FOREWORD

It is the policy of the Sheet Metal and Air Conditioning Contractors’ National Association (SMACNA) to continually re-visit and update the technical manuals that have become the backbone of the industry.

With this policy in mind, a Task Force was organized and tasked with the purpose of reviewing and bringing up to date the HVAC Systems Applications. This manual is often referred to as one-of-three manuals that constitute SMACNA’s approach to HVAC systems. The other two manuals making up this group are the “HVAC Systems – Duct Design” manual and the “HVAC Systems – Testing, Adjusting and Balancing” manual. It is recommended that this manual be used in conjunction with the latest ASHRAE (American Society of Heating, Refrigeration and Air-Conditioning Engineers) “HVAC Applications” handbook.

Accordingly, the Task Force found areas that were in need of updating, primarily due to the advancement of technology, since the manual was last revised. This included hardware as well as software improvements. In addition, the science and methodology of Heating, Ventilating and Air Conditioning has changed, with an increased emphasis on energy efficiency and sustainability.

As you review this 2nd edition, you will notice that some chapters have been re-named, others have been deleted, and the order of the remaining chapters presented has been adjusted. However the emphasis is still on Air and Hydronic systems. You will also notice new topics of discussion, i.e.:

- Displacement Ventilation
- Variable Flow Refrigerant systems
- Fan Wall Systems
- HVAC Systems as they pertain to Sustainable Buildings
- Updated review of modern Variable Frequency Drives
- Revised/Updated Figures
- Expanded chapters for Smoke Control, Cleanrooms and Laboratory HVAC systems

It is the hope of the Task Force that this revised manual will be of value to those contractors who participate in the Design Build arena, as well as those that are involved with retro-fitting existing building systems. For those working in the retro-fit market, references to systems that may be obsolete by today’s standards have been retained to offer an insight into these systems.

Finally, the Glossary has also been updated to reflect these changes.

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1.1 INTRODUCTION

This chapter provides an introduction to heating, ventilating, and air conditioning (HVAC) applications. This chapter starts with stating the purpose of HVAC systems followed by a discussion of human thermal comfort and the industry standards that are used to establish the operating parameters for HVAC systems. Basic HVAC system components are then identified and discussed and a typical central HVAC system is presented. The importance of HVAC operating efficiency is addressed along with discussion of energy codes and standards. Energy codes and standards, green building rating systems and their relationship to HVAC system applications are also covered. This chapter closes by addressing various issues that should be considered when selecting, designing, and installing HVAC systems.

1.2 HVAC SYSTEM PURPOSE

The purpose of an HVAC system is to provide a suitable thermal environment in a defined space that meets the needs of the occupants and the activity that takes place in the space. Most HVAC systems are installed to establish an indoor environment within which building occupants can live, work, and play. The indoor environment impacts the quality of life, productivity, and well being of building occupants. As people spend an increasing amount of time inside buildings, HVAC systems and their associated control systems are becoming more important. To address this growing need, this manual focuses on HVAC equipment that creates human comfort indoors. Energy use in buildings is becoming increasingly important and impacting the type of the HVAC distribution system design, the HVAC equipment specified, and how the HVAC operates. HVAC systems are also required to provide suitable environmental conditions in addition to providing human comfort. In addition, energy use in buildings is becoming increasingly important and impacting the type of the HVAC distribution system design, the HVAC equipment specified, and how the HVAC operates. HVAC systems are also required to provide suitable environmental conditions for purposes other than human comfort.

1.3 HUMAN THERMAL COMFORT

1.3.1 Variables That Determine Human Thermal Comfort

Human thermal comfort is determined by the following four variables:

- Temperature
- Humidity
- Air Movement
- Air Quality

The objective of an HVAC system installed for human comfort is to control these four variables within an acceptable range for the occupants in the zone served by the HVAC system. The zone can be an entire building, an enclosed space within a building such as a room, or an area within a building. The HVAC system must be capable of controlling these four variables considering the activity taking place in the zone as well as changes in the outside environment, changes in the occupancy of the zone, and changes in the activity taking place in the zone. All of these changes take place continuously throughout the day and the HVAC system must be able to adjust and adapt to the dynamic nature of building thermal loads.

1.3.2 Establishing Parameters For Human Thermal Comfort

There are a number of industry standards and recommended practices that provide recommendations and guidance in establishing the parameters for achieving human thermal comfort for a given occupancy that take into account the activity being performed in the zone served by the HVAC system. Two important industry standards that establish the general parameters for human thermal comfort are published by the American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc. (ASHRAE) and are as follows:


Both of these standards are referenced in building codes, project technical specifications, and green building rating systems and their requirements may be mandatory on a building project. The following sections will discuss each of these industry standards.

1.3.2.1 ASHRAE Standard 55

ASHRAE Standard 55 specifies the combinations of indoor space environment and personal factors that will produce thermal environmental conditions acceptable to 80 percent of the occupants in a space. The environmental factors addressed are temperature,
Comfort zones defined by dry bulb temperature and relative humidity in a space on a psychrometric chart are defined in ASHRAE Standard 55 for summer and winter, see Figure 1-1. The comfort zones provide a range of temperature and humidity conditions that will satisfy most people who are appropriately dressed and performing light activity such as office work. People involved in strenuous activity or those wearing heavier clothing may need cooler conditions than the comfort zones, see Figure 1-1. The summer and winter comfort zones are different because people dress differently in each season and differences in humidity levels “move” the acceptable comfort temperatures.

The ASHRAE summer and winter comfort zones illustrated in Figure 1-1 comprise only a small part of the entire psychrometric chart indicating that a very narrow range of temperature and humidity conditions defines the range ascribed to human comfort.

Within these boundaries of temperature and humidity, any combination of these two variables - temperature and humidity will provide acceptable conditions for human comfort, see Figure 1-1.

The winter comfort zone temperatures for human comfort are lower because people wear heavier clothes and humidity requirements are also less resulting in the comfort zone being shifted to the left of the summer comfort zone.

As indoor air temperature varies the humidity also varies. The indoor humidity is generally within the comfort zone but in dry climates HVAC systems often need a humidifier to increase the moisture level of the conditioned air being supplied to the space. In humid climates, HVAC systems may need to include a separate dehumidifier or a dehumidifier integrated into the HVAC operation to remove excess moisture from the air to achieve human comfort. Note: See HVAC Systems Duct Design Chapter 3 for more details related to human comfort.

1.3.2.2 ASHRAE Standard 62.1

Indoor air quality (IAQ) is also a very important factor both for human comfort and the safety and health of occupants. Air quality is addressed in ASHRAE Standard 62.1-2004. The purpose of this standard is to specify minimum ventilation rates and indoor air quality that will be acceptable to human occupants and are minimize the potential for adverse health effects. This standard applies to all indoor or enclosed spaces that people may occupy. This standard considers the chemical, physical, and biological contaminants that can affect air quality. Thermal comfort requirements are not included in this standard because they are addressed in ASHRAE Standard 55.

Indoor air quality can be contaminated by the presence of all different types of pollutants that are both gaseous and particulate. Gaseous pollutants occur naturally as in the case of carbon dioxide and carbon monoxide. Carbon dioxide enters the indoor atmosphere as a result of simple respiration by humans and animals occupying the space. Carbon monoxide is a byproduct of combustion and can enter the indoor atmosphere as a result of cooking or heating with natural gas or propane or other organic materials such as wood. The use of chemicals such as paints, adhesives, and pesticides can also pollute the indoor environment and impact occupant comfort due to odors or cause headaches and sickness. Lastly, there are both inanimate and living particulates that are present in the air. These particulates include dust, mites, molds, bacteria, viruses and other contaminants of concern.

Today, people spend most of their lives living, working, and playing inside of buildings. As a result, indoor air quality is a very important aspect of any HVAC system and pollutants must either be removed from the air stream as it passes through the HVAC system or diluted to acceptable levels by the introduction of outside air to mix with recirculated conditioned air. In the case of gaseous pollutants, such as carbon dioxide and carbon monoxide, it is very difficult to remove them from the air stream using chemical air cleaners and gas phase filtration methods. It is typically more effective and economical in commercial and institutional buildings to increase the amount of outside air brought into the building and dilutes the concentration of gaseous pollutants like carbon dioxide and carbon monoxide to acceptable levels. However, in areas with high levels of outdoor pollutants—often classified by the EPA as “non-attainment area”—the outside air must be filtered of these gaseous pollutants and fine particulates when the outdoor air contains high levels of contaminants.

1.4 BASIC HVAC SYSTEM COMPONENTS

1.4.1 Four Basic HVAC System Components

The primary purpose of HVAC systems is to provide thermal comfort and a healthy environment for build-
FIGURE 1-1 COMFORT ZONE
ing occupants by controlling the building’s temperature, humidity, air movement, and air quality. Figure 1-2 provides a simplified block diagram of an HVAC system. The four basic components that comprise this basic HVAC system are as follows:

- Air Cleaner
- Thermal Conditioner
- Air Mover
- Air Distribution System

These four basic elements are found in any HVAC system regardless of the size or complexity. A self-contained HVAC unit such as a common residential window air conditioner needs all four of these building blocks as much as a large commercial or institutional HVAC system that includes a central heating and cooling plant, extensive air and hydronic distribution systems, and air handling units located in mechanical equipment rooms throughout the building.

1.4.1.1 Air Cleaner

Whether contaminants are in the outside air brought into the building or introduced in the building itself, they can impact the comfort and health of building occupants. These contaminants or pollutants can be either gaseous or particulate and need to be removed from the air stream or have their concentration reduced to where they are both harmless and unnoticeable to occupants. The purpose of the air cleaner in this case is to remove particulate pollutants from the air stream and the building by filtering them out before they pass through the HVAC system to protect it from becoming contaminated. Air cleaners used to remove particulate pollutants include media air filters and electronic air filters that trap particulates as they pass through. In the case of living organisms it is often better to destroy them instead of just trapping them in a filter. This can be accomplished by particulate destruction systems such as ultraviolet germicidal irradiation (UVGI) systems that use ultraviolet light.

Gas-phase air filtration removes gaseous pollutants from the air stream by using a material referred to as a sorbent. A commonly used sorbent is activated carbon that absorbs the gaseous pollutants, typically detected as odors, as they pass through the gas-phase filter.

1.4.1.2 Thermal Conditioner

After passing through the air cleaner in Figure 1-2, the air stream then passes through the thermal conditioner. The purpose of the thermal conditioner is to condition the incoming air by altering the incoming air’s temperature and humidity so that it is suitable for distribution and delivery to the zone or zones served. The thermal conditioner may be capable of adding heat, removing heat, or both using heat exchangers. For heating, the temperature of the air can be increased by passing it through hot water or steam heat exchangers, heat pump refrigeration heat exchanger, electric resistance heating or other means that transfers thermal energy to the air and increases its temperature. Similarly, the incoming air can be cooled by removing heat energy from it by passing it through a chilled water or refrigerant heat exchanger.

Humidity control can be accomplished using humidity control equipment such as humidifiers and dehumidifiers for defined spaces and special occupancies like indoor pools and ice rinks. However, humidification and dehumidification can also be accomplished introducing water into the air stream in the thermal conditioner or removing it by lowering the temperature and condensing the water out before reheating it to the desired temperature.

The thermal conditioner for an HVAC system can be either an all-in-one package, as in the case of unitary HVAC equipment, or distributed throughout the building with a central heating and cooling plant. The distributed thermal conditioner can be local air handling units or convection terminal units serving designated building zones.

1.4.1.3 Air Mover

An air mover is required to pull the air through the air cleaner and thermal conditioner as well as push the air through the air distribution system at the right airflow to the zone or zones served by the HVAC system, see Figure 1-2. Air movement is also one of the four variables that determine thermal comfort. In HVAC systems, the air mover is a mechanical fan that can be either an axial or centrifugal fan. For ducted air distribution systems, fans are typically either a centrifugal fan or a vane-axial fan to overcome the static pressure of the duct system. For other HVAC applications, fans whose characteristics best match the application are used. For example, propeller fans are often used for exhaust purposes where there is little static pressure at the discharge.

1.4.1.4 Air Distribution System

The purpose of the air distribution system is to deliver air to the zone served by the HVAC system. It is done
FIGURE 1–2 BASIC HVAC SYSTEM COMPONENTS

AIR CLEANER

THERMAL CONDITIONER

AIR MOVER

CONDITIONED ZONE

EXHAUST AIR

OUTSIDE AIR

BEFORE AIR
by mixing supply air with existing air in the zone to achieve the desired temperature without drafts or thermal stratification. The system extracts and returns the existing air in the zone with outside air and reconditioning or exhausts all air that is unsuitable to the outside. Outside air is mixed with return air from the zone and this mixture flows through the air cleaner, thermal conditioner, and air mover where it is conditioned and then delivered to the zone as supply air via the supply air distribution system, see Figure 1-2. The supply air is mixed with the existing air in the space and then either exhausted to the outside, recirculated through the return air duct to be reconditioned, or a combination of both. The air distribution system consists of ducts and plenums along with duct accessories and air outlets and inlets. Examples of duct accessories include dampers, turning vanes, and silencers. Air outlets and inlets include diffusers, grilles, and registers.

1.4.2 HVAC System Control

The simplified block diagram provided in Figure 1-2 would be fine if the thermal conditions inside and outside the building were fixed and did not change. However, the thermal environment both inside and outside the building is constantly changing and for the HVAC system to be effective it must be dynamic and capable of automatically adapting to these changing conditions. Inside the building the demands on the HVAC system are constantly changing and for the HVAC system to be effective it must be capable of gathering data on environmental conditions and current HVAC system operation. It should then process this data, and direct changes to the system operation based on this data. The HVAC control system is the key to the effective and efficient operation of the HVAC system.

1.4.3 Central HVAC System

Figure 1-3 illustrates a central HVAC system with a central heating and cooling plant that supplies hot and chilled water throughout the building via a hydronic distribution system. The hot and chilled water serves as the primary heat transfer media that warms and cools the supply air as it passes through water-to-air heat exchangers before being delivered to the conditioned space.

The boiler (Item 14) supplies hot water via a pump (Item 15) through the return hydronic piping system (Item 16) to both the preheat coil (Item 11) and the reheat coil (Item 13) in the local air handling unit and then is recycled back to the boiler through the hot water return hydronic piping (Item 16) for reheating and resupply, see Figure 1-3. Similarly, the chiller (19) supplies chilled water to the cooling coil (Item 12) to cool and dehumidify the supply air by pumping (Item 17) it through the chilled water hydronic piping system (Item 18). The chiller (Item 19) also has a condenser water loop that pumps (Item 21) condenser water through condenser water piping (Item 22) and the cooling tower (Item 20).

Conditioned air is pulled through the local air-handling unit and pushed through the supply duct system (Item 6) by the supply air fan (Item 5). The conditioned air is supplied to the space from the local air-handling unit by the supply duct system (Item 6) through air terminal devices (Item 7) in the space. Return air exits the space via return air inlets (Item 8) and returns to the local air-handling unit through the return duct system (Item 10). At the intake of the air-handling unit it is mixed with the appropriate amount of outside air before passing through the filters (Item 4) for air cleaning. The amount of outside air introduced to the conditioned space in relation to the amount of return air is determined by outdoor air intake (Item 1).

Air that is not recycled is either exhausted through the relief air vent (Item 3) in the return duct system (Item 10) or exhausted directly from the conditioned space through the exhaust air system (Item 9) when required. Exhaust air systems are often used when it is not desirable to recycle returning air due to odors or contaminants. Examples of when exhaust air systems are used include restrooms, kitchens, manufacturing processes, and laboratories.

1.5 HVAC SYSTEM OPERATIONAL EFFICIENCY

1.5.1 HVAC System Energy Use

Buildings account for about 48 percent of the U.S. energy consumption and result in the generation of more greenhouse gas than transportation or any other individual sector. Figure 1-4 provides a breakdown of commercial building energy use from data gathered by the U.S. Department of Energy (DOE) Energy Inform-
A. Duct Systems & Return
1. Outdoor Air Intake
   - Louver
   - Screen
   - O.A. Damper
2. Return Air Damper
3. Relief Air Damper
4. Filter Bank(s)
5. Supply Air Fan
6. Supply Duct System
7. Air Terminal Devices
8. Return Air Inlets
9. Exhaust Air System
10. Return Air Duct System

B. Hydronic Systems
11. Preheat Coil
12. Cooling Coil
13. Reheat Coil
14. Boiler
15. Pump (heating)
16. Piping (heating)
17. Pump (chilled water)
18. Piping (chilled water)

C. Refrigeration System
19. Chiller
20. Cooling Tower
21. Pump (Condenser Water)
22. Piping (Condenser Water)

Regulation of
Air used for ventilation and/or
"economizer cycle" cooling.

Regulation of return air to HVAC unit
Regulation of relief air to outside
Removes contaminants from airstream
Force to overcome system resistance
Path of supply air distribution
Air distribution devices to space
Inlet devices from conditioned space
Contaminated air removal system
Path for air return to HVAC unit

Preheats outdoor or mixed air
Cools and dehumidifies air
Heating or humidity control
Source of heating hot water or steam
Circulating device or boiler return pump
Path of hot water or steam
Chilled water circulating device
Path of chilled water

Source of chilled water for cooling
Disposes heat from condensing water
Condenser water circulating device
Path of condenser water

FIGURE 1–3 TYPICAL CENTRAL HVAC SYSTEM
HVAC and refrigeration systems in the estimated 4.9 million commercial buildings in the U.S. account for about 65 percent of commercial building energy usage per the following breakdown (Figure 1-5):

<table>
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<tr>
<th>Category</th>
<th>Percentage</th>
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<tr>
<td>Space Heating</td>
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<tr>
<td>Ventilation</td>
<td>7%</td>
</tr>
<tr>
<td>Water Heating</td>
<td>8%</td>
</tr>
<tr>
<td>Refrigeration</td>
<td>6%</td>
</tr>
<tr>
<td>Cooling</td>
<td>8%</td>
</tr>
<tr>
<td>Total HVAC&amp;R</td>
<td>65%</td>
</tr>
<tr>
<td>Lighting</td>
<td>21%</td>
</tr>
<tr>
<td>Total Consumption</td>
<td>36%</td>
</tr>
<tr>
<td>Lighting</td>
<td>8%</td>
</tr>
<tr>
<td>Office Equipment</td>
<td>9%</td>
</tr>
<tr>
<td>Other</td>
<td>1%</td>
</tr>
<tr>
<td>Cooking</td>
<td>6%</td>
</tr>
<tr>
<td>Computers</td>
<td>9%</td>
</tr>
<tr>
<td>Refrigeration</td>
<td>9%</td>
</tr>
<tr>
<td>Ventilation</td>
<td>2%</td>
</tr>
<tr>
<td>Water Heating</td>
<td>7%</td>
</tr>
<tr>
<td>Total HVAC&amp;R</td>
<td>65%</td>
</tr>
<tr>
<td>Total Consumption</td>
<td>36%</td>
</tr>
<tr>
<td>Lighting</td>
<td>8%</td>
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<tr>
<td>Office Equipment</td>
<td>9%</td>
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<tr>
<td>Other</td>
<td>1%</td>
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<tr>
<td>Cooking</td>
<td>6%</td>
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<tr>
<td>Computers</td>
<td>9%</td>
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<tr>
<td>Refrigeration</td>
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<tr>
<td>Ventilation</td>
<td>2%</td>
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<tr>
<td>Water Heating</td>
<td>7%</td>
</tr>
<tr>
<td>Total HVAC&amp;R</td>
<td>65%</td>
</tr>
<tr>
<td>Total Consumption</td>
<td>36%</td>
</tr>
</tbody>
</table>

The operational efficiency of HVAC systems in commercial buildings is a very important consideration when selecting an HVAC system for a particular building. It is also dependent on occupancy within a building and procuring the equipment and materials that comprise that HVAC system. Energy standards and codes have been adopted by most federal, state, and local government entities and both public and private buildings must meet the requirements of those standards and codes. These energy standards and codes are adopted to ensure that energy efficiency is a primary consideration in the selection of both the HVAC system design and the equipment specified. These energy standards and codes are important because they not only establish minimum energy efficiency standards but also may restrict or ban the use of certain systems and equipment.

1.5.2 ASHRAE/IESNA Standard 90.1

ASHRAE publishes and continuously updates ASHRAE/IESNA Standard 90.1 entitled Energy Standard for Buildings Except Low-Rise Residential Buildings in conjunction with the Illuminating Engineering Society of North America (IESNA). This standard is the most commonly referenced energy standard in the U.S., ASHRAE/IESNA 90.1 provides both prescriptive and performance requirements for HVAC system operation and has been adopted directly as a requirement by federal, state, and local governments and agencies as well as included by reference in building codes, construction contracts, and green building rating systems. The purpose of this standard is to provide the minimum requirements for the energy-efficient design of buildings. ASHRAE/IESNA Standard 90.1
1.5.3 International Energy Conservation Code

There are a myriad of energy codes in use today throughout the U.S. One model energy code that is commonly encountered is the International Energy Conservation Code (IECC). The IECC is published by the International Code Council (ICC) as part of its family of building codes. Like ASHRAE/IESNA Standard 90.1, the IECC establishes minimum prescriptive and performance-based requirements for the design of energy-efficient buildings. It should be noted that meeting the requirements of ASHRAE Standard 90.1 is one of the compliance methods permitted by the IECC.

1.5.4 Other Energy Codes and Standards

There are many other building energy codes and standards promulgated by federal, state, and local governments as well as other industry organizations. One example is Title 24, Part 6 of the California Code of Regulations, California’s Energy Efficiency Standards for Residential and Nonresidential Buildings which was established in 1978 in response to a legislative mandate to reduce California’s energy consumption. As concern about the environment and energy use increases, energy codes and standards will become increasingly important and have a greater impact on HVAC system design, installation, operation, maintenance, and monitoring.

1.6 GREEN BUILDING RATING SYSTEMS

In addition to energy codes and standards, the green building movement is also impacting HVAC system and equipment selection, procurement, installation, operation, maintenance, and monitoring. HVAC systems are a key element of any green building from both an energy and indoor air quality standpoint. As a result, HVAC systems are a central part of any green building rating system.

The purpose of a green building rating system is to provide an objective standard for certifying that a building has minimal environmental impact, which is commonly referred to as “green.” As a result of the public’s increasing concern about the environment
and energy consumption there is a growing movement among both public and private building owners to have their buildings certified as “green” by an objective third-party rating system. Many federal, state, and local governments and governmental agencies are beginning to require that its buildings as well as private buildings under their jurisdiction either be certified or certifiable according to a third-party green building rating system.

There are a number of green building rating systems that are in use or being developed. These systems are being developed and applied at the international, national, and local levels. All of these rating systems are similar in the green building criteria they address but can be very different in their intent, criteria, emphasis, implementation, and in other important ways. The HVAC contractor needs to be aware of the specific requirements included in the green building rating system being used on a project prior to bidding. See the appendix for more specific information.

1.7 HVAC SYSTEM SELECTION PARAMETERS

The selection of the type of building HVAC system is one of the most critical decisions made during the design process. Many factors must be analyzed but the cost of installation and operation are always the most important except in rare circumstances. Figure 1-5 provides a list of parameters that should be considered when selecting an HVAC system for a particular building or occupancy.

When considering the type of system or systems to be selected, it is important to determine whether the building heating and cooling load is mostly sensible or latent, high or low per area of conditioned space, uniformly or irregularly distributed within the building spaces, and constant or variable as a regular pattern. These considerations are necessary for selecting the HVAC system, delineating HVAC system zones, and selecting the needed controls to maintain the desired comfort conditions under varying thermal loads.

1.8 SPACE CONDITIONS

1.8.1 Space Considerations

Each zone whether it is an individual room, designated space, or the entire building presents an individual challenge. The structure, its thermal behavior, and the interaction of individual zones or building with external and internal thermal loads, and the control of these loads by the system must be considered. The installed HVAC system, the system controls, and the building must perform as a total system. The wide range of spaces and buildings for application of air conditioning for human comfort may be divided into single-purpose occupancies and multi-purpose occupancies.

1.8.2 Single-Purpose Occupancies

Single-purpose occupancies involve either an individual or a multitude of individuals gathered for a common purpose with a single environmental control zone. Examples include a room, residence, or large open area with or without partitions. A larger area can be an office space, restaurant, beauty salon, or other occupancy. The largest area can be a church, theater, or auditorium, or similar space. The common feature is a building with one or more large open spaces as a major area to be conditioned.

1.8.3 Multi-Purpose Occupancies

Multi-purpose occupancies involve many people gathered for various purposes in one or more rooms or a multi-story building. These buildings may serve a single purpose such as a department store, a library, a museum, a laboratory, a school or a factory. Generally the multi-room, multi-story buildings may be office buildings, hotels, apartment buildings or hospitals. Other multi-purpose occupancies also include apartment houses, condominiums, schools, colleges, medical and shopping centers, and factories. The major characteristic of these occupancies is a multiplicity of environmental control zones served by a single or distributed HVAC system.

1.9 HVAC EQUIPMENT AND SPACE

HVAC equipment piping and duct systems require space. Higher energy efficiency typically means larger equipment. Reducing one of the major energy users—the HVAC fan—requires more efficient air distribution systems which can best be realized via lower operating pressures and more efficient air conveyance design. Air moves with less resistance along a path of least resistance which requires duct fittings with long smooth turns starting an appropriate distance from fans and accessories in the duct that create turbulence that interferes with airflow. The size, location and configuration of equipment rooms is one of the most important decisions that will impact HVAC system energy use for the entire life of the building, not just the life of the “first” HVAC systems equipment and distribution system. Also important is that sufficient space must be provided to allow equipment servicing and replacement. Designers need to recognize that, based on historic trends and simple physics, the
next generation of equipment will almost certainly be larger than what is installed today. The space available for proper service of HVAC equipment should not be overlooked. Maintained HVAC equipment means better operating efficiencies, better indoor air quality and overall occupant comfort.

Large central HVAC systems usually have large space requirements. Building space is needed for the air handling units, duct systems, transitions from air handlers to duct, hydronic piping, chillers, boilers, and other associated equipment. Much of this can be in or adjacent to the conditioned areas. Most cooling towers, evaporative or air-cooled condensers, and condensing units are located outdoors and similar allowances for future equipment replacements need to be considered.
2.1 INTRODUCTION

This purpose of this chapter is to introduce the fundamentals of HVAC system applications that will serve as the basis for the remainder of this manual. This chapter provides a brief discussion of air chemistry, moist air physical properties including the psychrometric chart, and moist air energy content. General terms that are used throughout the manual such as zone, space, and airflow are then defined in this chapter as they are used throughout this manual. Ways of categorizing HVAC systems based on space conditioning method, primary heat transfer media, and physical layout are also covered.

2.2 AIR CHEMISTRY

2.2.1 Air Ingredients

Air near the surface of the earth consists of a mixture of three ingredients that include the following:

- Standard Dry Air
- Water Vapor
- Other Constituents

2.2.1.1 Standard Dry Air

Standard dry air is composed of a variety of gases but it primarily consists of oxygen and nitrogen that make up about 99 percent of air by volume. A breakdown of dry air by gas as a percent of volume is as follows:

<table>
<thead>
<tr>
<th>GAS</th>
<th>PERCENT BY VOLUME</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nitrogen</td>
<td>78.080</td>
</tr>
<tr>
<td>Oxygen</td>
<td>20.950</td>
</tr>
<tr>
<td>Noble Gases</td>
<td>0.930</td>
</tr>
<tr>
<td>Carbon Dioxide</td>
<td>0.038</td>
</tr>
<tr>
<td>Total</td>
<td>100.000</td>
</tr>
</tbody>
</table>

Nitrogen and oxygen make up about 78 percent and 21 percent of the atmosphere by volume, respectively. The six noble gases are helium, neon, argon, krypton, xenon, and radon and they comprise about one percent of the air by volume. Carbon dioxide makes up only a fraction of one percent of dry air but it is a very important consideration in HVAC systems and impacts IAQ and ventilation requirements.

2.2.1.2 Water Vapor

Water vapor is the second ingredient in air and it can vary greatly from almost nothing to about 5 percent by volume. When the amount of water in air is very low which can get down as low as 0.5 percent by volume, it is referred to as dry air. When air contains a sufficient amount of water vapor that is noticeable to people, the air is referred to qualitatively as humid. Humidity is an important consideration in human thermal comfort and an important factor in HVAC system design, installation, and operation.

2.2.1.3 Other Constituents

Other constituents can also be found in the air that reflects local conditions. These other constituents can simply be an annoying odor or could be a chemical compound like carbon monoxide that can become an irritant or a health risk when they reach a certain concentration in the air. These chemical compounds can occur naturally, they can be the byproduct of combustion including cooking, result from off gassing by building materials and furnishings, or caused by the use of adhesives, coatings, pesticides, or other chemicals in a building. In addition, there can be particulate matter in the air that can also cause irritation and be a health risk to occupants. The particulate matter could be inanimate such as dust or it could be living organisms like bacteria, viruses, mold, and other small organisms in the air. When known to be present and possible, the HVAC system should mitigate the impact of these constituents on occupants by increasing ventilation to reduce their concentration or removing them from the air stream using air cleaning devices.

2.3 MOIST AIR PHYSICAL PROPERTIES

2.3.1 Thermophysical Properties For HVAC

The thermophysical properties of air-water mixtures can be described by the following five metrics:

- Dry-Bulb Temperature
- Wet-Bulb Temperature
- Relative Humidity
- Humidity Ratio
- Dew Point Temperature

2.3.1.1 Dry-Bulb Temperature

The dry-bulb (DB) temperature of the air is the temperature that is measured by an ordinary thermometer that
is freely exposed to the air but shielded from both radiation and moisture.

### 2.3.1.2 Wet-Bulb Temperature

Wet-bulb (WB) temperature is measured using a WB thermometer. A WB thermometer is a standard thermometer that has its bulb wrapped in a piece of material (cloth) that is typically referred to as a wick or sock. The wick is wetted with water and temperature is measured with air flowing rapidly over the thermometer. The water evaporates and cools the thermometer bulb and the measured temperature is referred to as the WB temperature. The WB temperature will always be less than the DB temperature except at the dew point when the WB temperature is equal to the DB temperature.

### 2.3.1.3 Percent Relative Humidity

Relative humidity (RH) is dimensionless and is typically expressed as a percentage. Relative humidity is the ratio of the amount of water vapor in the air to the water vapor present in saturated air at the same temperature and barometric pressure. The relative humidity of the air in a space can be determined from the measured DB and WB temperatures by calculation or using a psychrometric chart as will be discussed later in this section.

### 2.3.1.4 Humidity Ratio

Humidity ratio is usually expressed in grains of moisture per pound of dry air (grams of water per kilogram of dry air) and is the ratio of the mass of water vapor to the mass of dry air in a space.

### 2.3.1.5 Dew-Point Temperature

The dew-point (DP) temperature is the temperature at which the water vapor in a space will begin condensing for a given humidity and pressure. The DP temperature can also be defined as the temperature corresponding to saturation (100 percent relative humidity) for a given absolute humidity at constant pressure.

### 2.3.2 Psychrometric Chart

Psychrometry is the science of air-water mixtures. A psychrometric chart is a graphical representation that shows the relationship between the five metrics that can be used to describe the thermophysical properties of moist air at a particular barometric pressure. Figure 2-1 provides an example psychrometric chart at sea level with a standard barometric pressure of 29.921 inches of Mercury.

By knowing any two of the five metrics, the other three can be found from the psychrometric chart. For example, given a dry bulb temperature of 85°F and a wet bulb temperature of 70°F, the other four metrics can be read directly from the psychrometric chart (Figure 2-2) as follows:

Relative Humidity = 47%

Humidity Ratio = 42 Grains Of Moisture/Pound Of Dry Air

Dew Point Temperature = 37°F

A psychrometric chart represents the relationships between the physical properties of moist air at a constant atmospheric pressure that is usually expressed in terms of elevation above or below sea level. The psychrometric charts are for atmospheric pressure at sea level, see Figures 2-1 and 2-2. If the elevation of the building site is significantly different than sea level then a psychrometric chart for the actual elevation of the building site should be used. Psychrometric charts for elevations other than sea level are available from the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE), HVAC equipment manufacturers, and other sources.

### 2.4 MOIST AIR ENERGY CONTENT

#### 2.4.1 Sensible Heat

Sensible heat or the specific enthalpy of dry air is any heat transfer that causes a change in temperature. Heating and cooling of air and water that may be measured with a thermometer is sensible heat. Heating or cooling coils that simply increase or decrease the air temperature without a change in moisture content are example of sensible heat.

Sensible heat is energy that is transferred by an object that has a higher temperature than its surroundings. Sensible heat is transferred by conduction or convection, or a combination of both. Sensible heat transfer does not involve a change in state and is a function of the object’s mass, its specific heat capacity, and its temperature above its surroundings.

#### 2.4.2 Latent Heat

Latent heat or the specific enthalpy of water vapor is the amount of heat necessary to change a quantity of water to water vapor without changing either the tem-
FIGURE 2-1 TYPICAL HVAC PSYCHROMETRIC CHART
FIGURE 2-2 PSYCHROMETRIC CHART EXAMPLE

Chart Source: Trane Corporation
temperature or pressure. When water vapor is vaporized and passes into the air, the latent heat of vaporization passes into the air along with the vapor. Likewise, latent heat is removed when water vapor is condensed.

2.4.3 Total Heat

Total heat or the specific enthalpy of moist air is the sum of the sensible heat plus the humidity ratio times the latent heat. In many cases, the addition or subtraction of sensible and latent heat at terminal coils occurs simultaneously.

2.4.4 Moist Air Energy Content From The Psychrometric Chart

The total heat or specific enthalpy of moist air can be read directly from the psychrometric chart, see Figure 2-2 for the example where the dry bulb and wet bulb temperatures are known to be 85°F and 70°F, respectively. The enthalpy read from the scale running diagonally along the left side of the psychrometric chart is 34.2 Btu per pound of dry air.

2.5 ZONE DEFINED

2.5.1 Zone Versus Space

The area being conditioned by an HVAC system is often referred to as a “space” and could be a room within a building or the entire building. HVAC systems serve specific rooms as in the case of a stand alone HVAC unit in a hotel room or entire buildings as in the case of a central HVAC system for a residence or small commercial building. However, in most commercial and institutional HVAC systems different areas of the building have different requirements due to the number of people and activity being carried out in the area, the location of the area relative to outside walls, time of day and year, among many other variables. As a result, different areas in commercial and institutional buildings of any size have multiple areas with different HVAC requirements at any time. These areas may be individual rooms but are most likely to be groups of rooms with common coincident HVAC requirements or larger contiguous “rooms” within the building such as open office areas, atriums, and auditoriums that can be divided into smaller sub-areas that each have different coincident HVAC requirements.

The terms zone and space have very specific meanings in HVAC system design, installation, and operation. In order to be consistent throughout this manual, the term “zone” will be used when referring to the area that an HVAC system serves and the term “space” will be used to describe an architecturally definable area within a building. The definition of each of these terms as they are used throughout this manual is provided in the paragraphs that follow.

2.5.2 Zone Defined

A “zone” is a designated area of a building that has its own sensors to monitor the thermal conditions in that area and control the HVAC system serving that area. Zoning is not determined by the number of air outlets in a designated area but rather the control of those air outlets. One set of sensors can control multiple air outlets. For example, a single-family residence is typically single zone because there is usually a single thermostat centrally located in the home that controls the HVAC system that serves the entire home.

Figure 2-3 illustrates the floor plan of a commercial office building. Each office (e.g. Room 201) has its own thermostat that monitors and controls the temperature of the office, see Figure 2-3. Similarly the conference room (Room 204) also has its own thermostat making it a zone as well. The open office area is divided into four quadrants that are each monitored and controlled by its own thermostat (e.g. Column B4). As a result, the open office area consists of four zones even though it is a single contiguous space.

Typically, a thermostat controls the thermal condition in a commercial building zone. The thermostat monitors the temperature in the zone and controls the HVAC system serving that zone. This thermostat compares the measured temperature in the zone to its setpoint which is the desired temperature. However, zoning in commercial buildings can include sensors that track other variables including occupancy and carbon dioxide that work together with the thermostat to provide a comfortable and healthy environment for occupants as well as efficient operation of the HVAC system serving the zone.

2.5.3 Space Defined

A “space” is an area of a building that is defined by walls or other architectural features that restrict free air movement between that area and other parts of the building. A space can be a zone when there is only one set of sensors that monitor and control the thermal conditions in the space. An example of a space that might also be a zone in a commercial building would be the offices or conference room, see Figure 2-3. Similarly, a large space such as the open office area in Figure 2-3 might include multiple zones where multiple sets of sensors are used to control thermal conditions in different areas within the space.
FIGURE 2-3 COMMERCIAL OFFICE BUILDING FLOOR PLAN
2.6 AIRFLOW DEFINED

The term “airflow” is used throughout this manual to denote the volumetric rate of air supply through the air distribution system to a zone or space. The volumetric rate of air supply is usually expressed as either cubic feet per minute (cfm) in inch-pound (I-P) units or cubic liters per second (l/sec) in metric units.

2.7 SPACE CONDITIONING METHODS

An HVAC system conditions a space by introducing air with the needed combination of temperature and humidity into the zone to mix with existing air to ensure that the thermal conditions in the space are within the thermal comfort zone. In addition, sufficient outside air needs to be supplied to the space to ensure indoor air quality. The effectiveness of the HVAC system depends on the following three factors:

- The airflow supplied to the zone.
- The temperature and humidity of the air being supplied to the zone.
- Air displacement or diffuser throw to achieve the proper mixing of airstreams in the zone.

To heat or cool a space, these two factors are combined in different ways depending on the type of HVAC system installed. HVAC systems are often classified by how these two factors are combined to condition a zone as follows:

- Constant Air Volume – Variable Air Temperature (CAV-VAT)
- Variable Air Volume – Constant Air Temperature (VAV-CAT)
- Variable Air Volume – Variable Air Temperature (VAV-VAT)

2.7.1 Constant Air Volume – Variable Air Temperature

A constant air volume-variable air temperature (CAV-VAT) HVAC system supplies conditioned air to the HVAC zone that it serves at a constant airflow. The temperature in the zone is controlled by a CAV-VAT HVAC system by varying the temperature of the air supplied to the space. To cool the zone in response to an increase in the thermal load, cooler supply air is delivered by the CAV-VAT HVAC system at constant airflow to mix with the existing air in the zone and lower the temperature. Similarly, when the zone requires less cooling because the thermal load has decreased, the CAV-VAT HVAC system delivers warmer air at a constant airflow to increase the temperature of the zone.

CAV-VAT HVAC systems were the first type of systems that were used in commercial and institutional buildings for cooling. CAV-VAT systems are still being used in existing buildings but these systems are often being converted to or replaced by the more efficient variable air volume-constant air temperature (VAV-CAT) systems that will be discussed next. Additionally, the more efficient VAV-CAT systems are supplanting CAV-VAT systems in most new construction today.

2.7.2 Variable Air Volume – Constant Air Temperature

A VAV-CAT HVAC system works just the opposite of a CAV-VAT HVAC system to condition the zone served. With a VAV-CAT HVAC system, the temperature of the supply air remains constant and the airflow is varied. When cooling is required because the thermal load in the zone increases, a VAV-CAT cooling system delivers an increased airflow to the zone at a constant temperature. The increased conditioned air delivered to the zone mixes with the existing higher temperature air and the zone’s temperature decreases to the desired value. Similarly, if the thermal load in a zone served by a VAV-CAT HVAC system decreases, the amount of conditioned air delivered to the zone is reduced allowing the zone’s temperature to increase to the desired value. VAV-CAT systems are more efficient than CAV-VAT systems. Almost all HVAC systems installed in commercial and institutional buildings are VAV-CAT HVAC systems.

2.7.3 Variable Air Volume – Variable Air Temperature

A third method of maintaining the desired temperature in an HVAC zone would be to use a variable air volume-variable air temperature (VAV-VAT) system where both the airflow and temperature of the conditioned air delivered to the space is varied. While this is a possibility, HVAC systems for commercial and institutional buildings are not designed, installed, or operated as VAV-VAT systems. However, it could be argued that many HVAC systems that are classified as either CAV-VAT or VAV-CAT systems are actually VAV-VAT systems. Examples of this might include a terminal unit in a hotel room that is usually considered a CAT-VAV system but has a multi-speed fan whose speed can be manually adjusted by the occupant while the temperature of the air supply is also varied by the
operation of the unit. Similarly, a VAV system with reheat that would be classified as a VAV-CAT system actually varies the temperature of the air supplied by the reheat coil in the VAV box as well as the airflow. Therefore, the classification of an HVAC system based on whether airflow, temperature, or when both are varied is based on its dominant mode of operation which is typically either as a CAV-VAT or VAV-CAT system.

