as interrupting current rating, response characteristics, and methods of operation. For instance, molded-case breakers are compact and relatively inexpensive, but they have generally low interrupting ratings and thus cannot be applied to large systems. In addition, electronic solid state trip units are currently commonly used, especially for power circuit breakers. Indeed, solid state trip units provide more flexibility and accuracy than the mechanical dual-magnetic types.

It should be noted that circuit breakers can be installed either in single pole or multi-pole. Multi-pole breakers are generally gang-operated so that all the poles are closed and opened simultaneously by one common operating mechanism (such as a handle). Therefore, circuit breakers cannot cause single phasing in three-phase systems, as is the case with fuses.

Compared to fuses, the circuit breakers provide the following advantages:

- Can serve as means of both protecting and switching an electrical circuit
- Do not cause single phasing
- Can be remotely operated
- Can easily incorporate ground-fault protection

However, breakers have some disadvantages compared to fuses. In particular, circuit breakers

- Have higher initial cost
- Require more space since they are larger
- Require more maintenance because of their complexity in construction and operation
- Do not limit fault current and thus the electrical system is subject to higher thermal and magnetic stresses under fault conditions
- Are not a fail-safe device since the trip mechanism can be damaged and the breaker can be left in a closed position

Design Requirements for HVAC Systems

The NEC includes a number of articles specific to electrically driven heating, air conditioning, and refrigeration equipment. Some of the design requirements based on the NEC that apply to the HVAC systems are summarized below.

Article 424 applies for fixed electric space heating equipment. In particular, the following design requirements should be considered for electric heating equipment:

- The branch circuit conductor and the protective device should not be smaller than 125% of the total load (i.e., heater and motor).
- A disconnect means is required to disconnect the controller and the heater.
- The disconnect must be located within sight of the heater but may not be readily accessible.

Article 422 applies to any air conditioning and refrigerating equipment that does not have a hermetic refrigerant motor compressor, such as fan-coil units and evaporator coils. In fact, the scope of article 422 includes appliances that are fastened in place, permanently connected, or cord-and-plug-connected in any occupancy. Some of the design requirements of article 422 are listed below:

- The conductor ampacity to any individual appliance shall not be less than required by the appliance marking or instruction.
- The overcurrent protection for appliances must not exceed the protective device rating marked on the appliance.
- Circuit breakers or switches can serve as the disconnect means for permanently connected equipment rated over 300 VA.

Article 440 is specific to electrically driven air conditioning and refrigeration equipment that has hermetic refrigerant motor compressors. In particular, the NEC states that:

- The branch circuit conductors to a single motor compressor must have an ampacity not less than 125% of the motor compressor current. For several motors, conductors must have an ampacity of not less than 125% of the highest rated motor compressor current of the group, plus the sum of the other motor compressor currents of the group.
- The protection device rating must not exceed the manufacturer's values marked on the equipment. For instance, if the nameplate specifies "HACR Circuit Breaker," then the equipment must be protected by a circuit breaker that is rated for heating, air conditioning, or refrigeration equipment.
- The disconnecting means should be readily accessible and within sight of the equipment. Only room air conditioners, household refrigerators and freezers, drinking water coolers, and beverage dispensers are permitted to use the attachment plug and receptacle as the disconnecting means.

For motors, starters need to be specified in addition to the conductors, protective devices, and disconnecting means. A starter is a controller whose primary function is to start and stop the operation of the motor either manually or automatically. However, starters can have additional features, such as overload, under-voltage, single-phasing protection, reversal of direction of rotation, and reduced voltage starting. In addition to the manual starters that can be used only to start and stop small motors, magnetic starters are widely used because of their proven reliability. Some of the common types and features of magnetic starters are summarized below:

- Under-voltage protection to prevent motors from restarting whenever the power is restored
- Combination starters to provide disconnecting means and overcurrent protection to the electrical system
- Reversing starters to reverse the direction of the motor rotation. Reversing starters can also be used to plug a motor to a rapid stop.

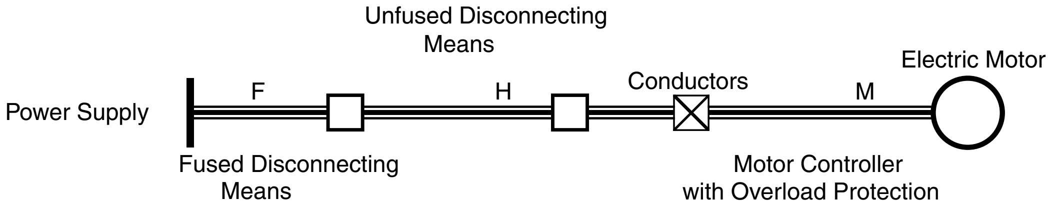

FIGURE 4.4.10 Typical branch circuit components for a single motor.

- Full-voltage starters to be used when the motor starting current does not cause serious disturbances on the electrical system
- Reduced-voltage starters to be used whenever full-voltage starting can cause unacceptable disturbances on the electrical system. Different mechanisms are used to reduce the starting voltage, including autotransformer arrangements and wye-delta connections.

Typically, motor control centers (MCCs) are used to house the motor starters in addition to protection devices, disconnecting means, and overload relays. MCCs are available in modular plug-in units allowing flexibility in arrangement and ease of maintenance.

In the U.S., the design of branch circuits, feeders, protective devices, and motor control centers for motors is based on NEC requirements. The following sections provide the design steps, through a series of examples, to illustrate the procedure to select the adequate conductor size, fuse rating, and MCC layout.

Branch Circuit for one Motor

To design the branch circuit for a single motor, four essential components need to be sized: the conductor, the protective device with the disconnecting means, the overload protection, and the unfused disconnect means, as shown in Figure 4.4.10. The following steps are suggested to size all the components of motor branch circuits. To better illustrate the design procedure, we consider the specific case of a branch circuit that supplies electrical power to a 40 hp, 460 V, three-phase squirrel-cage induction motor, with nameplate full-load current of 50 A. The motor is protected with a non-time-delay fuse and is supplied with THW conductor. Both the motor and its controller are out of sight of the branch circuit source of supply.

Step 1: Motor load

The rated current for a motor is determined from the NEC (1996) using Table 430-18 (for single-phase motors) or Table 430-150 (for three-phase motors). For a 40 hp, three-phase motor, the rated current is 52 A.

Step 2: Conductor size

Using the NEC requirement for motor branch circuits, the minimum ampacity of the conductor is 125% of 52 A or 65 A. Using Table 4.4.5 (Table NEC 310-16), the size of a THW conductor is No. 6 AWG (rated at 65 A). Since the motor required 3 conductors (for a three-phase motor with a delta connection), the conduit size is 1 in (Table 3A of chapter 9 from the NEC).

Step 3: Protective device with a disconnecting means

Since the protection device is a non-time delay fuse, a factor of 300% should be used to determine the rating of the fuse — that is 300% of 52 A or 156 A. It should be noted that NEC Table 430-152 should be used to obtain the rating factors. A standard size of fuse should be selected (the largest standard size after 156 A) or 175 A. The disconnecting size is based on the horsepower and the type of the protective device for the motor. Table 4.4.7 presents the standard rating of switch

TABLE 4.4.7 Standard Rating of Switch (in Amperes) for Three-Phase Motors (Rated at 480 Volt)

Horsepower Rating Range	Fused Switch With Non-time Delay Fuses	Fused Switch With Time Delay Fuses	Unfused Switch
Below 7.5 hp (5.6 kW)	30	30	30
7.5–15 hp (5.6–11.2 kW)	60	30	30
15–20 hp (11.2–14.9 kW)	100	60	60
20–25 hp (14.9–18.7 kW)	100	60	60
25–30 hp (18.7–22.4 kW)	200	60	60
30–50 hp (22.4–37.3 kW)	200	100	100
50–60 hp (37.3–44.8 kW)	400	100	100
60–100 hp (44.8–74.6 kW)	400	200	100
Above 100 hp (74.6 kW)	400	400	200

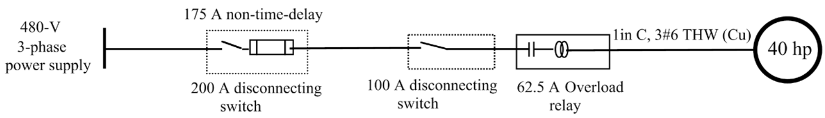

FIGURE 4.4.11 The branch circuit specifications for the 40 hp motor.

for 480-volt three-phase motors. For a 40 hp motor protected with a non-time delay fuse, the switch size is 200 A.