2.8 HVAC SYSTEMS CATEGORIZED BY PRIMARY HEAT TRANSFER MEDIA

One way to categorize HVAC systems is by the primary heat transfer medium that is used to remove or add heat to the air stream delivered to the HVAC zone. HVAC systems can be categorized using primary heat transfer media as follows:

- All-Air HVAC Systems
- Air-Hydraulic HVAC Systems
- All-Hydraulic HVAC Systems
- Direct Refrigerant HVAC Systems
- Variable Refrigerant Flow HVAC Systems
- Hybrid HVAC Systems

2.8.1 All-Air HVAC Systems

All-air HVAC systems use air as the primary heat transfer medium. Figure 2-4 provides a schematic diagram of an all-air HVAC system. A mixture of return air from the zone and outside air is conditioned by the air-handling unit and supplied to the zone. The air-handling unit cleans the air and then either cools or heats it using water-to-air heat exchangers shown as the cooling and heating coil, respectively. Air is forced through the system by the supply fan in the air-handling unit which is the air mover. Conditioned air is then supplied to one of more zones served by the air-handling unit through a supply duct and an air inlet.

Return air leaves the room through an air outlet and is routed back through a return duct, plenum or a combination of the two. A portion of the return air is recirculated through the air-handling unit and the remainder is exhausted depending on the zone thermal load, outside air temperature, and need for outside air to maintain for air quality.

In the schematic diagram of the all-air HVAC system illustrated in Figure 2-4, heat is added or removed from the air stream by the heating and cooling coils in the air-handling unit. The heating coil uses either hot water or steam supplied from a central heating plant that would typically be a boiler and the cooling coil is supplied by chilled water from a central cooling plant that would typically be a chiller.

The central heating plant is represented by a single box in Figure 2-4 but is comprised of boilers, boiler feed pumps, hot water circulating pumps and other mechanical equipment that will be covered in Chapter 10. Similarly, the central cooling plant block in the schematic diagram includes chillers, cooling towers, chilled water circulating pumps and other mechanical equipment that will be discussed in Chapter 9. The supply and return of the hot water or steam and the chilled water is accomplished using a hydronic piping system which is covered in Chapter 15.

Adding or removing heat from the air stream using water-to-air heat exchangers and either hot water or steam and chilled water generated by a central plant and transported to the air-handling unit using a hydronic piping system is only one way that an all-air HVAC system can operate. To cool the air stream, the air-handling unit could have its own refrigeration unit and instead of a water-to-air heat exchanger use a direct refrigerant-to-air heat exchanger, thereby, eliminating the need for chilled water and the central cooling plant. Similarly, heat can be added to the air stream by a variety of methods such as electric resistance heating or direct or indirect gas-fired heating. A common example of a standalone all-air HVAC system would be a rooftop packaged HVAC unit which is also categorized as a unitary system and will be covered in Chapter 8.

The schematic diagram of the HVAC system in Figure 2-4 also shows both a cooling coil for removing heat from the air stream and a heating coil for adding heat to the air stream. Depending on the building HVAC needs, climate where the building is located, and HVAC system design, both a cooling function and heating function may not be required. The all-air HVAC system may only be required to provide cooling and only need a cooling coil. On the other hand, the all-air HVAC system may be heating-only and only need a heating coil. Also, Figure 2-4 is intended to illustrate the basic layout and function of an all-air HVAC and a number of important components that might be needed to make the system work in a given application such as dampers and return air fans have been omitted to focus on the base system. These addi-
FIGURE 2-4 ALL-AIR HVAC SYSTEM

LEGEND

AC       AIR CLEANER
AI       AIR INLET
AM       AIR MOVER
AO       AIR OUTLET
CC       COOLING COIL
CCP      CENTRAL COOLING PLANT
CHP      CENTRAL HEATING PLANT
CWR      CHILLED WATER RETURN
CWS      CHILLED WATER SUPPLY
EA       EXHAUST AIR
HC       HEATING COIL
HWR      HOT WATER RETURN
HWS      HOT WATER SUPPLY
OA       OUTSIDE AIR
RA       RETURN AIR
SA       SUPPLY AIR
tional components will be added and discussed in later chapters of this manual.

2.8.2 Air-Hydronic HVAC Systems

Figure 2-5 provides a schematic diagram of an air-hydronic HVAC system. An air-hydronic HVAC system uses both air and water as the HVAC system’s primary heat transfer media, see Figure 2-5. The air stream is filtered and conditioned in the air-handling unit and then supplied to the space through a water-to-air heat exchanger. In this simple air-hydronic HVAC system, the air is cooled in the air handling unit by a water-to-air heat exchanger but this could as easily been a refrigerant-to-air heat exchanger as discussed in the previous section. When heat is required, the cooling coil can be shut down and the air stream supplied by the air-handling unit could be heated by the water-to-air heat exchanger in the zone’s terminal unit.

While the zone’s terminal unit is shown as a water-to-air heat exchanger in Figure 2-5, it could just as well be an electric resistance heater as discussed in the previous section. Similarly, the air-hydronic HVAC system shown in Figure 2-5 uses an induction unit where the supply air passes directly through the hydronic heating coil before being supplied to the zone. Induction HVAC systems using hot water or steam will be covered in Chapter 7. Other air-hydronic systems may not pass the supply air through the terminal unit but instead deliver it directly to the space as in the case with an all-air HVAC system. The zone's heating needs would then be met by radiators, finned tube radiators, and other similar water-air heat exchangers that are covered in Chapter 15 with hydronic piping.

2.8.3 All-Hydronic Systems

A schematic diagram of an all-hydronic system using a local fan coil unit to provide conditioned air to the zone, see Figure 2-6. In this case, the fan coil unit is dedicated to the zone and includes an air cleaner along with water-to-air heat exchangers for the cooling coil and the heating coil. The supply air is provided directly to the zone and there is no supply air duct system unlike the all-air and air-hydronic HVAC systems. A return air system recirculates air from the zone through the unit and mixes the recirculated air with outside air in the fan coil unit itself, see Figure 2-6.

An all-hydronic HVAC system eliminates the need for an air distribution system and instead uses water as the sole heat transfer media requiring a hydronic distribution system instead, see Figure 2-6. Also, if either heating or cooling is not required then those coils and their associated central plant and distribution system can be eliminated. All-hydronic HVAC systems will be addressed in Chapter 15 of this manual with hydronic distribution systems.

2.8.4 Direct Refrigerant HVAC Systems

Direct refrigerant HVAC systems are also typically referred to as direct expansion (DX) HVAC systems. Direct refrigerant HVAC systems eliminate the need for air and hydronic distribution systems because these HVAC units are self contained and intended to serve the zone in which they are located. Direct refrigerant systems are often cooling only but they can also provide a heating function by reversing the refrigeration cycle or including electric resistance heating. When the refrigeration cycle is reversible and the system can provide both cooling and heating it is commonly referred to as a heat pump. These self-contained HVAC systems are typically classified as unitary HVAC systems and these systems are covered in Chapter 8 of this manual.

An example of a direct refrigerant HVAC system would be a common window air conditioner or a through-wall HVAC unit that is often used in hotels, motels, and apartment buildings that have a number of rooms that are each independent zones and are located on an outside wall of the building. Other applications of direct refrigerant HVAC units are when there is a particular zone such as a data processing or telecommunications room that needs additional cooling. This room can either be located on an outside wall to permit the rejection of heat directly to the outside or use a split system using refrigerant piping to connect the supply fan and cooling coil inside with the condenser and compressor outside.

2.8.5 Variable Refrigerant Flow HVAC Systems

Variable refrigerant flow (VRF) or variable refrigerant volume (VRV) HVAC systems are a variation of the direct refrigerant HVAC systems discussed in the previous section. Both the VRF HVAC system and the standard direct refrigerant HVAC systems use refrigerant as their heat transfer media and operate using the mechanical refrigeration cycle described in Chapter 8. A VRF system can offer advantages over a standard direct refrigerant HVAC system in some instances where the system needs to serve multiple zones, space for installation of ductwork is restricted, and flexibility for future changes is desired.

A standard direct refrigerant HVAC system has one evaporator and one condenser that are either both con-
FIGURE 2–5 AIR-HYDRONIC HVAC SYSTEM
FIGURE 2-6 ALL-HYDRONIC HVAC SYSTEM

LEGEND

- AC: AIR CLEANER
- AI: AIR INLET
- AM: AIR MOVER
- AO: AIR OUTLET
- CC: COOLING COIL
- CCP: CENTRAL COOLING PLANT
- CHP: CENTRAL HEATING PLANT
- CWR: CHILLED WATER RETURN
- CWS: CHILLED WATER SUPPLY
- EA: EXHAUST AIR
- HC: HEATING COIL
- HWR: HOT WATER RETURN
- HWS: HOT WATER SUPPLY
- OA: OUTSIDE AIR
- RA: RETURN AIR
- SA: SUPPLY AIR

CENTRAL HEATING & COOLING PLANT

RETURN DUCT AND/OR PLENUM

HYDRONIC PIPING

ZONE
tained in a single enclosure as in the case of a common window air conditioning unit or separated with the evaporator located in the building and the condenser located outside the building as in the case of split direct refrigerant HVAC systems found in residential and light commercial HVAC systems. With a split direct refrigerant HVAC system the evaporator and condenser are located as close together as possible to limit the length of refrigerant piping connecting them and conditioned air is provided to the zone served through ductwork.

With a VRF HVAC system, multiple evaporators are interconnected to a single condenser by a refrigerant piping system. Each evaporator serves its own zone so the occupants in that zone can control the temperature of that zone and the evaporator can monitor and respond to changes in zone temperature automatically independent of the other evaporators served by the central condenser. Controllability is achieved by varying the refrigerant flow through the evaporator which is where the VRF HVAC system gets its name. This variable refrigerant flow and the ability of each evaporator to respond to thermal conditions in the zone it serves has the potential to make VRF systems more efficient than standard direct refrigerant HVAC systems in some applications. VRF HVAC systems with reversible refrigeration cycles can also operate as a heat pump supplying both cooling and heating to the zones served as described in Chapter 8.

A disadvantage of VRF HVAC systems is that they do not provide a means for delivering outside air to a space. If ventilation and indoor air quality (IAQ) are concerns, then a separate ventilation system may need to be provided to supply outdoor air to the zones served. The need to provide ventilation to zones served by a VRF HVAC system should be a consideration when evaluating its use over other HVAC systems.

### 2.8.6 Hybrid HVAC Systems

Commercial and institutional buildings often use a combination of these basic systems to provide an HVAC system to provide a comfortable and healthy environment for occupants as well as minimize HVAC system life-cycle cost. This is accomplished by utilizing systems based on their advantages for particular applications in buildings. For instance, an all-air HVAC system may be used to cool the interior of an office building while an air-hydraulic HVAC system might be used to provide both cooling and heating to the perimeter of the building which has the highest heat gain in the summer and the highest heat loss in the winter.
CHAPTER 3

VARIABLE-AIR-VOLUME HVAC SYSTEMS
CHAPTER 3

VARIABLE-AIR-VOLUME HVAC SYSTEMS

3.1 INTRODUCTION

This chapter addresses variable-air-volume (VAV) HVAC systems that are variable air volume – constant air temperature (VAV-CAT) HVAC systems as described in Chapter 2. VAV HVAC systems produce constant-temperature air and control the thermal condition of the building zones served by varying the constant-temperature airflow to each of these zones. VAV HVAC systems are the system of choice for commercial and institutional buildings because they provide a high level of occupant comfort and are more energy efficient than constant air volume – variable air temperature (CAV-VAT) systems.

There are a variety of VAV HVAC system configurations used in commercial and institutional buildings. The most common and preferred for new construction, retrofits, and major renovations is a VAV system that either provides cooling only or provides either cooling or heating depending on the needs of the zone served. These systems are the most energy efficient when compared to other VAV HVAC system configurations and CAV-VAT systems. However, there are other VAV HVAC system configurations that are not as energy efficient and typically not allowed to be used in new HVAC systems by energy codes and standards but are still used in existing buildings. These VAV HVAC systems include dual-duct systems that mix cold and hot airstreams and terminal reheat systems that heat conditioned air in order to provide supply air that meets the thermal requirements of the zone served. These mixing and reheat VAV HVAC systems are also addressed in this chapter for completeness.

3.2 VAV SYSTEM DESCRIPTION

A variable-air-volume or VAV HVAC system maintains the desired thermal conditions in a zone by varying the volumetric rate of constant temperature air that is delivered to that zone. In other words, the conditioned air that is supplied by the HVAC system is at constant temperature which is typically about 55°F (12.8°C). The temperature in a building zone is then maintained under varying load conditions by introducing varying amounts of this constant temperature air to mix with existing air in the zone to counteract load changes and maintain the desired temperature. The airflow to a zone in a VAV HVAC system is controlled by a VAV terminal unit often referred to as a “VAV box.” The position of the damper in a VAV terminal unit is automatically controlled by a thermostat in the zone in conjunction with other control devices or systems to provide the required airflow to the zone.

3.3 VAV VERSUS CONSTANT-AIR-VOLUME HVAC SYSTEMS

3.3.1 VAV HVAC System Advantages

A VAV HVAC system is the preferred system for many commercial and institutional building applications when compared to a constant-air-volume (CAV) HVAC system. Properly designed, installed, operated, and maintained, a VAV HVAC system can provide both a higher degree of building occupant comfort and significant energy savings when compared with a CAV HVAC system.

CAV HVAC systems operate with the supply fan providing an airflow designed to accommodate building heating and cooling requirements at full design load. However, most commercial and institutional buildings rarely operate at full design load and usually operate at significantly less that full design load. Since the airflow in a CAV HVAC system is constant, the supply fan horsepower requirements and the accompanying energy cost for operating the fan remain essentially constant regardless of the building HVAC load. If the CAV HVAC system operates for twelve to eighteen hours per day, a great deal of energy is wasted and the system is not very efficient.

On the other hand, VAV HVAC systems supply cold and sometimes hot air to building HVAC zones based on their cooling and heating needs. Since the amount of energy needed by a fan is proportional to airflow, a VAV HVAC system can significantly reduce building energy usage and costs. For example, if fan airflow could be reduced to 70 percent, the fan laws predict that the energy used by that fan will reduce to about 34 percent of the energy used if the fan operates at its rated airflow.

VAV HVAC systems are very flexible and can be reconfigured, expanded, or scaled back relatively easily depending on the building use and occupant needs. VAV terminal units are the basic building blocks of a VAV HVAC system and these units can be relocated, modified, or upgraded very easily to accommodate changing HVAC loads and air supply needs. In addition, VAV HVAC systems can address the need for any number of separate building zones because VAV HVAC systems are both modular and scalable.

Through zoning, VAV HVAC systems can simultaneously provide heating to those zones needing heating and those zones needing cooling. This feature results in increased occupant comfort throughout the building. Savings can also be achieved in sizing central heating, cooling, and air distribution equipment.
through zoning. Not all zones will require either maximum heating or cooling at any one time which results in diversity in the building’s HVAC load. This load diversity can be taken into account when sizing central equipment for a VAV HVAC system and it may be possible to select smaller equipment than would otherwise be required for a CAV HVAC system. Smaller central heating, cooling, and air distribution equipment will not only result in energy savings over the life of the building but may also result in a lower HVAC system first cost.

### 3.3.2 CAV HVAC System Advantages

Despite the advantages of VAV HVAC systems over CAV HVAC systems discussed in the previous section, there are applications where CAV HVAC systems are the better choice over VAV HVAC systems. CAV HVAC systems are generally better suited than VAV HVAC systems for applications that require a minimum number of air changes per time, positive pressure in one space in relationship to surrounding spaces, or where odor or contaminant removal is a requirement. Therefore, all of the space and occupant requirements should be considered when selecting an HVAC system and the HVAC system that best meets all of the criteria should be selected.

Where a space requires a minimum number of air changes per unit time, a CAV HVAC system may be the best choice for the application. When the VAV HVAC system reduces airflow to the space in response to falling temperature, the required minimum number of air changes per unit time may not be met. However, as will be discussed later in this chapter, a VAV HVAC system using dual-duct VAV terminal units can provide the needed airflow under varying space temperature conditions and still provide the energy benefits of a VAV HVAC system. It should be noted however, that dual duct systems that include a hot deck and cold deck may only be found in existing HVAC systems because the mixing of hot and cold air to maintain zone temperature is energy inefficient.

Similarly, a VAV HVAC system may not be the best choice when positive pressure needs to be maintained in a space to prevent the migration of airborne contaminants such as dust from adjacent spaces. A CAV HVAC system delivers a constant airflow into the space that will maintain a constant pressure in the space to prevent the infiltration of airborne contaminants. Again, by its very nature a VAV HVAC system reduces airflow into a space in response to falling temperature which will necessarily result in a drop in pressure. This drop in pressure could result in the space having a negative pressure in relation to surrounding spaces and allow the infiltration of airborne contaminants into the space. However, a VAV HVAC system using either dual-duct VAV terminal units or fan-powered VAV terminal units may be able to provide constant airflow to ensure that positive pressure is maintained in the space and also take advantage of VAV HVAC system energy savings.

Again, when odor or contaminant removal is required in a space a CAV HVAC system may be the best choice over a VAV HVAC system. Reduced VAV HVAC system airflow in response to a falling space temperature may compromise the HVAC system’s ability to remove odors and airborne contaminants from the space. However, a VAV HVAC system using either dual-duct or fan-powered VAV terminal units may be able to meet the constant airflow requirements as well as provide the energy savings associated with VAV HVAC systems. Dual duct systems that maintain zone temperature by mixing hot and cold supply air will only be found in existing HVAC systems because these systems are energy.

### 3.4 VAV System Operation

#### 3.4.1 Simple Single-Zone VAV System

A VAV system is essentially a variable-air-volume, constant air temperature (VAV-CAT) HVAC system that maintains the desired temperature and humidity of a space over a specified range of load conditions by varying the volume of constant-temperature air that is delivered to the space. Figure 3-1 illustrates the operation of a basic single-zone VAV system that provides cooling only and employs a dedicated air-handling unit (AHU) to serve a single space.

In Figure 3-1, space cooling is accomplished by supplying conditioned air from the dedicated AHU that consists of a cooling coil and fan. A mixture of return air and outside air is drawn across the cooling coil by the fan and cooled to the desired supply air temperature. The fan also forces the conditioned air through the supply duct and into the space. Air from the space is then returned to the AHU or exhausted to the outside through the return air duct. Return air does not have to be ducted and could be returned via a return air plenum, exhausted directly to the outside from the space, or a combination of these return air methods.

#### 3.4.2 Adjusting Airflow

The amount of conditioned air that is supplied to the space in Figure 3-1 can be controlled in any of the following three ways:
3.4.2.1 Adjusting Damper Position

Opening or closing the damper in the supply duct will result in more or less supply air being introduced into the space, respectively. If the thermal load in the space were constant, a manual damper could be installed and this damper could be adjusted once during building commissioning and never changed. Unfortunately, the thermal load in each zone is dynamic and changes constantly due to season, time of day, daily weather patterns, number of occupants, occupant activity, equipment use, and other variables. In addition, different occupants may want the space warmer or cooler depending on the activity being performed in the space. As a result, a simple manual damper will not provide the dynamic control needed to maintain thermal comfort in a building zone. In order to be effective, the damper needs to have the ability to respond automatically to changes in the zone’s thermal load. Therefore, VAV terminal units that are usually referred to simply as VAV boxes are used in place of a manual damper to control the amount of conditioned air supplied to the space. VAV terminal units automatically modulate the airflow through them in order to maintain the desired temperature of the space. VAV terminal units are an important component of VAV systems and will be discussed in detail in this chapter.

3.4.2.2 Adjusting Fan Airflow

The second way to vary the amount of conditioned air that is delivered to the space is by adjusting the fan airflow. This can be accomplished by varying the fan operating characteristics such as speed or blade pitch in response to the space’s changing thermal load. Restricting the fan inlet air supply using variable inlet vanes or blocking the fan discharge using variable discharge dampers can also be used to adjust fan airflow. Today, variable frequency drives (VFD) are used almost exclusively to adjust fan airflow because they are more efficient, require less maintenance, and are more economical in most cases when compared with other methods of adjusting fan airflow. The use of VFDs in VAV systems as well as other mechanical and electrical methods that can be used to control the supply fan airflow will be covered in this chapter.
3.4.2.3 Combination Of Methods

It should be pointed out that in the simple VAV system illustrated in Figure 3-1, the AHU only supplies one space and so varying the airflow through the fan only affects the amount of conditioned air supplied to that space. If the AHU supplies multiple spaces, see Figure 3-2 and each space has a different thermal load, then it is nearly impossible to ensure that each of the spaces served by the AHU will remain within its thermal comfort zone. Historically, VAV systems first attempted to control the amount of conditioned air delivered to spaces by varying the airflow through the fan but it was found that this didn’t work well for multiple zones and VAV terminal units were introduced to overcome this limitation.

Today, most VAV systems control the volume of air delivered to a space using both VAV terminal units at the space and vary fan speed to adjust the primary conditioned airflow based on the total demand on the AHU by all of the spaces it serves. This combination approach ensures that each space receives the supply air it needs to maintain thermal comfort under varying thermal load conditions and increases the system efficiency by allowing the supply fan to be throttled back when the system is operating at less than full load. Throttling back the supply fan in this simple system reduces electrical energy usage and avoids the need to bypass or dump primary conditioned air that is not needed when the thermal load in the spaces requires less than full fan airflow.

3.4.3 Controlling Airflow

If the thermal load in a space did not change, the damper position, fan operating characteristic, or both could be set during the system commissioning process and only adjusted as required over the life of the building. This is not the case because the thermal load is dynamic and this requires nearly constant adjustment in the amount of conditioned air supplied to the space in response to thermal load changes. The VAV system accomplishes this by monitoring selected environmental variables in a space such as temperature and adjusting the amount and characteristics of the supply air delivered to the space in response to deviations from desired conditions such as space temperature. Local control of VAV systems will be covered in this chapter and HVAC control systems including sensors and control systems will be covered in Chapter 18.

3.5 VAV TERMINAL UNITS

3.5.1 VAV Terminal Unit Purpose

The purpose of a VAV terminal unit is to control the airflow of conditioned air that is delivered to a zone in response to that zone’s thermal load. The ability of a VAV terminal unit to vary the volume of constant temperature air supplied to a space in response to the zone’s changing thermal load allows the VAV system to maintain the desired temperature in the zone.

3.5.2 VAV Terminal Unit Operation

A basic VAV terminal unit is a device that is installed in ductwork downstream from the primary air supply and upstream from the air outlets that serve the zone, see Figure 3-3. For simplicity, Figures 3-1 and 3-2 show a single air outlet associated with a single VAV terminal unit. In practical applications, it is common for a single VAV terminal unit to serve more than one air outlet in a zone, see Figure 3-3.

Figure 3-3 illustrates a VAV system with ducted return where the primary air supply is an air handling unit (AHU), a basic VAV terminal unit is inserted in the branch ductwork to control the amount of conditioned air supplied to the space, and three ceiling mounted diffusers are used to introduce the supply air into the space.

A basic VAV terminal unit is a sheet-metal box with an air inlet and outlet with the following two operational components (Figures 3-4 and 3-5):

- Airflow Throttling Device
- Throttling Device Control Mechanism

The airflow throttling device is typically a rotating blade damper that is positioned by the control mechanism to allow the required volume of conditioned air to pass through the VAV terminal unit and into the zone served by the VAV terminal unit. The position of the damper in this basic VAV terminal unit can vary from fully closed where no air is supplied to the space to fully open which allows the maximum amount of supply air through the VAV terminal unit. The maximum amount of conditioned air that can pass through the VAV terminal unit is a function of not only the physical design and dimensions of the VAV terminal unit but also the primary air pressure at the inlet of the VAV terminal unit and the static pressure at the outlet of the VAV terminal unit.
FIGURE 3–2 BASIC MULTI-ZONE COOLING-ONLY VAV SYSTEM
FIGURE 3–3  VAV AIR TERMINAL UNIT SERVING MULTIPLE AIR OUTLETS
FIGURE 3-4 BASIC VAV SINGLE-DUCT TERMINAL UNIT – FUNCTIONAL DIAGRAM

FIGURE 3-5 BASIC VAV SINGLE-DUCT TERMINAL UNIT – CUTAWAY VIEW
The VAV terminal unit illustrated in Figures 3-4 and 3-5 is referred to as a pressure-independent VAV terminal unit because it is not capable of sensing and adjusting for changes in duct static pressure in order to regulate airflow into the zone. However, with no flow or static sensing, this unit would become pressure dependent. The need for VAV terminal units to sense and adjust for duct static pressure as well as how pressure independence is accomplished will be addressed later in this chapter.

The throttling device control mechanism in the basic VAV terminal unit responds to a signal from one or more sensors in the zone, see Figures 3-4 and 3-5. When thermal conditions or other variables being monitored deviate from the desired conditions, a signal is sent to the VAV terminal unit controller that adjusts the position of the VAV terminal unit damper so that the right amount of conditioned air can be supplied to the space. In Figure 3-3, the sensor is a thermostat located in the zone that monitors temperature. When the ambient temperature in the space deviates from the thermostat setting or setpoint, a signal is sent to the VAV terminal unit controller and the controller automatically adjusts the damper position to allow more or less conditioned air to be delivered to the space.

3.5.3 VAV Terminal Unit Types, Configurations, and Features

There are a number of different types and configurations of VAV terminal units available to meet specific VAV system needs and application requirements. In addition, there are a number of features that can also be specified when selecting a VAV terminal unit. Figure 3-6 provides a breakdown of VAV terminal unit types, configurations, and features. Each of these types, configurations and features will be addressed in the sections that follow.

3.6 Basic VAV Terminal Units

3.6.1 Single-Duct VAV Terminal Units

The VAV terminal unit described in the previous section is referred to as a single-duct VAV terminal unit and is illustrated in Figures 3-4 and 3-5. This type of terminal unit is usually used on cooling-only systems where any heat needed in the zone is provided by a separate heating system or generated by the activity taking place in the space itself. For example, a single-duct VAV terminal unit could be used in the interior of an office building where people, lights, and office equipment generate needed heat.

The airflow through the single-duct VAV terminal unit is controlled by the damper within the preset range.
range. A signal from a sensor such as a thermostat in the zone is sent to the terminal unit’s controller causing the damper to change its position resulting in more or less airflow into the space. As the temperature rises in the zone, the damper opens allowing greater airflow into the space for cooling. Conversely, when the temperature is falling in the zone, the damper closes reducing airflow and cooling.

### 3.6.2 Single-Duct VAV Terminal Unit Control Scheme

Figure 3-7 illustrates the control scheme for a single-duct VAV terminal unit controlled by a thermostat in the zone that it serves. When the zone temperature increases above the thermostat setpoint, the terminal unit damper opens increasing the cold airflow through the terminal unit that is delivered to the zone. When the damper is fully open, the airflow is at its maximum, and as illustrated in Figure 3-7, the terminal has reached its maximum cooling capability. Similarly, as the temperature in the zone drops the terminal unit damper closes to reduce the cold airflow to the zone. If the temperature in the zone continues to drop, the damper will eventually close to its minimum setting. The minimum damper setting could be fully closed which would completely cut off the cold air supply to the zone. However, the minimum damper setting is usually not fully closed, see Figure 3-7 and is set to ensure minimum air movement in the zone for comfort and the introduction of outside air into the zone to maintain indoor air quality.

### 3.6.3 Dual-Duct VAV Terminal Units

A dual-duct VAV terminal unit is essentially two single-duct VAV terminal units combined into a single unit as illustrated in Figures 3-8 and 3-9. Figure 3-8 provides a functional diagram of a basic dual-duct VAV terminal unit and Figure 3-9 provides a three-dimensional cutaway view of that unit. As can be seen from these figures, a dual-duct VAV terminal unit has two separate primary air inlets and dampers allowing it to perform both a cooling and heating function. Incoming cold and hot primary air supplied by the HVAC system is mixed in the mixing chamber of the terminal unit and the mixed air is discharged into the zone served by the terminal unit.

Dual-duct VAV terminal units can be used in any application where both primary cold and hot air are available and the zone served has a need for both cold and hot supply air. These terminals have the ability to provide a constant air volume at varying air temperatures which allows a VAV HVAC system to be used...
FIGURE 3–8 BASIC VAV DUAL-DUCT TERMINAL UNIT – FUNCTIONAL DIAGRAM

FIGURE 3–9 BASIC VAV DUAL-DUCT TERMINAL UNIT – CUTAWAY VIEW
where a CAV HVAC system might normally be used such as health care facilities and laboratories.

The CAV application of dual-duct VAV terminal units provides the needed constant airflow in the zone with some possible energy savings when compared to a CAV HVAC system. By coordinating the control of the cold and hot dampers, the cold and hot airflow entering the terminal unit’s mixing chamber can be adjusted to provide constant airflow at the right temperature to the space. When full cooling is required, the cold damper is fully open and the hot damper would be fully closed. Conversely, where full heating is required, the cold damper would be fully closed and the hot damper would be fully open. This arrangement will probably only be seen in existing HVAC systems because mixing cold and hot airstreams is energy inefficient.

3.6.4 Dual-Duct VAV Terminal Unit Control Schemes

Figure 3-10 provides a schematic diagram of a three-zone HVAC system using dual-duct VAV terminal units. Two ducts are used to supply air to the VAV terminal units, see Figure 3-10. One delivers cool, dehumidified air when the refrigeration system is operating. The other delivers warm air, which may either be heated air or return air from the conditioned space during warmer weather. Heat could also be used during the summer cooling season to better control humidity. The volumes of cool and warm air circulated throughout the VAV system vary in relation to the changing ratio of cooling and heating loads. Equipment configurations are available both for horizontal and vertical installations.

3.6.4.1 Non-Blending Control Strategy

A non-blending control strategy for dual-duct VAV terminal units is illustrated in Figure 3-11. In the cooling mode, as the zone temperature approaches the thermostat setpoint the cold airflow is throttled back to zero flow. As the zone temperature drops below the thermostat setpoint, the hot airflow increases from zero to maximum heating. For this control strategy, cooling and heating flow rates can be different.

3.6.4.2 Maximum Heating Blending Control Strategy

Figure 3-12 shows the maximum heating blending control strategy. In the cooling mode, as the temperature approaches the setpoint the cold air is throttled back from its maximum airflow to its minimum. As the temperature continues to drop, the hot airflow increases as the cold air decreases in the mixing zone to maintain a minimum airflow to the zone.

3.6.4.3 Unequal Flow Blending Control Strategy

This control strategy is illustrated in Figure 3-13 and is similar to the maximum heating blending control strategy discussed in the previous paragraph. This strategy can be employed where the heating mode airflow is higher than the minimum airflow required.

3.6.4.4 Constant-Volume Blending Control Strategy

This strategy results in a constant-volume airflow from the dual-duct VAV terminal unit outlet, see Figure 3-14. Both the cooling and heating maximum airflow are the same at their respective maximums. With increasing zone temperature, the heating hot damper in the VAV dual-duct terminal unit begins to close and the cold damper begins to open. The operation of these two dampers is coordinated so that the total airflow out of the terminal unit is held constant until the heating damper is completely closed and the cooling damper is completely open. This control strategy should only be found on existing systems because mixing cold and hot airstreams to achieve the desired zone temperature is energy inefficient and not allowed by energy codes and standards.

3.7 FAN-POWERED VAV TERMINAL UNITS

3.7.1 Fan-Powered VAV Terminal Unit Purpose

Fan-powered VAV terminal units provide the capability of adjusting the primary airflow to the zone similar to a basic single-duct VAV terminal unit as well as providing heat to the zone by recirculating local return air from the zone. Figure 3-15 provides a schematic diagram of a VAV system using fan-powered VAV terminal units. Primary cool air is supplied directly to the terminal unit and a damper modulates primary cold airflow through it, see Figure 3-15. A fan in parallel with the cold air stream draws local return air into the terminal unit and mixes it with the primary cold air. The mixed conditioned air is then supplied to the space.

The amount of primary air and local return air mixed in the terminal unit and delivered to the zone depends on the setpoint of the controlling thermostat and zone temperature. When full cooling is required, the
FIGURE 3–10 DUAL-DUCT VAV TERMINAL UNIT SCHEMATIC DIAGRAM
FIGURE 3–11 DUAL-DUCT VAV TERMINAL UNIT – NON-BLENDING CONTROL STRATEGY

FIGURE 3–12 DUAL-DUCT VAV TERMINAL UNIT – MAXIMUM HEATING BLENDING CONTROL STRATEGY
FIGURE 3-13 DUAL-DUCT VAV TERMINAL UNIT - UNEQUAL FLOW BLENDING CONTROL STRATEGY

ZONE TEMPERATURE
MAXIMUM COOLING
MAXIMUM HEATING
MIXING

FIGURE 3-14 DUAL-DUCT VAV TERMINAL UNIT - CONSTANT VOLUME BLENDING CONTROL STRATEGY

ZONE TEMPERATURE
MAXIMUM COOLING
MAXIMUM HEATING
MIXING
FIGURE 3–15  FAN-POWERED VAV TERMINAL UNIT SCHEMATIC DIAGRAM
damper moves to its full open position and the fan shuts down allowing only cool air to enter the zone to provide maximum system cooling. Conversely, when the zone temperature drops below the thermostat set-point and heat is required the cold air damper closed to its minimum position and the local return air fan speed is increased to its maximum in order to deliver the maximum amount of warm return air to the zone. Between these two extremes, the terminal unit controller adjusts both the damper position and fan speed to supply the needed air temperature and airflow to the space.

Fan-powered VAV terminal units are applied to the perimeter zones of commercial buildings that exhibit one or more of the following characteristics:

- Unacceptable part-load airflow rates from modulating shutoff VAV terminal units.
- Zones that have seasonal heating and cooling requirements.
- Perimeter heating loads that can be offset largely by recirculated warm plenum air.
- Heat losses are such that heating can be handled by overhead diffusers.

In addition to perimeter areas, fan-powered VAV terminal units can also be used in those interior zone applications where the required air movement cannot be maintained by basic single-duct VAV terminal unit at shutoff.

There are two types of fan-powered VAV terminal units used. The fan-powered terminal unit (Figure 3-15) is referred to as a parallel-flow fan-powered VAV terminal unit because the local return air is discharged from the terminal unit fan in parallel with the primary cold air flowing through the damper. The other type of fan-powered VAV terminal unit is a series-flow fan-powered VAV terminal unit where the terminal unit fan is in series with both the primary cold air and local return air streams and runs all the time. Both of these types of fan-powered terminal units are used and have their own unique construction, operating characteristics, and applications. The following sections will address each of these types of fan-powered VAV terminal units.

### 3.7.2 Parallel Flow Fan-Powered VAV Terminal Units

Figure 3-16 provides a functional diagram of a parallel flow fan-powered VAV terminal unit and Figure 3-17 provides a three-dimensional cutaway view of a unit. A parallel flow fan-powered VAV terminal unit is divided into two parallel sections that provide parallel paths for primary air supplied by the upstream air handling unit and return air being recirculated from the zone. The primary air section looks and operates like a basic single-duct VAV terminal unit that regulates the primary airflow using a damper. The return air section includes a fan to pull air from the space for recirculation back to the space. The return air section also includes a backdraft damper at the discharge of the fan that prevents primary air from being back fed into the return air section when the fan is not running. A parallel flow fan-powered VAV terminal unit also includes a mixing chamber that allows primary air and recirculated air to be mixed before being supplied to the zone.

When no heat is required and the space requires maximum cooling, the fan in the parallel flow fan-powered VAV terminal unit is turned off and all the conditioned air supplied to the zone is primary air. As the zone cools, the primary air damper closes reducing the primary airflow to the space. When the primary airflow is reduced to its minimum and the space cools below its temperature setpoint, the fan is turned on and the backdraft damper opens which allows warm air from the zone to be recirculated to raise the zone temperature. While the return air is being recirculated, the primary air damper is open to allow a constant volume of primary air to be supplied to the space to maintain required air quality.

An advantage of parallel flow fan-powered VAV terminal units is that during nighttime or weekend operation the upstream air handling units can be shut down and the terminal unit fan will cycle on and off according to the space heat requirements. Additionally, a heating coil can be installed at the outlet of these terminal units if additional heating is required.

Another advantage of a parallel flow fan-powered VAV terminal unit is that the fan only runs when the space requires heat. Unlike the series flow fan-powered VAV terminal unit, the terminal unit fan operates outside of the primary air stream and does not need to be interlocked with the air handling unit, simplifying controls. This arrangement allows the heat generated by the lighting that is captured by the air in the plenum to be recirculated by a parallel-flow...
FIGURE 3−16 PARALLEL FLOW FAN-POWERED VAV TERMINAL UNIT − FUNCTIONAL DIAGRAM

FIGURE 3−17 PARALLEL FLOW FAN-POWERED VAV TERMINAL UNIT − CUTAWAY VIEW
fan-powered VAV terminal unit making the overall system more efficient.

### 3.7.3 Series Flow Fan-Powered VAV Terminal Units

Series flow fan-powered VAV terminal units differ from parallel flow fan-powered VAV terminal units in both the way they are constructed and the way that they operate. Figure 3-18 provides a functional diagram of a series flow fan-powered VAV terminal unit and Figure 3-19 provides a three-dimensional cutaway view. As can be seen from these figures, the fan is located in series with both the primary and return airstreams. The fan in a series flow fan-powered VAV terminal unit is on all the time and not just when the zone served by the series flow fan-powered VAV terminal unit is calling for heat. The fan in a series flow fan-powered VAV terminal unit should be sized for full zone airflow since the primary air passes directly through it. As a result, series flow fan-powered VAV terminal units are less efficient than parallel flow fan-powered VAV terminal units.

When the zone served by the series flow fan-powered VAV terminal unit requires cooling, the primary air damper opens and the fan gets its airflow from the primary air. When the zone temperature reaches or drops below its desired value, the primary air damper closes to its minimum position for air quality purposes and the fan draws return air from the zone. If additional heating is required, a heating coil can be installed at the outlet of the series flow fan-powered VAV terminal unit.

The design airflow for both the upstream air-handling unit and the series flow fan-powered VAV terminal unit fan need to be the same. If the primary air static pressure is too high at the terminal unit, conditioned air supplied by the upstream air-handling unit will be dumped into the return air system through the return air inlet. Similarly, too low of a primary air static pressure will result in the fan drawing both primary and return air through it when the zone served needs to be cooled. As a result, the combined primary and return air will have a higher temperature than the primary air alone and this will limit the system’s ability to cool the zone.

Series flow fan-powered VAV terminal units may be used throughout the building or in specific building areas that will benefit from constant airflow. These terminal units are typically used in zones that require heat during occupied hours and will also benefit from a constant airflow. Some examples of spaces that might use series flow fan-powered VAV terminal units are conference rooms, entrance areas, atria, and bathrooms.

### 3.8 VAV TERMINAL UNITS WITH REHEAT

Figure 3-20 provides a functional diagram of a single-duct VAV terminal unit with reheat. The only difference between a standard single-duct VAV terminal unit and one with reheat is the heating coil installed at the outlet of the terminal unit. VAV terminal units with reheat heat the air as it exits the terminal unit outlet using a hot water, steam, or an electric coil. As noted in the sections covering fan-powered VAV terminal units, reheat can be used with these terminal units as well. The use of reheat is energy inefficient and should be minimized.

Reheat VAV terminal units are typically used in zones that require cooling only when occupied and may need some supplemental heat when unoccupied. Reheating conditioned air supplied by the upstream air-handling unit is not an efficient method of providing needed heat to an occupied zone. When heat is required during occupancy, the cooling-only VAV system should be supplemented by another more economical method of providing zone heating such as a radiant heating system using hot water or steam. For example, in perimeter zones where heat is required at outside walls to counter heat loss and under windows to prevent cold down drafts, perimeter radiant heating can be provided to work in conjunction with the cooling-only VAV system. The VAV system discharge air temperature could be reset in the winter to provide a hotter supply air temperature from the primary system since cold air would not be required for dehumidification.

### 3.9 BYPASS VAV TERMINAL UNITS

With bypass VAV terminal units the excess primary supply air is diverted at the terminal unit from entering the zone by "dumping" it into the return air system. A functional diagram of a bypass or dump VAV terminal unit is provided in Figure 3-21. Primary air enters the terminal unit inlet and either exits through the terminal unit outlet is bypassed or “dumped” through the return air plenum or duct through the bypass damper.

The airflow to the zone served by the bypass VAV terminal unit is controlled by a damper at the output of the terminal unit, a single-duct VAV terminal unit downstream, or a VAV diffuser in the zone. Bypass VAV terminal units are used with constant volume supply systems and there is no reduction in the primary air-
FIGURE 3−18 SERIES FLOW FAN-POWERED VAV TERMINAL UNIT – FUNCTIONAL DIAGRAM

FIGURE 3−19 SERIES FLOW FAN-POWERED VAV TERMINAL UNIT – CUT-AWAY VIEW
### Figure 3-20 VAV Terminal Unit with Reheat – Functional Diagram

- **Rotating Blade Damper**
- **Reheat Coil**
- **Primary Air Inlet**
- **Supply Air Outlet**
- **Damper Control Shaft**

### Figure 3-21 Bypass (Dump) VAV Terminal Unit – Functional Diagram

- **Bypass Air**
- **Sheet Metal Box**
- **Bypass Damper Control**
- **Rotating Blade Damper**
- **Outlet**
- **Supply Air**
- **Damper Control Shaft**
- **Primary Air Inlet**
flow the terminal. Bypass systems are not as efficient as other VAV terminal units discussed in this chapter and their use may be restricted by energy codes and standards.

The bypass VAV terminal unit is totally different from all other types of VAV terminal units in that it does not actually throttle or reduce the airflow. A bypass VAV terminal unit simply bypasses or “dumps” the excess air back into the return air system. Space temperatures are maintained by varying the ratio of air entering the space and being recycled back to the central system at the terminal unit. This method of modulating airflow to a zone is usually only used on small direct expansion (DX) systems where a minimum airflow is required across the DX coil to prevent freezing.

The amount of supply air being delivered to the VAV terminal unit is constant. Therefore, the fan airflow also remains relatively constant and no energy is saved by reducing the airflow to the zone served. Also, it is not possible to take advantage of system diversity with bypass terminal units because the supply airflow remains constant instead of being throttled back as it is with other VAV terminal units. Bypass boxes can be used with integral heating coils where conditions require more heating than that obtained from internal loads.

### 3.10 VAV TERMINAL UNIT

#### 3.10.1 VAV Terminal Unit Inlet Pressure

Up to this point, the effect of the supply duct static pressure on the airflow through a VAV terminal unit has not been considered. It has been assumed that the airflow through the VAV terminal unit is a function only of the position of its damper and that the static pressure in the supply duct serving the VAV terminal unit is constant. In reality, this is not the case. VAV systems typically supply multiple zones and their associated VAV terminal units through a common supply duct using a common supply fan, see Figure 3-2. Each of the zones served by the supply system can have a changing thermal load that requires more or less conditioned air to be supplied to the zone in response to a signal from the zone’s thermostat. As a result, each VAV terminal unit damper is opening and closing in response to its zone’s thermal load which in turn impacts the static pressure in the primary supply duct. Even if the thermal load in Space #1 of Figure 3-2 remained unchanged and its VAV terminal unit damper remained in a fixed position, the opening and closing of VAV terminal unit dampers serving Spaces #2 and #3 would impact the supply duct static pressure and result in a changing airflow through the VAV terminal unit serving Space #1. The varying volume of air flowing into Space #1 as a result of changing supply duct static pressure will cause the temperature to unnecessarily fluctuate in the space and impact both occupant comfort and energy efficiency.

#### 3.10.2 Pressure-Dependent VAV Terminal Units

The situation described above is what happens when a pressure-dependent VAV terminal unit is used. Pressure-dependent VAV terminal units were used when VAV systems were first introduced. Pressure-dependent VAV terminal units do not have the capability to measure and adjust their damper position in response to the supply duct static pressure in order to maintain the required airflow. The airflow supplied by a pressure-dependent VAV terminal unit to its zone depends on the static pressure in the supply duct and can vary depending on what is happening in other zones. For example, VAV terminal units that are close to the supply air fan are likely to supply too much air and VAV terminal units located further downstream from the supply air fan will probably supply too little air.

A pressure-dependent VAV terminal unit’s damper is controlled solely by the thermostat located in the zone that it serves. When the supply duct static pressure increases, the damper remains in the same position and the airflow into the zone increases. This increase in cool air results in a gradual drop in space temperature until it is sensed by the thermostat and a signal is sent to the VAV terminal unit to close its damper. Similarly, when the static pressure drops in the supply air duct, the airflow through the VAV terminal unit also decreases and the temperature in the zone will increase if the thermal load remains constant. Again, when the temperature increase is sensed by the thermostat, it will send a signal to the VAV terminal unit and it will open its damper allowing for greater airflow.

Pressure-dependent VAV terminal units respond to the changing supply duct static pressure through its impact on the temperature of the zone. Even though the thermostat continually monitors the temperature of the zone and adjusts the VAV terminal unit’s damper accordingly, the response can be sluggish and result in unacceptable temperature variations within the space. As a result, VAV HVAC systems seldom use pressure dependent VAV terminal units that depend on a change in space temperature to adjust the VAV terminal unit’s damper in response to a change in the supply duct static pressure. Instead, VAV HVAC systems utilize pressure independent VAV terminal units that measure the supply duct static pressure directly and
adjust the VAV terminal unit’s damper directly in response to changes in supply duct static pressure.

### 3.10.3 Pressure-Independent VAV Terminal Units

Nearly all VAV terminal units that are installed in new VAV HVAC systems or retrofitted into existing VAV HVAC systems are pressure independent. Pressure independent control of airflow through VAV terminal units requires accurate measurement of the pressure differential between the inlet and outlet of the terminal unit. This measurement is usually accomplished using an integral airflow-measuring device such as a multipoint airflow sensor located at the inlet of the VAV terminal unit, see Figure 3-22.

The output of the flow sensor at the VAV terminal unit inlet is fed into the terminal unit controller and used in conjunction with the signal from the zone thermostat to modulate the terminal unit damper in order to provide constant airflow through the terminal unit. As a result, the airflow through the VAV terminal unit is directly controlled and is independent of the supply duct static pressure at its inlet. Pressure independent VAV terminal units result in a more stable airflow through the VAV air terminal unit as well as result in minimum and maximum airflow settings that correspond to actual airflows rather than just the physical position of the VAV air terminal unit damper.

Figure 3-23 provides a section view of a typical multipoint pressure sensor. From Figure 3-23 it can be seen that a multipoint pressure sensor consists of two independent sensors with ports facing in opposite directions. The sensor with ports facing upstream into the incoming primary air stream measures the total pressure at the inlet of the terminal unit and the sensor with ports facing downstream measures the static pressure. The difference between the total and static pressure relates directly to the airflow through the VAV terminal unit. Multipoint pressure sensors come in a variety of configurations including the common circular sensor (Figure 3-22) as well as diamond, linear, and cross, among others.

### 3.11 HVAC SYSTEMS INCORPORATING VAV

#### 3.11.1 Types Of HVAC Systems Incorporating VAV

There are a number of VAV system variations that are used to achieve thermal comfort in the spaces served by the HVAC system. These VAV system variations include the following:

- Cooling- Or Heating-Only VAV Systems
- VAV Reheat Systems
- Dual-Duct VAV Systems
- Separate Interior and Perimeter Systems
- Combined VAV Interior and Perimeter Systems
- Bypass Or Dump Systems

#### 3.11.1.1 Cooling- Or Heating-Only VAV Systems

Cooling-only VAV systems often are used in interior spaces where there are some variations in cooling loads required due to lights, people, equipment etc., but where air conditioning is needed constantly throughout the year.

#### 3.11.1.2 VAV Reheat Systems

The variable air volume concept, when applied to reheat systems, permits a reduction in the airflow thereby suspending the application of heat until flow conditions reach a predetermined minimum. VAV reheat systems are not energy efficient because they heat conditioned supply air and the use of reheat should be minimized.

With an air volume selected for maximum instantaneous peak loads rather than the sum of all peaks, the total system air volume is reached. Also, any additional system diversity, such as areas with intermittent loads (conference rooms, office equipment rooms etc.) may be included in the total volume reduction. With volume reduction as a first step of the control system, reheat is not applied until the minimum volume is reached. This procedure reduces system operating cost for summer and intermediate weather appreciably.

Terminal units with a pair of regulators are available for reheat VAV reheat use. There are two types, one that is thermostat-controlled and the other that maintains a constant minimum volume. Some overlapping of energy requirements may still occur at intermediate loads, depending on the amount of flow reduction selected or permitted.

#### 3.11.1.3 Dual-Duct VAV Systems

Another type system that can handle both the interior and perimeter simultaneously is the dual-duct VAV system. A VAV fan supplying a separate "cold" duct feeds cold air to interior cooling only VAV terminal units and if used, perimeter heating and cooling ter-
FIGURE 3-22 VAV TERMINAL UNIT INLET MULTIPOINT PRESSURE SENSOR

FIGURE 3-23 MULTIPOINT PRESSURE SENSOR
terminal units. The second duct is a heating duct with its own VAV fan that supplies heat to the perimeter dual duct terminal units.

Some existing dual-duct VAV system accomplishes zone temperature control by blending of cold and warm air after volume reduction of the total supply air to each zone. This system is energy inefficient is likely to only be encountered in existing HVAC installations. The following describes the sequence of operation for a dual-duct VAV system:

- At maximum cooling (when zone cooling governs), the room supply temperature and air quantity requirements are identical with constant volume systems; the volume regulators deliver maximum volume and the cold port is wide open.
- As the cooling load drops, the volume regulator reduces the supply volume to the room to the minimum acceptable value. Upon further load decrease, the warm port begins to open as the cold port continues to close.
- Minimum volume is maintained below this cooling level and during the entire heating cycle.
- Smaller air supply volumes in off-peak zones, whose loads are met with reduced cold air instead of a higher temperature blend of warm and cold air, tend to lower the supply temperature to such zones, thereby lowering the average system supply temperature and humidity.

A typical dual duct VAV system uses two supply fans, one for the hot deck and one for the cold deck. Each fan's volume is controlled independently by the static pressure in its respective duct. The return fan is controlled relative to the sum of the hot and cold fan volumes using flow-measuring stations (FMS). The return air fan must track with the two supply air fans that are discussed later in this chapter.

Each fan is sized for the anticipated maximum coincident hot or cold volume, not the sum of the instantaneous peaks. The cold deck can be maintained at a constant temperature either by using the cooling coil with a minimal of fresh air or economizer cycle when the outside air is below the cold deck set point. This does not affect the hot deck, which can use the recovered heat from the internal load for hot deck heating. The heating coil is used only when heating requirements cannot be met with return air or when the hot deck air supply temperature is reset with the outside temperature to reduce the required size of the hot system fan and duct system.

### 3.11.1.4 Separate Interior and Exterior Systems

A very common approach in building HVAC systems is to have separate interior and perimeter systems, such as a cooling-only VAV system for the interior and a heating-only or combined heating and cooling system for the perimeter. A VAV system with an independent hydronic perimeter heating system accomplishes all cooling in all zones with air, on an all-season cooling cycle, while the perimeter heating system offsets the transmission heat losses but not the summer transmission heat gains. Hydronic radiation heating that is customarily scheduled either for the entire building or for selected exposures need not have manual or automatic individual room control, although it may to increase flexibility and operating economy.

Alternatively, it is possible to develop the necessary heating with a constant volume perimeter air system where the air temperature is varied with the outside air temperature, providing cooling in hot weather and heating in cold weather. Other types of perimeter systems that can be used include hydronic fintube radiation, fan coil units, induction units, among others.

### 3.11.1.5 Combined VAV Interior and Perimeter Systems

Another approach is to combine the interior and perimeter terminals into one system. The interior cooling-only system in this case is expanded to also handle the perimeter by adding perimeter reheat VAV boxes to the system.

### 3.11.1.6 Bypass Or Dump Systems

A bypass or dump VAV HVAC system is typically used on smaller systems that bypass or cycle the air at the terminal unit or at the central unit back to a constant volume fan. To vary the volume of air in the spaces, the bypass or dump system bleeds off air at the VAV terminal units and dumps or cycles the excess conditioned air into the return air duct or ceiling plenum rather than actually throttling down the airflow. This type of VAV system maintains constant fan volume and constant airflow in all supply ducts to the terminal units. Energy usually is wasted by mixing the conditioned air with the return air. This system, however, has practical advantages when direct expansion (DX) cooling coils are used because there is constant airflow through the DX coils regardless of the load. Often this is a necessary requirement for correct DX coil operation.
4.1 INTRODUCTION

A conventional multizone HVAC system is a constant air volume and variable air temperature (CAV-VAT) system where both cold and hot air are mixed to provide the needed supply-air temperature required by the zone at a constant airflow. Mixing of the cold- and hot-air supplies is accomplished using a zone damper located at the air handling unit or at the space that simultaneously modulates the cold- and hot-air supplies to meet the associated zone’s space conditioning requirements. Local zone thermostats usually control the mixing of cold and hot air via the zone damper.

4.2 MULTIZONE HVAC SYSTEM DESCRIPTION

Figure 4-1 provides a diagram of a multizone HVAC system. A multizone HVAC system conditions a zone by mixing a cold and a hot air stream together to provide supply air to a zone at the desired temperature at constant volume. The mixing is accomplished by zone dampers that can be located at the air-handling unit, at the zone served, or anywhere in between. A thermostat that responds to zone temperature normally controls the zone damper that adjusts the cold- and hot-air supplies simultaneously to provide the required supply air temperature at constant airflow.

The HVAC unit supplying both cold and hot air in Figure 4-1 is an air-handling unit with a fan, a cooling coil, and a heating coil. The cooling and heating coils in the air handling unit would use a hydronic distribution system to supply chilled and hot water through a central cooling and heating plant. However, the HVAC unit supplying the cold and hot air to the multizone HVAC system could also be a unitary HVAC system such as a self-contained rooftop unit. A unitary HVAC unit used on a multizone HVAC system could include its own direct expansion (DX) cooling coils and a section for heating using natural gas, electric, or waste heat recovery.

The air supply system for the multizone HVAC system illustrated in Figure 4-1 is very similar to the dual-duct HVAC system that will be discussed in Chapter 6. The difference is the location of the zone damper in the multizone system. The multizone system allows the zone dampers to be located anywhere between the air-handling unit and the zone served. Figure 4-1 shows the zone dampers close to the zone served which requires a similar duct layout as a dual-duct HVAC system. However, once the zone damper mixes the cold and warm air only a single duct is required to supply conditioned air to the zone. Therefore, as the zone dampers are moved closer to the air-handling unit there is a savings in ductwork with a multizone system as well as space required to accommodate the air distribution system.

A multizone system can include a preheat coil to heat the outside air when the system design requires a higher design temperature of the incoming outdoor air temperature. Where humidity control is required, a dehumidification coil can be installed in the outdoor air supply. Similarly, when humidification is required, a humidifier can be installed.

4.3 USE OF A MULTIZONE HVAC SYSTEM

Multizone HVAC systems control the temperature of the air delivered to a zone by mixing the hot and cold air streams together before delivering them to the zone. This mixing of the cold and hot air streams is not energy efficient and may be prohibited by energy codes. Therefore, unless an HVAC application requires constant volume air delivery, multizone systems are not used in new construction. Variable-air-volume (VAV) HVAC systems that are discussed in Chapter 3 have replaced multizone HVAC systems in most applications because VAV HVAC systems are more efficient. However, there are existing buildings that still use multizone HVAC systems despite their inherent inefficiency.

Multizone HVAC systems are typically used in buildings that require a constant-volume air supply to each zone and have the following characteristics:

- The area to be conditioned consists of several large or small zones that need to be individually controlled. Examples of this would be a school, a suite of offices, or an interior zone combining several individual open floors of a multi-story building.
- The area includes zones with different exterior exposures and different internal loads. An example of this would be the floor of a building with multiple exposures that all vary throughout the day and around the year.
- The area combines a large interior zone with a small number of small enclosed zones such as a large open office area with some enclosed offices.
- The area consists of several adjacent zones that each has their own unique load characteristics.

If a constant volume air supply is not needed, then a VAV HVAC system should be considered for the application.
FIGURE 4-1 MULTIZONE HVAC SYSTEM
5.1 INTRODUCTION

Terminal reheat HVAC systems usually reheat preconditioned supply air before the air is delivered to the zone that the system serves. The process of reheating the preconditioned air can be used to control supply air temperature, humidity, or both. In fact, the ability to closely control the temperature and humidity of the air supplied to a zone is an advantage of a terminal reheat HVAC systems. However, the process of reheating preconditioned air is energy inefficient. Despite the inefficiency inherent in a typical terminal reheat HVAC system, there are still existing buildings that use terminal reheat HVAC systems, special purpose HVAC systems that use terminal reheat because of this systems ability to closely control temperature and humidity, and terminal reheat HVAC systems that use 100 percent outside air for cooling and only need reheat for incidental zone heating during unoccupied times.

5.2 SYSTEM DESCRIPTION

A terminal reheat HVAC system is a constant-volume, single-zone central HVAC system that reheats preconditioned air before that air is supplied to the zone that it serves. Terminal reheat HVAC systems can be used to do the following:

- Provide zone or space control for areas of unequal loading
- Allow for the simultaneous heating and cooling of perimeter building areas with different exposures using the same central HVAC unit.
- Accommodate process or comfort HVAC zones where close control of space conditions is desired.

Terminal reheat HVAC systems typically reheat previously cooled air to provide supply air that meets required temperature and humidity criteria. In the case of a terminal reheat HVAC system, the application of heat is a secondary process to either preconditioned primary air or recirculated room air. This reheating of previously cooled air in a typical terminal reheat HVAC is energy inefficient and discouraged except in special cases where this system is preferred for an application due to its inherent capabilities and characteristics.

Terminal reheat HVAC systems are generally used in hospitals, laboratories, or spaces where wide thermal load variations are expected. Terminal units can be used to achieve space conditioning objectives by reheating either primary air or secondary air returned from the conditioned space.

Conditioned air is supplied from a central HVAC unit at a fixed, cold air temperature that is capable of offsetting the maximum cooling load in the zone or zones served. The thermostat monitoring the temperature in the zone calls for heat as the cooling load in the space drops below the maximum.

A terminal reheat HVAC system is very flexible and offers infinite zoning capability in the initial design stages. In addition, zoning can be readily revised during construction if the space use or HVAC requirements change. Changes to a terminal reheat HVAC system to accommodate revisions during or after construction often only require the addition of a heating coil or a terminal unit.

Reheat systems are not economical when energy is expended to cool the air by mechanical refrigeration and then new energy is added to reheat the same air. If reheat energy can be obtained from other sources such as the utilization of internal heat recovered from other areas or rejected refrigeration heat, then the use of reheat would be more energy economical in terms of energy savings. However, some energy is spent in the recovery and transport processes.

5.3 SYSTEM FEATURES

The principle advantage of a terminal reheat HVAC system is its ability to maintain close control of the temperature and humidity conditions in a zone. Terminal reheat HVAC systems can be used in laboratories that have high exhaust requirements, in process applications where close control of space conditions is important to product quality such as in textile mills, or for human activities that require close control of space conditions such as in hospital operating rooms.

Under window terminal reheat units allow night heating with the central air system shutdown. Low temperature types offer excellent control capability of the reheat system at reduced initial cost. For hospitals, return air systems can be eliminated reducing system complexity and air recirculation between patients' rooms. However, low humidity conditions are possible using these systems and are a possible drawback to using the terminal reheat system in a hospital application.

5.4 SYSTEM LAYOUT

Figure 5-1 illustrates a functional diagram of a typical HVAC terminal reheat system. The typical terminal...
A terminal reheat HVAC system consists of a central HVAC system and an air distribution system. The HVAC unit can be either factory-assembled or built-up and can be either a roof-top unit (RTU) or an air-handling unit (AHU) coupled with a central cooling plant. Return and exhaust air fans are optional depending on the system size and building pressurization requirements.

The air distribution system for a terminal reheat HVAC system is usually a constant-volume single- or multi-zone duct system. The cooling coil on a terminal reheat HVAC system is typically located at the central HVAC unit, see Figure 5-1. For a single-zone terminal reheat HVAC system, the reheat coil can be located either at the zone, see Figure 5-1 or at the HVAC unit. For a multizone terminal reheat HVAC system, the reheat coils will be located in the branch ductwork or terminal unit serving the zone per Figure 5-1.

For low-pressure multizone applications the reheat coils are typically located in the supply duct serving the zone in. These duct-mounted coils can be hot water, steam, or electric resistance. Single duct boxes that include the reheat coil are used with medium pressure systems, see Figure 5-2. The single duct reheat terminal unit has a manually operated inlet damper that is balanced prior to occupancy and then left in that position to maintain a constant airflow through the box. However, some terminal reheat units have a constant volume regulator that is preset at the factory. This type of flow regulating device depends on the airflow energy to actuate it. While it is sometimes thought that these systems do not require balancing, these systems require both setup and calibration after they are installed. This setup and calibration includes both the measurement of airflow and the proportionate balancing of the air outlets.

5.5 SYSTEM OPERATION

Terminal reheat HVAC systems can provide very good control of space conditions relative to both temperature and humidity. Very close temperature control can be obtained by adding heat at the terminal unit to increase the temperature of the supply air and maintain a constant space temperature. However, unless the HVAC system is using 100 percent outside air for cooling, a terminal reheat HVAC system is energy inefficient.

Similarly, required space humidity conditions can be achieved by maintaining the supply air at a constant dewpoint temperature that ensures that the supply air has constant moisture content. During hours of partial sensible and latent heat loads, the space humidity is lowered. This lowering can be considerable in the case
of applications using water-chilling cycles where the water is circulated continuously and results in a lower dewpoint and supply air temperatures. The operation of the central cooling plant can be controlled from either the return or supply water temperature. Room conditions are maintained by controlling either the HVAC unit reheat coil in the case of a single zone system or the duct reheat coils in the case of a multizone system. The reheat coils can also provide winter heating, as required.

5.6 VAV HVAC SYSTEM WITH TERMINAL REHEAT

Terminal reheat can be used in conjunction with VAV HVAC systems as discussed in Section 3.9. Figure 3-21 in Chapter 3 provides a functional diagram of a single-duct VAV terminal unit with reheat. The only difference between a standard, single-duct VAV terminal unit and one with reheat is the heating coil installed at the outlet of the terminal unit. VAV terminal units with reheat are used to heat the air as it leaves the terminal unit outlet using a hot water, steam, or electric resistance. Fan-powered VAV terminal units can incorporate reheat as well.

Reheat VAV terminal units are typically used in zones that require cooling only when occupied and may need some supplemental heat when occupied. Reheating conditioned air supplied by an upstream air-handling unit is not an efficient method of providing needed heat to an occupied zone. When heat is required during occupancy, the cooling-only VAV HVAC system should be supplemented by another more economical method of providing zone heating such as a dual-duct VAV HVAC system as discussed in Chapter 6 or a convection heating system using hot water or steam as discussed in Chapter 15. For example, in perimeter zones where heat is required at outside walls to prevent cold down drafts, perimeter convection terminal units can be used in conjunction with a cooling-only VAV HVAC system to ensure thermal comfort.
6.1 INTRODUCTION

This chapter covers dual-duct HVAC systems that are sometimes referred to as double-duct HVAC systems. A dual-duct HVAC system is in many ways a special case of a multizone HVAC system that was covered in Chapter 4. A multizone HVAC system mixes hot and cold conditioned air streams centrally at the supply unit to provide the desired temperature of supply air and then delivers this fixed-temperature supply air through a single duct to the zones served. A dual-duct HVAC system, on the other hand, uses a two-duct system that distributes both cold and hot air through separate parallel ducts throughout the building or portion of building served by the dual-duct HVAC system. The cold and hot air supplies are mixed proportionately in the air terminal unit serving each zone to provide the conditioned air needed to maintain the desired temperature in the zone. For this reason the air terminal unit in a dual-duct HVAC system is typically referred to as a “mixing box.” A dual-duct HVAC system is more complex and requires more ductwork than a multizone HVAC system but provides the ability to more easily and accurately control individual zone temperature.

6.2 DUAL-DUCT HVAC SYSTEM DESCRIPTION

Figure 6-1 illustrates a diagram of a dual-duct HVAC system. A dual-duct HVAC system supplies conditioned air to the zones that it serves using two parallel ducts. The heating duct has a continuously operating heating coil that heats the supply air and the cooling duct has a continuously operating cooling coil that cools the supply air. At each zone served by the HVAC system, the two main supply ducts are tapped and the cold and hot air streams are mixed proportionately in an air terminal unit typically referred to as a “mixing box.” The temperature and volumetric rate of the combined air stream delivered to the zone maintains the desired temperature in the zone. Automatic dampers in the air terminal unit control the proportionate mixing of the two air streams. A thermostat in the zone served by the terminal unit usually controls the setting of the dampers in the terminal unit.

Dual-duct HVAC systems are constant-air-volume, variable-air-temperature systems. Dual-duct HVAC systems control the temperature of the air delivered to the zone by mixing the hot and cold airstreams before delivering the conditioned air to the zone. This mixing of the cold and hot airstreams is not energy efficient and, unless an HVAC application requires constant volume air delivery, dual-duct systems are not used in new construction. VAV HVAC systems covered in Chapter 3 have replaced dual-duct HVAC systems in most applications because they are more energy efficient. However, there are existing buildings that where dual-duct HVAC systems are still in use despite their inherent inefficiency.

Figure 6-1 is a diagrammatic illustration of the dual-duct HVAC system. The same system is shown in schematic form in Figure 6-2. The dual-duct HVAC system illustrated in both Figures 6-1 and 6-2 is a single-fan system. The single fan supplies air to both the heating and cooling coils that increases and decreases the supply air, respectively. The heated and cooled airstreams are then distributed to the building zones served by a parallel duct system. One duct is the hot air supply duct and the other is the cold air supply duct. At each zone, both supply ducts are tapped and the two air streams are fed into a dual duct air terminal unit that mixes the two air streams before delivery to the zone by supply air diffusers. A thermostat located in the zone controls the mixing box based on the desired versus actual temperature in the space, see Figure 6-1.

Figure 6-3 illustrates a variation on the single-fan system, see Figures 6-1 and 6-2. The dual-duct HVAC system uses two fans instead of one, see Figure 6-3. One fan is dedicated to the hot air supply system and the other fan is dedicated to the cold air supply system. The dual-fan, dual-duct HVAC system provides greater control of airflow and energy savings than the single-fan system. With the dual-fan system, each fan can be better matched with the system it serves making operation more efficient. A variable frequency drive can also be coupled with each of the two fans to better control airflow based on duct static pressure which will improve both the energy efficiency of the system and the airflow to the zone.

6.3 DUAL-DUCT HVAC SYSTEM APPLICATION

A building or an area within a building that has multiple zones that each have different thermal load profiles or comfort requirements and will be served from a single central HVAC system or air handling unit. This is an example of where a dual-duct HVAC system would be a better choice than a multizone HVAC system. Dual-duct HVAC systems have been installed in office buildings, hotels, apartment houses, hospitals, schools laboratories and other commercial and institutional buildings. A common characteristic of these multi-zone buildings is a highly variable sensible heat load that matches well with the operational characteristics of a dual-duct HVAC system.
FIGURE 6–1 DUAL-DUCT SINGLE-FAN HVAC SYSTEM
FIGURE 6-2 DUAL-DUCT SINGLE-FAN HVAC SYSTEM SCHEMATIC DIAGRAM
FIGURE 6–3 DUAL-DUCT DUAL-FAN HVAC SYSTEM SCHEMATIC DIAGRAM
VAV HVAC systems have supplanted the use of dual-duct HVAC in many applications, such as office buildings, where the volumetric rate of air delivery at constant temperature to zones served can be varied in order to maintain the desired temperature. In general, VAV HVAC systems are more efficient than dual-duct HVAC systems and require less ductwork. However, dual-duct HVAC systems are still used where a constant volume of conditioned air is required to be delivered to zones from a central HVAC unit or air-handling unit as in the case of some locations within hospitals and laboratories. In applications where a central HVAC unit or air handling unit serves multiple zones and constant volume air delivery is required along with individual zone control, a dual-duct HVAC system is a good choice.

6.4 DUAL-DUCT HVAC SYSTEM OPERATION

6.4.1 Dual-Duct HVAC System Classification By Air Velocity

Dual-duct HVAC systems can be classified by air velocity as either low or high velocity dual-duct HVAC systems.

6.4.2 Low-Velocity Dual-Duct HVAC Systems

An HVAC system conditions all the air in a central unit and distributes it to the zones served through two parallel main air supply ducts. One duct carries cold air and other hot air. As a result, sources of both heating and cooling are provided to all zones at all times. In each conditioned space or zone, an air terminal unit that is responsive to a room thermostat mixes the hot and cold air in proper proportions to satisfy the prevailing load of the space. The central air-handling unit in Figure 6-4 may be either factory-assembled and delivered to the jobsite as a unit or built-up from components at the jobsite.

The return air system in Figure 6-4 is normally low-pressure and can be either a plenum or ducted return. For small systems, the return fans can be eliminated if provisions are made to relieve the excess outdoor air from the conditioned spaces. Return air fans are used in dual-duct HVAC systems that have economizer cooling cycles with substantial return air ductwork.

Cold chamber sprays can also be used to produce evaporative cooling during intermediate seasons to improve system efficiency and reduce cooling costs. In dry climates, evaporative cooling by sprays will result in a substantial savings by reducing the amount of mechanical refrigeration required for cooling. Water sprays in the warm chamber are installed only when control of humidity is required for winter operation. The use of water sprays on all-air comfort installations has greatly diminished because of operating and maintenance problems associated with sprays. If water sprays are used, regular water analysis and treatment is recommended.

6.4.3 High-Velocity Dual-Duct HVAC Systems

Dual-duct high velocity or high pressure systems operate in a manner similar to low velocity systems except that the supply fan runs at a much higher pressure and each zone requires a mixing box with sound attenuation, see Figure 6-5. At higher pressures, more energy is required to operate the supply air fan and return air fan than at low pressures. Therefore, high-velocity dual duct HVAC systems are not recommended for new buildings. Where high-velocity dual duct HVAC systems are used in existing buildings, a close analysis of the pressure drops within the duct system should be made after the building loads have been reduced. When building loads are reduced, fan pressures can often also be reduced to the minimum pressures required to operate either the existing mixing boxes or the retrofitted VAV boxes.

6.5 DUAL-DUCT HVAC SYSTEM FEATURES

6.5.1 Dual-Duct HVAC System Advantages

The advantages of a dual-duct HVAC system are very similar to those of a multizone system as discussed in Chapter 4. The following are some of the advantages of a dual-duct HVAC system.

- When the outside air temperature is low enough, the dual-duct system can use 100 percent outdoor air for cooling without using the refrigeration equipment.

- A recommended combination return air and exhaust air fan can be used to exhaust excess air to the outdoors as well as return air to the central air handling unit. This aids in maintaining the proper building pressurization.

- All of the supply air can be filtered to the required efficiency.
FIGURE 6–4 DUAL-DUCT LOW VELOCITY SYSTEM
FIGURE 6-5 DUAL-DUCT HIGH VELOCITY SYSTEM
• Ventilation air may be preheated where required.

• The degree of dehumidification or humidification is determined by the apparatus arrangement and the building occupancy requirements. Spray systems are optional and may be included.

• When 100 percent air systems are used, there is complete absence of water, steam, and drain piping; electrical equipment; wiring; and filters in conditioned spaces.

• The system can be used in combination with direct radiation or with other conditioning methods to avoid the necessity of placing mechanical equipment at the perimeter of the building, saving valuable floor area. This feature is especially valuable in existing buildings when the removal of an existing heating system is unnecessary or not otherwise justified. It also reduces fan power operating costs by permitting unoccupied period heating with the perimeter radiation instead of the fan system.

• Zoning of central equipment is seldom required. Under some design and operating conditions, however, it may be advantageous to provide separate air handling equipment for exterior and interior zones.

With the heating coil available on a year-round basis, the system is highly rated in its ability to maintain temperature and humidity conditions in lightly loaded and no-load zones. However, dual-duct HVAC systems are not considered as good as terminal reheat systems for humidity control.

6.5.2 Dual-Duct HVAC System Disadvantages

The main disadvantages of a dual-duct HVAC system are:

• To make the system mechanically stable, adequate provisions must be incorporated in the terminal units for control of volumetric delivery.

• Due to space limitations in the majority of installations, velocities and pressures are higher in the dual-duct system than those required in other types of systems.

• The arrangement of two parallel ducts with cross-overs to terminal points will require special attention, study, and technique on the part of the duct designer and installer.

• Operating economy of constant volume dual-duct systems is not as good as for VAV dual-duct systems.

6.6 CENTRAL DUAL-DUCT HVAC SYSTEM EQUIPMENT

The central dual-duct HVAC system can be either a unitary HVAC system or an air-handling unit served by a central cooling and heating plant. The central unit can be a factory built unit or it could be a built-up unit that is assembled at the jobsite. Either type of unit could be in a blow-through or draw-through configuration.

Since the dual-duct system may be designed in either blow-through or draw-through arrangements, several special aspects in equipment selection peculiar to these arrangements must be considered. Fans in a blow-through arrangement should have perforated plates placed in front of the air discharge and at the heating and cooling coils to distribute the air evenly. To regulate the fan discharge air pattern in a more efficient manner, an evasé section should be used. An evasé is a cone-shaped exhaust stack that recaptures static pressure from velocity pressure. Its length should be 1.5 to 2 times the fan discharge equivalent diameter.

The dehumidifiers will usually be a dry coil type with sprays, if needed or desired. These dehumidifiers should be selected to cool and dehumidify the air mixture from entering conditions to the apparatus dew-point. At times it may be desirable to operate the selected dehumidifier at a lower dewpoint to compensate for possible irregularities of apparatus arrangements that may contribute to excessive bypassing of outdoor humid air.

In either a blow-through or a draw-through arrangement, the re heater should be selected to heat the warm air quantity from the design winter cold air temperature to the required warm air temperature. This selection provides enough capacity in a draw-through arrangement to reheat the air in summer from the fan discharge temperature to room temperature. It is recommended that reheaters be selected with about 15 to 25 percent excess capacity to provide for morning pick-up and duct heat losses.

The minimum outdoor air precooling coils should be designed to cool the minimum outdoor air from out-
door design conditions to the room dewpoint level. The outdoor air intake screen, louvers, minimum and maximum dampers, minimum outdoor air preheaters, if required, return air fan and dampers, and air filters are selected for both blow- and draw-through arrangements, using standard procedures. The degree of filtration desired determines the filter selection.

6.7 DUAL-DUCT HVAC SYSTEM AIR TERMINAL UNITS

6.7.1 Constant Volume Air Terminal Units

There are two types of constant volume air terminal units or mixing boxes used in dual-duct HVAC systems:

- Air-terminal units that control space temperature only.
- Air-terminal units that maintain both the temperature and the desired airflow volume requirements.

A thermostat located in the zone controls an operator that modulates the valving device in the air terminal unit to correctly proportion the warm and cold air delivered to the zone. However, without volume regulation the airflow to the conditioned spaces served by the central HVAC system equipment can vary with changes of pressure in the two ducts. The potential for airflow variation increases with the size and pressure of the distribution system. As a result, the air terminal unit used should be a constant volume type of unit with thermostatic mixing of the warm and cold air.

Figures 6-6 and 6-7 show common electromechanical methods of controlling this type of unit in order to satisfy both the temperature and volume requirements. The mixing is similar in each case with operator-controlled valves positioned by a room thermostat.

In Figure 6-6, a spring-loaded regulator that closes as the system pressure rises controls the airflow. The spring and movement may be calibrated to maintain constant airflow and the adjustment may be preset for any prescribed air volume. The space thermostat usually prevents air from bypassing from one duct to the other when unequal pressures exist in the two ducts. However, air check valves at the inlets to the unit may be required under certain conditions.

In Figure 6-7, the cold and warm air dampers have separate operators. The thermostat controls the flow of warm air and a static pressure regulator compensates for changes in warm air supply by varying the cold airflow. When the static pressure regulator is used as a flow control, the sensitivity required depends on the permissible air volume variations.

With a constant volume air terminal unit (Figure 6-7), care should be taken in design of the system and in operating practices not to excessively reduce warm air pressures at the inlet to the unit. If the warm air pressure becomes excessively low, the heating function of the unit will be impaired or entirely lost. Inserting a relay in the control circuit of the unit can eliminate this deficiency in the control arrangement. This relay will allow the thermostat to override the action of the static pressure regulator and to close the cold damper when and if pressure in the warm duct is reduced excessively during the heating cycle.

Stratification caused by improperly designed terminal units can occur and that causes uneven temperature conditions near the unit discharge. If multiple outlets are connected to a mixing box by low-pressure ducts branching horizontally from the main duct connecting to an overhead unit discharge, different outlet delivery temperatures can occur during partial cooling demand as a result of stratification. Splitting the main discharge duct horizontally into branch ducts will usually avoid this condition.

Air terminal units, particularly those with capacity over 1000 cfm (472 L/s), can generate sufficient internal noise to require installation of downstream attenuators or duct lining. Sound power spectrums of noise generation for the range of unit sizes at various air deliveries and inlet static pressures are available from many air terminal manufacturers. When units are installed within the occupied spaces with no intervening sound barrier, noise will be radiated directly from the terminal unit casing into the room. The amount of this noise will approximate that emitted from the discharge. The sound from both sources in these situations should be added logarithmically to determine the resultant room sound levels.

6.7.2 Air-Terminal Unit Location

Dual-duct air terminal units are also referred to as mixing boxes. Dual-duct air terminal units are designed to be installed in either of the following two locations:

- Under-Window Unit
- Ceiling-Mounted Unit
FIGURE 6-6 MIXING AND VOLUME CONTROL METHOD USING SELF-ACTUATED SPRING-LOADED VOLUME REGULATOR FOR CONSTANT VOLUME SYSTEM

FROM THERMOSTAT

WARM AIR

VOLUME CONTROL DAMPER

COLD AIR

MIXING VALVE
FIGURE 6–7  MIXING AND VOLUME CONTROL METHOD USING FLOW REGULATOR FOR CONSTANT VOLUME SYSTEM
Under-window units are limited in size, capacity, and architectural adaptability to the conditioned space in which it is installed. The space limitations and location also limit the acoustical treatment that is an especially important consideration for air terminals with large air capacities.

Ceiling-mounted units are usually concealed or isolated from the space it conditions. The fact that they are mounted above the ceiling and out of sight eliminates the architectural concerns with their appearance as well as acoustical concerns. While location of the air-terminal units above the ceiling or outside the space that they serve should reduce acoustical issues associated with their operation, acoustics should still be considered in the location of ceiling-mounted units as well as the supply air velocity through them.

6.8 SYSTEM OPERATION

The dual-duct terminal unit is designed to:

- Supply the correct proportions of the cold and warm air through thermostatically controlled air valves or dampers in the air terminal unit serving the zone.
- Mix the two airstreams and discharge them into the secondary duct system at acceptable sound levels.
- Provide a constant volume of discharge air by compensating for varying inlet duct static pressures.

Varying amounts of cold, dehumidified air from the cold duct or moderately heated air or air slightly above room temperature from the warm duct are supplied to the air terminal units to satisfy the demands of the zone thermostat. The cold air supply at 100 percent volume is designed to offset the sensible and latent heat loads and the ventilation requirements of the space. The warm air is supplied to keep the total supply air volume constant whenever the space or zone thermostat reduces the cold airflow.

There are three methods of operating the warm air portion of the system.

- Warm air temperature is maintained slightly above the temperature of the zone served.
- Warm air temperature is controlled by a return air hygrostat that raises the temperature as the relative humidity increases. As the warm air temperature is raised, the space or zone thermostat calls for less warm air and more dehumidified air, thus reducing the rising relative humidity.
- Warm air temperature is constantly maintained at a higher level.

The dual-duct terminal units are equipped with valves, valve actuators, and volume compensators to provide constant volume regardless of varying pressure within the cold air and warm air supply ducts. The terminal unit warm air damper is normally open. Space temperature is controlled by a thermostat operating the cold and warm air valve actuators to provide the needed mixture of the two airstreams to satisfy the load.

6.9 IMPROVING EXISTING DUAL-DUCT HVAC SYSTEM PERFORMANCE

The following are recommendations for improving existing dual-duct HVAC system performance:

- Under conditions when there is no cooling load, install controls that will close off the cold air duct, shut down the chilled water supply and operate the system as a single duct heating system, and adjust the hot air supply temperature accordingly. Make sure outdoor air requirements are still met.
- Under conditions where there is no heating load, install controls that will close off the hot air duct, shut down the hot water supply, steam supply, or electricity to the hot air duct and operate the system as a single duct cooling system, and adjust the cold air supply temperature accordingly. Make sure outdoor air requirements are still met.
- Provide volume control for the supply air fan by installing a variable frequency drive (VFD).
- Change the dual-duct HVAC systems to VAV HVAC system when an energy analysis shows that adding VAV air terminal units and fan controls is economically feasible.
- Return air is a mixture of all zones and reflects the average building temperature. In some existing dual-duct HVAC systems, it is possible to stratify the return air and the outdoor air by installing splitters so that the hottest air favors the hot deck and coldest air favors the cold deck. This will reduce both heating and cooling loads and improve system efficiency.
• A hot gas coil or condenser water coil can be installed in the warm air duct to supplement the new heat requirements with recovered heat. If a condenser water coil is used, it is best to use a dual circuit condenser.