Step 4: Overload protection

The rating of the overload protection is 125% of 50 A (nameplate full-load current) or 62.5 A.

Step 5: Unprotected disconnecting means

The rating of unfused switch depends on the size of the motor. For a 40 hp motor, a 100 A rated unfused switch is needed (see Table 4.4.7).

A one-line diagram for the branch circuit is shown in Figure 4.4.11.

Feeder for Several Motors

In general, two components need to be sized to design a motor feeder: the conductor and the protective device with the disconnecting means. Typically, overload protection is not provided to the feeder since individual motors are cleared by their own overload relays in case any excessive overloads occur. The design procedure for motor feeders is illustrated below using a feeder that supplies electric power to the following three-phase, 460-V motors: one 40 hp, two 30 hp, three 20 hp, and six 10 hp. All the motors have full voltage nonreversing starters (FVNR) except the 40 hp motor with full voltage reversing (FVR) starter. The feeder and all the branch circuits are protected with non-time delay fuses and are supplied with THW conductors.

Step 1: Motor load

The rated load current for all three-phase motors is determined from the NEC Table 430-150. The rated current loads for 40 hp, 30 hp, 20 hp, and 10 hp 460-V motors are 52 A, 40 A, 27 A, and 14 A, respectively.

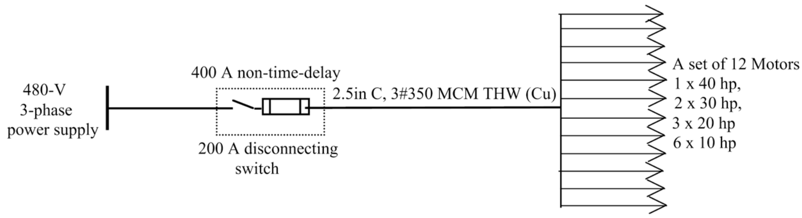

FIGURE 4.4.12 The feeder specifications to supply several motors.

Step 2: Conductor size

Using the NEC requirement for motor branch circuit, the minimum ampacity of the conductor is 125% of 52 A plus the sum of 40 A × 2, 27 A × 3, and 14 A × 6. Thus the ampacity of the conductor is 297 A. Therefore, the size of THW feeder conductors is No. 350 MCM (rated at 310 A) based on Table 4.4.5 or Table NEC 310-16. Since the motor required 3 conductors (for a three-phase motor with a delta connection), the conduit size is 2.5 in (Table 3A of chapter 9 from the NEC).

Step 3: Protective device with a disconnecting means

Since the protection device is a non-time delay fuse, a factor of 300% should be used for the largest motor. According to the NEC (1996), the fuse should be set at a rating no greater than the value calculated by taking the largest rating of the largest motor current multiplied by the appropriate factor (here, 300%) plus the sum of the full-load currents of all the remaining motors — that is, the nearest standard fuse size (300% × 52 A + 40 A × 2 + 27 A × 3 + 14 A × 6 = 401 A) or 400 A. From Table 4.4.7, the switch size is 200 A.

A one-line diagram for the feeder is shown in Figure 4.4.12.

Motor Control Centers

Motor control centers (MCCs) are recommended when several motors need to be controlled from one location. The centralized location of the control units can offer several benefits, including convenience of operation and ease of maintenance. For commercial buildings, MCCs are typically located in the mechanical rooms that house the fans, pumps, and HVAC equipment.

The overall size of MCCs depends on several factors including the number of motors, the type of the motor starters, and the rating of the protective devices. In the U.S., MCCs are typically arranged in modules of 20 in × 20 in × 90 in. The procedure for determining the space requirements of MCCs is illustrated below, using a motor control center that feeds the motors defined in the previous section.