Other alternative methods of improving existing dual-duct HVAC system performance include:

• The unit fan is placed on a variable frequency drive (VFD) and the VFD modulates the fan to maintain the temperature setpoint in the zones served. Box decks are controlled independently as variable-air-volume (VAV) outlets. The cooling deck modulates from maximum down to minimum and the heating deck modulates from minimum to maximum. Minimum box airflow is sum of hot and cold deck minimums that is normally used to maintain ventilation requirements.

• The dual-duct system uses two air-handling units (AHU) with one using 100-percent return air and the other with using 100-percent outside air and an economizer. Both units operate under setpoint control using VFDs. The boxes are programmed with multiple setpoints on each deck. The deck connected to the outside air AHU is the ventilation and free cooling deck with both occupied and unoccupied setpoints. The other deck that is connected to the return air unit is used for both heating and mechanical cooling.
7.1 INTRODUCTION

Induction reheat HVAC systems have been used in exterior rooms of multi-story, multi-room buildings such as office buildings, hotels, hospital patient rooms and apartments. Specifically, these systems have been used in buildings where cooling may be required in one room and heating may be required in an adjacent room. These systems are able to handle the loads with minimum space requirements for mechanical equipment by reheating or recooling the conditioned primary air supplied to the space using induction terminal units. Like the terminal reheat HVAC systems discussed in Chapter 5, induction reheat HVAC systems are inefficient. Despite the inherent inefficiency of induction reheat HVAC systems, there are existing buildings that use induction reheat HVAC systems and there may be special applications where the advantages these systems provide outweigh the disadvantage of their inefficient energy use.

7.2 SYSTEM DESCRIPTION

An induction reheat HVAC system is very similar to a terminal reheat HVAC system. An induction reheat HVAC system is a constant volume, single-zone HVAC system that reheats or recools the preconditioned air supplied by the single-duct central HVAC system in an induction terminal unit before supplying conditioned air to the zone served. Figure 7-1 provides a functional diagram of an induction reheat HVAC system. An induction reheat HVAC system consists of the following components:

- Central air conditioning system that could be a local air handling unit served by central heating and cooling plants or a unitary HVAC system such as a roof top unit (RTU).
- Air distribution system consisting of a single supply duct.
- Hydronic distribution system capable of supplying hot or chilled water, or both, to induction terminal units located in the zones served.
- Induction terminal units located in the zones served.

The components that comprise an induction reheat HVAC system are addressed in detail throughout this manual. Air-handling units (AHU) are covered in Chapter 13 and unitary HVAC systems are addressed in Chapter 8. Air distribution systems are discussed in Chapter 12 and hydronic distribution systems including induction terminal units are covered in Chapter 15. Central cooling and heating plants are addressed in Chapters 9 and 10, respectively.

The primary air that is preconditioned by the central HVAC system is supplied to the space through induction terminal units, see Figure 7-1. These induction terminal units mix the preconditioned primary air with the secondary air returned from the zone served. The mixed primary and secondary air streams are then reheated or recooled by a water-to-air heat exchanger coil in the induction terminal unit. The temperature in the zone determines the amount of reconditioning of the mixed air stream that is required. A thermostat located in the zone controls the temperature of the mixed supply air delivered by modulating the volumetric flow of hot or cold water through the water-to-air heat exchanger via a control valve contained in the induction terminal unit.

The primary purposes for using a terminal reheat HVAC system are as follows:

- Permit zone or space control for areas of unequal loading.
- Provide heating or cooling of perimeter areas with different exposures.
- For process or comfort application where close control of space temperature and humidity is required.

The central air conditioning system can either supply a mixture of outdoor and return air or 100 percent outdoor air to the induction terminal unit. Some primary air systems are designed to operate with 100 percent outdoor air at all times while other systems, like the system illustrated in Figure 7-1, have return air systems that can include an economizer cycle for use during intermediate seasons. When the quantity of the primary air that is supplied exceeds the ventilation or exhaust requirements, the excess air is often recirculated by a return system common with the interior system.

7.3 INDUCTION TERMINAL UNITS

Figure 7-2 shows the basic arrangement for an induction terminal unit. Primary air, conditioned by the central HVAC system is supplied to the unit plenum through supply air ductwork. The plenum is acoustically treated to attenuate part of the noise generated in the duct system and in the unit. A balancing damper is used to adjust the primary air quantity within limits.
FIGURE 7-1 INDUCTION REHEAT SYSTEM
FIGURE 7-2 INDUCTION TERMINAL UNIT FUNCTIONAL DIAGRAM
The high-pressure air flows through the induction nozzles and induces secondary air from the room through the secondary coil. This secondary air is either heated or cooled by the coil depending on the season, the room requirements, or both. Ordinarily, no latent cooling is performed in the induction terminal unit but a drain pan is typically provided to collect condensed moisture resulting from unusual latent loads of short duration. The primary and secondary air are then mixed and delivered to the zone served by the induction terminal unit.

Induction terminal units are usually floor mounted and installed at the building perimeter under windows to offset winter downdrafts. When installed under windows, an induction terminal unit has the advantage of providing gravity heating during off-hour operation that allows the shutdown of the primary air system. Induction terminal units are also available for overhead installation. Overhead installation of induction terminal units is uncommon because the duct connections conveying induced air have limited static pressure and result in both decreased airflow and reduced unit capacity.

Induction terminal unit operation is typically controlled by valves that regulate hot and chilled water flow through the water-to-air heat exchanger in response to a room thermostat or a unit mounted thermostat with its sensing element located in the induced air stream. Where multiple units are installed to serve a single zone, it is best to have a single thermostat to control the flow in all units. As an alternative to water flow control, induction units that are designed for use on two-pipe hydronic distribution systems can include an automatic damper arrangement that bypasses induced air around the secondary coil to achieve the desired supply air temperature, see Figure 7-3.

### 7.4 SYSTEM OPERATION

#### 7.4.1 Normal Temperature Systems

The primary air stream provides full cooling capacity to the induction terminal units in the induction reheat HVAC system, see Figure 7-1. Heating or cooling the secondary or induced air stream recirculated from the zone served achieves zone control. This type of induction terminal unit is used when it is desired to introduce supply air to the space at a higher temperature or permit higher space air movement without increasing the quantity of primary air above that required for cooling.

The primary air is discharged from nozzles arranged to induce room air into the terminal unit at a rate approximately four times the volume of the primary air.

![Figure 7-3 Two-Pipe Induction Terminal Unit Bypass Control](image-url)
The induced air is cooled or heated by the secondary water-to-air heat exchanger. This heat exchanger coil may be supplied by any of the following types of hydronic distribution systems:

- A two-pipe system where either hot or chilled water is available.

- A three-pipe system where there is a separate supply of both hot or chilled water to the induction terminal unit and, after passing through the unit, the two streams are mixed into a common return.

- A four-pipe system where supply and return loops of both hot and chilled water are continuously available.

During cooling, the primary air system provides some or all of the sensible cooling capacity and all of the required dehumidification. If it is needed, additional sensible cooling can be provided by cooling the induced air from the zone using the secondary water-to-air heat exchanger. The conditioned mixture of primary and secondary air is then discharged to the zone. In winter when the primary air is provided at a low temperature and additional moisture is needed, the air can be humidified. Under this condition, the induction terminal unit’s secondary coil supplies all the required room heating.

### 7.4.2 Low Temperature Systems

A modification of the induction reheat HVAC system provides a very low supply air temperature for cooling at high load conditions along with the resetting of the supply air to a higher temperature for lighter load conditions. The reheat coil is located in the primary air system and room air is induced but not heated. A low-temperature system permits a reduction in both equipment and duct sizes. Room air is induced at a fixed ratio to raise the supply air temperature that prevents the introduction of extreme temperatures to the zone as well as condensation on supply grille surfaces. With the air volume selected for a high temperature difference between the supply and space air, the total air volume for the system is reduced considerably.

### 7.5 SYSTEM ADVANTAGES AND DISADVANTAGES

#### 7.5.1 System Advantages

Despite the inherent disadvantage of being energy inefficient, induction reheat HVAC systems have the following advantages:

- Individual zone temperature control is possible at relatively low cost with the capability of adjusting each thermostat in the zone for a different temperature.

- The provision of separate hot and chilled water sources makes it possible to heat or cool each induction terminal unit based on occupant needs during differing weather conditions where a choice may be necessary.

- A minimum of space is required for the primary air duct system as a result of using secondary water system for cooling and heating. The return air duct system is reduced in size and sometimes can be eliminated or combined with the return air system provided for other building areas such as the interior spaces.

- The size of the central air handling apparatus can be smaller since less air must be conditioned.

- Dehumidification, filtration, and humidification are performed in a central location remote from conditioned spaces.

- Ventilation air supply is positive.

- The space can be heated without operating the primary air system by using the secondary water system when the building is unoccupied.

- Annual energy consumption for induction systems, when applied in exterior spaces with high solar, electrical, and other thermal loadings, can be lower than that of other system types.

- The individual induction units do not contain fans, motors or compressors. Routine service is generally limited to the temperature controls, cleaning of the lint screen or filters, and occasional cleaning of the induction nozzles.
When properly selected, induction units can operate relatively quietly, and the acoustical performance will not vary over the life of the installation.

7.5.2 System Disadvantages

Disadvantages of induction reheat HVAC systems, in addition to being energy inefficient include the following:

- The relatively low primary air quantities make the design of the system for operation during the intermediate fall and winter seasons more critical than with other system types.
- The induction system has operating complexities not present in other systems. Accordingly, an operator who understands the system cycles and changeover procedures is essential.
- For most buildings, the application of these systems is limited to perimeter spaces. Separate systems are required for other building areas.
- Controls tend to be more complex than for many all-air systems.
- Secondary airflow can cause the coils of induction units to become dirty to the point that screens and low efficiency filters require frequent in-room maintenance in order to avoid reduced thermal performance.
- The primary air supply usually is constant with no provision for shutoff. This is a disadvantage where room occupants or the building operator may prefer to cut off the air conditioning to selected spaces to reduce energy costs.
- A low chilled water temperature is needed to control space humidity.
- The system is not applicable to spaces with high exhaust requirements such as research laboratories unless supplementary ventilation air is provided from other systems.
- Central dehumidification eliminates condensation on the secondary water heat transfer surface under maximum design latent load. However, abnormal moisture sources resulting from open windows or people concentration can cause condensation. This could be damaging if the terminal units do not have condensate drain pans.
- Due to space limitations and duct routing, measurement of airflows to the induction unit (primary air) could be difficult to measure and balance. In addition, the induced airflow is difficult to measure and balance.
8.1 **INTRODUCTION**

Unitary HVAC systems include a wide variety of different types of air-conditioning units that include their own integral refrigeration cycle. Unitary HVAC systems include the following types of packaged HVAC systems that are commonly used in commercial, institutional, and residential buildings:

- Room Air Conditioners
- Split-system Air-Conditioners
- Packaged Terminal HVAC Units
- Computer-Room Air-Conditioners
- Air-Source Unitary Heat Pumps
- Water-Source Unitary Heat Pumps

Unitary HVAC systems are manufactured and shipped as a unit with all components selected and coordinated by the manufacturer. This allows standardized performance testing and rating of these units as a system unlike larger central HVAC systems that are custom designed and built for a specific building. In addition, unitary HVAC systems usually have a lower initial cost and are simpler to install than custom component HVAC systems. Note: For editorial convenience, this chapter’s descriptions are primarily specific to cooling equipment until Section 8.9 where heat pumps are detailed.

8.2 **UNITARY HVAC SYSTEM CHARACTERISTICS**

Unitary HVAC systems are also referred to as packaged air-conditioning systems. These systems are typically manufactured as a complete standardized unit that can provide heating, cooling, and ventilation for a complete building or a zone within a building. Unitary HVAC system components are matched and assembled to achieve specific performance objectives by the manufacturer. These systems are factory assembled into an integrated package that includes fans, filters, heating coil, cooling coil, refrigeration compressors, controls, and other components as required. As a result, unitary HVAC systems are only available in pre-established increments of capacity with set performance parameters unless they are custom ordered and built.

Unitary HVAC systems are used in a variety of ways to meet commercial, institutional, and residential building HVAC needs. In general, unitary HVAC systems are used as follows:

- Central or sole HVAC system that serves an entire building. This use is typical for small stand alone commercial buildings and single-family residential buildings.
- Multiple unitary HVAC systems where each individual system serves its own zone within a larger commercial or institutional building or multi-family residential building. An example of this application would be an assisted living facility where each resident had their own living space and dedicated unitary HVAC system to serve their space.
- Supplemental HVAC system in a commercial or institutional building that has specific HVAC requirements that cannot be met by the building’s central HVAC system. For example, a computer room may require closer control of temperature and humidity than can be attained by the building’s central HVAC system and a unitary HVAC system can be used to provide the required environment.
- Complement a commercial or institutional building’s central HVAC system where it is more effective and economical to use a hybrid HVAC system that takes advantage of the characteristics of both systems. For example, a large hotel might use a central HVAC system for public and operational spaces and unitary systems for guest rooms where individual control is desired.

8.3 **UNITARY HVAC SYSTEM**

8.3.1 **Unitary HVAC System Operation**

Each unitary HVAC system includes its own self-contained mechanical refrigeration cycle, see Figure 8-1. From Figure 8-1 it can be seen that mechanical refrigeration is achieved by continuously circulating a fixed amount of refrigerant in a closed loop where it is systematically evaporated, compressed, condensed, and finally expanded to transfer heat from one place to another. For conventional unitary HVAC systems, heat is removed from the zone being cooled and rejected to the outside environment resulting in air conditioning. This is the same process used in mechanical refrigeration where heat is removed from the inside of a refrigerator or freezer and rejected to the atmosphere outside the refrigerator or freezer enclosure. Unitary heat pumps are also capable of reversing this process and absorbing heat from the outside environment and delivering it to the zone being heated.
FIGURE 8-1 MECHANICAL REFRIGERATION CYCLE
8.3.2 Mechanical Refrigeration Cycle

The mechanical refrigeration cycle starts with the evaporator, see Figure 8-1. Evaporation or the change of state from liquid to gas by the refrigerant occurs at low temperature and low pressure by absorbing heat. Heat is absorbed by the refrigerant passing through the evaporator as it changes phase from a low-temperature, low-pressure liquid to a low-temperature, low-pressure gas. The evaporator in a unitary HVAC system is typically an air-to-refrigerant heat exchanger that transfers heat from the air to the refrigerant by passing air through a coil using a motor-driven fan. The cooled air exits the unitary HVAC system and cools the zone that it serves. The warm liquid refrigerant exits the evaporator through refrigerant piping and enters the next stage of the mechanical refrigeration cycle, the compressor.

Upon leaving the evaporator, the refrigerant travels through the connecting refrigeration piping and enters the compressor as a low-pressure, low-temperature gas. The compressor on a unitary HVAC system is typically a hermetically sealed, motor-compressor unit where the motor operates in the same enclosure as the compressor surrounded by refrigerant. The compressor unit compresses the incoming low-pressure, low-temperature refrigerant gas and turns it into a high-pressure and high-temperature gas at its output.

From the compressor outlet, the high pressure and high temperature refrigerant gas travels through the refrigerant piping to the condenser. Like the evaporator, the condenser is typically a refrigerant-to-air heat exchanger in unitary HVAC equipment. The high temperature refrigerant transfers heat to the lower temperature air by passing air through the condenser coil and exits the condenser as a high-pressure, low-temperature liquid. The phase change from gas to liquid by the refrigerant occurs because the refrigerant is cooled as it passes through the condenser and condenses from a gaseous vapor to liquid.

The final stop for the refrigerant in the mechanical refrigeration cycle is the expansion valve which causes the high-pressure, low-temperature liquid refrigerant to expand resulting in a low-pressure, low-temperature liquid, where the discussion of the mechanical refrigeration cycle started.

8.3.3 Direct Expansion Unitary HVAC Systems

Unitary HVAC systems are sometimes referred to as direct-expansion (DX) units. Most unitary HVAC systems are DX units. In a DX unitary HVAC system, the evaporator is in direct contact with the supply air. The evaporator serves as both the evaporator in the mechanical refrigeration cycle and as the cooling coil.

The term “direct” in the designation “direct expansion” refers to the fact that the evaporator is in direct contact with the supply air. The term “expansion” in the designation “direct expansion” refers to the fact that the refrigerant passes through the expansion valve before entering the evaporator that doubles as the cooling coil. The expansion valve reduces the pressure and temperature of the refrigerant passing through the evaporator or cooling coil so that the refrigerant’s temperature is lower than the air passing through the evaporator allowing the refrigerant to absorb heat from the air passing through the evaporator. Typical DX systems include residential window and central air conditioning units as well as rooftop units.

8.3.4 Six Primary Unitary System Components

Each unitary system has its own integral refrigeration cycle and is comprised of the following eight primary functional components:

- Compressor
- Condenser or Condensing Coil
- Expansion Valve
- Evaporator or Cooling Coil
- Fan
- Refrigerant
- Refrigerant Piping
- Controls

8.3.4.1 Compressor

The compressor is an electromechanical device that compresses the refrigerant as it leaves the evaporator in a gaseous form. The compressor in unitary HVAC systems is typically a hermetically sealed motor-compressor unit where the motor operates with the com-
compressor inside a sealed case in the refrigerant. The compressor increases both the temperature and pressure of the refrigerant.

8.3.4.2 Condenser Or Condensing Coil

The condenser transfers heat from the hot refrigerant to the outside air through convection. The condenser is typically a refrigerant-to-air heat exchanger referred to as the condensing coil. A mechanical fan is normally used by unitary HVAC systems to force air through the condensing coil to affect the heat transfer from the refrigerant to the environment using forced convection.

8.3.4.3 Expansion Valve

The cooled refrigerant that is in gas or liquid state after passing through the condenser flows through the expansion valve. The expansion valve allows the refrigerant to quickly expand in the evaporator coil. This rapid expansion of the refrigerant makes it very cold. The refrigerant’s low temperature gives it the ability to absorb heat from the air in the zone it serves. Note: The expansion valve can be a modulating valve that meters refrigerant flow under the control of a pressure/temperature sensing device or a simple “capillary” tube that meters refrigerant flow through a long, small-diameter tube. Energy efficiency concerns are driving the development of more sophisticated expansion devices and away from capillary tubes except in the smaller equipment sizes.

8.3.4.4 Evaporator Or Cooling Coil

The evaporator also serves as the cooling coil in DX units. A mechanical fan is used in unitary HVAC systems to force air from the zone being conditioned through the evaporator that transfers heat from the warmer air to the cooler refrigerant by forced convection and transfers it to the refrigerant. This process both lowers the temperature of the air passing through it and causes the moisture in the air to condense. Reducing the air temperature removes sensible heat and removing moisture from the air removes latent heat. The transfer of heat energy between the refrigerant in the evaporator coil and the supply air increases the temperature of the refrigerant and converts it into gas. The refrigerant enters the compressor in a gaseous state and begins the cycle again.

8.3.4.5 Fan

One or more fans are incorporated into unitary HVAC systems to force air through air-to-refrigerant and refrigerant-to-air heat exchangers that are typically referred to as cooling and condensing coils, respectively. In addition to forced convection, these fans are used to provide necessary ventilation to the building or zone served as well as draw in cool air for free cooling when outside air conditions permit.

8.3.4.6 Refrigerant

The refrigerant is the primary heat transfer medium used in a unitary HVAC system.

8.3.4.7 Refrigerant Piping

Refrigerant piping that connects the compressor, condenser, expansion valve, and evaporator and provides a closed loop for the refrigerant.

8.3.4.8 Controls

Controls for unitary HVAC systems control system operation and typically consist of a single thermostatic control located in the building or zone served by the unitary HVAC system.

Other components may also be added to the six primary components such as heating coils, controlled or fixed outside air intakes, economizers, and humidifiers to tailor a unitary HVAC system to the occupancy’s particular HVAC needs and provide the desired energy efficiency.

8.4 UNITARY HVAC SYSTEM ADVANTAGES AND DISADVANTAGES

8.4.1 Advantages

The advantages of using unitary HVAC systems in commercial, institutional, and residential buildings are as follows:

- Individual room control is provided.
- Heating and cooling capability can be provided at all times, independent of the mode of operation of other spaces in the building.
- Individual ventilation air can be provided.
- Manufacturer matched components have certified ratings and performance data.
- For improved energy control, equipment serving unoccupied spaces can be turned off.
locally or from a central point, without affecting occupied spaces.

- System operation is simple. Trained operators are not required.
- Less mechanical and electrical room space is required than with central systems.
- Initial system cost is generally low.

### 8.4.2 Disadvantages

The disadvantages of using unitary HVAC systems in commercial, institutional, and residential buildings are as follows:

- Limited performance options available because of fixed air flow and cooling coil and condenser sizing.
- These systems are not generally suited for close humidity control except when using special-purpose equipment such as packaged units for computer rooms.
- Temperature control is usually two-position, causing swings in room temperature.
- Equipment life has historically been less than that of central systems.
- Energy usage may be greater than central systems because fixed unit size increments require oversizing for some applications.
- Free cooling by outdoor air economizers is not always available and may be difficult to provide due to limited access to outside air.
- Air distribution control is limited in room units.
- Operating sound levels can be high and should be considered.
- Ventilation capabilities are fixed by equipment design.
- Overall appearance can be unappealing.
- Air filtration options are limited.
- Condensate removal in air-cooled room conditioners can cause dripping on walls, balconies or sidewalks unless the unit provides for removal.
- Maintenance may be difficult because of the many pieces of equipment and the equipment location, frequently in occupied spaces or above ceilings. Maintenance may also be increased because of a greater number individual pieces of equipment.
- Ability to provide ventilation air to a zone or space can also be compromised by fan cycles.

### 8.5 CONVENTIONAL UNITARY HVAC SYSTEM TYPES

Conventional unitary HVAC systems are those systems that are designed and manufactured to cool a building or zone by absorbing heat from the supply air stream. This supply air stream can consist of both return air from the building or zone served and additional outside air intake. These conventional unitary HVAC systems provide ventilation through the intake of outside air. Heating is done by heating the air stream using hot water, steam, gas-fired heat exchangers, or electric heating coils. In contrast to conventional unitary HVAC systems, unitary heat pump systems provide the heating function by reversing the mechanical refrigeration cycle and absorbing heat from the outside environment and delivering it to the building or zone served eliminating the need for a separate heat source in the form of hot water, steam, or electricity. Unitary heat pumps will be addressed later in this chapter.

Conventional unitary HVAC systems can be categorized as follows based on the way that they are physically installed:

- Single-Packaged Units
- Split-Systems
- Packaged Terminal Air Conditioners

### 8.6 SINGLE-PACKAGED UNITS

#### 8.6.1 Installation And Operation

A single-packaged unit is an outdoor unitary HVAC system that is installed to cool, heat, and ventilate an entire building or a portion of it. The complete system consists of the packaged HVAC unit itself along with a ducted air distribution system and a temperature control system. The packaged HVAC unit can be installed on a building roof where it is referred to as a rooftop unit (RTU), mounted on a wall where it is referred to as a wall pack, or on grade outside the building. When a single unit is used to condition an entire building, the unitary HVAC system and associated ductwork constitute a central station all-air system. Rooftop units
are designed as central station equipment for single-zone, multizone, and variable air volume applications. Figure 8-2 provides a diagram of a rooftop unitary HVAC system.

8.6.2 Application

Unitary HVAC systems that use multiple outdoor single-package units to serve a single building are of the single-zone type. The number of units is determined by the temperature control zoning and includes a unit for each zone. The zones are determined by the cooling and heating loads for the space served, occupancy considerations, allowable roof structural loads, flexibility requirements, appearance, duct size limitations, equipment size availability, and other factors. Multiple unit systems have been installed in manufacturing plants, warehouses, schools, shopping centers, office buildings and department stores. These units serve core areas of buildings whose perimeter spaces are served by packaged terminal air conditioners. These systems are usually applied to low-rise buildings of one or two floors, but have been used for conditioning multistory buildings.

8.6.3 Design Considerations

Design considerations include the following factors:

- HVAC Equipment Location
- Duct Insulation
- Zoning
- Return Air Fans
- Heating
- Controls
- Mounting and Isolation
- Vibration
- Noise
- Outdoor Airflow Measurement Capabilities
- Special Considerations
- Accessories

8.6.3.1 HVAC Equipment Location

Centering a rooftop unit over the conditioned space results in reduced fan power, ducting, and wiring resulting in a more efficient and economical installation.

8.6.3.2 Duct Insulation

All outdoor ductwork should be insulated. Refer to local codes for information on the required insulation R value. All exterior ductwork should be sealed to prevent condensation in the insulation and the ductwork insulation should be covered and weatherproofed to keep insulation from getting wet.

8.6.3.3 Zoning

Single-zone unitary HVAC systems should be used instead of multizone systems whenever possible. For large areas such as manufacturing plants, warehouses, gymnasiums, and other similar building types, single-zone units are less expensive and provide protection against total system failure. The loss of one single-zone unitary HVAC system has less impact than the loss of a single, large multizone unitary HVAC system.

8.6.3.4 Return Air Fans

Use units with return air fans whenever there is static pressure loss or when the unit is used to introduce a large amount of outdoor air through an economizer.

8.6.3.5 Heating

Options are available using natural gas, propane, oil, electricity, hot water, steam and refrigerant gas.

8.6.3.6 Controls

The equipment manufacturer provides most operating and safety controls. Although remote monitoring panels are optional, remote monitoring panels should be considered because they permit operating personnel to monitor system performance.

8.6.3.7 Mounting and Isolation

Rooftop units are generally mounted using integral support frames or lightweight steel structures. Integral support frames are designed by the manufacturer to connect to the base of the unit. The completed installation must assure that condensed water is adequately drained. Lightweight steel support structures allow the
FIGURE 8–2 ROOFTOP UNITARY HVAC SYSTEM
unit to be installed above the roof using separately flashed duct openings. Any condensed water can be drained through the roof drainage system.

**8.6.3.8 Vibration**

Rotating equipment such as compressors and centrifugal fans that are an integral part of unitary HVAC system should have vibration isolation. Since most unitary HVAC systems provide vibration isolation for individual pieces of rotating equipment, vibration isolation for the entire unit casing is seldom required.

**8.6.3.9 Noise**

Outdoor noise from unitary equipment should be minimized. Airborne noise can be attenuated by silencers in the supply and return air ducts or with acoustically lined ductwork.

**8.6.3.10 Outdoor Airflow Measurement Capabilities**

Historically, most rooftop installations do not allow for measurement of outdoor airflow. This can create issues with building pressurization and meeting ventilation requirements. Additional duct for a duct traverse of the outdoor air may be required on the outdoor intake of the rooftop unit. The exhaust airflow from the rooftop typically cannot be measured in most installations. That said, HVAC equipment manufacturers are being required by every stricter codes and standards to provide equipment with more features, such as economizers and monitored outside air intakes.

**8.6.3.11 Special Considerations**

In a rooftop system, the air handler is outdoors and requires special installation considerations. Units need to be weatherproofed against rain and snow as well as blowing dirt and sand in some areas. In cold climates, fuel oil will not atomize and must be warmed to burn properly. Hot water or steam heating coils and piping must be protected against freezing. In some areas, enclosures are necessary to permit effective maintenance of units during inclement weather. A permanent access to the roof as well as a roof walkway to protect against roof damage should also be provided.

**8.6.3.12 Accessories**

Accessories such as economizers, special filters and humidifiers are available. Economizer packages are available which are factory-installed and wired. Other options offered are return and exhaust fans, smoke and fire detectors, portable external service enclosures and special filters.

**8.6.4 Advantages and Disadvantages**

**8.6.4.1 Advantages**

The advantages of using single packaged units include the following:

- Equipment location allows for shorter duct runs, reduced duct space requirements and lower initial cost.
- Installation is simplified.
- Valuable building space required for mechanical equipment is conserved.
- Use of multiple units minimizes chances of total system failure.
- Multiple units allow system operation if only a portion of the building requires air conditioning. Only the unit serving that part need operate.

**8.6.4.2 Disadvantages**

The disadvantages of using single packaged units include the following:

- Maintenance of outdoor units is difficult and may be postponed or delayed due to weather.
- Frequent removal of panels for access often destroys the weatherproofing of the unit, causing electrical component failures, rusting, and water leakage.
- Rusting of casings is a problem. Many manufacturers prevent rusting by using vinyl coating and other protective measures.
- Equipment life is reduced by operating in exposed outdoor installations.

**8.7 SPLIT SYSTEMS**

**8.7.1 Operation**

Unitary HVAC systems for indoor locations can be used to cool and heat an entire building or a zone within the building. A split unitary HVAC system consists of an indoor unit, a remote air-cooled condensing unit, a duct distribution system, and a temperature control system. A water-cooled condenser could be used in place of the air-cooled condensing unit but air-cooled
condensers are the most common. The indoor equipment is generally located in service areas adjacent to the conditioned space.

A split DX system is one that divides the components of the refrigeration system into two subsystems allowing the air-cooled condenser and compressor subsystem to be located outdoors and the evaporator to be located indoors. The evaporator is located near or within the conditioned space. The two subsystems are linked by refrigeration piping which limits the distance between them. Figure 8-3 illustrates a split DX system which is commonly used in small commercial buildings and residential applications.

8.7.2 Application

Multiple unit systems generally use single-zone unitary HVAC systems with a unit for each zone. This arrangement includes unitary HVAC equipment supplying variable air volume systems. The following considerations determine the zoning:

- The cooling and heating loads.
- Occupancy considerations.
- Flexibility requirements.
- Appearance considerations.
- Equipment and duct space availability.

Multiple unit systems are popular air conditioning approaches for office buildings, manufacturing plants, shopping centers, department stores and apartment buildings.

8.7.3 Design Considerations

Design considerations for split unitary HVAC systems include the following:

- Core Systems
- Separate Unit for Dedicated Outside Air Systems
- Economizer Cycle
- Specialized Systems

8.7.3.1 Core Systems

Unitary systems supplement perimeter area packaged terminal air-conditioning systems to serve interior spaces. Except for floors with heat loss through the floor or the roof or where tempering of ventilation air would require it, no heating is required by the core area system. Since core areas frequently have little or no heat loss, unitary equipment with water-cooled condensers can be applied with water-source heat pump units serving the perimeter.

8.7.3.2 Separate Unit For Dedicated Outside Air Systems

In this multiple unit system, one unitary HVAC system is an all-outdoor air unit used to precondition outside air for a group of systems. This unit will prevent the introduction of hot, humid air into the conditioned space under periods of light loading. In selecting equipment for this system, the outdoor unit should have sufficient capacity to cool the required ventilation air from outdoor design conditions to interior design temperature and humidity. Zone units are then sized to handle only the internal load for their particular area. This preconditioned outdoor air feature can be incorporated into unitary systems for existing high-rise buildings to allow tenants to control their own systems.

8.7.3.3 Economizer Cycle

Energy use can be reduced in many locales by using outdoor air for cooling in lieu of mechanical refrigeration when outdoor temperature permits. Ideally, units should be located close to an outside wall or outside air duct shaft.

8.7.3.4 Specialized Systems

Special purpose unitary equipment is frequently used to condition computer areas to provide cooling, dehumidification, humidification, and reheat to maintain close control of space temperature and humidity.

8.8 PACKAGED TERMINAL AIR CONDITIONERS

8.8.1 Operation

A packaged terminal air conditioner (PTAC) is a single-zone, constant-volume unitary HVAC system. PTACs contain several components of the airside loop and all the components of the refrigeration, heat-rejection, and controls loops, inside a common casing. PTAC units are typically installed in the perimeter wall of the building which allows the air-cooled condenser to reject heat directly to the outdoors. PTACs are often used in hotels, dormitories, nursing homes, and apartments.

Return air from occupied space is drawn through the front return grille of the PTAC unit by the fan. The un-
FIGURE 8–3 SPLIT-SYSTEM UNITARY HVAC SYSTEM
conditioned air passes through a filter and DX cooling coil before the supply fan discharges the conditioned air from the top of the unit directly into the occupied space. Outdoor air for the ventilation can enter through a separate damper and mix with the recirculated air or it can be delivered to the conditioned space by a dedicated outdoor air system.

8.8.2 Through-The-Wall Air Conditioning

The through-the-wall PTAC includes a complete air-cooled refrigeration and air handling system in an individual package. Each room is an individual, occupant-controlled zone. Cooled or warmed air is discharged in response to thermostatic control to meet room requirements. Controls can be both individual and aggregate for the building. In off-schedule hours, individual room systems can be controlled by occupants and during scheduled hours the individual systems can be centrally controlled.

Each through-the-wall PTAC has a self contained, air-cooled, direct-expansion cooling system, and a heating system using hot water, steam, or electric heating coils, and controls. The two basic configurations are as follows:

- Figure 8-4 shows a wall box, an outdoor louver, heater section, cooling chassis and cabinet enclosure.
- Figure 8-5 shows a combination wall sleeve and combination heating and cooling chassis and outdoor louver.

Room air conditioners are often used in parts of buildings primarily conditioned by other systems. This application is desirable where spaces to be conditioned are:

- Isolated physically from the rest of the building.
- Occupied on a different time schedule such as a clergy office in a church or a ticket office in a theater.

8.8.3 Application

Through-the-wall air conditioner systems are applied in buildings requiring many temperature control zones such as:

- Office Buildings
- Motels and Hotels
- Apartments and Dormitories
- Schools and Other Educational Buildings
- Hospitals and Nursing Homes

The system lends itself to both low- and high-rise buildings. In buildings where stack effect is present, application should be limited to those units that can provide dependable ventilation and have a tight wall of separation between the interior and exterior. This system is applicable for renovation of existing buildings since existing heating systems can still be used.

8.8.4 Advantages and Disadvantages

8.8.4.1 Advantages

The advantages of using packaged terminal air conditioning units include the following:

- Individual room control is provided.
- Heating and cooling capability can be provided at all times independent of the mode of operation of other spaces in the building.
- Individual ventilation air is provided and operates whenever the conditioner operates.
- Manufacturer matched components have certified ratings and performance data.
- For improved energy control, equipment serving vacant spaces can be turned off locally or from a central point, without affecting occupied spaces.
- System operation is simple. Occupants normally control the system and trained operators are not required.
- Less mechanical and electrical room space is required than with central systems.
- Initial system cost is usually low.

8.8.4.2 Disadvantages

The disadvantages of using packaged terminal air conditioning units include the following:

- The system is distributed throughout the building and maintaining all of these individual unitary HVAC systems can be difficult. Note: Since the equipment tends to be of the same size, spare units can be kept to replace units that are not operating properly.
FIGURE 8–4 THROUGH-THE-WALL PTAC WITH SEPARATE HEATING AND COOLING CHASSIS

FIGURE 8–5 THROUGH-THE-WALL PTAC WITH COMBINED CHASSIS
Those removed can be repaired and placed back in storage until needed.

- There is typically no central control of all of the individual unitary HVAC systems making it difficult to optimize building performance.
- Accurate measurement or adjustment of ventilation air is totally dependent on the individual units.
- Outdoor noise generated by some many units operating simultaneously may be an issue.

8.9 UNITARY HEAT PUMPS

8.9.1 Unitary Heat Pump Operation

Unitary heat pumps differ from conventional unitary HVAC systems in that their mechanical refrigeration cycle can be reversed. Reversal of the mechanical refrigeration cycle allows heat to be absorbed by the refrigerant from the outside environment and transferred to the building or zone served by the unit. As a result, unitary heat pumps may not need a separate heating coil and associated source of heat energy such as hot water or steam, or electric resistance to heat the supply air to a building or zone.

8.9.2 Unitary Heat Pump Performance

Unitary heat pumps are rated based on the following three metrics:

- Coefficient Of Performance
- Energy Efficiency Ratio
- Seasonal Energy Efficiency Ratio

8.9.2.1 Coefficient Of Performance

Thermodynamically, a unitary heat pump can be described as a heat engine operating in reverse. This means that it requires a work input to remove heat from a low-temperature source and deliver it to a higher temperature sink. The classical Carnot Cycle defines the maximum efficiency at which a unitary heat pump cycle can operate. This efficiency is referred to as the coefficient of performance (COP) which is dimensionless and depends only on operating temperatures. Coefficient of Performance is defined by ASHRAE as follows:

The ratio of heat removal to the rate of energy input, in consistent units, for a complete refrigerating system or some specific portion of that system under designated operating conditions.

8.9.2.2 Energy Efficiency Ratio

The cooling performance used by the industry is not the dimensionless COP but the energy efficiency ratio (EER). The energy efficiency ratio is defined by ASHRAE as follows:

The ratio of the net cooling capacity in British thermal units per hour (Btu/h) to the total rate of electric input in watts under designated operating conditions.

The COP and EER normally apply only to steady-state operation. Cycling during heating and cooling, frosting effect, and defrosting are transient conditions that decrease the steady-state performance.

8.9.2.3 Seasonal Energy Efficiency Ratio

The Seasonal Energy Efficiency Ratio (SEER) is similar to EER but EER is used when referring to values based on a full annual season of operation. The SEER focus is on the unitary heat pump used during the cooling season and is defined by ASHRAE as follows:

The total cooling output of an air conditioner during its normal annual usage period for cooling (in Btu) divided by the total electric energy input during the same period.

8.9.3 Unitary Heat Pump Types

Like other unitary HVAC equipment, unitary heat pumps are produced and delivered by the manufacturer as a complete unit. Engineered heat pump systems are built-up systems that are an assembly of components like other central heating and cooling systems. Engineered heat pump systems are covered in Chapter 11. This chapter covers the following three types of unitary heat pump systems that are commonly found in residential and commercial buildings:

- Air-Source Unitary Heat Pumps
- Water-Source Unitary Heat Pumps
- Ground-Source Unitary Heat Pumps

8.9.3.1 Air-Source Unitary Heat Pumps

Air-source unitary heat pumps are also referred to as an air-to-air unitary heat pumps. Air-source unitary heat pumps use air-to-refrigerant heat exchangers to reject heat to outdoor air when cooling indoor air and extract heat from outdoor air when heating indoor air.
Air-to-air unitary heat pumps consist of factory-matched refrigerant cycle components. When this equipment is provided in more than one assembly, the separate assemblies are designed and manufactured for use as a unit. Air-to-air unitary heat pumps are shipped from the factory as a complete preassembled unit or units and include internal wiring, controls, and piping. Only the ductwork, external power wiring, control wiring, and condensate piping are required to complete the installation of an air-to-air unitary heat pump. For split systems, it is also necessary to connect the refrigerant piping between the indoor and outdoor units of the air-to-air unitary heat pump system. The appearance and dimensions of the enclosures closely resemble those of conventional air-conditioning units of equivalent cooling capacity.

The selection and installation of an air-to-air unitary heat pump is similar to that described for a conventional unitary HVAC system. Factors that are unique to an air-to-air unitary heat pump include the following:

- The unitary heat pump normally fulfills a dual function, heating and cooling. As a result, only a single piece of equipment is required for year-round comfort. Some manufacturers offer heating-only heat pumps for special applications.
- A single source of energy can supply both heating and cooling requirements.
- Heat output can be as much as two to three times that of the purchased energy input.
- Vents or chimneys may be eliminated which reduces building costs.
- Moderate supply air temperatures in the heating cycle require careful attention and close adherence to air distribution good duct design principles.

Many of the air-to-air unitary heat pump designs and installations are very similar to conventional unitary HVAC systems described earlier in this chapter. Most residential applications consist of an indoor fan and coil unit and an outdoor fan-coil unit. The compressor is usually located in the outdoor unit. Electric heaters are commonly provided within the indoor unit to provide heat during defrost cycles and periods of high heating demand that cannot be satisfied by the refrigeration system when the outdoor temperature is low. Consideration should be given to the disposal of the defrost condensate.

### 8.9.3.2 Water-Source Heat Pumps

Water-source unitary heat pumps are also referred to as water-to-air unitary heat pumps. Water-source unitary heat pumps use water as the heat source in the heating mode and as the heat sink in the cooling mode. The water supply may be a closed water loop, a lake, or a well. A refrigerant-to-water heat exchanger is used as a refrigerant heat sink when cooling indoor air and as a refrigerant heat source when heating indoor air.

A water-source unitary heat pump system has the following advantages over an air-source unitary heat pump:

- A water-to-air unitary heat pump maintains a fairly constant capacity since the heat source is available at a more constant temperature range than the outside air.
- A water-to-air unitary heat pump can operate at a higher seasonal Coefficient of Performance (COP) if the source water has a constant temperature that is relatively high compared to the outside air.

A water-to-air unitary heat pump is able to operate without the need for defrosting.

Storage tanks with solar assisted heating offer an alternative for residential applications that do not have an adequate supply of ground source water. This system has the advantage of operating the collectors at relatively low temperatures where the collector efficiency is the highest. However, the economics associated with solar collectors and storage may be prohibitive when the total system cost effectiveness is considered.