Step 1: Space requirements for motor starters

Each module (20 in × 20 in × 90 in) of a motor control center can hold a variable number of drawout control units. Each drawout control unit houses the starter, the protection device (circuit breaker or fused switch), and a disconnecting means for one motor. The drawout control units have standard incremental dimensions. Commonly, the smallest unit has a dimension of 12 in (30 cm) with an increment of 6 in (15 cm) for larger units. Other increments such as 6.5 or 7 in (16.5 or 17.7 cm) are also available. Typically, the increments are referred to as space factors. Table 4.4.8 indicates the number of space factors required for motor control units protected by either circuit breakers or fusible switches for full voltage nonreversing (FVNR) and full voltage reversing (FVR) controllers for 480-volt three-phase motors.

Table 4.4.9 provides the space factor requirements for various drawout control units used in the MCC that serve the motors considered in the previous section.

TABLE 4.4.8 Typical Space Factors Required for MCCs (Based on a Space Factor = 6 in = 15 cm)

Motor Horsepower Range	Circuit Breaker (FVNR)	Fusible Switch (FVNR)	Circuit Breaker (FVR)	Fusible Switch
Below 10 hp (7.5 kW)	2	2	3	3
10–25 hp (7.5–19 kW)	2	2	3	3
25–50 hp (19–37 kW)	4	4	4	5
50–100 hp (37–75 kW)	4	6	4	6
100–200 hp (75–149 kW)	6	7	10	12
200–400 hp (149–298 kW)	12	12	12	12

Note: Manufacturers' specifications should be consulted.

TABLE 4.4.9 Number of Space Factors Required by the Motor Control Units

Motor hp	Type of Starter	Space Factor per One Motor	Number of Motors	Number of Space Factors
40	FVR	5	1	5
30	FVNR	4	2	8
20	FVNR	2	3	6
10	FVNR	2	6	12

Therefore, the total number of space factors required for all the motor control units is 31. In addition, one space factor is typically allocated to the incoming feeder cables. Thus, 32 space factors are needed for the MCC that serves 40 hp, 2 × 30 hp, 3 × 20 hp, and 6 × 10 hp motors. Since each module can hold 12 space factors, the number of modules needed for the MCC is 32/12 or 3 modules with 4 spare space factors.

Step 2: Layout of the MCC

Figure 4.4.13 shows a possible layout for the MCC, including the position of the drawout control units for all the motors.

The MCC can be free standing or mounted against a wall. To allow for easy access to the drawout control units, sufficient working space in the front of the MCC should be available. For an MCC rated at 480 volt, the NEC requires a minimum clearance of 3.5 ft (1.07 m) from the front face of the MCC to the nearest grounded surface such as a wall.

4.4.5 Power Quality

Under ideal operation conditions, the electrical current and voltage vary as a sine function of time. However, utility generator or distribution system problems such as voltage drops, spikes, or transients can cause fluctuations in the electricity, which can reduce the life of electrical equipment including motors and lighting systems. Moreover, an increasing number of electrical devices operating on the system can cause distortion of the sine waveform of the current and/or voltage. This distortion leads to poor power quality which can waste energy and harm both electrical distribution and devices operating on the systems.

Total Harmonic Distortion

The power quality can be defined as the extent to which an electrical system distorts the voltage or current sine waveform. The voltage and current for an electrical system with ideal power quality vary as a simple sine function of time, often referred to as the fundamental harmonic, and are expressed by Equations 4.4.1 and 4.4.2, respectively. When the power is distorted, due for instance to electronic ballasts (which change the frequency of the electricity supplied to the lighting systems), several harmonics need to be