### 8.9.3.3 Ground-Source Unitary Heat Pumps

Ground-source unitary heat pumps use the earth as both a heat sink for cooling and a heat source for heating. The earth’s temperature is nearly constant year around a few feet below the surface and does not vary greatly throughout the United States. Heat transfer between the unitary heat pump and the earth is accomplished by circulating water through a buried piping system that serves as a water-to-earth heat exchanger. When operating in the cooling mode, the water circulating through the underground piping system transfers the heat extracted from the building by the unitary heat pump to the earth. Similarly, when operating in the heating mode the water circulating in the underground piping system extracts heat from the earth and the unitary heat pump uses this heat to heat the building.
The size of the unitary heat pump, needed heat transfer capability, and soil conditions at the site will determine the size and layout of the underground piping system. In general, underground piping coils are spaced horizontally 3 to 6 feet and have a burial depth of 3 to 6 feet for residential and small commercial buildings. However, where greater heat transfer capability is required or the area where the underground coils will be installed is small, vertical coils may need to be installed which go deeper into the earth. Vertical coils that are installed deeper in the earth provide greater heat transfer capability and require less surface area than horizontal coils but the cost of installing vertical coils is higher than horizontal coils. Basically, vertical coils are installed in drilled wells that have the annular space grouted to improve heat transfer.

8.10 COMBINATION UNITARY AND CENTRAL HVAC SYSTEMS

Unitary HVAC equipment that heats and cools perimeter spaces is often used as part of a combination system that also includes a central HVAC system. Combination systems can provide improved humidity control, air purity, and ventilation than can be obtained with unitary systems alone. The central system may also serve the interior building space that could not be conditioned by wall or window-mounted units.
CHAPTER 9

CENTRAL COOLING PLANT
9.1 INTRODUCTION

Central cooling plant is covered in this chapter including chillers and cooling towers. This chapter starts with a discussion of chiller operation with reference to chiller refrigeration cycle and chiller components. It also describes the types of chillers commonly used to supply chilled water in commercial and institutional buildings. The chapter then covers the cooling tower operation, types, and construction. Evaporative cooling systems are covered last.

9.2 CENTRAL COOLING PLANT OPERATION AND COMPONENTS

Figure 9-1 is a schematic diagram of a central cooling plant showing the major components and how they are related. The central cooling plant illustrated uses a water-cooled chiller and a cooling tower to extract heat from the interior of the building. The central cooling plant could have also used an air-cooled chiller. If the air-cooled chiller was used, a condenser water loop circulates through the cooling tower and the cooling tower itself would not be needed. Instead the heat from the building would be rejected to the outside environment from the condenser using a refrigerant-to-air heat exchanger.

The central cooling plant in Figure 9-1 circulates chilled water throughout the building using a hydronic distribution system as discussed in Chapter 15. The system is designed so that the chilled water is supplied at 44°F (6.7°C) as the output of the chiller. It then absorbs heat as the chilled water passes through air-to-water heat exchangers, such as air handling unit cooling coils throughout the building, and then water returns to the chiller at 54°F (12.2°C). The chiller extracts heat from the returning chilled water and starts the cycle again by supplying 44°F (6.7°C) chilled water to the hydronic distribution system.

The heat removed from the returning chilled water is absorbed by refrigerant circulating in the chiller and transferred to the cooling tower water loop using the vapor-compression refrigeration cycle. The cooling tower is supplied with 95°F (35°C) condenser water and, by evaporation, reduces the temperature of the condenser water to 85°F (29.5°C) and then recycles it back to the condenser to start the cycle again.

9.3 CHILLER PURPOSE

Chillers are the key element in a central commercial cooling plant. Chillers are machines that extract heat from the facility by cooling the water that is circulated around the facility to absorb the heat. The water circulated around the facility is typically referred to as “chilled water” and because it is the heat transfer medium for the building’s cooling system it is also often referred to as the “secondary fluid” with the chiller refrigerant being the “primary fluid” in the heat transfer process. Both sensible and latent heat are transferred to the chilled water when it is pumped through cooling coils within air handling units (AHUs), fan coils, or any component of the air-conditioning system that acts as an air-to-liquid heat exchanger whose purpose is to remove the heat from the air passing through it. The affect of this air-to-water heat transfer is to condition the air delivered to a zone by cooling and dehumidifying it. Chiller size is typically expressed in tons of cooling capacity and chillers manufactured for commercial HVAC applications range in size from about 15 to more than 1500 tons.

9.4 CHILLER REFRIGERATION CYCLE

Chillers are essentially refrigeration systems that extract heat from the circulating chilled water using a refrigeration cycle. Chillers use a vapor-compression refrigeration cycle that is also known as a reverse Rankine refrigeration cycle to cool the circulating chilled water in a building. This refrigeration cycle is similar to the refrigeration cycle used in unitary HVAC systems that was described in Chapter 8. The main difference between the chiller refrigeration cycle and the unitary HVAC system is that chillers chill water that is used to cool the supply air to a building zone using a water-to-air heat exchanger. The unitary HVAC systems, on the other hand, chills the air supplied to the zone directly using a refrigerant-to-air heat exchanger.

Figure 9-2 provides a schematic diagram of the vapor-compression refrigeration cycle for a chiller that contains the same four components as the unitary air-conditioning system refrigeration cycle, see Figure 8-1. Each of these components provides the same function as in a unitary air-conditioning system. The refrigeration cycle extracts heat from the building using a refrigerant circulating in a closed loop. The heat absorbed by the refrigerant is then rejected to the outside atmosphere using either a water-to-air or refrigerant-to-air heat exchanger. The four mechanical components that comprise a chiller and allow it to chill water through the vapor-compression refrigeration cycle are:
FIGURE 9-1 CENTRAL COOLING PLANT SCHEMATIC DIAGRAM
FIGURE 9-2 VAPOR-COMPRESSION REFRIGERATION CYCLE
Each of these four components operates exactly the same as it does in a unitary air conditioning system. There are a number of alternative chiller configurations that provide an efficient system which economically meets the unique air conditioning needs of the facility it serves.

### 9.4.1 Compressor

#### 9.4.1.1 Compressor Function and Types

The function of a compressor in the vapor-compression refrigeration cycle is to compress the refrigerant from a low-pressure vapor to a high-pressure vapor. There are two major categories of vapor-compression refrigeration cycle water chillers used to produce chilled water for air conditioning in commercial buildings today:

- **Mechanical Chillers**
- **Absorption Chillers**

Mechanical chillers employ a mechanical compressor to compress the refrigerant. The mechanical compressor that is used to compress the refrigerant is usually a centrifugal, reciprocating, scroll, or rotary screw compressor. These mechanical compressors are driven by a variety of prime movers that include electric motors, steam turbines, diesel engines, and others. However, the most common means of driving a chiller compressor in a commercial building air-conditioning application is a three phase electric induction motor although large chillers sometimes use synchronous motors. For greater detail on electric motor operation and characteristics as applied to HVAC equipment, refer to Chapter 17.

Each of these types of mechanical compressors has unique operating characteristics that make it the preferred choice for a particular chiller size or application. In fact, a mechanical chiller is often classified and referred to by the type of compressor that it employs. For example, a mechanical chiller employing a centrifugal compressor will often be referred to generically as a centrifugal chiller. However, when selecting a chiller for a particular application, manufacturers should be consulted because compressor technologies, ratings, and efficiencies are constantly improving and a compressor type that may not have been suitable for an application in the past may be the preferred alternative today.

Absorption chillers do not have a mechanical compressor but instead use a thermochemical compressor. Absorption chillers are classified according to the heat source that they use to drive their thermochemical compressor. Absorption chillers are classified by their heat source as either:

- **Direct-Fired**
- **Indirect-Fired**

Direct-fired absorption chillers include a burner that burns a fuel to produce the heat needed for the absorption refrigeration cycle. Typical fuels used to provide the necessary heat for the absorption refrigeration cycle include natural gas and fuel oil. Indirect-fired absorption chillers use either waste hot water or steam produced by an external boiler, industrial process or other source to drive the absorption refrigeration cycle. Indirect-fired absorption chillers are often used in combined heat and power (CHP) or cogeneration applications.

A second classification of absorption chillers comes from the number of refrigeration cycles that are incorporated into the chiller. Absorption chillers can be classified as either:

- **Single-Effect**
- **Multiple-Effect**

Single-effect absorption chillers incorporate one refrigeration cycle into their operation whereas multiple-effect absorption chillers incorporate two or more refrigeration cycles into their operation. Multiple-effect absorption chillers are more efficient than single-stage absorption chillers but multiple-effect absorption chillers require more thermal energy in the first stage that restricts many applications to a single-effect absorption chiller. Absorption chiller operation will be discussed in more detail later in this chapter.

### 9.4.2 Condenser

#### 9.4.2.1 Condenser Function and Types

Following the compressor in the refrigeration cycle is the condenser. The function of the condenser is to reject the heat from the chiller so the condenser is essentially a heat exchanger. Chiller condensers for com-
mmercial air conditioning applications remove heat from the refrigerant using either a refrigerant-to-water heat exchanger or a refrigerant-to-air heat exchanger. The type of condenser or heat rejection method is another way that a chiller is classified. There are two classifications of chillers used in commercial air-conditioning systems based on heat rejection method.

- Water-Cooled Chillers
- Air-Cooled Chillers

For a water-cooled chiller, heat is normally rejected to the atmosphere through a cooling tower located outside the building. Cooling towers are discussed later in this chapter. For an air-cooled chiller, heat is rejected to the atmosphere by an air-cooled condenser that is located outside the building. In either case, the refrigerant enters the condenser as a vapor and leaves as a liquid.

Water-cooled chillers use water to remove heat from the condenser. In a water-cooled chiller, water is circulated through the condenser and heat from the refrigerant is transferred from the refrigerant to the water by a refrigerant-to-water heat exchanger. Typically, water-cooled chillers capture and reject building heat by transferring it first from the refrigerant to the water and then running the water through an outside cooling tower. The cooling tower then extracts the heat from the water and transfers it to the atmosphere. The cooled water is then returned to the chiller to begin the heat transfer cycle again. Alternatively, other sources of water can be used to cool the refrigerant such as water supplied by the building’s water utility, water from a well, water from a nearby pond established for this purpose. Water-cooled chillers are typically 100 tons of cooling capacity or greater and are usually located indoors and their associated cooling towers are located outdoors.

Air-cooled chillers reject the heat extracted from the building directly to the atmosphere using a condenser that is essentially a refrigerant-to-air heat exchanger. Air is drawn across the condenser by fans and heat is transferred to the atmosphere by the refrigerant-to-air heat exchange. Air-cooled chillers are typically less efficient than water-cooled chillers because heat rejection for a water-cooled chiller with a cooling tower takes place at or near the outside air’s wet-bulb temperature whereas heat rejection for an air-cooled chiller occurs at the higher dry-bulb temperature.

Air-cooled chillers are manufactured in two basic configurations. In one configuration, the air-cooled chiller is manufactured as a unit with all components in a single enclosure that is either located outside on grade near the building or on the building roof. Integrated air-cooled chillers are typically referred to as packaged units and are the most common configuration. However, the air-cooled chiller could be supplied into two parts like a split unitary air-conditioning unit. With a split air-cooled chiller, the condenser is installed somewhere outside the building like a packaged unit and the other components that comprise the air-cooled chiller are located inside the building. Refrigerant piping connects the inside unit with the outside condenser which usually requires that the two be located close to one another such as having the condenser unit installed on the roof and the remaining equipment installed on the floor directly below.

9.4.3 Expansion Valve

The next mechanical device in the refrigeration cycle is the expansion valve that is also sometimes referred to as the metering device. The purpose of the expansion valve is to control the transfer or meter the high-pressure refrigerant from the condenser into the low-pressure evaporator.

9.4.4 Evaporator

The last major component in the chiller’s vapor-compression refrigeration cycle is the evaporator or water cooler. The evaporator is the mechanical component that cools or chills the water that is circulated throughout the building to collect heat. Like the condenser, the evaporator is a refrigerant-to-water heat exchanger whose function is to transfer the heat from the circulating water to the refrigerant. The evaporator is typically a shell and tube heat exchanger. The incoming refrigerant is in a liquid-vapor mixture and which allows it to absorb heat from the water by changing state and exiting the evaporator as a vapor.

9.5 REFRIGERANT

The refrigerant is the primary heat transfer medium that is circulated in a closed loop within the chiller. The selection of refrigerant is an important consideration in determining equipment and operating costs. In choosing refrigerant, consider coefficient of performance, operating pressures, flow rate, heat transfer properties, stability, toxicity, and flammability. The thermal stability of the refrigerant and its compatibility with materials in contact with it are also important considerations. Special attention needs to be given to the selection of elastomers and electrical insulating materials because many common materials are affected by the refrigerants.
9.6 MOTOR-COMPRESSOR UNITS

9.6.1 Motor-Compressor Unit Types

Compressors require a prime mover that provides rotating mechanical energy that allows them to do the work of compressing the refrigerant in the vapor-compression refrigeration cycle. These prime movers can be any type of energy conversion device that has the power to drive the compressor but are typically electric motors. The physical construction of motor-compressor units used in HVAC equipment impacts the type of prime mover that can be used, chiller operation, and maintenance costs. Mechanical chiller motor-compressor units can be constructed in either of the following two ways:

- External Drive Motor-Compressor Units
- Hermetically Sealed Motor-Compressor Units

Both external drive and hermetically sealed motor compressor units are available.

9.6.1.1 External Drive Motor-Compressor Units

An external drive machine uses a compressor that is driven by a prime mover that is external to the compressor and transfers rotating mechanical energy to it through a common external shaft. External drive motor-compressor units are typically used when a prime mover other than an induction motor is used to drive the compressor. External drive motor-compressor units can use a steam or gas turbine, internal combustion engine, synchronous electric motor, induction electric motor, or other prime mover to drive the compressor. The advantage of an external drive motor-compressor unit over a hermetically sealed motor-compressor unit is that the prime mover is easily accessible for maintenance, repair, or replacement. The disadvantage is that a drive shaft seal is required in the compressor to contain the refrigerant and oil from escaping into the atmosphere and that seal requires regular inspection and maintenance.

9.6.1.2 Hermetically Sealed Motor-Compressor Units

A hermetically sealed motor-compressor unit encloses both the compressor and electric drive motor in the same enclosure the refrigerant, eliminating the need for a drive shaft seal. The electric drive motor operates in the refrigerant atmosphere. As a result, the possibility of refrigerant leakage to the outside atmosphere through the shaft seal is eliminated and the housing reduces motor noise. Since forced refrigerant cooling of the motor is very effective, smaller, less expensive motors can be used with hermetically sealed motor-compressor units. The need for a heavy external base to maintain motor-compressor shaft alignment is also eliminated. As a result, hermetic machines are less expensive than external drive machines to build and install, have slightly greater power consumption than a similarly rated external drive machine, and are quieter. However, if the motor should fail the repair costs for a hermetically sealed motor-compressor unit will be higher.

9.7 MECHANICAL COMPRESSOR OPERATION AND CHARACTERISTICS

9.7.1 Compressor Types

Mechanical chillers employ a variety of mechanical compressors that convert the refrigerant from a low-pressure to a high-pressure vapor. The type of compressor used impacts the mechanical chiller’s operating characteristics. Chillers use four common types of mechanical compressors:

- Centrifugal
- Reciprocating
- Rotary Screw
- Scroll

9.7.1.1 Centrifugal Compressors and Chillers

Centrifugal compressors raise the pressure of refrigerant by using a rotating impeller to impart velocity to the refrigerant and then convert that velocity to pressure. Chillers using centrifugal compressors are typically large chillers rated above 500 tons of capacity. Centrifugal chiller efficiencies are high at full load but centrifugal chillers do not perform as well at partial load as other chiller types.

Since the centrifugal compressor is not of the constant displacement type, it offers a wide range of capacities continuously modulated over a limited range of pressure ratios. By altering built-in design items including number of stages, compressor speed, impeller diameters, and choice of refrigerant, it can be used in liquid chillers having a wide range of design chilled liquid temperatures and design cooling fluid temperatures. Its ability to continuously vary capacity to match a wide range of load conditions with nearly proportion-
ate changes in power consumption makes it desirable for both close temperature control and energy conservation. Its ability to operate at greatly reduced capacity makes for more on-the-line time with infrequent stopping/starting cycles.

Centrifugal packages are currently available from about 80 to 2400 tons (281 to 8440 kWs) at nominal conditions of 44°F (6.7°C) chilled water temperature and 95°F (35°C) condenser water temperature. This upper limit is continually increasing. Field-assembled machines extend to about 10,000 tons. Single-stage, two-stage internally geared machines, and two-stage direct-drive machines are commonly used in packaged units. Electric motor-driven machines constitute the majority of units sold.

Units with hermetically sealed motor-compressor units that are cooled by refrigerant gas or liquid are offered from about 80 to 2000 tons. Open-drive units are not offered by all manufacturers in the same size increments but are generally available from 80 to 10,000 tons. Packaged electric-drive chillers may be of the open- or hermetically sealed motor-compressor type and use three-phase electric motors, with or without speed-increasing gears. Hermetic units use only three-phase induction electric motors.

9.7.1.2 Reciprocating Compressors and Chillers

Reciprocating compressors are piston-style, positive displacement compressors. Reciprocating compressors are used in chillers because they are typically less expensive than other types of compressors. Chillers with reciprocating compressors are also more economical in smaller sizes than chillers using other types of compressors. With a reciprocating compressor, the chiller’s efficiency can be increased at partial load operation by using a variable frequency drive (VFD) or step control. Offsetting their efficiency is the fact that reciprocating compressors have a number of moving parts that can increase their maintenance costs when compared to chillers with other types of compressors.

A distinguishing feature of the reciprocating compressor is its pressure rise versus capacity characteristic. Pressure rise has only a slight influence on the volume flow rate of the compressor and, therefore, a reciprocating liquid chiller retains nearly full cooling capacity even when the actual wet-bulb temperature is above the design wet-bulb temperature. Reciprocating condensers are well suited for air-cooled condenser application and low temperature refrigeration. Available capacities range from about 2 to 200 tons. The use of multiple reciprocating compressor units have become popular for two reasons:

- The number of capacity increments is greater resulting in closer liquid temperature control, lower power consumption, less current inrush during starting, and extra standby capacity.
- Multiple refrigerant circuits are employed, resulting in the potential for limited servicing or maintenance of some components while maintaining cooling.

9.7.1.3 Rotary Screw Compressors and Chillers

Like the reciprocating compressor, a rotary screw or helical rotary compressor is also a positive displacement compressor with nearly constant flow performance. Rotary screw compressors operate by meshing two screw rotors rotating in opposite directions that trap and compress the refrigerant vapor along the rotors to the discharge point. Mid-sized chillers with ratings in the range of 150 to 1,500 tons of chilling capacity often use rotary screw compressors. Rotary screw compressor chillers have fewer moving parts than a comparable reciprocating compressor and typically have lower maintenance costs and a longer useful life. Rotary screw compressors are also efficient at partial load and are capable of adjusting to load swings.

When rotary screw compressors are used for liquid chillers, they are typically oil-injected. When compared to non oil-injected rotary screw compressors, the use oil-injected rotary screw compressors provide the following advantages:

- Reduced operating noise.
- Lower operating speed.
- Increased thermal and volumetric efficiencies.
- The ability to operate at very high pressure ratios.
- Elimination of timing gears.
- Lower discharge temperature that results in lower thermal stress in the rotors.
- Smaller condensers when a portion of the total heat rejection is accomplished using an oil cooler.

Screw compressor liquid chillers are available as factory-packaged units from about 40 to 850 tons. Both
open and hermetically sealed motor-compressor units are manufactured. Additionally, compressor units, comprised of a compressor, hermetic or open motor, oil separator, and oil system, are available from 20 to 2000 tons (70 to 7033 kWs). These compressor units are for use with remote evaporators and condensers for low, medium, or high evaporating temperature applications. Condensing units, similar to compressor units with water-cooled condensers, are also built in the same range and capacity as compressor units.

9.7.1.4 Scroll Compressors and Chillers

Scroll compressors are also positive displacement compressors where the refrigerant is compressed when one spiral rotates around a second stationary spiral resulting in high refrigerant pressures at the compressor discharge point. Scroll compressors are found in smaller chillers such as rotary heat pumps. Scroll compressors can be up to 10 percent more efficient than a comparably sized reciprocating compressor.

9.8 ABSORPTION CHILLERS

9.8.1 Use of Absorption Chillers in the United States

Absorption chillers were first developed, patented, and used in the late 1800’s for cooling. In some parts of the world today, absorption chillers dominate the chiller market because of available energy sources. In the United States, a readily available and reliable electric energy supply and inexpensive electric rates have resulted in limited use of absorption chillers. In the past, absorption chillers have typically been used in commercial building air-conditioning applications under the following conditions:

- Where waste heat energy is available resulting in essentially free chilled water.
- Where utility-supplied electricity is unavailable or expensive than other forms of energy.
- Where other forms of energy such as natural gas, geothermal, or solar are competitive with utility-supplied electric energy.

However, today there is renewed interest in absorption chillers in the United States as the cost of utility supplied electric energy continues to increase at the same time that the availability and reliability of utility-supplied electric energy decreases. Growing interest in the environment, the drive to sustainable construction and high performance HVAC systems for commercial buildings has resulted is a growing demand for absorption chillers in the United States.

9.8.2 Absorption Chillers – Factors for Consideration

Electric motor driven mechanical chillers are the most common type of chiller used in the United States because of ready availability, inexpensive electric energy. However, an absorption chiller may be the better choice under certain circumstances.

- There is a source of waste steam or hot water available in the facility that can be used to provide essentially free chilled water.
- There is a source of thermal energy available such as geothermal or solar that can be harnessed and is more economical than utility-supplied electric energy on a life cycle cost basis.
- Electric demand energy charges are high for utility-supplied electric energy.
- Seasonal electricity and natural gas rates that result in higher electric rates and lower natural gas rates during the facility’s cooling season.
- Government, utility, or manufacturer financial incentives and rebates that lower the first cost, ongoing operating cost, or both making an absorption chiller more economical than a mechanical chiller on a life cycle cost basis.

Absorption chillers can also be an economical alternative in central plants that have multiple chillers where the advantages of both mechanical chillers and absorption chillers can used to optimize overall central chiller plant’s year round operation. For example, absorption chillers could be used during the summer when electricity prices are high and natural gas prices are low and mechanical chillers can be used in the winter when the opposite energy pricing structure is in effect. Also, there may be waste heat available in the winter from boilers that could be used by the absorption chiller to provide any needed chilled water.

9.8.3 Absorption Chiller Operation

An absorption chiller operates based on a vapor-compression refrigeration cycle just like a mechanical chiller. The difference is in the compressor. Rather than using a mechanical compressor, an absorption chiller uses a thermochemical compressor that requires a refrigerant and an absorbent to operate. For
commercial building air-conditioning applications, the refrigerant is usually water that can easily change phase between liquid and vapor. The absorbent is usually either lithium bromide or ammonia both of which have a high affinity for water meaning that either the refrigerant or absorbent can dissolve easily in the other. This characteristic allows water to boil at a lower temperature and pressure than it normally would at standard temperature and pressure (STP) making it possible to absorb heat from the returning chilled water in the evaporator and then release heat in the condenser.

9.8.4 Direct- Versus Indirect-Fired Absorption Chillers

Direct-fired absorption chillers get the thermal energy they need directly from the combustion of fossil fuels such as natural gas or fuel oil. Indirect-fired absorption chillers use waste heat for operation that includes steam and hot water that are byproducts of other processes that effectively provides free energy for chilling water. Indirect-fired absorption chillers are used where heat recovery or cogeneration is possible.

9.8.5 Multiple-Effect Absorption Chillers

The efficiency of an absorption chiller can be improved by adding absorption stages to the refrigeration cycle referred to as “effects.” A basic absorption chiller has one refrigeration cycle and multiple-effect absorption chillers have additional refrigeration cycles cascaded on the primary refrigeration cycle to improve chiller efficiency. Each of the cascaded refrigeration cycles uses the waste energy from the previous cycle for operation. Currently, the following absorption chillers are available:

- Single-Effect Absorption Chillers
- Double-Effect Absorption Chillers
- Triple-Effect Absorption Chillers

9.8.5.1 Single-Effect Absorption Chillers

A single-effect absorption chiller is the simplest and the least efficient because it uses only a single refrigeration cycle. With a single-effect absorption chiller, the refrigerant and absorbent only make one pass through the system to absorb heat. However, a single-effect absorption chiller does not need the refrigerant and absorbent to be at high temperature to operate which makes single-effect absorption chillers more suited for indirect-fired systems that are using hot exhaust gases, water, or steam. Single-effect absorption chillers also work best for systems that are using solar and low temperature geothermal hot water or steam to operate. Single-effect absorption chillers are the most common absorption chillers used today due to their simplicity and low initial cost when compared to double- and triple-effect absorption chillers.

9.8.5.2 Double-Effect Absorption Chillers

Double-effect absorption chillers are more efficient than single-effect absorption chillers because the refrigerant and absorbent make two passes and absorb more heat. Double-effect absorption chillers utilize two refrigeration cycles in tandem with the first driven by the primary heat source and the second driven by waste heat from the first refrigeration cycle. Even though double-effect absorption chillers are more efficient, they are more expensive and the higher first cost of a double-effect absorption chiller must be offset by the energy savings associated with the increased efficiency over the life of the absorption chiller. For double-effect absorption chillers to operate efficiently they need a higher temperature heat source than is usually available from waste or naturally generated heat. Therefore, double-effect absorption chillers are typically direct fired.

9.8.5.3 Triple-Effect Absorption Chillers

Triple-effect absorption chillers can further increase operating efficiency above a double-effect absorption chiller. A third refrigeration cycle is used in a triple-effect absorption chiller which means it will need to operate at a higher operating temperature than a double-effect absorption chiller, require more expensive materials to build, and be more complex to operate and maintain. As in the case of the double-effect absorption chiller, the savings that result from the incremental increase in efficiency over a double-effect absorption chiller will need to offset the increased first cost of the triple-effect absorption chiller.

9.8.6 Absorption Chiller Operating Characteristics

Absorption chillers have capacities from about 10 tons to over 1500 tons of cooling capacity. Their coefficients of performance range from 0.7 to 1.2 and electricity usage from 0.004 to 0.04 kilowatts per ton (kW/ton) of cooling capacity. An electric pump is typically needed by absorption chillers to move the refrigerant and absorbent through the cycle but the amount of energy required to pump the mixture is less than the energy required by the compressor in a mechanical chiller. This is because pumping the liquid mixture
through an absorption chiller requires a lot less energy than compressing gas.

9.9 CENTRAL COOLING PLANTS WITH MULTIPLE CHILLERS

9.9.1 Multiple Chiller Central Cooling Plant Advantages

Central cooling plants with multiple chillers are usually designed to provide standby capacity if a chiller is out of service for repair or routine maintenance. An advantage of this system is that maintenance can be scheduled for one chiller during part-load times and the remaining chiller or chillers can still provide sufficient cooling. Multiple smaller chillers can result in reduced inrush currents at startup, which will avoid the high demand charges that are charged by some electric utilities. Similarly, energy efficiency at partial-load conditions is greater with multiple smaller chillers that can be brought on and off line as needed. There are operational advantages for central cooling plants with multiple chillers having partial or full redundancy but the tradeoff is greater complexity, increased maintenance, and higher initial cost when compared to a single chiller central cooling plant or a multiple chiller central cooling plant without redundancy.

9.9.2 Basic Chilled Water Flow Arrangements

There are two basic chilled water flow arrangements used in commercial building air-conditioning central plant that have multiple chillers:

- Parallel Chilled Water Flow
- Series Chilled Water Flow

9.9.2.1 Parallel Chilled Water Flow

In the parallel chilled water flow arrangement, the warmed returning chilled water is divided among the liquid chillers through a return header. The supply chilled water streams are then combined again at the output of the chiller using a chilled water supply header. As the cooling load decreases, one or more chillers may be shut down and the remaining chiller or chillers must provide colder-than-design chilled water so that when all streams combine including the streams from idle machines the design chilled water supply temperature is maintained in the common hydronic supply piping.

When the design leaving chilled water temperature is above 45°F (7.2°C), all units should be controlled by

9.9.2.2 Series Chilled Water Flow

The series chilled water flow arrangement is preferred over the parallel chilled water flow arrangement in most applications. With series chilled water flow no over chilling is ever required because the chilled water supply passes through each chiller in series and there are no unconditioned water streams though inactive chillers that are mixed with the output of operating chillers before being supplied to the building. In addition, compressor energy consumption is less than it is for parallel chilled water flow arrangement at partial loads. Since the evaporator temperature never drops below the design value because no over chilling is necessary, the chances of evaporator freeze up are minimized with series chilled water flow. However, the chiller should still be protected by a low-temperature safety control. The drawback to series chilled water flow is that chilled water pressure drop may be higher than the parallel chilled water flow arrangement if shells with fewer liquid-side passes or baffles are not available which will increase pumping requirements and pressure.

When the condensers are water-cooled, it is best to pipe them in a series counter flow so that the lead machine is provided with a warmer condenser and chilled water and the lag machine has a colder condenser and
chilled water. Refrigerant compression for each unit is also made nearly the same. If about 55 percent of design cooling capacity is assigned to the lead machine and about 45 percent to the lag machine, then identical units can be used. In this way, either machine can provide the same standby capacity if the other is down and lead and lag machines can be interchanged to equalize the number of operating hours on each.

9.10 COOLING TOWERS

9.10.1 Cooling Tower Operation

A cooling tower is essentially an evaporative cooler that cools the water coming from the chiller condenser to near the outside wet-bulb air temperature by evaporation and then recycles it back to the condenser to start the cycle again. The heat rejected is comprised of two components:

- Heat extracted from the chilled water.
- Heat added by the compressor.

The compressor adds about 20 to 25 percent to the cooling load. Cooling towers rated to operate at 78°F (25.6°C) wet bulb temperature cool water over a 10°F (-12.2°C) range from 95°F (35°C) to 85°F (29.4°C).

Cooling towers are used to dissipate heat from water-cooled refrigeration, air-conditioning, and industrial process systems. Cooling towers can economically cool water to within 5°F (15°C) to 10°F (12.2°C) of the ambient wet bulb temperature or about 35°F (1.7°C) lower than air-cooled systems of comparable size. A cooling tower uses a combination of heat and mass transfer to cool water. The water to be cooled is distributed in the tower by spray nozzles, splash bars, or film-type fill in a manner that exposes a very large water surface area to atmospheric air. Circulation of atmospheric air is accomplished by one of the following methods:

- Mechanical Fans
- Natural Convective Air Currents
- Natural Wind Currents
- Induction Effect From Sprays
- Combination Of Methods

The entering air wet bulb temperature affects the thermal performance of a cooling tower. Entering air dry bulb temperature and relative humidity have an insignificant effect on thermal performance but they do affect the rate of water evaporation. The evaporation rate at typical design conditions is approximately one percent of the water flow rate for each 12.6°F (10.8°C) of water temperature range. The actual annual evaporation rate is less than the design rate because the sensible component of total heat transfer increases as the entering air temperature decreases. In addition to water loss from evaporation, losses also occur because of liquid carryover into the discharge air stream, also referred to as drift, and from blowdown required to maintain acceptable water quality.

9.10.3 Thermal Capacity

The thermal capability of any cooling tower may be defined by the following three parameters:

- Entering and leaving water temperatures.
- Entering air wet bulb temperature.
- Water flow rate.

The thermal capability of cooling towers for air conditioning applications is stated in terms of nominal tonnage based on a heat dissipation of 15,000 Btu per hour.
per ton and a water circulation rate of 3 gallons per minute (gpm) per ton cooled from 95°F (35°C) to 85°F (29.4°C) at a 78°F (25.6°C) wet bulb temperature. For industrial applications, nominal tonnage ratings are not used and the performance capability of the cooling tower is stated in terms of flow rate at specified operating conditions that include entering and leaving water temperature and entering air wet bulb temperature.

9.10.4 Cooling Tower Types

9.10.4.1 Direct Versus Indirect Evaporative Cooling Towers

The two basic types of evaporative cooling towers are:

- Direct-Contact Evaporative Cooling Tower
- Indirect-Contact Evaporative Cooling Tower

The direct-contact evaporative cooling tower is an open system that cools the hot water from the condenser by direct contact with the atmosphere. By exposing water directly to the cooling atmosphere, there is a transfer of heat directly to the air. Spray-filled cooling towers expose water to air without utilizing a heat-transfer medium. Spray-filled cooling towers are the most rudimentary method of exposing water to air. The amount of water surface exposed to the air is dependent on the efficiency of the sprays alone and the time of contact is a function of the elevation and pressure of the water distribution system.

To increase the contact surface as well as time of exposure, a heat-transfer medium called “fill” is installed below the water distribution system in the path of the air. Figure 9-3 illustrates a direct-contact evaporative cooling tower with fill. The two types of fill used in direct-contact evaporative cooling towers are splash type and film type. Splash-type fill maximizes contact area and time by causing the water to cascade through successive elevations of splash bars arranged in staggered rows. Film-type fill achieves the same effect by causing the water to flow in a thin layer over closed-spaced sheets that are arranged vertically.

An indirect-contact evaporative cooling tower on the other hand is a closed system where the hot water from the condenser is circulated through a water-to-air heat exchanger in the cooling tower. Figure 9-4 illustrates an indirect contact evaporative cooling tower. Heat from the condenser water is transferred to the atmosphere through the heat exchanger and the condenser water never comes in direct contact with the atmosphere. Indirect-contact towers require a closed-circuit heat exchanger that is usually a tubular serpentine coil bundles. The closed-circuit heat exchanger is exposed to air/water cascades similar to the fill of a cooling tower. Some indirect-contact evaporative cooling towers include supplemental film or splash fill sections to augment the external heat-exchange surface area.

9.10.4.2 Nonmechanical Draft Cooling Towers

Nonmechanical draft towers are aspirated by sprays or density differential, contain no fill, and utilize no mechanical fans for the movement of air. The aspirating effect of the water spray, either vertically or horizontally, induces airflow through the tower in a parallel-flow pattern. Without fans, both entering and leaving air velocities are relatively low for nonmechanical draft towers that make them susceptible to adverse wind effects. As a result, nonmechanical draft towers are normally used to satisfy a low cost requirement when operating temperatures are not critical to the system.

9.10.4.3 Chimney Cooling Towers

Chimney or hyperbolic towers are the giants of the cooling tower industry. These towers are used primarily for larger power plant installations but are included here for completeness. The heat transfer mode may be counter flow, cross flow, or parallel flow. Air is induced through the tower by the air density differentials that exist between the lighter heat-humidified chimney air and the outside atmosphere. When fills are used they are typically either splash- or film-type fills.

9.10.4.4 Mechanical Draft Cooling Towers

The fans on a mechanical draft cooling tower can be installed on either the inlet air side or exit air side. When fans are installed on the inlet air side of a mechanical draft cooling tower it is referred to as a forced draft cooling tower. Similarly, when fans are installed on the exit air side of a mechanical draft cooling tower it is referred to as an induced draft cooling tower. The fans used on mechanical draft cooling towers are typically either centrifugal- or propeller-type fans depending on external pressure needs, permissible sound levels, and energy usage requirements.

The relative direction of air and water through a mechanical draft cooling tower is used for categorization of the system. When water is flowing downward and air is flowing upward, the cooling tower is classified as having counterflow heat transfer. Crossflow or horizontal-flow heat transfer occurs when the water is flowing downward and the air is moving horizontally.
FIGURE 9-3 DIRECT-CONTACT EVAPORATIVE COOLING TOWER
FIGURE 9-4 INDIRECT-CONTACT EVAPORATIVE COOLING TOWER
through the cooling tower. All four combinations have been used in mechanical draft cooling towers of various sizes resulting in mechanical draft cooling towers that are:

- Forced-Draft Counterflow (Figure 9-5)
- Induced-Draft Counterflow (Figure 9-6)
- Forced-Draft Crossflow (Figure 9-7)
- Induced-Draft Crossflow (Figure 9-8)

Air can also be single entry and introduced through one side of tower or double entry where the air is introduced through two sides of the tower. A double entry induced draft crossflow mechanical draft cooling tower is illustrated in Figure 9-9.

Mechanical draft cooling towers are also classified as to whether they are factory assembled or field erected. Factory assembled mechanical draft cooling towers have the entire tower or a few large components factory assembled and shipped to the site for final installation. With field erected cooling towers the tower is completely constructed on site.

9.10.4.5 Closed-Circuit Mechanical Draft Cooling Towers

Counterflow- and crossflow- types are used in forced and induced fan arrangements for closed-circuit mechanical draft cooling towers. The tubular heat exchangers are typically serpentine bundles that are arranged for free-gravity internal drainage. Pumps are integrated by the manufacturer to transport water from the lower collection basin to the upper distribution basins or sprays. The internal coils can be fabricated from any of several materials but galvanized steel or copper is commonly used because they are capable of sustaining the required internal pressures.

Closed-circuit mechanical draft cooling towers are increasingly used for heat pump systems. These systems are typically multi-zone water source heat pumps that have intermediate water heat exchangers coupled to a closed water loop. The closed-circuit mechanical draft cooling tower is installed in this loop in series with a boiler so that heat may be either rejected or added to the system as required to maintain the temperature of the water loop within specified limits.

9.10.5 Cooling Tower Control

Cooling towers must be integrated into the overall central cooling plant control system because cooling towers are an integral part of the refrigeration cycle and the operation of the cooling tower must be matched to chiller operation if the central cooling plant is to operate optimally. Typically, the cooling tower, chiller, and condenser pump control must be considered if the overall plant is to be stable and energy-efficient. There are several types of packaged, mechanical-draft cooling towers but the counterflow induced-draft and forced-draft types are the most common for commercial air-conditioning applications. Both are controlled similarly depending on the manufacturer's recommendations. Variable frequency drives are often used to reduce fan power consumption at part-load conditions.

A bypass valve can be used to control condenser water temperature and conserve energy. With centrifugal chillers, the condenser supply water temperature is allowed to "float" as long as the temperature remains above a lower limit. The manufacturer should be consulted regarding the minimum entering condenser water temperature required for satisfactory chiller performance. Minimum condenser water temperatures for centrifugal chillers usually range from 55°F (12.8°C) to 65°F (18.3°C). In colder areas that require year-round air-conditioning, cooling towers may require sump heating to prevent freeze-up and continuous full flow over the tower to prevent ice formation. Then, the condenser water thermostat would control a hot water or steam valve in sequence with the bypass valve.

9.11 CONDENSER WATER SYSTEMS

9.11.1 Open System Overview

Condenser water systems for can be classified either as cooling tower systems or as once-through systems that use utility-supplied water, well water, or pond or lake water systems. Condenser water systems used in commercial air-conditioning systems are typically open systems meaning that air is continuously in contact with the water. Open condenser water systems require a different approach to pump selection and pipe sizing than do closed heating and cooling systems.

Some heat conservation systems rely on a split condenser heating system that includes a two-section condenser. Heat from one section of the condenser is used for heating in closed circuit systems and is occasionally interconnected with chilled water systems. The
FIGURE 9–5 FORCED-DRAFT COOLING TOWER WITH COUNTERFLOW

FIGURE 9–6 INDUCED-DRAFT COOLING TOWER WITH COUNTERFLOW
FIGURE 9–7 FORCED-DRAFT COOLING TOWER WITH CROSSFLOW

FIGURE 9–8 INDUCED-DRAFT COOLING TOWER WITH CROSSFLOW
FIGURE 9-9 DOUBLE-ENTRY INDUCED-DRAFT COOLING TOWER WITH CROSSFLOW
other section of the condenser serves as a heat-rejection circuit and is an open system that is connected to a cooling tower.

In selecting a pump for a condenser water system, consideration must be given to the static head as well as to the piping system friction loss in sizing the pump. Proper provision must be made to assure an adequate net positive suction head at the pump inlet. In addition, continuous contact with air in an open system introduces impurities that can result in scale and corrosion on a continuing basis. For this reason, aging of the piping that will result in an increased pressure drop over time can be anticipated. As a result, fouling factors must be included in the condenser design.

Depending on tower performance, cooling tower water is available at a temperature several degrees above the design wet bulb temperature. Where utility-supplied water, well water, or other open reservoir water such as a manmade pond, the maximum water temperature occurring during the operating season must be used for equipment selection. From manufacturer performance data and using known condenser water temperature, the required flow rate can be determined for any condensing temperature and capacity. A condensing temperature and corresponding flow rate may then be selected to produce the required capacity with a minimum of energy input and purchased water.

9.11.2 Once-Through Systems

A water-cooled condenser using utility-supplied water, well water, or water from a manmade or natural reservoir such as a pond is a once-through system. The return is run higher than the condenser so that the condenser is always full of water. Water flow through the condenser is modulated by a control valve in the supply line that is usually actuated from condenser head pressure to maintain a constant condensing temperature with variations in load. Utility-supplied water systems require check valves and open sight drains. When more than one condenser is used on the same circuit, individual control valves are used to avoid balance problems.

Piping should be sized in accordance with velocities of 5 (1.5 m/s) to 10 fps (3 m/s) for the flow rates. Where utility-supplied water is used, a pump is normally not required. For well or outside reservoir systems, pumps may be necessary.

9.11.3 Cooling Tower Systems

In a cooling tower system, water flows to the pump from the tower basin and is discharged under pressure to the condenser and back to the tower. Since it is usually desirable to maintain condenser water temperature above a predetermined minimum, water is diverted through a control valve to maintain minimum sump temperature. Piping from the tower sump to the pump requires some cautions. Sump level should be above the top of the pump casing to provide positive prime. Piping pressure drop should be minimized. All piping must pitch up either to the tower or the pump suction to eliminate air pockets. Suction strainers should be equipped with inlet and outlet gauges to indicate when cleaning is required.

Vortexing in the tower basin is prevented by piping connections at the tower in accordance with manufacturer's specification and by limiting flow to the maximum allowed by the sump design. A straight section of suction pipe five times the diameter in length contributes to achieving expected pump performance.

Since there is an equal head of water between the level in the tower sump and the pump on both the suction and discharge sides, these heads cancel each other and may be disregarded. The elements of pump head are:

- Static head from tower sump to the tower header.
- Friction loss in suction and discharge piping.
- Pressure loss in condenser.
- Control valves.
- Strainer and tower nozzles if used.

These elements added together determine the required pump total dynamic head. Normally, piping is sized to yield water velocities between 5 (1.5 m/s) and 12 fps (3.7 m/s). Friction factors for 15-year-old pipe are commonly used. The pressure drops for condenser, cooling tower, control valves, and strainers are obtained from manufacturer data. If condensers are installed in parallel, only the one with the highest pressure drop is counted. Combination flow measuring and balancing valves can be used to equalize pressure drops.

If multiple cooling towers are to be connected, the piping should be designed so that the pressure loss from the tower to the pump suction is approximately equal for each tower. Large equalizing lines or a common
reservoir are used to maintain the same water level in each tower.

Evaporation in a cooling tower results in a concentration of dissolved solids in the circulating water. This concentration is held within limits by wasting a portion of the water as overflow or blow down. The drift loss that results from droplets of water drifting out of the tower helps to maintain a limited concentration.

Special precautions are required if cooling towers are to be operated in subfreezing weather. Ice formation can hinder the performance of the tower by obstructing air flow and periodic manual shutdown of the tower fan may be necessary for de-icing. Longer operating periods in cold weather can be achieved if the tower fan is thermostatically controlled with the bypass so that gravity airflow is achieved and larger water quantities are circulated through the tower.

9.12 EVAPORATIVE COOLING SYSTEMS

9.12.1 Indirect Evaporative Cooling

9.12.1.1 Description And Operation

An indirect or dry air evaporative cooler-exchanger is designed with two air passages. A dry one through the interior of the heat exchange tubes and a wet one over the exterior of the heat exchange tubes. Water is circulated over the exterior of these heat exchange tubes while air is moved over them creating an evaporative cooling effect that reduces the temperature of the tubes. At the same time, air is moved through the dry air passage in the interior of the cooled tubes releasing its heat to the cooled tube surfaces and creating cool air that has not had moisture added to it. This is the cooled, dry-side air stream.

Certain types of fixed plate heat exchangers can also be used as dry air evaporative coolers by spraying a mist of water vapor into the exhaust air stream prior to entering the heat exchanger. The mist deposits on the exhaust side surfaces and is evaporated which lowers the temperature of the plates to approximately the exhaust air stream wet bulb temperature.

The major use of the dry air evaporative cooler is to pre-cool the outside air stream to reduce the load of the air conditioning system. The cooled, dry-side air may also be used as an exclusive cooling source in all applications where design conditions permit. The temperature range is in the range of approximately 35°F (1.7°C) to a maximum of 125°F (51.7°C).

9.12.1.2 Dry Air Cooler Exchangers

In order to obtain the maximum cooling capability from the dry air cooler, the system design should provide that the building exhaust air be ducted to supply the air source for the wet-side air passage. This pre-conditioned exhaust air can then serve a useful purpose prior to wasting its energy into the atmosphere. It provides a useable low wet bulb temperature in the cooling season and increases the capacity of the unit for cooling. In lieu of building exhaust air, outside ambient air may be used as the wet-side air source.

The wet-side air is cooled as it passes over the outside of the tubes and becomes relatively saturated. Using this air to pre-cool air conditioning system condensers when near saturated air is satisfactory can provide additional energy savings.

During the cooling operation of the unit, the dry bulb temperature leaving the wet-side will be depressed. The depression will vary somewhat depending on dry-side air temperature and the evaporation rate. This depression can be expressed as approach to wet bulb temperature. The unit, based on the minimum recommended wet-side air, will achieve an approximate 10°F (5.6°C) dry bulb approach. When building, exhaust air is provided as the wet-side air source, the maximum cooling capacity is obtained.

During the heating season, the unit acts as a plate type heat exchanger as the outdoor air is pre-heated by recovering the heat of the exhausted building air. De-energizing the unit’s circulating pump and allowing the knitted tube sleeves to dry out accomplish this heat recovery. Automatic controls can energize and de-energize the water pump based on the entering air temperature of the outdoor air stream.

Well over 90 percent of most cooling system operating hours are at partial load conditions. It is important to evaluate the capability of dry air evaporative cooling units at these partial load conditions. The cooling work performed as the wet-bulb temperature decreases. As the outside dry bulb temperature decreases, the outside wet bulb temperature usually decreases also. As a result, units will provide a larger portion of the total HVAC system load than during full load operation. As wet bulb temperatures decline, the percentage increases until a point is reached where the HVAC system cooling load and outside wet bulb temperatures can allow for the total cooling to be provided by the dry air evaporative cooler.

9.12.1.3 Fixed Plate Exchangers

Fixed plate or sensible energy recovery devices can also use dry evaporative cooling. Normal evaporative
cooling processes reduce the supply air temperature while increasing the moisture content and are adiabatic. Indirect evaporative cooling reduces the enthalpy level of the supply air and under certain conditions can condense moisture out of the supply air stream. This is accomplished by spraying a mist of water vapor into the exhaust air stream prior to the fixed plate heat exchanger where it is deposited on the exhaust side surface and is evaporated. The temperature of the plates is lowered to approximately the exhaust wet bulb temperature. The outside air entering the heat exchanger is exposed to the cooled plates and can approach within approximately 25 percent of the difference between the outside dry bulb and exhaust side wet bulb temperatures. The effectiveness of recovery can be in the 50 to 65 percent range as compared to 25 to 40 percent for straight sensible recovery.

9.12.1.4 Filters

Air filtration of the outside stream on the dry side is normally required. In very dirty areas it may be advisable to consider infiltration of the wet-side air stream also. Polyester media filters are recommended because they are not damaged by dampness resulting from rain or fog. High capacity filters will provide a long life prior to change out. Select filters at a face velocity of approximately 400 fpm (122 m/s).

9.12.1.5 Condensation And Freeze Up

In areas where temperatures consistently drop below freezing, it is advisable to consider freeze protection of the dry air cooler unit pump and basin similar to what would be done for cooling towers or evaporative condensers. Typical methods include the automatic or manual draining of the water pan, the use of electric pan heaters, or incorporation of steam or hot water coils. It is recommended that a 40°F (4.4°C) basin water temperature be maintained.

9.12.1.6 Controls

Cooling is achieved by energizing the water circulating pump and wet-side fan if it is used. A temperature sensing control element is normally located in the entering dry-side outside air stream to control the pump and wet-side fan. Heat recovery from the building exhaust air is achieved by thermostatically turning off the water pump. When exhaust air temperatures are higher than entering outdoor air temperatures, automatic heat recovery operation is obtained by deenergizing the pump. Many other varieties of control operation may be applied such as face and bypass dampers.

9.12.1.7 Water Treatment

Water treatment is normally not required. However, installation of a dry air evaporative cooling unit in some areas may require a periodic or continuous water treatment program to obtain maximum performance. Algae growth can be non-existent in some areas and quite severe in other areas. Periodic treatment can control algae. In areas where the make-up water has a high percentage of dissolved solids, a program to prevent scaling should also be considered. Mineral scale deposits in the unit should be minimal due to the small temperature difference between the tube and water.

9.12.2 Indirect/Direct Evaporative Cooling

9.12.2.1 Description And Operation

The growing need for energy efficient cooling systems resulted in the development of indirect/direct evaporative cooling that cools air by evaporation with very high-energy efficiencies. These systems are two-stage evaporative systems employing efficient indirect evaporative or sensible cooling with a reduction in total heat content using direct evaporative cooling. In some cases a third or standby stage that includes refrigeration is also included in the process. These systems are typically used in the western United States and normally provide annual cooling at a fraction of the energy required for conventional compression cooling systems.

Indirect/direct evaporative cooling is a system concept that uses indirect evaporative cooling in the first stage of cooling system air and direct adiabatic cooling as the second stage of cooling air. Many systems may also include a third stage of conventional refrigerated cooling using a chilled water or a direct expansion (DX) coil due to higher design ambient wet bulb conditions. This third stage may also be needed to reduce the supply air quantity required to maintain room conditions.

9.12.2.2 Transfer Process

The first stage of indirect/direct evaporative cooling uses indirect evaporative cooling and the second stage use direct evaporative cooling. This is accomplished with a fill of air washer device that provides a very efficient approach of about 90 percent plus to total saturation. Conventional evaporative or swamp coolers will not achieve this. While on the cooling cycle, the system uses 100 percent outside air similar to an economizer cycle. The capability of the system to offset space heat gain is based on leaving supply air temperature
and quantity as with a conventional air conditioning system.

Three factors affect the supply air temperature:

- The dry bulb temperature of air entering the indirect evaporative cooler. At a lower dry bulb temperature, less adiabatic cooling is required to achieve a given supply air temperature.
- The wet bulb temperature of air entering the secondary side of the indirect evaporative cooler. At a given outside air dry bulb temperature, a decrease in wet bulb temperature achieves a greater wet bulb depression and lower supply air temperature.
- The efficiency of the indirect and direct evaporative coolers. The indirect evaporative cooler efficiency is measured as a percent of wet bulb depression which is outside air dry bulb less wet bulb temperature. Most design conditions will require minimum sixty percent efficiency. Therefore, at conditions of 100°F (37.8°C) dry bulb and 70°F (21.1°C) wet bulb, the wet bulb depression is 30°F (-1.1°C). Sixty percent efficiency will provide a temperature drop of 18°F (7.8°C) or a leaving system air temperature from the indirect cooler of 82°F (27.8°C). Selecting the indirect cooler at a higher efficiency condition will produce a lower leaving supply air temperature.

The second stage of cooling, provided by the direct evaporative cooler, will cool the first stage air to the saturation efficiency of the direct cooler. This process is adiabatic and will follow the wet bulb line of the psychrometric chart. If the entering conditions to the direct cooler are 82°F (27.8°C) dry bulb and 64°F (17.8°C) wet bulb, the saturation point is 64°F (17.8°C). As a result, a 90 percent efficient direct cooler will achieve a temperature drop of 16.2°F (8.8°C) or a leaving air temperature of 65.8°F (18.8°C). Seventy percent efficiency will achieve a temperature drop of 12.6°F (10.8°C) or a leaving air temperature of 69.4°F (20.8°C). This decrease in efficiency is detrimental to system performance.

In all cases, the relative humidity of the leaving air is close to or above 90 percent relative humidity. This leaving air condition, in relation to saturation, is similar to leaving air conditions from conventional chilled water or direct-expansion coils. The fact that the air is relatively saturated is not detrimental to achieving satisfactory relative humidity in a zone.

9.12.2.3 Controls

The control of a simple one-zone heating and cooling system may be a single two-stage cooling and one-stage heating thermostat. The first stage cooling closes the return air damper, opens the outside air damper, and energizes the indirect cooler pump and secondary fan. The second stage cooling energizes the direct cooler pump. The heating stage will reverse the outside air and return air damper settings and energize the heating section.

VAV system control should be considered because this can be an excellent means of control as well as a way to reduce the supply air fan horsepower required during partial loads due to the reduction in supply air quantity and room cooling load. As supply air volume reduces, the indirect and direct cooler efficiency increases and the applicable pressure drop to the supply air fan decreases.

Where VAV system control is not applied, constant air volume and supply air temperature control can be applied. The system control should be set up for two-stage supply air temperature control with the indirect sensible cooling as stage number one and the direct cooler as stage number two. Energizing the indirect cooler pump and secondary fan on a rise in supply air temperature will control the indirect cooler. Energizing the direct cooler pump on a rise in supply air temperature will control the direct cooler. Many of the hours of cooling operation can be satisfied by the operation of the indirect cooler alone.

When using the refrigerated third stage, a three-stage thermostat control can be used. Once again based on achieving a selected system supply air temperature, the indirect unit will be controlled as stage one and the direct evaporative unit will be controlled as stage two.
CHAPTER 10

CENTRAL HEATING PLANT
10.1 INTRODUCTION

Central heating plant refers to centrally located heating equipment that increases the temperature of a heat transfer medium such as air or water for distribution throughout the entire building or to a designated portion of the building. The heat transfer media most often used for commercial or institutional buildings is either air or water. If the heat transfer medium is air, the heated air is distributed throughout the building by an air distribution system as discussed in Chapter 12. Similarly, if water is used as the heat transfer medium the hot water or steam is distributed using a hydronic distribution system covered in Chapter 15. Once delivered to the heating zone being served, the hot air mixes with the existing air to raise the overall temperature of the zone. Similarly, hot water and steam heat the space using convection terminal units that are also covered in Chapter 15 and are essentially water- or steam-to-air heat exchangers.

This chapter focuses on the equipment that comprises a central heating plant. The key element of any central heating plant is either a furnace or boiler. Furnaces use air as their heat transfer medium and will be covered first. Boilers use water as their heat transfer medium and boiler operation, construction, and associated equipment will be discussed following furnaces.

10.2 FURNACES

Furnaces are used in residential and small commercial buildings for heating. A furnace is essentially an all-air heating system that uses air as its primary heat transport medium. A furnace is a central heating plant that heats incoming air through combustion and then delivers the heated air to the heating zones served via ducts. Central heating plants that use a furnace for heat normally include a fan to pull return air from the zones served, push air through the furnace where it is heated, and then through the air distribution system. Air is delivered to the space and removed from the space via registers and grilles that are discussed in Chapter 12 along with ducts and other air distribution equipment.

The most common combustion fuel source for furnaces is natural gas although other combustion fuel sources include liquefied petroleum (LP) gas, fuel oil, coal, wood, or wood pellets. Furnaces can be either single- or multi-fuel. With multi-fuel furnaces, the most economical or available fuel is typically used as the primary fuel source and the secondary fuel is used as a backup in the event that the primary fuel is unavailable. For example, a building located in a rural area where wood is readily available and inexpensive might use wood as the primary fuel but also use LP gas as the backup fuel source. Furnaces using combustion as their heat source always require venting to exhaust combustion gases and avoid the buildup of gases such as carbon monoxide in the building.

Electric furnaces are also available and are used either where electricity is less expensive than available combustion fuels, or when combustion fuels are not available. Electric furnaces also do not have the issue of byproducts of combustion entering air stream and contaminating the zone served. Electric furnaces heat the air passing through them using electric elements. These electric elements are usually just simple electric resistance heating coils but could be other materials that heat up when electric current passes through them. As a result, electric furnaces do not rely on combustion to heat the air and do not produce gases that require venting. This feature makes electric furnaces a good choice where additional heat is needed for an interior building zone or space and the routing of gas piping through the building to a conventional natural gas furnace along with the need to exhaust combustion gases is impractical.

Furnaces, like boilers, can be classified as either condensing or non-condensing. Condensing furnaces are more efficient than non-condensing or traditional furnaces. A condensing furnace extracts heat from flue gases before they are exhausted and uses this heat to improve the efficiency of the combustion process. What is waste heat in a conventional furnace is used in a condensing furnace to improve its efficiency. However, condensing furnaces must be designed to withstand the highly corrosive condensate that results from the cooling of exhaust combustion gases. Therefore, the first cost of a condensing furnace is higher and the decision to use a condensing furnace over a traditional or non-condensing furnace should be based on a life-cycle cost analysis.

10.3 BOILERS

Where furnaces use air as their primary heat transfer medium; central heating plants with boilers use water as their primary heat transfer medium. Boiler-based central heating plants are used almost exclusively in larger commercial and institutional buildings over furnaces because hot water or steam is easier and more economical to distribute than air. Hydronic piping takes up less room in a building than ductwork, has less heat loss than duct and can be routed through the building much easier. As a result, central heating plants use boilers and hydronic distribution systems in all but the smallest of buildings or where the function of larger commercial buildings requires them to be broken up
into small, independently-heated zones as in the case of a high-rise condominium or apartment building.

The purpose of a boiler is to transfer heat from a heat source to water by conduction that can then be circulated through a building to provide heat. This is a hydronic heating system because heated water either as liquid or vapor is the primary heat transport medium. Water in vapor form is usually referred to as steam. Steam produced by central heating plants for the purpose of heating commercial and institutional buildings is typically low-pressure and low-temperature steam. This is opposed to steam generated by boilers for manufacturing and process purposes that is both high temperature and high pressure. An example of a high-pressure, high-temperature steam process application is the use of super-heated steam to drive a steam turbine in a coal-fired or nuclear power plant.

Steam used in commercial or institutional building heating is exclusively low-temperature, low-pressure steam generation and distribution. High-pressure and high-temperature steam is rarely used to heat commercial and institutional buildings with the exception of central heating plants that serve multiple buildings. This situation can occur on an educational, office building, medical, or similar campus or with a district heating system where either a municipal or private utility distributes steam to a number of commercial and institutional buildings from a central location using an underground steam distribution system similar to electric power distribution. However, even when the steam is generated and distributed at high pressures and temperatures, it is converted to low-pressure, low-temperature steam or hot water when it enters the building using a heat exchanger. This heat exchanger operates much like a conventional fossil-fueled boiler except that it uses the steam supply instead of combustion to raise the temperature of the circulating hot water or convert it to low-pressure, low-temperature steam. Distributing low-pressure, low-temperature steam in a commercial or institutional building requires a much simpler and less expensive hydronic distribution system. It is safer, cheaper to operate and maintain than a high-pressure, high-temperature steam distribution system.

10.4 HOT WATER VERSUS STEAM

Boilers can produce either hot water or steam. Hot water is the most common heat transfer medium in commercial and institutional central heating plants today. Boilers used to generate hot water for these systems are referred to as hot water boilers. When the boiler produces steam, it is referred to as a steam boiler. Even though hot water generation and distribution is commonly used in central heating applications today, there are situations when steam generation and distribution in modern buildings is the better choice as will be discussed in this section.

In general, hot water boilers are more efficient than steam boilers because they operate at lower temperatures and they are able to operate at higher fuel conversion efficiencies than steam boilers. Steam boilers typically operate in an efficiency range of between 75 and 83 percent and hot water boilers usually operate in an efficiency range of between 80 and 93 percent. Steam is supplied at higher temperatures than hot water resulting in greater heat loss through the boiler shell and from the steam distribution system piping further lowering steam central heating-plant efficiency.

Even though hot water central heating plants are normally more efficient than steam systems, steam does have advantages in some commercial and institutional building heating applications. In most cases, steam is selected over hot water when there is a need for large amounts of heat to be delivered quickly as in the case of a building that requires 100 percent outside air. Similarly, steam is also the better choice for multi-building complexes that have a heating plant that serves the entire complex, for buildings that cover a very large area, or for tall buildings where the central heating plant is in the basement or lower floor. The larger the heating load and the further it is from the central heating plant, the more attractive steam is because of its higher energy density. Steam is also better able to handle fluctuating building loads than hot water. However, steam generation and distribution systems require more maintenance and are generally more costly than hot water systems both on a first cost and ongoing basis. Steam heating systems should be avoided for commercial and institutional buildings unless there are other overriding reasons to select a steam system.

10.5 BOILER PLANT OPERATION

A boiler is a heat exchanger in a cast-iron or steel pressure vessel. It is designed to efficiently transfer heat supplied from the combustion of carbon-based fuels, electricity, or other heat sources such as solar or geothermal to circulating water in order to raise the temperature of the water or convert it to steam. The hot water or steam is then circulated through a hydronic distribution system and warms the zones that it supplies by transferring its heat to the surrounding air through water-to-air or steam-to-air heat exchangers.

In a hydronic-to-air heating system, the heat exchanger is located in the air stream of the zone to be
heated and the air is heated when it passes through the hydronic-to-air heat exchanger. The primary mode of heat transfer for a hydronic-to-air heating system is convection. As will be discussed in Chapter 15, this heat transfer can either result from natural convection where air circulates through the terminal unit as the result of natural buoyancy associated with heated air or forced convection where a mechanical fan or other device forces air through the heat exchanger. An example of a natural convection terminal unit would be a simple hot water or steam radiator. Similarly, an air-handling unit where the air is forced through the heating coil by a fan would be an example of forced convection. As the hot water or steam circulates through the building providing heat, it gives up energy and cools. The water, steam, or combination of the two is then returned to the boiler for reheating by the hydronic distribution system and the cycle starts again.

In addition to heating zones by convection, hot water can also be circulated through hydronic piping systems embedded in the surfaces of the zone or space being heated. These surfaces could be the floor, walls, ceiling, or combination of surfaces. By running hot water through these surfaces, heat is transferred to the surface resulting in radiant heating.

### 10.6 BOILER CONSTRUCTION

#### 10.6.1 Boiler Classification

Boilers are classified by construction types. The construction types depend on how the water is circulated and heated inside the boiler. The classifications are:

- Fire Tube
- Water Tube

##### 10.6.1.1 Fire Tube Boilers

Fire-tube boilers are typically constructed with a cylindrical outer shell and a number of tubes passing through the outer shell. With a fire-tube boiler, hot gases from the combustion of fuel are passed through the tubes that are surrounded by water in the outer shell. The water surrounding the fire tubes is often referred to as the water jacket. Heat is transferred from the tubes to the water that is then circulated through the building as either hot water or steam.

Fire-tube boilers are also classified by the number of times that the combustion gases pass through the boiler heating the water before being exhausted. The number of times that the hot gases pass through the boiler before being exhausted is referred to as the number of passes through the boiler. Heat naturally flows from high temperature to low temperature but the rate at which heat is transferred is determined by a variety of variables such as the surface area in which the two fluids are in contact as well as the amount of time that the two fluids are in contact. Multiple passes through the boiler increases both the surface area that the hot combustion gases are in contact with the water through the fire tube and the time over which the two fluids are in contact. Multiple passes increase both the amount of heat transferred and the efficiency of the boiler because more of the energy supplied by combustion is converted into increased water or steam energy.

At the end of their pass through the boiler, the fire tubes make a 180-degree turn and pass back through the boiler releasing additional energy and making the boiler more efficient. The turnaround can either be dry-back where the 180-degree turn is made outside the boiler shell or water-back where the 180-degree turn is made within the boiler shell. As steam generators, fire-tube boilers are limited to the amount of pressure that they can produce because the water is located in the cylindrical shell that has a large surface area.

##### 10.6.1.2 Water Tube Boilers

Water tube boilers operate just the opposite of fire-tube boilers. With water tube boilers, the water passes through the boiler tubes and the hot combustion gases pass over the boiler tubes in the outer shell of the boiler. With their smaller surface area, the water tubes can be built to withstand higher pressures than the shell in a fire-tube boiler. As a result, water-tube boilers are typically used where steam is required. Water-tube boilers are usually used in manufacturing and industrial process applications where high-pressure steam is required such as in electric power generation. Water-tube boilers are seldom used in commercial or institutional central heating plants.

#### 10.6.2 Boiler Materials and Construction

Most boilers, other than special or unusual models, are made from cast iron or steel. Some boilers in the smaller sizes are made of copper or copper-clad steel.

Cast-iron boilers are constructed of individually cast sections, assembled into blocks of sections using push or screw nipples, gaskets, or an external header to join the sections pressure tight and provide passages for the water, steam, and products of combustion. The number of sections assembled determines boiler size and energy rating. Sections may be vertical or horizontal with the vertical design being the most common. The boiler may be dry-base where the firebox is beneath
the fluid-backed sections, wet-leg where the firebox top and sides are enclosed by fluid-backed sections, or wet-base where fluid-backed sections surround the firebox.

Steel boilers are fabricated into one assembly of a given size and rating by welding the sections together. The heat exchange surface past the firebox usually is an assembly of vertical, horizontal, or slanted tubes. The tubes may be either fire tube with combustion gases inside the tubes and the heated fluid outside or water tube with the fluid to be heated inside and combustion gases outside. As with cast-iron boilers, dry-base, wet-leg, or wet-base design may be used. Most small steel heating boilers are of dry-base, vertical fire tube design. Larger boilers usually have horizontal or slanted tubes and both fire tube and water tube designs can be used.

10.6.3 Scotch Marine Boilers

Scotch marine boilers are the most common fire-tube boiler because of their low initial cost and high efficiency and durability. These boilers are characterized by a central, fluid-backed cylindrical firebox that is surrounded by fire tubes in one or more passes with everything contained in a single outer shell. Scotch marine boilers typically contain a large volume of water that allows them to respond to load changes in the building.

10.7 BOILER FUELS

Like furnaces, boilers can be designed to operate on a variety of fuels. Just about anything that can be burned to produce heat can be used as a boiler fuel. Boilers in commercial and institutional central heating plants almost exclusively burn natural gas today due to its availability in most urban areas, reliable supply, relatively clean combustion, and low cost. However, boilers can also be designed to burn coal, oil, LP gas, wood, waste materials and trash, among other fuels. Further, like furnaces, boilers can be designed to be multi-fueled or adapted after installation to burn other carbon-based fuels. The selection of a boiler fuel should be based on fuel availability, economics, environmental considerations, local codes, and regulations.

10.8 ELECTRIC BOILERS

Electric boilers, like electric furnaces, use electric heating elements to heat or boil the water to heat the building. The heating surface is the surface of the electric elements or electrodes immersed in the boiler water. These heating elements can either be resistance or immersion-type elements that can produce hot water or steam depending on the type of boiler used. The design of electric boilers is largely determined by the shape and heat release rate of the electric heating elements used. By their nature, electric boilers are effectively 100 percent efficient.

Electric boilers do not need to be vented since they do not produce any combustion gases (such as carbon monoxide) that need to be exhausted. Electric boilers may be the right choice where hot water or steam is required in an existing facility but there is no way to vent the combustion gases or in locations where electric rates are low. Also, electric boilers may be used in conjunction with boilers using natural gas and other fuels where the cost of off-peak electricity or an interruptible electric rate makes the electric boiler the more economical source of hot water or steam at night or other off-peak times.

10.9 BOILER RATING

Boilers are rated based on their heat generating capacity. Smaller boilers are usually rated in British thermal units (Btu) per hour and larger boilers are typically rated in boiler horsepower. Since Btu is a unit of energy as discussed in Chapter 2, the rate of energy usage in Btu per hour is power and can be related to horsepower. One boiler horsepower is equivalent to 33,475 Btu per hour. Table 10-1 provides typical boiler size ranges for various types of facilities.

<table>
<thead>
<tr>
<th>Facility Type</th>
<th>Heat Rate (Btu/hr)</th>
<th>Horsepower (bhp)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Small Commercial or Residential</td>
<td>67,000 – 3,400,000</td>
<td>2 – 100</td>
</tr>
<tr>
<td>Medium Commercial and Small Industrial</td>
<td>2,500,000 – 10,000,000</td>
<td>100 – 300</td>
</tr>
<tr>
<td>Large Commercial and Large Industrial</td>
<td>10,000,000 – 33,500,000</td>
<td>300 – 1000</td>
</tr>
</tbody>
</table>

Table 10–1 Boiler Rating by Facility Type
Low-pressure boilers are constructed for maximum working pressures of 15 psi steam and up to 160 psi hot water. Hot water boilers are limited to 250°F (121°C) operating temperature. The controls and relief valves that limit temperature and pressure are usually not part of the boiler and must be installed to protect the boiler. Medium- and high-pressure boilers are designed to operate above 15 psi for steam boilers or above 160 psi for water boilers.

Steam boilers are available in standard sizes of up to 50,000 lbs of steam per hour or about 60 MBtu/h to 50,000 MBtu/h. As discussed previously, steam boilers can be used for space heating in both new and existing commercial and institutional facilities. On larger installations, steam boilers can also provide steam for auxiliary uses such as supplying the facility’s domestic hot water using a steam-to-water heat exchanger, chilled water using an absorption chiller as discussed in Chapter 9, or steam for laundries, sterilizers, and similar applications.

Hot water boilers are available in standard sizes of up to 50,000 MBtu/h. Most hot water boilers are in the low-pressure class and are used for space heating in both new and existing facilities. Hot water boilers can also be equipped with either internal or external heat exchangers to supply domestic hot water to the facility.

Every steam or water boiler is rated at the maximum working pressure and it must be equipped with safety controls and pressure-relief devices to ensure safe operation.

10.10 BOILER CONTROLS

10.10.1 Boiler Control Methods

Boiler controls are designed to control either the on/off operation or the rate of fuel input in response to some control signal. The objective of boiler control is to have the average boiler output equal heating load within some acceptable control tolerance. Boiler controls also include safety controls that act to shut off fuel flow when unsafe conditions develop. Boiler operation can be controlled in the following three ways:

- On-Off Control
- Low-High-Low Control
- Modulating Control

10.10.1 On-Off Control

On-off control is basically a binary control where the boiler cycles on and off depending on the temperature of the water in the boiler. This is the least expensive first cost control system but is also the least efficient control system over the life of the installation. On-off control is the least efficient because of the losses associated with each cycle. Also, oversized boilers will result in frequent cycling further reducing efficiency.

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10.10.1.2 Low-High-Low Control

Low-high-low control is more efficient and effective than just a simple on-off control by cycling between a high and low burner setting to maintain the hot water temperature and meet the building's changing heating load.

10.10.1.3 Modulating Control

Modulating control is the most efficient boiler control method of the three and best at meeting the building's changing heating load. With modulating control, the burner is adjusted or modulated on a continuous basis to meet the building’s heating load. At low fire conditions the flow of combustion gases through the boiler is less and more of the heat from these combustion gases can be captured and used.

10.10.2 Hot Water Boiler Control

Hot water boilers typically utilize burner control based on the temperature of the water. The control is set to maintain a 140° to 180°F (60 to 82°C) water temperature at the boilers output. Hot-water boilers are operated by temperature-actuated controls that are usually mounted on the boiler. These controls are very similar to those used for steam boilers and are typically either on-off, high-low-off, or modulating.

Due to the need to shutdown excess boiler capacity in the central heating plant for extended periods at night and during weekends for energy conservation, the water in shutdown boilers may cool well below the normal operating temperature for both steam and hot-water boilers. In these cases, a low-fire start thermostat control is recommended. The low-fire start thermostat control senses the water temperature in the shutdown boiler and keeps the firing control at low fire until a minimum temperature of 176° to 194°F (80 to 90°C) is reached. The purpose is to heat the water in the boiler slowly in order to minimize thermal stresses in the boiler.

It is important to know that anytime during startup or in normal operation when the water temperature in a
steam boiler or water boiler is below the dew point of the flue-gas, condensation of flue gas will occur on the fire side of the boiler. This condensate is corrosive to most boiler surfaces. Water boiler controls should not be set lower than the dew point unless the boiler is constructed using materials that are resistant to this corrosion. The boiler manufacturer should be consulted if low operating temperatures or frequent cold starts are anticipated.

10.10.3 Steam Boiler Control

Like hot-water boilers, steam boilers are operated by boiler-mounted, pressure-actuated controls that vary the input of fuel to the boiler. Steam boilers are typically controlled based on a pressure set point that is typically 5 to 10 psi. Common examples of controls are on-off, high-low-off, and modulating. Modulating controls can continuously vary the fuel input from 100 percent down to a selected minimum point. The ratio of maximum to minimum is called the turn-down ratio. The minimum is usually between 5 percent and 25 percent or alternately between 20 to 1 and 4 to 1 depending on the size and type of fuel-burning apparatus.

10.11 BOILER FEEDWATER AND CIRCULATING PUMPS

Feedwater and circulating pumps in central boiler plants circulate hot water throughout the building by way of the hydronic piping system and return it to the boiler for reheating. Hydronic pumps and their characteristics are covered in Chapter 16.

Typically, the hydronic piping system will be designed and installed with a primary and secondary loop that requires a circulating water pump for both loops. It is common for the primary loop pump to circulate hot water through the boiler continuously while the secondary loop pump circulates the hot water throughout the portion of the building that it serves. Multi-loop hydronic systems are discussed and illustrated in Chapter 15 with hydronic distribution systems.

10.12 DEAERATORS

A deaerator is a device that removes corrosive gasses from boiler feedwater to make it noncorrosive. Oxygen, carbon dioxide, and other dissolved gases can be very corrosive and need to be removed from boiler feedwater in order to prevent damage to the boiler. Oxygen is the most corrosive gas and it can be introduced into the boiler feedwater through the addition of makeup water or leakage in the return hydronic system. Dissolved gases along with suspended solids should be removed from boiler feedwater to avoid corrosion in the boiler. Deaeration can be accomplished by either mechanical or chemical means. Dissolved gases including oxygen can be removed chemically by water treatment but are most often removed using a mechanical deaerator.

10.13 INCREASING CONVENTIONAL BOILER PLANT EFFICIENCY

10.13.1 Methods Of Increasing Conventional Boiler Efficiency

There are a number of ways that the efficiency of a conventional boiler plant can be increased. They include the following:

- Jacket Insulation
- Recuperator Or Air Preheater
- Turbulators
- Blowdown Heat Recovery
- Stack Gas Economizer
- Combustion Control
- Supply Water Temperature Reset
- Feedwater and Circulating Pump Speed Control

10.13.1.1 Jacket Insulation

Insulating the boiler jacket is an easy and effective means to improve boiler efficiency. Jacket insulation will reduce radiation losses from the boiler shell.

10.13.1.2 Recuperator Or Air Preheater.

The use of a recuperator or air preheater increases the efficiency of a boiler by transferring some of the heat from the exhaust combustion gases to the intake combustion air. The preheated air will result in improved combustion conditions. However, care must be taken to avoid condensation of the exhaust gases and possible corrosion resulting from this condensation. Also, increased combustion air temperatures can result in higher nitrous oxide (NOx) emissions.

10.13.1.3 Turbulators

Turbulators increase the heat transfer of combustion gases when they are inserted in the fire tubes of fire-tube boilers. Turbulators are baffles that cause tur-
bullence in the hot gas stream that in turn increases the convective heat transfer to the surface of the fire tube. By increasing the heat transfer, boiler efficiency is increased.

10.13.1.4 Blowdown Heat Recovery

Blowdown heat recovery extracts heat energy from the blowdown water and uses the recovered energy to improve the efficiency of the boiler. Blowdown is a process that is done to remove boiler water impurities that can cause scale buildup in boiler tubes.

10.13.1.5 Stack Gas Economizer

A stack gas economizer can be used on a conventional boiler to capture combustion gas heat that would normally be exhausted with the combustion gases and wasted. A stack gas economizer is essentially a heat exchanger that absorbs heat from exhaust gases and transfers it to the return water entering the boiler. Economizers usually have a lower first cost than recuperators and don’t impact boiler emissions so they are usually preferred over recuperators.

Preheating the boiler feedwater results in reduced fuel usage and improved boiler efficiency. Stack gas economizers are particularly effective where boilers require a large amount of make-up water. The possible savings associated with the use of a stack gas economizer will be affected by the amount of make-up water needed as well as the hours of operation. As with recuperators, condensation of the exhaust gases needs to be avoided in order to prevent corrosion.

10.13.1.6 Combustion Control

With flue gas monitoring, the flow of combustion air into the boiler can be controlled. Optimizing combustion air results in more complete combustion of the fuel and reduced nitrous oxide (NOₓ) emissions. Optimizing combustion oxygen results in hotter combustion gases, greater heat transfer, and increased combustion efficiency that in turn produces an improvement in overall boiler efficiency.

10.13.1.7 Supply Water Temperature Reset

Another method of improving the overall efficiency of a boiler is supply water temperature reset. Supply water temperature reset the temperature of the hot water based on outside conditions. Supply water temperature reset improves the overall efficiency of the boiler system, provides better temperature control, and reduces the possibility of overheating, among other advantages. A multi-loop hydronic distribution system can be divided into decoupled, primary and secondary loops making it possible for the temperature in the secondary loop to be reset while maintaining the required return water temperature in the primary loop for the boiler.

10.13.1.8 Feedwater and Circulating Pump Speed Control

Feedwater and circulating pumps should use energy efficient motors and variable frequency drives (VFDs), where appropriate. Feedwater and circulating pumps with VFDs can increase system efficiency in the same way that VFDs increase the efficiency of fans and air distribution systems. Energy efficient motors and VFDs for pumps are covered in Chapter 17.

10.14 CONDENSING BOILERS

Conventional boilers have an efficiency of better than 80 percent whereas energy-efficient condensing boilers have a rated efficiency above 90 percent. Condensing boilers are more efficient because they absorb more heat from the combustion gases using very high efficiency heat exchangers. However, extracting this much heat from the combustion exhaust stream causes the water and other gases contained in the exhaust combustion gases to condense into a very corrosive fluid.

Conventional boilers are constructed using materials that cannot withstand the corrosive affects of the condensing flue or stack gases as discussed in the previous section. Any hydrocarbon fuel that is burned in the boiler including natural gas produces water vapor during the combustion process. The difference between a conventional boiler and a high-efficiency condensing boiler is the ability of the condensing boiler to withstand the corrosive affects of the flue gas condensate through their design and the corrosive-resistant materials used to construct them.

Like condensing furnaces, condensing boilers have a higher first cost because they have larger heat transfer surfaces and are constructed of materials that need to be able to withstand the corrosive flue gas condensate in order to achieve the higher efficiencies. The increased energy savings that result from its higher efficiency over its useful life must offset the higher first cost of a condensing boiler over a conventional boiler. A life cycle cost analysis should be performed when considering a condensing boiler.
10.15 HEAT EXCHANGERS

Heat exchangers or converters are used as sources of heating or cooling for many hydronic systems and are of three general types:

- Steam-To-Water
- Water-To-Water
- Water-To-Steam

Steam-to-water heat exchangers are usually shell-and-tube units. Steam is admitted to the shell, and water is heated as it circulates through the tubes. Steam-to-water converters are useful where an addition is to be made to an existing steam system and where hot water heating is desired.

Heat exchangers are used in multi-building campuses or where district steam is available and individual buildings are to be heated with a hot water or low-pressure, low temperature steam. High-rise buildings can be zoned vertically by using steam distribution and installing converters at various levels to serve several floors that limit the maximum operating temperatures and pressures in the zone. Also, heat exchangers can be used with alternative heating systems such as solar and geothermal.
CHAPTER 11

ENGINEERED HEAT PUMP SYSTEMS
11.1 INTRODUCTION

This chapter covers engineered heat pump systems that are used to heat and cool large multifamily, commercial, and institutional buildings. Engineered heat pump systems are designed for a specific building as opposed to manufacturer-designed unitary heat pumps covered in Chapter 8 that are installed as a unit to provide heating and cooling in residential and small commercial buildings. Engineered heat pumps systems often consist of multiple heat pumps that are distributed throughout the building and share a common heat source and sink as well as a common heat transfer media and distribution system. This chapter will discuss engineered heat pump system operation, characteristics, arrangements, types, and design considerations.

11.2 HEAT PUMP SYSTEMS

A heat pump system essentially “pumps” heat out of the building, into the building, or moves it around within the building, as required, using a reversible refrigeration cycle. Chapter 9 covered central cooling systems and described the chiller refrigeration cycle where heat is extracted from the building by circulating chilled water. The heat is absorbed by circulating chilled water through air-to-water heat exchangers. The heat is then extracted from the returning chilled water by the chiller and transferred to the cooling tower water circuit by the chiller’s refrigeration cycle. This heat from the cooling tower circuit is then rejected to the atmosphere by the cooling tower. If heating is also required, then a central heating plant is included as described in Chapter 10. The central heating plant heats the supply water that is then circulated and heats each zone using water-to-air heat exchangers.

With a heat pump system, one system provides both cooling and heating by reversing the refrigeration cycle. Heat pump systems operate in either of the following two modes:

- Cooling Mode
- Heating Mode

In the cooling mode, the engineered heat pump system operates very much like the cooling plant with the refrigeration cycle operating. In the heating mode, the mechanical refrigeration cycle is reversed and heat is extracted from an outside source and delivered to the interior of the building. When the building’s heating load exceeds the heat pump system’s heating capacity a supplemental boiler is often incorporated into the system.

Engineered heat pump systems are often decentralized systems with individual heat pumps distributed throughout the building in or near the zones that they serve. These systems typically use multiple water-source heat pumps that are connected to a common hydronic piping system that circulates water to the connected heat pumps similar to the hydronic distribution systems described in Chapter 15.