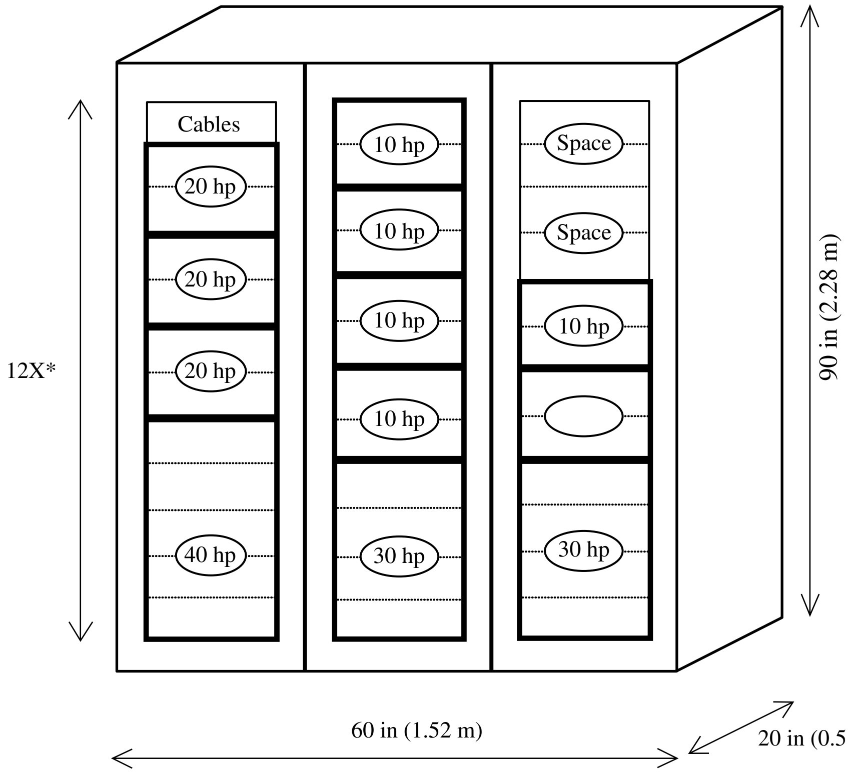

FIGURE 4.4.13 Layout for a Motor Control Center. *Note: the symbol X is used to designate one space factor [that is 6 in (15 cm)].

used in addition to the fundamental harmonic to represent the voltage or current time-variation as shown in Equations 4.4.26 and 4.4.27:

$$v(t) = \sum_{k=1}^{N_V} V_k \cos(k\omega - \theta_k) \tag{4.4.26}$$

$$i(t) = \sum_{k=1}^{N_I} I_k \cos(k\omega - \phi_k) \tag{4.4.27}$$

Highly distorted waveforms contain numerous harmonics. While the even harmonics (i.e., second, fourth, etc.) tend to cancel each other's effects, the odd harmonics (i.e., third, fifth, etc.) have peaks that coincide and significantly increase the distortion effects. To quantify the level of distortion for both voltage and current, a dimensionless number, referred to as the total harmonic distortion (THD), is determined through a Fourier series analysis of the voltage and current waveforms. The THD for voltage and current are respectively defined as follows:

$$THD_V = \sqrt{\frac{\sum_{k=2}^{N_V} V_k^2}{V_1^2}} \tag{4.4.28}$$

TABLE 4.4.10 Typical Power Quality Characteristics (Power Factor and Current THD) for Selected Electrical Loads

Electrical Load	Real Power Used (W)	Power Factor	Current THD (%)
Incandescent Lighting Systems			
100 W incandescent lamp	101	1.0	1
Compact Fluorescent Lighting Systems			
13 W lamp with magnetic ballast	16	0.54	13
13 W lamp with electronic ballast	13	0.50	153
Full-Sized Fluorescent Lighting Systems			
(2 lamps per ballast)	87	0.98	17
T12 40 W lamp with magnetic ballast	72	0.99	5
T12 40 W lamp with electronic ballast	93	0.98	22
T10 40 W lamp with magnetic ballast	75	0.99	5
T10 40 W lamp with electronic ballast	63	0.98	6
T8 32 W lamp with electronic ballast			
High Intensity Discharge Lighting Systems			
400 W high-pressure sodium lamp	425	0.99	14
with magnetic ballast	450	0.94	19
400 W metal halide lamp			
with magnetic ballast			
Office Equipment			
Desktop computer without monitor	33	0.56	139
Color monitor for desktop computer	49	0.56	138
Laser printer (in standby mode)	29	0.40	224
Laser printer (printing)	799	0.98	15
External fax/modem	5	0.73	47

Source: Adapted from NLPIP (1995).