While water is the most common heat transfer medium in engineered heat pump systems, air or a refrigerant could also be used as the heat transfer media. However, as discussed in Chapter 2, using water as a heat transfer medium with a hydronic distribution system has advantages over either air or a refrigerant and their associated distribution systems for distributed HVAC systems with central equipment.

Unlike unitary heat pump systems, engineered heat pump systems often include cooling towers and dry coolers that reject the heat to the outdoors and supplementary heaters such as boilers and waste heat exchangers to increase the heat pump systems heating capabilities. In addition, large heat pump systems can also include solar collection devices and thermal storage. The characteristics and operation of these auxiliary systems were discussed in Chapters 9 and 10 that cover central cooling and heating plants, respectively.

11.3 HEAT PUMP SYSTEM CHARACTERISTICS

Heat pump systems are often characterized by the following three parameters:

- Heat Source and Sink
- Heating and Cooling Distribution Fluid
- Thermodynamic Cycle

11.3.1 Heat Source and Sink

11.3.1.1 Definition

Heat pump systems require both a heat source for heating mode operation and a heat sink for cooling mode operation. The heat source provides the necessary heat energy when operating in the heating mode and the heat sink provides a means for rejecting heat when operating in the cooling mode. The heat source and sink are usually the same for an engineered heat pump system and simply referred to as the “source” when describing a heat pump system. The selection of the heat source and heat sink for an engineered heat pump system is a design issue that is determined by considering a number of parameters associated with the building and building site including the following:
11.3.1.2 Types

Common heat sources and sinks include the following:

- Air
- Water
- Earth
- Solar

11.3.1.2.1 Air Heat Source

Outdoor air can be used as the heat source and sink for an engineered heat pump system. Outdoor air is the typical heat source and sink for unitary heat pumps and smaller engineered heat pump systems but it is not commonly used for large engineered heat pump systems due to its temperature variability. When selecting or designing an air-source heat pump, two factors should be taken into consideration:

- Temperature variation in the given location
- Frost formation possibility.

As the outdoor temperature decreases, the heating capacity of an air-source heat pump also decreases. Selecting equipment for a given outdoor heating design temperature is therefore more critical than for a conventional fuel-fired heating system. Consequently, the equipment must be sized for as low a balance point as is practical for heating without having excessive and unnecessary cooling capacity during the summer.

When the surface temperature of an outdoor air coil is 32°F (0°C) or lower, frost may form on the coil that will interfere with heat transfer. The number of defrosting operations required for a particular installation is influenced by the climate, air-coil design, and the hours of operation. Experience shows that little defrosting is generally required below 20°F (-7°C) and below 60 percent relative humidity.

11.3.1.2.2 Water Heat Source

Water is also used as a heat source and sink for engineered heat pump systems. City water is seldom used because of cost and municipal restrictions. Well water is particularly attractive because of its relatively high and nearly constant temperature that is generally about 50°F (10°C) in Northern areas and 60°F (26°C) in Southern areas of the United States. Frequently, sufficient water is available from wells where the return water is reinjected into the aquifer after use. The use of well water for closed-loop heat pump systems results in negligible water being lost with the only change in the return water being a change in temperature. When well water is used as the heat source and sink for an engineered heat pump system, the water quality should be analyzed and the possible effect of scale formation and corrosion should be minimized.

In some instances, it may be necessary to separate the well water from the system equipment with an additional heat exchanger. Special consideration must also be given to filtering and settling ponds for specific fluoride. Other considerations are the costs of drilling, piping, and pumping and means for disposal of used water. Information on well water availability, temperature, chemical and physical analysis is available from U.S. Geological Survey.

Surface water can also be used as a heat source and heat sink for heat pump systems. However, surface water extracted from ponds, streams, rivers, and lakes will not have the near constant temperature throughout the year as well water. This variability is similar to the use of outdoor air as a heat source and sink and needs to be taken into account when designing the engineered heat pump system. Additionally, the cooling spread between inlet and outlet must be limited to prevent freeze up in the water chiller that is absorbing the heat during the winter in some locations.

Heated process water that is available in some commercial, industrial, and institutional facilities can also be used as a heat source and sink for heat pump systems. Waste process water can include spent warm water from on-site laundry operations, warm condenser water, or other on-site or off-site source. The use of waste heat from facility operations can save energy and make heat pump systems very economical.

Water-to-refrigerant heat exchangers that are sometimes used in engineered water-source heat pump systems are generally direct-expansion or flooded water coolers. These systems can either be shell-and-coil or shell-and-tube type. In small capacity refrigerant changeover systems, water-to-refrigerant heat ex-
changers are often designed to be used as a refrigerant condenser during the heating cycle and as a refrigerant evaporator during the cooling cycle.

11.3.1.2.3 Earth Heat Source

The earth can be used as a heat source and sink because of its near constant temperature a few feet below the surface. Buried coils accomplish heat transfer between the heat pump system and earth. The size of the system and the needed heat transfer capacity will determine the extent and configuration of the buried coils.

Soil composition varies widely throughout the United States from wet clay to sandy soil. The thermal properties of the soil in a particular location will determine the heat transfer capability of buried coils and must be taken into account during design. The poorer the heat transfer capability of the soil the more extensive the underground piping system for the heat pump system will need to be to either reject heat during the cooling mode or absorb heat in the heating mode. Additionally, soil moisture content that varies not only by location but also season of the year in most locations also impacts the soil heat transfer capabilities and the heat pump system performance.

In general, earth coils are usually spaced horizontally 3 to 6 feet and buried horizontally 3 to 6 feet below the surface for smaller residential and commercial buildings. Larger heat pump systems used for larger commercial and institutional facilities will often use vertical coils to go deeper into the earth and provide greater heat transfer area for a given surface area of land. In general, deeper burial depths are better for heat pump systems but excavation and drilling costs may require a compromise.

11.3.1.2.4 Solar Heat Source

A solar system can also be used as a heat source and sink for a heat pump systems. The principal advantage of using solar radiation as a heat pump heat source is that a solar system typically provides heat at a higher temperature than other heat sources. Higher temperatures result in an increase in coefficient of performance for the heat pump system. Compared to a solar heating system without a heat pump, the collector efficiency and capacity are materially increased when coupled with a heat pump system because of the lower collector temperature required.

Solar heat source are typically used on either a direct or indirect heat pump systems. With a direct system, the refrigerant evaporator tubes built into the solar collector that is usually a flat-plat collector. With a direct system, the solar collector can also serve as a condenser using outdoor air as a heat sink for cooling if the collector is open to allow outdoor air to pass through it.

An indirect heat pump system relies on either water or air as the heat transfer medium. Water or air is circulated through the solar collector that is typically used as a heat source. When air is used as the heat transfer medium, the solar collector can serve the following functions:

- The collector can serve as an outdoor-air pre-heater.
- The outdoor-air loop can be closed so that all source heat is derived from the sun.
- The collector can be disconnected from the outdoor air serving as the source or sink.

Heat pump systems using solar energy as the primary heat source usually require either an alternate heating system or a means of storing heat during periods of insufficient solar radiation.

11.3.2 Heating and Cooling Distribution Fluid

Like other HVAC systems discussed throughout this manual, heat pump systems use either one or a combination of the following distribution fluids or heat transfer media:

- Air
- Water
- Refrigerant

The advantages and disadvantages of using any one of these distribution fluids is the same for other systems as discussed in Chapter 2 for all-air, air-hydronic, all-hydronic, and direct refrigerant HVAC systems.

11.3.3 Thermodynamic Cycle

Heat pump systems include a reversible refrigeration cycle that allows them to operate in either a cooling or heating mode. This reversible refrigeration cycle is accomplished by the “changeover” or reversal of the heating and cooling distribution fluid used by the system. The thermodynamic cycle used for a heat pump system is characterized as one of the following:

- Refrigerant Changeover
11.4 HVAC Systems Applications

• Air Changeover
• Water Changeover

In each case, the changeover is accomplished by reversing flow through the refrigeration cycle using valves with refrigerant and water or dampers with air as the distribution fluid.

11.4 BASIC HEAT PUMP SYSTEM ARRANGEMENTS

Heat pump systems can be arranged in a variety of ways and categorized based on the characteristics discussed in the previous section. Figure 11-1 illustrates five basic heat pump system arrangements and categorizes these basic heat pump system arrangements based on the heat source and sink, distribution fluid, and thermal cycle used. Table 11-1 shows the five basic heat pump system arrangements and each system’s associated figure number.

11.4.1 Figure 11-1(a) Heat Pump System Arrangement

Figure 11-1(a) illustrates a heat pump system where the heat source and sink is outdoor air that includes refrigerant changeover. In Figure 11-1(a), refrigerant is the heat transfer medium for this heat pump system and indoor heating or cooling is accomplished using a refrigerant-to-air heat exchanger that operates as the evaporator in the cooling mode. In the heating mode with the refrigeration cycle reversed, this same indoor refrigerant-to-air heat exchanger operates as the condenser. The following paragraphs describe the cooling mode and heating mode refrigeration cycles for this system. A detailed description of the mechanical refrigeration cycle along with a diagram of the cycle discussed in this section is provided in Chapter 8.

In the cooling mode, refrigerant in a low-pressure liquid state passes through the indoor refrigerant-to-air heat exchanger and absorbs heat from the air forced through the heat exchanger by the fan. The addition of heat from the air converts the low-pressure liquid refrigerant into a low-pressure gas. The low-pressure gas refrigerant is then directed through the compressor by the “reversing” or “four-way” valve where it is compressed into a high-pressure gas. The outdoor refrigerant-to-air heat exchanger operates as a condenser in the cooling mode and transfers heat from the incoming high-pressure refrigerant to the outside atmosphere changing it into a high-pressure liquid. The high-pressure liquid refrigerant then passes through an expansion valve turning it to a low-pressure liquid. This low-pressure liquid refrigerant then enters the indoor refrigerant-to-air heat exchanger and the entire cooling cycle starts again.

In the heating mode, the reversing valve reverses the cooling-mode cycle described in the previous paragraph. Changing the position of the reversing valve reverses the direction of refrigerant flow in the system so that the indoor refrigerant-to-air heat exchanger operates as the condenser and the outdoor refrigerant-to-air heat exchanger becomes the evaporator in the refrigeration cycle. As a result, the building is heated by heat energy harvested from the outdoor air that is delivered to the interior of the building by the circulating refrigerant.

In addition to the reversing valve in Figure 11-1(a), there are also two expansion valves and each of these expansion valves is in parallel with a check valve. Two expansion valves are required because expansion valves are unidirectional and only work when high-pressure liquid refrigerant is flowing into the input of the valve. The parallel check valve allows the refrigerant to bypass its associated expansion valve when it is flowing opposite to the operating direction of the expansion valve. In this way, only the reversing valve position needs to be changed in order to switch the system between cooling and heating modes.

<table>
<thead>
<tr>
<th>Figure No.</th>
<th>Heat Source and Sink</th>
<th>Distribution Fluid</th>
<th>Thermal Cycle</th>
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<td>11-1(a)</td>
<td>Air</td>
<td>Air</td>
<td>Refrigerant Changeover</td>
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<td>11-1(b)</td>
<td>Air</td>
<td>Air</td>
<td>Air Changeover</td>
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<td>11-1(c)</td>
<td>Water</td>
<td>Air</td>
<td>Refrigerant Changeover</td>
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<td>11-1(d)</td>
<td>Earth</td>
<td>Air</td>
<td>Refrigerant Changeover</td>
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<td>11-1(e)</td>
<td>Water</td>
<td>Water</td>
<td>Water Changeover</td>
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Table 11-1 Basic Heat Pump System Arrangements
<table>
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<tr>
<th>HEAT SOURCE AND SINK</th>
<th>DISTR. FLUID</th>
<th>THERMAL CYCLE</th>
<th>DIAGRAM</th>
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<tr>
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**FIGURE 11–1 BASIC HEAT PUMP SYSTEM ARRANGEMENTS**
Figure 11-1(b) illustrates a heat pump system that uses air as its distribution fluid and a damper system instead of a reversing valve to switch between cooling and heating modes. Outside air is used as the heat source and heat sink in this heat pump system arrangement.

Figure 11-1(b) shows the heat pump system operating in the heating mode where the outdoor air is directed through the air-to-refrigerant heat exchanger labeled “evaporator” in the diagram by dampers. Heat is absorbed by the refrigerant from the outside air passing through the evaporator coil and enters the compressor as a low-pressure gas. The refrigerant exits the compressor as a high-pressure gas and then passes through the air-to-refrigerant heat exchanger labeled “condenser” in the diagram where it transfers heat to the indoor air passing through it. This heated air is then used to heat the interior of the building. The refrigerant leaves the condenser coil as a high-pressure liquid that is converted to a low-pressure liquid by the expansion valve. The low-pressure liquid refrigerant reenters the evaporator coil and the refrigeration cycle starts again.

In the cooling mode, the damper positions in Figure 11-1(b) are reversed and the indoor air is directed through the bottom air-to-refrigerant heat exchanger and the outside air is directed through the upper air-to-refrigerant heat exchanger. Unlike the previous system where reversing the refrigerant flow resulted in changing the function of the two air-to-refrigerant heat exchangers, the air changeover in this system does not change the function of the air-to-refrigerant heat exchangers. The upper air-to-refrigerant heater exchanger in Figure 11-1(b) remains the condenser and the bottom air-to-refrigerant heat exchanger remains the evaporator as labeled in the diagram.

The air changeover in Figure 11-1(b) is affected by changing damper positions that result in the indoor air passing through the evaporator coil and outdoor air passing through the condenser coil. When mixed return air and outside makeup air pass through the evaporator coil, heat is absorbed by the refrigerant and the air supplied to indoor spaces is cooled. The heat absorbed by the refrigerant is then transferred to the outside air that passes through the condenser coil via the refrigeration cycle. The heated air is then exhausted to the outdoors and the building is cooled by the heat pump system.

Figure 11-1(c) operates the same as Figure 11-1(a) except that this heat pump arrangement includes a chiller-condenser. With this heat pump arrangement the heat source and sink could either be water or air with the respective distribution fluid being air or water respectively. The chiller-condenser included in Figure 11-1(c) operates as a water-to-refrigerant heat exchanger when water is the heat source and sink and as a refrigerant-to-water heat exchanger when air is the heat source and heat sink. In other words, if water was used as the heat source and sink then air would be used to heat or cool the building via air passing through the refrigerant-to-air heat exchanger. Similarly, if air is the heat source and sink in this heat pump arrangement then the building would be conditioned by chilled or hot water provided by the chiller-condenser.

Figure 11-1(d) illustrates an earth-source heat pump system that uses two refrigerant-to-water heat exchangers that are labeled “evaporator” and “condenser” in the diagram. This system operates exactly the same as the heat pump system arrangement illustrated in Figure 11-1(b) except that it uses water instead of air. The changeover between cooling and heating modes is accomplished in Figure 11-1(d) using valves instead of the dampers that are used for changeover in Figure 11-1(b).

Figure 11-1(e) illustrates a heat pump system arrangement that uses two refrigerant-to-water heat exchangers that are labeled “evaporator” and “condenser” in the diagram. This system operates exactly the same as the heat pump system arrangement illustrated in Figure 11-1(b) except that it uses water instead of air. The changeover between cooling and heating modes is accomplished in Figure 11-1(e) using valves instead of the dampers that are used for changeover in Figure 11-1(b).

The three most common types of heat pump systems are commonly referred to as follows:

- Air-Source Heat Pump Systems
- Water-Source Heat Pump Systems
- Water-To-Water Heat Pump Systems
11.5.1 Air-Source Heat Pump Systems

Air-source heat pump systems are also known as air-to-air heat pump systems. Figure 11-2 provides a schematic diagram of an air-source heat pump system operating in both its heating and cooling mode. This air-source heat pump system operates the same as the engineered heat pump system illustrated in Figure 11-1(a) that uses air for both its heat source and sink and its distribution fluid.

Air-source heat pumps are typically unitary systems as discussed earlier in this chapter and described in detail in Chapter 8. An air-source heat pump transfers heat from indoor air to outdoor air in the cooling mode and from outdoor air to indoor air in the heating mode using refrigerant as the heat transfer medium. Air-source heat pumps can be obtained in the following configurations:

- Through-The-Wall Or Window Unit
- Rooftop Unit
- Split System Unit

Air-source heat pump systems are the most common type of heat pump systems and are usually factory-built unitary heat pumps. These systems are widely used in residential and small commercial building applications.

11.5.2 Water-Source Heat Pump Systems

A water-source heat pump system uses water as both its heat source and sink. Water-source heat pump systems are commonly found in large commercial and institutional buildings where zone control is needed. Water-source heat pump systems also used to produce hot or cold water in industrial applications. Figure 11-3 provides a schematic diagram of a water-source heat pump operating in both its cooling and heating mode. This water-source heat pump system operates the same as the engineered heat pump system illustrated in Figure 11-1(c) with water as the heat source and sink and air as the distribution fluid.

11.5.3 Water-To-Water Heat Pump Systems

Water-to-water heat pump systems transfer heat from one water loop to another water loop similar to a chiller in a central cooling plant. As a result, water-to-water heat pump systems are often referred to as a reverse cycle chiller since a water-to-water heat pump system can pump heat in both directions. A conventional chiller can only move heat in one direction. A water-to-water heat pump system uses water as the heat source and sink for both cooling and heating. The changeover between the cooling and heating modes for a water-to-water heat pump system can be accomplished in the refrigerant circuit but it is often more convenient to perform this changeover in the water circuits. A water-to-water engineered heat pump system with water changeover is illustrated in Figure 11-1(e).

11.6 WATER-TO-AIR HEAT PUMP SYSTEMS

Water-to-air heat pumps are the most common engineered heat pump systems for commercial and institutional buildings. Water-to-air heat pump systems use water as the heat source when in the heating mode and as the heat sink when in the cooling mode as illustrated in Figures 11-1(c) and 11-3. The water supply may be closed water loop, a lake, or ground source such as a well. The temperature and flow rate of the water source must be taken into account when designing the system to ensure that the system will operate as planned.

A distinct advantage of the water-source heat pump is its ability to operate without the need for defrosting. Its performance is superior to an air-source heat pump because:

- It maintains a fairly constant capacity since the heat source is available over a limited temperature range.
- It can operate at a seasonal coefficient of performance (COP) of 3 to 3.5 if the source water is constant at approximately 50°F (10°C).

Storage tanks with solar assisted heating offer an alternative for residential applications that do not have an adequate supply of ground source water. This system has the advantage of operating the collectors at relatively low temperatures where the collector efficiency is the highest. However, the economics associated with solar collectors and storage may be prohibitive when the total system cost effectiveness is considered.

11.6.1 Closed-Loop Systems

Figure 11-4 illustrates a closed-loop engineered water-to-air heat pump system. A number of individual heat pumps can be connected to the hydronic distribution system, see Figure 11-4. Depending on the building and the application, any number of individual heat
FIGURE 11–2 AIR-SOURCE HEAT PUMP SYSTEM SCHEMATIC DIAGRAM
FIGURE 11−3 WATER-SOURCE HEAT PUMP SYSTEM SCHEMATIC DIAGRAM
FIGURE 11–4 CLOSED LOOP WATER-TO-AIR HEAT PUMP SYSTEM
SCHEMATIC DIAGRAM
pumps can be connected to one common water loop if it is sized for them.

In large closed-loop applications such as office buildings that have substantial interior areas, some heat pumps may operate in a cooling mode while other units may simultaneously operate in the heating mode. Under this scenario, the heat pumps operating in the heating mode are removing heat from the water loop while those heat pumps operating in the cooling mode units are adding heat. A heating means such as a boiler and a cooling means such as a closed circuit cooling tower are often required to add or reject heat from the loop when heating and cooling loads in the building are unbalanced.

11.7 WATER-TO-AIR HEAT PUMP SYSTEM DESIGN CONSIDERATIONS

The following are design considerations for water-to-air heat pump systems:

- Zoning
- Heat Recovery and Heat Storage
- Concealed Units
- Ventilation
- Secondary Heat Source
- Boiler Capacity
- Heat Rejection Selection
- Ductwork Layout
- Piping Layout
- System Controls
- Unit location – condensate drain, maintenance accessibility, sound concerns

11.7.1 Zoning

The multiple water-loop heat pump system illustrated in Figure 11-4 offers excellent zoning capability. Since equipment can be placed in interior areas, the system can accommodate the future relocation of partitions with minimum duct changes.

11.7.2 Heat Recovery and Heat Storage

Many applications of water-to-air heat pump systems lend themselves well to heat storage. Installations that operate in winter on the cooling cycle most of the day and the heating cycle at night such as school buildings are good candidates for heat storage.

Heat storage may be accomplished by installing a large storage tank in the closed loop circuit ahead of the boiler and allowing the loop temperature to build up to 95°F (35°C) during the day. The stored water at 95°F (35°C) can be used during unoccupied hours to maintain heat in the building allowing the loop temperature to drop to 60°F (16°C). The boiler would not be used until the loop had dropped the entire 35°F (2°C). The storage tank provides a “flywheel” effect to prolong the period of operation where neither heat makeup nor heat rejection is required.

11.7.3 Concealed Units

When distributed heat pump units are located in the ceiling space, access for service and maintenance must be provided. Access is required for servicing filters, control panels, compressors and other components. Unless vertical space for the condensate drain lines is provided between the drain connection at the unit and the top of the installed ceiling, separate condensate pumps would be required.

11.7.4 Ventilation

With water-to-air heat pump systems, outdoor air for ventilation may be:

- Ducted from a ventilation supply system to the units.
- Drawn in directly through a motorized damper into the individual units.

To operate satisfactorily, the entering air to water-source heat pumps should be above 60°F (16°C). In cold climates it will be necessary to preheat the ventilation air supply. Stack effect, wind, and balancing can impact the quantity of ventilation air. Balancing the ventilation system for multiple heat pumps can be a challenge as airflow in the system is affected by individual units turning on and off and the operation of any motorized dampers utilized in the system.

11.7.5 Secondary Heat Source

The secondary heat source for heat makeup may be electric, natural gas, or other fuel. If the energy source for secondary heat is natural gas then a boiler is typic-
ally used. However, if electricity will be used then electric resistance elements can be installed in the individual heat pumps with suitable changeover controls. The changeover control could be an aquastat set to switch from heat pump to resistance air heaters when the loop water reaches the minimum design temperature. When loop temperature returns to an acceptable operating temperature, the resistance heat is cut off and the heat pump is again operated for heating.

11.7.6 Boiler Capacity

The boiler should be sized based on the heating load served by the individual heat pumps plus the heating load of the rejector.

11.7.7 Heat Rejector Selection

To have a closed loop circuit, the heat rejector must be a heat exchanger used to transfer heat between loop water and cooling tower water or a closed-circuit evaporative cooler. Most installations use the closed-circuit evaporative cooler. This cooler is selected in accordance with manufacturer’s recommendations using the following parameters:

- Water Flow Rate
- Water Temperature Range
- Approach

11.7.8 Ductwork Layout

Concealed heat pump units that are used in engineered water-to-air heat pump systems often use the space above the ceiling as the return plenum. Air-handling lighting fixtures should not be used to return air to the ceiling plenum in lieu of return air grilles because this “reheats” return air to the heat pump. Note: Reheat should never be introduced into the return air path of heat pumps that are operating in the heating cycle as this raises the condensing head pressure, which can decrease equipment life, and reduces the heat pump’s operating efficiency. The return duct connections to the heat pumps are often not airtight and many have severe leakage that allows return airflow to bypass the filter. In addition, if ventilation air is ducted to the return ductwork this problem is compounded by the lack of ventilation air returning to the unit. This creates difficulty in set up and proper balancing of the system.

Air system supply from the heat pumps should be designed for quiet operation. Heat pumps that are designed for ductwork require external static pressure. The heat pump manufacturer’s recommendations should be consulted for the minimum external static pressure required for proper operation and unit selection. Many heat pump manufacturers offer multiple fan speed and operating scenarios that can create system setup and unit operation issues.

11.7.9 Piping Layout

A reverse-return piping system as discussed in Chapter 15 should be used with a closed-loop water-to-air heat pump system. This is especially true where all heat pump units are approximately the same capacity. With a reverse-return piping system, balancing would only be required for each of the system branches.

If a direct return system is used, balancing the water flow is required at each individual heat pump. The entire system flow may circulate through the boiler and heat rejector in series, see Figure 11-4. Water makeup should be at the constant pressure point of the entire loop water system. Many considerations of piping system design are similar to the secondary water distribution of central hydronic systems.

11.7.10 System Controls

The system controls are simpler on the closed-loop heat pump system than for other central HVAC systems. There are only two control points for the water:

- Add heat when the water temperature drops to 60°F (16°C).
- Reject heat when the water temperature goes above 90°F (32°C).

Maintaining lower loop temperatures in warm weather can conserve energy and reduce operating costs.
12.1 INTRODUCTION

This chapter covers air distribution systems, associated accessories and components.

12.2 AIR DISTRIBUTION SYSTEM PURPOSE

The ideal purpose of an air distribution system is to provide a system to accomplish the following functions:

- Delivering conditioned supply air to the zones served by the air-handling unit (AHU).
- Recovering return air from zones.
- Providing outdoor ventilation air for mixing with return air to be reconditioned by the AHU.
- Exhausting air to the outdoors, as required.

Air distribution systems provide a way of controlling the airflow and temperature in zones throughout the building. Properly designed, installed, and maintained air distribution systems help provide a healthy, comfortable, and productive environment for building occupants.

12.3 AIR DISTRIBUTION SYSTEM COMPONENTS

The air distribution system is not just the air ducts that direct air to zones throughout the building. The air distribution system includes fans that move the air and ensure that it is delivered to the zones served at the correct airflow as well as air cleaners that remove contaminants from the air stream.

Thermal conditioners, in the form of heat exchangers that are typically referred to as coils, are also part of the air distribution system, when air is conditioned prior to delivery to the zone served. In addition, other devices such as humidifiers and dehumidifiers, that add or remove moisture from the supply air stream, are also part of the air distribution system, when required.

12.4 SMACNA AIR DISTRIBUTION SYSTEM STANDARDS

12.4.1 Objective of this Chapter

This chapter is not intended to replace information contained in other SMACNA standards and manuals with recommended practices that specifically address the design and construction of various types of duct. Appendix ? provides a list of SMACNA HVAC references and resources that include a number of publications that address air distribution system planning, design, installation, testing, and maintenance. SMACNA separates the design and construction of duct into two discrete activities. The following provides brief overviews of the contents of the following SMACNA publications:

- HVAC Systems Duct Design
- HVAC Duct Construction Standards
- HVAC Systems – Testing, Adjusting, and Balancing
- Other Relevant SMACNA Publications

12.4.1.1 HVAC Systems Duct Design

SMACNA’s HVAC Systems Duct Design manual provides the basic engineering guidelines for the sizing of HVAC ductwork systems along with the following related information:

- Materials
- Methods of Construction
- Economics of Duct Systems
- Duct System Layout
- Pressure Losses
- Fan Selection
- Duct Leakage
- Acoustic Considerations
- Duct Heat Transfer
- Testing, Adjusting and Balancing

12.4.1.2 HVAC Duct Construction Standards – Metal and Flexible

The third edition of the HVAC Duct Construction Standards – Metal and Flexible is intended primarily for commercial and institutional duct construction. This American National Standard contains tables and details for constructing duct for $\frac{1}{2}$ to 10 in. wg (125 to 2500 Pa) positive and negative pressures. An engineering and design chapter provides additional information and guidance to design professionals. The standard is applicable for construction using uncoated steel, galvanized and stainless steels and a limited range of...
aluminum ducts. This standard has been adopted in the ICC International Mechanical Code.

The SMACNA HVAC Duct Construction Standards manual addresses the construction of both rigid and flexible metal duct. Specific topics covered in this manual include the following:

- Straight Duct Construction
- Duct Fitting Construction
- Round, Oval, and Flexible Duct
- Duct Hangers and Supports
- Exterior Components
- Duct Equipment and Casings
- Duct Functional Criteria

12.4.1.3 HVAC Systems – Testing, Adjusting and Balancing

HVAC system air and hydronic system testing, adjusting, and balancing (TAB) are addressed in SMACNA’s HVAC Systems – Testing, Adjusting and Balancing manual. In addition to an introduction to TAB and HVAC system operating fundamentals, this manual addresses the following topics:

- TAB Instruments
- Preliminary Tab Procedures
- Air System Tab Procedures
- Hydronic System Tab Procedures

12.4.1.4 Other Relevant SMACNA Publications

Other SMACNA publications addressing the planning, design, construction, installation, and testing of air distribution systems include the following:

- Fibrous Glass Duct Construction Standards
- Fire, Smoke And Radiation Damper Installation Guide For HVAC Systems
- HVAC Air Duct Leakage Test Manual
- HVAC Duct Systems Inspection Guide
- Residential Sheet Metal Guidelines
- Seismic Restraint Manual: Guidelines For Mechanical Systems
- Thermoset FRP Duct Construction Manual
- Thermoplastic Duct (PVC) Construction Manual

12.5 AIR DUCTS AND PLENUMS

12.5.1 Air Duct Purpose

Air ducts provide a means for moving the following types of air throughout a building:

- Supply Air
- Return Air
- Outdoor Air
- Exhaust Air

Supply air is usually conditioned air supplied by the air-handling unit or unitary HVAC equipment to a zone.

Return air is the air extracted from the zone that is either returned to the HVAC equipment for reconditioning or relieved to the outdoors.

Outdoor air is air brought in from the outdoors and supplied to HVAC equipment or directly to zones for indoor air quality (IAQ) purposes. This outdoor air supply is typically referred to as ventilation air. Usually, outdoor air is mixed with return air and reconditioned before being supplied to the space but in some instances outdoor air may be cleaned and conditioned and supplied directly to the space via separate and independent systems and equipment. Dedicated outdoor air systems (DOAS) are one example of this type of system.

In locations where air is contaminated by the activity that takes place in the zone, the air may be exhausted directly from the building or cleaned and then exhausted directly from the building. Examples of areas that might require a ducted exhaust air system would be paint spray booths, commercial kitchen range hoods, and laboratory fume hoods.

12.5.2 Air Duct Construction

Air ducts can either be rigid or flexible. Rigid air ducts are usually main and branch duct runs from the air handling unit or unitary HVAC equipment to the zone served. Flexible duct is normally used to connect
between the main or branch duct and the air outlet or inlet that is located in the ceiling, wall, or floor of the zone served. The purpose of flexible duct is to provide flexibility when connecting to air outlets and inlets. The best practice from an energy efficiency standpoint is to keep the use of flexible duct to a minimum because it has a higher pressure drop compared to rigid metal duct.

Rigid duct in commercial and institutional buildings is usually made of galvanized sheet metal but stainless steel, aluminum, or even copper are also used for specific needs and applications. However, ducts can be made of other materials when the application, aesthetics, or atmosphere warrant. Ducts can also be constructed from a variety of other non-metallic materials for specific applications such as the following:

- Fibrous-Glass
- Thermoset Fiberglass-Reinforced Plastic
- Concrete
- Fabric

These materials are typically chosen for specific needs and applications.

Rigid metal ducts can also be internally lined to reduce sound transmission.

Rigid air ducts can be constructed as rectangular, square, round, oval, or any other geometric shape. Other geometric shapes are sometimes used when there are space restrictions or aesthetic requirements when the duct is exposed. Flexible air duct is always round.

12.5.3 Air Duct Accessories

Duct accessories include a number of different components such as:

- Volume Dampers
- Access Doors
- Flexible Duct Connectors
- Silencers
- Turning Vanes
- Duct Liners
- Fire, Smoke, Fire/Smoke Dampers

12.5.3.1 Volume Dampers

Dampers can be used to adjust or completely close off airflow and vary in design from single-blade dampers rotating on a fixed axis to multi-blade opposing- or parallel-blade dampers.

12.5.3.2 Access Doors

Access doors are provided to service accessories such as fire and smoke dampers, to clean air coils, to service in-duct sensors and controls, and to provide access to other in-duct operational devices for maintenance.

12.5.3.3 Flexible Duct Connectors

Flexible duct connectors are used to isolate the vibration of fans from rigid metal duct so that vibration is not transmitted directly from the fan to the duct.

12.5.3.4 Silencers

Silencers are typically used near the main fan to reduce sound transmission into the distribution system or in locations where low sound levels must be maintained.

12.5.3.5 Turning Vanes

Properly installed, turning vanes reduce the pressure drop through elbows. Single-thickness vanes present lower pressure losses to the air stream but double-thickness vanes provide more rigidity to the fitting and are less prone to rattle in high-velocity air streams.

12.5.3.6 Duct Liners

Duct liners reduce sound transmission along and through the walls of lined duct and provide some thermal benefits.

12.5.3.7 Fire, Smoke, Fire/Smoke Dampers

These dampers prevent fire and smoke from traveling through the building’s air distribution system.

12.5.4 Air Plenums

Air plenums are spaces that are used to move air from a conditioned zone to HVAC equipment without the use of duct. Most often, air plenums are used as the return air path. The space above an architectural ceiling and below the structural overhead is an example of the most commonly used return air plenum. However, air plenums may also be created by lining shafts with sheet rock, these are only allowed to be used for return
air paths as conditioned air could condense moisture and create mold growth on the paper covering on sheet rock.

Careful design is required when return air plenums are used as a component of an HVAC system because they create extensive negative pressure zones within the building and this negative pressure can create undesirable air paths between the conditioned interior and unconditioned exterior air. Conduit, piping, floor-to-floor barriers, and other penetrations into or through an air plenum need to be sealed to prevent air from being pulled from the wrong spaces—such as warm humid outdoor air being drawn into the building during cooling—into spaces where moisture-laden air can condense. Where moisture is condensing and accumulating in hidden, unseen spaces mold growth can occur which may simply cause objectionable odors or serious damage to a building’s materials and structure.

Ventilation plenums—used to draw in outdoor air to be distributed by HVAC equipment—are typically located in the same areas as the building’s air handling units and mechanical equipment room. Ventilation plenums are fabricated from sheet metal and mounted on the interior side of the ventilation louver. Due to the inherent size and performance of mechanical louver, these plenums can be quite large in order to provide the needed volume of outdoor air for ventilation.

An example of an air plenum being used to supply condition air is an under floor air distribution (UFAD) systems. This method of distributing conditioned air uses the space below a raised floor and above the structural floor as a supply air plenum. Initially, these systems were only found in high-tech office spaces, data processing centers, data and communications rooms, and other areas that included a raised floor in the design scheme because the computer and servers required a lot of interconnecting data, communications, and power cabling that frequently needed to be reconfigured. The design and construction of supply air plenums requires significant attention to detail in terms of sealing and blocking air paths that can cause energy loss send cool moist air into concealed spaces. The types of materials require careful selection and specification, for example all cabling must meet specific requirements and sheet rock is not permitted in supply plenums. See Appendix?—Displacement Ventilation for more details

12.6 AIR TERMINAL UNITS

Air terminal units for both variable-air-volume and constant-air-volume HVAC systems are an important part of the air distribution system. This is particularly true of VAV air terminal units that regulate the amount of conditioned air delivered to each zone. Air terminal units can be obtained in a variety of configurations including the following:

- Single-Duct VAV Air Terminal Units
- Dual-Duct VAV Air Terminal Units
- Parallel Flow Fan-Powered VAV Air Terminal Units
- Series Flow Fan-Powered VAV Air Terminal Units
- Other VAV Air Terminal Unit Configurations

12.7 AIR OUTLETS AND INLETS

12.7.1 Air Outlet and Inlet Function

Air outlets and inlets are the interface points between the HVAC air distribution system and the zone served. Air outlets represent the point at which conditioned air is introduced into the space from the air distribution system and air inlets are the point at which the mixed air is extracted from the space and either returned to the HVAC system for reconditioning or exhausted to the outdoors. In commercial and institutional buildings, air outlets and inlets are normally installed in the ceiling because that is where the supply duct is typically routed and where either the return duct or air plenum is located. However, air inlets and outlets can also be installed in walls and floors depending on the location of the supply and return air distribution system and the required zone air distribution patterns.

12.7.2 Air Outlet and Inlet Types

Air outlets and inlets are typically categorized as one of the follow types:

- Diffusers
- Grilles
- Registers

12.7.2.1 Diffusers

A diffuser is an air distribution system outlet. The purpose of a diffuser is to direct supply air in various dir-
sections and planes within a zone to promote the mixing of the conditioned supply air with the existing air. A diffuser is comprised of deflecting vanes used to direct the supply airflow into the zone. Diffusers are typically mounted in the ceiling and can be circular, square, or rectangular.

12.7.2.2 Grilles

A grille is simply a louvered or perforated covering for an air passage opening that can be installed in the ceiling, wall, or floor.

12.7.2.3 Registers

A register is a grille that is equipped with an internal damper that is usually manually operated to control the airflow through the register. Note: Internal dampers in registers can be noisy, are ineffective, create drafts, and affect the air throw. The primary air volume adjustments should be made at the distribution duct damper supplying air to the register.

12.7.3 Air Outlet and Inlet Selection and Installation

The type, size, throw and location of air outlets and inlets in a zone are very important. Air outlets and inlets need to be sized so that they can supply the required airflow without generating too much noise or introducing the air at too high or too low of a velocity. Excessive pressure loss through the air outlets or inlets will require more fan power. Air outlets and inlets must be located to allow the necessary mixing of conditioned supply air in the zone before it is extracted and to avoid air stratification in the zone.

Air outlets and inlets as well as their location in the zone are very important to occupant thermal and acoustic comfort as well as the perceived quality of the HVAC system. Unlike the rest of the HVAC system that is hidden from building occupants’ sight, HVAC air outlets and inlets can be seen and heard by building occupants. Installing the wrong air outlets and inlets or choosing the wrong location can result in an otherwise outstanding HVAC system being judged as inadequate or poor by building occupants. Poor placement of air outlets and inlets can result in unnecessary complaints and call backs.

12.8 PROVISIONS FOR TESTING, ADJUSTING, AND BALANCING

Provisions must be included in the air distribution system for testing, adjusting, and balancing (TAB). This includes providing an adequate number of the following measuring instruments, sensors, and TAB access points:

- Pressure Gages or Ports
- Thermometers
- Flow Meters
- Capped Duct Test Wells
- Volume Dampers
- Other Instrumentation and Test Points

These measuring instruments, sensors, and test points need to be specified by the duct designer and provided at strategic locations at the time of the air distribution system installation. SMACNA’s HVAC Systems – Testing, Adjusting and Balancing manual and Chapter 9 of SMACNA’s HVAC Systems Duct Design manual provide specific information on the provisions for the air distribution system to allow for proper TAB procedures.

In addition, access to these devices must be provided for TAB. This includes consideration in the building’s design process all the way through the installation process. Examples where special attention to access is required are:

- High ceiling heights
- Duct or piping to be located high above a ceiling
- Duct and piping routed above sheetrock ceilings
- Duct and piping is located above sloped or stepped auditorium seating
- Architectural features that prevent access such as atrium outlets located high above stairs

In all cases, reasonable access and provisions for TAB requirements must be considered and incorporated into the design and installation of the air distribution system.
CHAPTER 13
FANS AND AIR-HANDLING UNITS
13.1 INTRODUCTION

This chapter covers fans and air-handling units that are used to supply conditioned air to zones. This chapter starts with a discussion of fans and fan types that leads to an introduction to fan operation and the fan curve. Fan curves for a backward blade centrifugal fan are then used to illustrate the fan laws and the impact of varying static pressure, airflow, and fan speed on fan performance including power and energy use. The air distribution system curve is then presented and superimposed on the fan curves to illustrate air distribution system dynamics as a function of thermal load. Various methods of fan airflow control are then discussed with a focus on variable frequency drives being the most common method used today. This chapter ends by covering air-handling units which are commonly used to condition and deliver air to zones served in distributed HVAC systems in large commercial buildings with a central cooling and heating plant.

13.2 FANS

13.2.1 Fan Purpose

The purpose of a fan is to move air through the air distribution system so that it can be delivered to a zone at a given airflow. In this respect, a fan can be thought of as a pump and the pressure determines the rate at which air is pumped by the fan. By creating a pressure differential, air moves through the air distribution system. In other words, the greater the pressure differential that the fan creates, the greater the volume of airflow through the system. The airflow through a fan is determined by the resistance to flow in the air distribution system which is also referred to as static pressure.

13.2.2 Fan Categories

There are two basic types of fans used in HVAC applications. These two types of fans are categorized according to the direction of airflow through the fan and are as follows:

- Axial Fans
- Centrifugal Fans

13.2.2.1 Axial Fans

Axial fans get their name from the fact that the airflow through the fan is along the axis of the fan body and perpendicular to the impeller or blade. Axial flow fans are typically used in applications that require high volumes of airflow and low static pressure. An example of an application for an axial flow fan would be an exhaust fan. Axial fans also have the ability to reverse airflow by changing the direction of their rotation which centrifugal fans are not able to do.