$$THD_I = \sqrt{\frac{\sum_{k=2}^{N_I} I_k^2}{I_1^2}} \tag{4.4.29}$$

Table 4.4.10 provides current THD for selected but specific lighting and office equipment loads (NLPIP, 1995). Generally, it is found that devices with high current THD contribute to voltage THD in proportion to their share of the total building electrical load. Therefore, the engineer should consider the higher-wattage devices before the lower devices to reduce the voltage THD for the entire building or facility. Example 4.4.6 shows a simple calculation procedure that can be followed to assess the impact of an electrical device on the current THD. Thus, the engineer can determine which devices to correct first to improve the power quality of the overall electric system. Typically, harmonic filters are added to electrical devices to reduce the current THD values.

Example 4.4.6

Problem

Assess the impact on the current THD of a building of two devices: the 13 W compact fluorescent lamp (CFL) with an electronic ballast and the laser printer while printing. Use the data from Table 4.4.13.

Solution

Both devices have an rms voltage of 120 V (i.e., V_{rms} = 120 V); their rms current can be determined using the real power used and the the power factor given in Table 4.4.10 and Equation 4.4.30 (see Equation 4.4.5):

$$I_{rms} = \frac{P_R}{V_{rms} \cdot pf} \tag{4.4.30}$$

The above equation gives an rms current of 0.22 A for the CFL and 6.79 A for the printer. These values are the rms of each device's fundamental current waveform and can be used in the THD equation, Equation 4.4.27, to estimate the total harmonic current of each device:

$$I_{tot} = I_{rms} \cdot THD_I \tag{4.4.31}$$

The resultant values of 0.33 A for the CFL and 1.02 A for the printer show that although the printer has relatively low current THD (15%), the actual distortion current produced by the printer is more than three times that of the CFL because the printer uses more power.

IEEE (1992) recommends a maximum allowable voltage THD of 5% at the building service entrance (i.e., point where the utility distribution system is connected to the building electrical system). Based on a study by Verderber et al. (1993), the voltage THD reaches the 5% limit when about 50% of the building electrical load has a current THD of 55% or when 25% of the building electrical load has a current THD of 115%.

It should be noted that when the electrical device has a power factor of unity (i.e., $pf = 1$), there is little or no current THD (i.e., $THD_I = 0\%$) since the device has only a resistive load and effectively converts input current and voltage into useful electric power. As shown in Table 4.4.10, the power factor and the current THD are interrelated, and both define the characteristics of power quality. In particular, Table 4.4.10 indicates that lighting systems with electronic ballast have typically high power factor and low current THD. This good power quality is achieved using capacitors to reduce the phase lag between the current and voltage (thus improving the power factor as discussed in Section 4.4.2) and filters to reduce harmonics (and therefore increase the current THD value).

The possible problems that have been reported due to poor power quality include:

- Overload of neutral conductors in three-phase with four wires. In a system with no THD, the neutral wire carries no current if the system is well balanced. However, when the current THD becomes significant, the currents due to the odd harmonics do not cancel each other but rather add up on the neutral wire which can overheat and be a fire hazard.
- Reduction in the life of transformers and capacitors. This effect is mostly caused by distortion in voltage.
- Interference with communication systems. Electrical devices that operate with high frequencies, such as electronic ballasts (that operate at frequencies ranging from 20 to 40 kHz), can interfere and disturb the normal operation of communication systems such as radios, phones, and energy management systems (EMS).

4.4.6 Summary

This chapter provides an overview of the basic characteristics of electrical systems in HVAC applications for buildings. In particular, the operation principles of motors are emphasized. Throughout the chapter, several measures are described to improve the energy performance of existing or new electrical installations. Moreover, illustrative examples are presented to evaluate the cost effectiveness of selected energy efficiency measures. For instance, it is shown that the use of larger conductors for branch circuits can be justified based on the reduction of energy losses and thus operating costs. Moreover, the chapter provides suggestions to improve the power quality, increase the power factor, and reduce lighting energy use in buildings. These suggestions are presented to illustrate the wide range of issues that an engineer should address when designing, analyzing, or retrofitting electrical systems for buildings.

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