Axial fans can be further divided into the following three fan types:

- Propeller Fans
- Tube-Axial Fans
- Vane-Axial Fans

13.2.2.1.1 Propeller Fans

Propeller fans get their name from the fact that their impeller looks like an airplane propeller. Figure 13-1 shows a diagram of a propeller fan. Propeller fans are typically found in applications where a large volume of air needs to be moved against little or no static pressure. Propeller fans are commonly found in residential applications such as a simple box fan or an attic fan used for ventilating or cooling a room or the entire home. In commercial, industrial, and institutional applications, propeller fans are most often used as exhaust fans and can be either roof or wall mounted. Propeller fans are also sometimes used for ventilating unconditioned buildings such as warehouses.

13.2.2.1.2 Tube-Axial Fans

Tube axial fans encase their impeller in a cylindrical enclosure, see Figure 13-2. By placing the impeller in a cylindrical enclosure that guides the air through the fan, tube-axial fans are able to deliver higher airflows against higher static pressures than simple propeller fans more efficiently.

13.2.2.1.3 Vane-Axial Fans

Figure 13-3 shows a diagram of a vane-axial fan. Vane axial fans differ from tube-axial fans in that they have a set of vanes located either at the inlet or outlet of the fan to guide the air through the fan. Due to their design, vane-axial fans can be used in applications that require airflow delivered at higher pressures through supply ducts. Vane-axial fans are not as efficient as propeller fans but tend to be the most efficient fans available for HVAC air-handling units (AHU) with efficiencies in the upper 80 percents largely because the direction of the airflow is not changed as it passes through the fan.

13.2.2.1.4 Axial Fan Blade Pitch

The pitch of axial-flow fan blades can be:

- Fixed
FIGURE 13-1 AXIAL-FLOW FAN: PROPELLER TYPE
FIGURE 13-2 AXIAL-FLOW FAN: TUBE-AXIAL TYPE

FIGURE 13-3 AXIAL-FLOW FAN: VANE-AXIAL TYPE
With fixed-pitch blades, the fan blade pitch cannot be changed. However, adjustable-pitch blades allow the user to manually adjust the blade pitch to set the fan’s supply airflow and not penalize efficiency. Adjustable-pitch blades can be a useful feature for building commissioning. Variable-pitch blades allow the fan blade pitch to be varied during operation by pneumatic or electric actuators that permits them to provide efficient airflow without changing the speed of the fan. However, the mechanism that varies the blade pitch must be maintained to ensure proper operation over time.

13.2.2 Centrifugal Fans

Centrifugal fans are built and operate similar to centrifugal pumps covered in Chapter 16 with the fan impeller enclosed in a casing. Unlike an axial fan where the air stream passes straight through the fan, the air stream makes a right angle from where it enters the fan. Centrifugal fans get their name from the fact that the air is spun around by the impeller and ejected by centrifugal action. Centrifugal fans are less efficient than axial fans with efficiencies up to about 80 percent for backward-inclined airfoil blade centrifugal fans compared to up to 90 percent efficiency for axial fans. However, centrifugal fans are typically quieter than axial fans which is important in HVAC applications but have more inertia and require greater motor starting torque than axial flow fans.

Centrifugal fans are further classified by the their impeller blade construction as follows:

- Backward Inclined Blade
- Radial Blade
- Forward-Curved Blade

13.2.2.1 Backward Inclined Blade

Backward inclined (BI) blade centrifugal fans have fan blades that are inclined backward. BI blade centrifugal fans can have either airfoil or flat blades. Airfoil BI blade centrifugal fans are also referred to as backward-curved blade centrifugal fans. Airfoil BI blade centrifugal fan blades are curved backward and look like an airplane wing. Airfoil BI blade centrifugal fans are the most efficient blade configuration and are commonly used in commercial and institutional HVAC systems as supply and return air fans. Figure 13-4 provides a diagram of an airfoil BI blade centrifugal fan. Flat-blade BI blade centrifugal fans lack the curved airfoil design and are typically used as exhaust fans in commercial and institutional HVAC systems.

13.2.2.2 Radial Blade

Radial or straight blade centrifugal fans are most often found in manufacturing and industrial operations and used in exhaust and ventilation applications. Figure 13-5 provides a diagram of a radial blade centrifugal fan. The main advantage of a radial blade impeller over backward and forward blade impellers is that a radial blade impeller will allow small airborne particulates such as sawdust and metal shavings to pass through the fan in the air stream. Radial blade centrifugal fans are not used in commercial and institutional HVAC systems because they are not as efficient as backward curved blade centrifugal fans and their ability to pass small airborne particulates found in the air stream is not an advantage for HVAC systems. Radial blade centrifugal fans have typical efficiencies of around 50 to 60 percent.

13.2.2.3 Forward Curved Blade

Forward curved blade centrifugal fans are the least efficient type of centrifugal fan but they can be built smaller and less expensively than backward curved blade centrifugal fans. Forward curved blade centrifugal fans are used primarily in smaller unitary HVAC equipment. Figure 13-6 provides a diagram of a forward curved blade centrifugal fan.

13.2.3 Summary Of Fan Types and Operation

Figures 13-7(a) and 13-7(b) provide a concise summary of the fan categories and types that are encountered in commercial and institutional facility HVAC systems. In addition to the axial and centrifugal fan categories discussed above, the characteristics of some special fan designs including tubular centrifugal both centrifugal and axial roof ventilators are also provided. Figures 13-7(a) and 13-7(b) are taken from the 2008 ASHRAE Handbook – HVAC Systems and Equipment that is published by the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) and provides the following summary information about each fan type:

- Impeller Design
- Housing Design
- Performance Curves
FIGURE 13-4 CENTRIFUGAL FAN: BACKWARD INCLINED (AIRFOIL) BLADE

FIGURE 13-5 CENTRIFUGAL FAN: RADIAL (STRAIGHT) BLADE
13.2.4 Fan Operation

A fan is an air pump. The rate at which a fan can "pump" air depends on the pressure the fan must overcome. For a fan, the flow rate corresponds to a specific resistance to flow or static pressure. The series of airflow and static pressure points for a fan operating at a constant speed defines the fan curve.

Figure 13-8 illustrates a fan curve for a typical centrifugal fan with backward curved blades. A fan curve provides the unique operating characteristics for a fan that is running at a given speed or revolutions per minute (rpm). The fan curve describes the relationship graphically between airflow and static pressure for a particular fan running at a given speed. Fan manufacturers determine fan curves by testing the fan under controlled conditions.

Figure 13-8 illustrates a fan curve for a typical centrifugal fan with backward curved blades. A fan curve provides the unique operating characteristics for a fan that is running at a given speed or revolutions per minute (rpm). The fan curve describes the relationship graphically between airflow and static pressure for a particular fan running at a given speed. Fan manufacturers determine fan curves by testing the fan under controlled conditions.

It should be noted that fan curves are not all developed using test data. Instead, fan manufacturers sometimes develop fan curves using calculations instead of testing. This and other factors like system effect can create issues with fan curve accuracy in the field. Unlike pumps, it is usually not possible to shut off head on a fan to correct the fan curve.

In addition, the altitude above or below sea level should be considered when selecting a fan for a particular location. Fan curves are typically prepared by manufacturers assuming air density at sea level. At higher elevations, such as Colorado Springs, where the air is less dense than sea level fan performance can change significantly from what it is at sea level. Fan performance should be adjusted based on manufacturer recommendations for altitudes that differ significantly from the altitude that the fan curve is based on.

Given the fan curve for a particular speed, the fan's operating point for a specified airflow or static pressure can be determined. For example, given a static pressure, the resulting airflow can be read from the fan curve by drawing a horizontal line from the static pressure on the vertical axis to the intersection of the fan curve. From the intersection, draw a vertical line downward and the resulting airflow can be read from the horizontal axis. The static pressure for a given airflow can also be determined from the fan curve by reversing this process.

From Figure 13-8, it can be seen that if the fan is being driven at 1000 rpm and has a static pressure of 2.5 in. wg at its outlet, the airflow through the fan will be
<table>
<thead>
<tr>
<th>TYPE</th>
<th>IMPELLER DESIGN</th>
<th>HOUSING DESIGN</th>
</tr>
</thead>
<tbody>
<tr>
<td>CENTRIFUGAL FANS</td>
<td></td>
<td></td>
</tr>
<tr>
<td>AIRFOIL</td>
<td>Highest efficiency of all centrifugal fan designs. Ten to 16 blades of airfoil contour curved away from direction of rotation. Deep blades allow efficient expansion within blade passages. Air leaves impeller at velocity less than tip speed. For given duty, has highest speed of centrifugal fan designs.</td>
<td>Scroll design for efficient conversion of velocity pressure to static pressure. Maximum efficiency requires close clearance and alignment between wheel and inlet.</td>
</tr>
<tr>
<td>BACKWARD-CURVED</td>
<td>Efficiency only slightly less than airfoil fan. Ten to 16 single-thickness blades curved or inclined away from direction of rotation. Efficient for same reasons as airfoil fan.</td>
<td>Uses same housing configuration as airfoil design.</td>
</tr>
<tr>
<td>RADIAL</td>
<td>Higher pressure characteristics than airfoil, backward-curved, and backward-inclined fans. Curve may have a break to left of peak pressure and fan should not be operated in this area. Power rises continually to free delivery.</td>
<td>Scroll. Usually narrowest of all centrifugal designs. Because wheel design is less efficient, housing dimensions are not as critical as for airfoil and backward-inclined fans.</td>
</tr>
<tr>
<td>FORWARD-CURVED</td>
<td>Flatter pressure curve and lower efficiency than the airfoil, backward-curved, and backward-inclined. Do not rate fan in the pressure curve dip to the left of peak pressure. Power rises continually toward free delivery. Motor selection must take this into account.</td>
<td>Scroll similar to and often identical to other centrifugal fan designs. Fit between wheel and inlet not as critical as for airfoil and backward-inclined fans.</td>
</tr>
<tr>
<td>PROPELLER</td>
<td>Low efficiency. Limited to low-pressure applications. Usually low-blade impellers have two or more blades of single thickness attached to relatively small hub. Primary energy transfer by velocity pressure.</td>
<td>Simple circular ring, orifice plate, or venturi. Optimum design is close to blade tips and forms smooth airflow inside wheel.</td>
</tr>
<tr>
<td>TUBERICAL</td>
<td>Good blade design gives medium-to-high-pressure capability at good efficiency. Most efficient have airfoil blades. Blades may have fixed, adjustable, or controllable pitch. Hub is usually greater than half fan tip diameter.</td>
<td>Cylindrical tube with close clearance to blade tips. Guide vanes upstream or downstream from impeller increase pressure capability and efficiency.</td>
</tr>
<tr>
<td>AXIAL FANS</td>
<td>Performance similar to backward-curved fan except capacity and pressure are lower. Lower efficiency than backward-curved fan. Performance curve may have a dip to the left of peak pressure.</td>
<td>Cylindrical tube similar to vaneaxial fan, except close to wheel is not as close. Air discharges radially from wheel and turns 90° to flow through guide vanes.</td>
</tr>
<tr>
<td>VANEAXIAL</td>
<td>Low-pressure exhaust systems such as general factory, kitchen, warehouse, and some commercial installations. Provides positive exhaust ventilation, which is an advantage over gravity-type exhaust units. Centrifugal units are slightly quieter than axial units.</td>
<td>Normal housing not used, because air discharges from impeller in full circle. Usually does not include configuration to recover velocity pressure component.</td>
</tr>
<tr>
<td>TUBULAR CENTRIFUGAL</td>
<td>Low-pressure exhaust systems such as general factory, kitchen, warehouse, and some commercial installations. Provides positive exhaust ventilation, which is an advantage over gravity-type exhaust units.</td>
<td>Essentially a propeller fan mounted in a supporting structure. Hood protects fan from weather and acts as safety guard. Air discharges from annular space at bottom of weather hood.</td>
</tr>
</tbody>
</table>

**FIGURE 13-7 (a) SUMMARY OF FAN CATEGORIES, TYPES, AND CHARACTERISTICS**
These performance curves reflect general characteristics of various fans as commonly applied. They are not intended to provide complete selection criteria, because other parameters, such as diameter and speed, are not defined.

**FIGURE 13-7 (b) SUMMARY OF FAN CATEGORIES, TYPES, AND CHARACTERISTICS**
FIGURE 13.8 FAN CURVE FOR TYPICAL CENTRIFUGAL FAN: BACKWARD INCLINED BLADE

- Brake Horsepower (BHP)
- Static Pressure (in. wg)
- CFM (in 1,000s)

- 14.5 BHP at 24,000 CFM
- 2.5 in wg
- 2000 RPM
24000 cfm. If the static pressure at the outlet of the fan increases to 3.0 in. wg at the outlet, the airflow drops to about 22000 cfm. Similarly, the airflow will increase to 25500 cfm if the static pressure at the outlet of the fan decreases to 2.0 in. wg at the outlet.

Also shown in Figure 13-8 is the fan brake horsepower (BHp) curve. This curve provides brake horsepower as a function of cfm for the fan. At an operating point of 24000 cfm and 2.5 in. wg, the fan’s required brake horsepower at 1000 rpm is 14.25.

### 13.2.5 Varying Fan Speed

A fan curve illustrates the relationship between static pressure and airflow for a fan at a given speed. At a constant speed of 1000 rpm, the fan’s operating point will move along the fan curve with airflow and static pressure changing, see Figure 13-8. In order to achieve a static pressure and airflow different from those defined by the 1000 rpm fan curve, the speed of the fan must be changed.

Changing the fan speed results in a new fan curve that provides a new set of unique relationships between static pressure and airflow for the fan at the new speed. Allowing fan speed to vary results in a family of fan curves, see Figure 13-9. From Figure 13-9, it can be seen that by increasing the fan speed from 1000 rpm to 1050 rpm and keeping the static pressure constant at 2.5 in. wg airflow increases from 24000 cfm to about 26000 cfm. Similarly, decreasing the fan speed to 950 rpm from 1000 rpm for a static pressure of 2.5 in. wg results in the airflow from the fan decreasing from 24000 cfm to 26000 cfm. In general, for a constant static pressure, increasing the fan speed increases airflow and decreasing static pressure decreases airflow.

Similarly, by holding airflow constant at 24000 cfm and increasing speed from 1000 rpm to 1050 rpm results in an increase in static pressure for the fan from 2.5 in. wg to 3.2 in. wg. Decreasing fan speed from 1000 rpm down to 950 rpm at 24000 cfm results in a decrease in static pressure from 2.5 in. wg to about 1.8 in. wg in Figure 13-9. In general, for a constant airflow, increasing fan speed increases static pressure and decreasing fan speed decreases static pressure.

### 13.2.6 Fan Laws

Fan laws express the relationship between airflow, static pressure, and brake horsepower as a function of fan speed for a fan.

In the previous section, the fan curves were used to show the relationship between static pressure and airflow at a given fan speed. The fan laws can also be used to predict the new operating point of a fan if the speed is changed. From the first fan law, it can be seen that the ratio of the fan speed is equal to the ratio of the fan speed. Given that at 1000 rpm the airflow is 24000 cfm from Figure 13-8, the first fan law would predict that the airflow for this fan operating at 1050 rpm would be as follows:

\[
\text{cfm}_2 = \frac{\text{cfm}_1 \cdot \text{rpm}_2}{\text{rpm}_1}
\]

Similarly, the second fan law would predict that the static pressure resulting from an increase in fan speed would increase as the square of the ratio of the change in fan speed. For a fan with the fan curve illustrated in Figure 13-8, the static pressure would be predicted to increase as follows:

\[
\frac{\text{SP}_1}{\text{SP}_2} = \left(\frac{\text{rpm}_1}{\text{rpm}_2}\right)^2
\]

Similarly, the third fan law would predict that the brake horsepower required to drive the fan. The cube of the ratio of the speed change is inversely related to the ratio of the change in fan brake horsepower as shown in the third fan law. The change in fan speed from 1000 rpm to 1050 rpm will result in the following increase in fan brake horsepower:

\[
\frac{\text{Bhp}_1}{\text{Bhp}_2} = \left(\frac{\text{rpm}_1}{\text{rpm}_2}\right)^3
\]

This increase in airflow through the fan does not come without a cost. The fan laws predict that increasing the speed of the fan to increase airflow and static pressure also increases the required power input or brake horsepower required to drive the fan. The cube of the ratio of the speed change is inversely related to the ratio of the change in fan brake horsepower as shown in the third fan law. The change in fan speed from 1000 rpm to 1050 rpm will result in the following increase in fan brake horsepower:
FIGURE 13-9 FAMILY OF FAN CURVES
Therefore, increasing the fan speed from 1000 rpm to 1050 rpm will result an increase in power to drive the fan from 14.5 Bhp to 16.8 Bhp. This is a 16 percent increase in power to drive the fan in order to increase the airflow from 24000 cfm to 25200 cfm or 5 percent. Since power is energy per time, this increase in speed also results in an increase in energy required to run the fan for a given period of time by 16 percent. Therefore, this increase in fan speed may not only require a larger motor that will increase the initial installation cost but also will increase the operating cost over the life of the installation.

Figure 13-10 illustrates this change in operating point. The original operating point for the fan at 1000 rpm was 24000 cfm at a static pressure of 2.5 in. wg. By increasing the fan speed to 1050 rpm, the airflow increased to 25200 cfm and the static pressure increased to 2.8 in. wg. In addition, the change in speed from 1000 rpm to 1050 rpm also resulted in an increase in required brake horsepower to drive the fan from 14.5 Bhp to 16.8 Bhp.

\[
\frac{14.5 \text{ Bhp}}{1000 \text{ rpm}} = \left(\frac{1000 \text{ rpm}}{1050 \text{ rpm}}\right)^3 \quad \text{Bhp}_2 = (14.5 \text{ Bhp}) \left(\frac{1050 \text{ rpm}}{1000 \text{ rpm}}\right)^3 \quad \text{Bhp}_2 = 16.8 \text{ Bhp}
\]

Where:
- \(SP\) = Static Pressure
- \(K\) = Constant that determines the Steepness of the Curve
- \(cfm\) = Airflow

Figure 13-11 shows an air distribution system curve that is based on the above equation. As can be seen from the equation and Figure 13-11, the air distribution system static pressure is directly proportional to the square of the airflow. Therefore, doubling the airflow through the distribution system will result in the static pressure increasing by four times.

The above equation relates static pressure and airflow for a given air distribution system under specified conditions. Any physical change in the air distribution system will result in a shift to a new system resistance curve. Specifically, a change in the air distribution system will result in a new constant “K” in the above equation. If a change increases the air distribution system resistance then the constant K will increase resulting in a new and steeper system resistance curve as illustrated by System Curve #2 in Figure 13-12. Similarly, if the air distribution system resistance decreases then the constant “K” will also decrease resulting in a new system resistance curve that is not as steep as the original System Curve #1.

To illustrate how a system distribution curve can change in an actual HVAC system, consider a VAV system. If the dampers in downstream VAV air terminals close to reduce airflow to a zone, additional resistance is introduced in the air distribution system and the system resistance curve will shift from the original System Curve #1 to the steeper System Curve #2 in Figure 13-12. The same shift to a new system curve can occur when system filters become dirty or when anything else occurs that increases the air distribution system’s resistance to airflow. Similarly, if the dampers in downstream VAV air terminals open to increase airflow to a zone then resistance to airflow will decrease and the system resistance curve will which will shift from the original System Curve #1 to the new flatter system curve.

13.4 AIR DISTRIBUTION SYSTEM OPERATING POINT

As discussed in Section 13.2.6, a fan curve describes the operating characteristics of a particular fan and consists of a series of airflow and static pressure points that a fan can operate at for a given fan speed. Simil-
FIGURE 13–10 FAN LAW EXAMPLE ILLUSTRATED WITH FAN CURVES
FIGURE 13-11 SYSTEM CURVE

SYSTEM CURVE EQUATION:

\[ SP = \frac{4.34 \times 10^{-6} \times CFM^2}{\text{CFM}} \]
FIGURE 13–12 SYSTEM CURVE CHANGE DUE TO INCREASED RESISTANCE TO FLOW
arly, the system resistance curve was defined in Section 13.3 as a series of airflow and static pressure points at which the air distribution system will operate. By superimposing the system resistance curve shown in Figure 13-12 on the fan performance curve provided in Figure 13-9, the air distribution system operating point can be determined. The air distribution system operating point occurs at the intersection of the system resistance curve and the fan performance curve, see Figure 13-13. Under the conditions specified, the air distribution system is delivering 24,000 cfm at a static pressure of 2.5 in. wg with the fan operating at 1000 rpm.

13.5 AIR DISTRIBUTION SYSTEM DYNAMICS

Opening or closing a damper in a supply duct will result in more or less air being supplied to the zone that the air distribution system is serving. As discussed in Chapter 3, VAV HVAC systems supply variable amounts of constant temperature air to a zone in order to maintain the desired temperature in that zone. The amount of constant temperature air supplied to the space is determined by the damper settings in the VAV terminal units serving the zone, see Figure 1-3.

If the thermal load in the zone never changed, a manual damper could be installed and this damper could be adjusted once during the testing, adjusting, and balancing (TAB) of the HVAC systems and never changed. This would result in a constant system operating point, see Figure 13-13. Unfortunately, this is not the case and the thermal load in any HVAC zone is always changing due to season, time of day, daily weather patterns, number of people in the zone, occupant activity, equipment use, and other variables. Therefore, to maintain the desired temperature in the zone the volumetric rate of constant-temperature air delivered to the zone must varied in response to changes in zone thermal load. This is accomplished automatically by sending a signal from the zone thermostat that directs the VAV terminal units to open to cool the space or close to allow the space to warm.

As discussed in Section 13.4, changes in the VAV terminal unit damper settings not only change the amount of air delivered to the zone but also change the air distribution system resistance curve, see Figure 13-12. By overlaying the system operating curves shown in Figure 13-12 on the fan-operating curve provided in Figure 13-8, the impact of a changing system-operating curve on air distribution system operation is illustrated in Figure 13-14.

As the VAV terminal unit dampers close the air distribution system it offers more resistance to airflow and the system resistance curve shifts from System Curve #1 to System Curve #2 with a corresponding shift in fan operating point along the fan operating curve, see Figure 13-14. This shift results in the desired reduction in airflow but also results in an increase in static pressure. Because this method of adjusting airflow is in response to changing zone thermal conditions results from the system traversing from System Operating Point #1 to System Operating Point #2 it is referred to as “riding the fan curve.” Adjusting supply fan airflow by “riding the fan curve” can be used with any centrifugal fan but it is most efficient when a forward-curved centrifugal fan is used. However, “riding the fan curve” can result in problems if the airflow is required to vary more than just a little during operation. Problems that can result from “riding the fan curve” over a wide range of airflows can include the following:

- Excessive duct pressure.
- Excessive duct leakage.
- Excessive noise at VAV terminal units.
- Erratic VAV terminal unit performance.
- Fan surge causing air pulsation in duct system.
- Negligible energy savings.

13.6 SYSTEM OPERATING POINT AND FAN SPEED

Figure 13-15 superimposes the system operating curves shown in Figure 13-12 on fan curve for 1000 rpm shown in Figure 13-8 along with the fan curve for the same fan operating at 900 rpm. Instead of riding the fan curve from System Operating Point #1 to System Operating Point #2 as discussed in the previous section, the reduced system airflow could also be achieved by reducing the fan speed. Reducing the fan speed from 1000 rpm to 900 rpm results in a shift from System Operating Point #1 to System Operating Point #3 along the system operating curve, see Figure 13-15. At System Operating Point #3, the airflow is now the desired 21200 cfm and the static pressure is 2.0 in wg. Unlike riding the fan curve to achieve reduced system airflow, reducing fan speed results in not only reduced airflow but also reduced static pressure which eliminates most of the problems with riding the fan curve. In addition, reducing the fan speed from 1000 rpm to 900 rpm reduced the power required by the fan from 14.5 Bhp down to 10.5 Bhp or about 28 percent.
FIGURE 13-13 SYSTEM OPERATING POINT
FIGURE 13–14 FAN AIRFLOW MODULATION “RIDING THE FAN CURVE”
The reduction in fan power can also be approximated using the third fan law:

\[
\frac{Bhp_1}{Bhp_2} = \left(\frac{rpm_1}{rpm_2}\right)^3
\]

\[
\frac{14.5 \ Bhp}{Bhp_2} = \left(\frac{1000 \ rpm}{900 \ rpm}\right)^3
\]

\[
Bhp_2 = (14.5 \ Bhp) \left(\frac{900 \ rpm}{1000 \ rpm}\right)^3
\]

\[
Bhp_2 = 10.6 \ Bhp
\]

13.7 SUPPLY FAN AIRFLOW CONTROL

13.7.1 Introduction and Overview

So far in this chapter, the airflow to a zone has been controlled solely by the VAV terminal unit dedicated to that zone. The supply air fan has been assumed to be a constant-air-volume system that supplies a constant airflow of conditioned air through the supply duct to the VAV air terminal units. This system will work but it is both difficult to control and inefficient. To improve the VAV HVAC system operation and efficiency, the airflow needs to be controlled both at the VAV air terminal unit and at the supply air fan.

This section will address why it is important to control supply airflow in a VAV HVAC system and methods used to control supply fan airflow. Many of the methods discussed in this section are no longer used and have been replaced by variable frequency drives (VFDs). VFDs provide better system control, are more efficient, and more economical on a life cycle cost basis than other methods of supply fan airflow control. However, these other methods of controlling supply fan airflow were used extensively before the advent of modern VFDs and may be encountered in existing VAV systems. As a result, these legacy systems are covered in this section because they need to be understood when maintaining, operating, or modifying an existing VAV HVAC system that uses them or when an existing VAV HVAC system is retrofitted with a VFD.

13.7.2 Why Control Supply Fan Airflow?

In a VAV system having a constant speed supply fan, as the VAV terminal units reduce the total airflow, the total and static pressures of the system will increase as the fan moves up its curve. If uncontrolled, ductwork can be damaged. Even in strong ducts, the terminal unit dampers must work against the higher pressure, resulting in poor control and increased noise. In addition to the impact of high duct pressures on duct construction and system acoustics, control of fan airflow also results in increased system efficiency and energy savings especially when using VFDs.

13.7.3 Supply Fan Airflow Control Methods

Methods used to control supply fan airflow can be categorized as follows:

- Fan Inlet/Outlet Control
- Fan Characteristic Control
- Fan Speed Control

Fan inlet and outlet control refers to the use of mechanical air valves placed either at the inlet or outlet of the fan to control the amount of air entering or exiting the fan. Common methods of fan inlet and outlet control include inlet vanes and discharge dampers. Fan characteristic control refers to mechanically adjusting the physical characteristics of the fan itself to achieve the desired airflow. Common methods of fan characteristic control include adjustable and controllable fan blade pitch as well the use of fan shrouds and fan bypass. Fan speed control can be accomplished by either controlling fan speed through the mechanical linkage between the fan drive motor and fan shaft or electrically by controlling the speed of the fan drive motor shaft. Controlling fan speed through a mechanical linkage includes the use of variable pitch sheaves and eddy current couplings. Two methods used to control the fan drive motor speed are DC motors and VFDs. Figure 13-16 summarizes the common methods used to control supply fan airflow.

13.7.4 Inlet Vanes and Discharge Dampers

Both inlet vanes and discharge dampers can be used to control supply fan airflow. Figure 13-17 compares the input power for a fan equipped with an inlet vane versus the same fan with a discharge damper as a function of rated fan airflow. The fan input power required by a fan operating below its rated airflow using an inlet vane is always less than the same fan using a discharge damper, see Figure 13-17. Since energy is the product of power and time, it can be seen that the use of an inlet vane results in an energy savings over the use of a discharge damper when the supply fan is operated below its rated airflow which is most of the time for a typical VAV system. This is why inlet vanes are more commonly used than discharge dampers and more likely to be encountered on existing VAV systems where inlet/outlet control is used to control the supply fan airflow.
13.7.5 Adjustable and Controllable Fan Blade Pitch

Supply fan airflow control can also be accomplished by physically changing the fan characteristics to meet the VAV system airflow requirements. This is usually accomplished by changing the pitch of the fan blades. This method can be used on any type of fan but it is most commonly used with axial fans because of their construction.

Changing the fan blade pitch or angle results in a new fan curve and supply airflow. The ability to change the pitch of the fan blades allows the fan to operate efficiently over a wide range of operating conditions. Even though the fan speed remains constant, changing the pitch of the blades reduces the needed fan power when the fan is operating below its rated airflow. This results in energy savings when compared to fan inlet/outlet control methods but the initial cost of a variable pitch axial fan along with maintenance costs over its useful life typically makes them less economical than inlet/outlet control on a life cycle cost basis.

Variable pitch blades on axial fans can be either:

- Adjustable Pitch
- Controllable Pitch

The angle of adjustable pitch blades is set manually to balance the fan against the systems. As a result, adjustable pitch blades are used where adjustments to blade angle are needed infrequently. This is not the case for VAV systems so adjustable pitch blades are not used extensively on VAV systems. As a result, axial fans with controllable pitch blades whose angle can be changed automatically by a blade controller during operation are most common on VAV systems.
FIGURE 13-17 FAN POWER INPUT VERSUS RATED AIRFLOW
13.7.6 Internal Centrifugal Fan Shrouds

The volume of a centrifugal fan can be controlled with cylindrical dampers that automatically slide over the airfoil or backward inclined wheel reducing the effective widths of the wheel that decreases the airflow and static pressure. This method reduces power consumption because the different fan curves have lower static pressures. The fan runs at a constant speed and the motor efficiencies remain the same.

13.7.7 Fan Bypass

With a fan bypass, air is cycled in the fan housing with an external bypass fitting. Fan bypasses do not provide any net savings in power since the fan is still running at the same speed and conveying the same total volume of air. Their only function is to vary the fan airflow as required by the system.

13.7.8 Variable Pitch Sheaves

One method of controlling supply fan airflow is by varying the fan speed using variable pitch sheaves. Variable pitch sheaves change the fan speed by varying the motor pulley diameter during operation using a mechanical gearbox. Variable pitch sheaves are more efficient than fan inlet/outlet control methods but do not operate the motor at its optimal efficiency.

13.7.9 Eddy Current Couplings

Eddy current couplings and other slip devices can also be used to control supply fan airflow by changing the fan speed. Eddy current couplings modify the supply air fan-operating curve by allowing the coupling between the motor shaft and the fan shaft to "slip." The fan drive motor operates at close to rated speed and the slip between shafts results in the fan operating at a slower speed. This system of fan speed control requires a relatively high horsepower and a resulting increase in energy consumption.

Eddy current drives use standard induction motors and an eddy current coupling. The constant speed input rotor generates a rotating magnetic flux for the output rotor. This flux generates eddy currents in the output rotor when differences in output and input rotor speeds exist. The eddy currents generate magnetic fields that are attracted to the originating field and this results in an output torque. Eddy current drives are less efficient than either DC motors or VFDs.

13.7.10 DC Motors

Varying supply fan airflow can also be accomplished by using a direct current (DC) motor to drive the fan. Standard alternating current (AC) supply voltage is converted to DC using an inverter and the resulting DC voltage is then used to drive the DC motor.

13.7.11 Variable Frequency Drives

Varying fan airflow can be accomplished using variable frequency drives or VFDs. VFDs work with alternating-current (AC) induction motors whose shaft speed is a function of the frequency of the applied AC voltage and the motor’s construction. VFDs electronically convert the applied 60 Hertz (Hz) or cycles per second voltage supplied by the building power distribution system to a higher or lower frequency waveform which is used to power the induction motor and results in a corresponding change in motor shaft speed. The motor’s shaft is coupled to the supply fan and results in a corresponding change in fan speed and operating characteristics. VFDs provide better system control, are more efficient, and more economical on a lifecycle cost basis than other methods of supply fan airflow control including riding the fan curve. VFDs are used extensively in commercial and institutional VAV HVAC systems today. Fan speed control using VFDs is covered in Chapter 17.

13.8 AIR HANDLING UNITS

An air-handling unit (AHU) is used to condition supply air to zones in central HVAC systems that include central cooling and heating equipment in a central plant. Air handling units are usually located throughout commercial and institutional buildings in fan or mechanical equipment rooms near the zones that they serve. Figure 13-18 shows a schematic diagram of a simple air-handling unit. The air handling unit contains an air filter for cleaning outside air and return air, a cooling coil that is essentially a water-to-air heat exchanger for cooling incoming air, a heating coil which is also a water-to-air heat exchanger for warming incoming air, and a supply fan in a single enclosure, see Figure 13-18. Chilled water and hot water are supplied by central cooling and heating equipment located in a central plant via chillers and boilers along with associated equipment such as pumps and cooling towers. A hydronic distribution system supplies the needed chilled and hot water to the air-handling unit cooling and heating coils.

Air-handling units can be supplied by the manufacturer as a pre-engineered packaged unit or custom built by the HVAC contractor and supplied to the field as a
preassembled unit or field built from components at the project site. In addition to the basic air-handling unit components shown in Figure 13-18, air-handling units often include other components such as controls; motor starters and variable frequency drives (VFD); additional fans for outside and return air as well as multiple supply air fans for staging airflow; economizers and energy recovery equipment; various types of dampers; sound attenuators; air filtration and cleaning systems, humidification and dehumidification systems; among other options. The options selected for the air-handling unit will depend on the application. In addition, air-handling units can be built and supplied in a variety of configurations to meet space constraints in fan rooms and mechanical equipment rooms.
FIGURE 13-18 AIR HANDLING UNIT SCHEMATIC DIAGRAM
CHAPTER 14

AIR FILTRATION AND CLEANING
14.1 INTRODUCTION

As discussed in Chapter 1, one of the four variables that determine human comfort is air quality. Up until this chapter, the primary focus of this manual has been on maintaining the temperature, humidity, and air movement in an HVAC zone for human comfort. An air cleaner is one of the four basic components that comprise a basic HVAC system, see Figure 1-3. This chapter will address air filtration and cleaning devices that can be used to ensure air quality.

14.2 AIR FILTRATION AND CLEANING

14.2.1 Need For Air Filtration and Cleaning

Indoor air pollutants are unwanted and sometimes dangerous to building occupants. The best way to eliminate the risk of indoor air pollutants is to control the use of chemicals and other contaminants that could become indoor pollutants and ventilate the building using outside air. Unfortunately, total building ventilation with outside air may be limited due to economics, HVAC system capabilities during weather extremes, and the fact that there may be objectionable or harmful levels of contaminants in the outside air. Air filtration and cleaning is intended to reduce the need for outside air by physically removing pollutants from both HVAC return air (recirculated air) and outside air (ventilation air) before supplying the conditioned air to the zone. Air filtration and cleaning should be used in conjunction with source control and ventilation but not as a substitute. Air filtration and cleaning has limitations and these limitations must be considered and accounted for in the design of HVAC systems.

14.2.2 Categories Of Air Pollutants

Indoor air pollutants can be classified as either:

- Particulate
- Gaseous

Particulate pollutants include inanimate particulates like dust, smoke, and pollen as well as organic particulates such as mites, molds, bacteria, and viruses. Gaseous pollutants are typically byproducts of combustion or chemicals being used in the building or off gassed from building furnishings and materials. Gaseous pollutants that result from combustion includes gas cooking and vehicle exhaust. Gaseous pollutants from building materials include adhesives, paints, cleaning products, and pesticides, among others.

14.2.3 Air Filtration and Cleaning Devices

Different applications require different degrees of air filtration and cleaning effectiveness. Unfortunately, the smaller components of atmospheric dust are the worst offenders in smudging and discoloring building interiors. Electronic air cleaners or high efficiency dry filters are required for small particle removal. In clean room applications or when radioactive or other dangerous particles are present, extremely high efficiency mechanical filters should be selected.

Atmospheric dust is a complex mixture of smokes, mists, fumes, dry granular particles, and fibers. When suspended in a gas, this mixture is generically classified as an aerosol. The characteristics of aerosols most affecting the performance of an air filter include particle size and shape, specific gravity, concentration, and electrical properties. The most important of these is particle size. Air filtration and cleaning efficiency are also affected by the velocity of the air stream. The degree of air cleanliness required and aerosol concentration are both major factors influencing the design and selection of air filters and cleaners. Removal of particles becomes progressively more difficult and expensive as particle size decreases.

In addition to wide-ranging criteria affecting the degree of air cleanliness, considerations of life-cycle cost that accounts for the initial investment and ongoing operation and maintenance costs, space requirements, and airflow resistance have resulted in a wide variety of commercially available air filters and cleaners.

14.3 RATING AIR FILTERS AND CLEANERS

14.3.1 Air Filter and Cleaner Operating Characteristics

The three operating characteristics that distinguish the various types of air filters and cleaners are the following:

- Efficiency
- Airflow Resistance
- Life or Dust-Holding Capacity

14.3.1.1 Efficiency

Efficiency measures the ability of the air cleaner to remove particulate matter from an air stream. Average efficiency over the life of the filter is the most meaningful for most types and applications.
In general there are three types of tests that are employed for the determination of air cleaner efficiency:

- A standardized synthetic dust consisting of various particle sizes is fed into the air cleaner and the weight fraction of the dust removed is determined. This type of efficiency measurement is named synthetic dust weight arrestance to distinguish it from other efficiency values. The term is often abbreviated as arrestance.

- Atmospheric dust is passed into the air cleaner and the discoloration effect of the cleaned air is compared with that of the incoming air. This type of measurement is named dust spot efficiency.

- Uniform-sized particles are fed into the air cleaner and the percentage removed by the cleaner is determined. For example, particle size concentration is measured upstream and downstream of the air cleaner. The results of this test are referred to as fractional efficiency.

### 14.3.1.2 Airflow Resistance

Airflow resistance or merely resistance is the static pressure drop across the filter at a given airflow rate. The term pressure drop is used interchangeably with resistance. Dust-holding capacity defines the amount of a particular type of dust that an air cleaner can hold when it is operated at a specified airflow rate to some maximum resistance value or before its arrestance is seriously reduced as a result of the collected dust.

When rating automatic renewable media devices such as roll filters, the rating system must evaluate the rate of media supply that is needed to maintain constant resistance under a specified feed rate of standard dust. As applied in the case of ionizer-plate electronic air cleaners, the effect of dust buildup on arrestance must be evaluated. Also, it is essential in this case that the manufacturer supply information on the maintenance needed to keep the electronic air cleaner at its rated efficiency.

The measurement of pressure drop is relatively simple but the exact measurement of true dust-holding capacity is complicated by the variability of atmospheric dust. As a result, a standardized synthetic dust is normally used. The use of a standardized synthetic dust shortens the dust holding cycle to hours instead of weeks.

### 14.3.1.3 Life or Dust-Holding Capacity

Since synthetic dusts are not exactly the same as atmospheric dusts, the dust-holding capacity that is measured by these accelerated tests may be different from that achieved by tests using atmospheric dust. It is impossible to determine the exact life of a filter that is used in an actual application by laboratory testing. However, tests of filters under standard conditions do provide a reasonable guide to the relative effect of dust on performance of various units.

### 14.4 AIR POLLUTANT CAPTURE METHODS

Methods used to capture air pollutants by air filters and cleaners include the following:

- Impingement
- Straining
- Diffusional Effects
- Electrostatic Effects

#### 14.4.1 Impingement

Impingement refers to capturing particle air pollutants in the air stream when they collide with and adhere to the filter fiber as the air stream passes through an air filter. Impingement works with all size particles because there is a chance that any size pollutant particle in the air stream will collide with and adhere from a filter fiber as the air stream carrying it passes through the filter. However, impingement is most effective for larger size pollutant particles and relies on the following two mechanisms for removing these pollutant particles from the air stream.

- Interception
- Inertial Impaction

#### 14.4.1.1 Interception

Interception is the impingement mechanism where pollutant particles are captured and removed from the air stream by colliding directly with filter fibers. Interception works best when the size of the pollutant particles and the filter fiber mesh are comparable in size. Interception is probably the simplest capture method used in air filtration.
14.4.1.2 Inertial Impaction

Inertial impaction is similar to interception in that it is also removes pollutant particles from the air stream when they collide with and adhere to filter fiber. The air stream being filtered makes abrupt changes of direction as it passes around each fiber in a filter. Due to inertia, the larger pollutant particles are not always able to make these abrupt changes in direction around filter fiber as the air stream does. As a result of this inertia, the trajectory of the pollutant particles diverges from that of the air stream in which they are being carried and the pollutant particles collide with a filter fiber and are removed from the air stream. Like interception, inertial impaction works for all size particles to some degree but is most effective for larger particles at high air velocities. This is the predominant means of collection for viscous-impingement filters.

14.4.2 Straining

If the air stream must pass between fibers where the width of passage is less than particle diameter then the particle will be stopped and held. This method is most often used in the collection of large particles and lint on the surface of filters.

14.4.3 Diffusional Effects

Very fine particles are bombarded by the random motion of air molecules and are driven to the filter fibers across the general motion of the air stream. A similar effect takes place as a result of any turbulence present in the gas stream. These effects increase with decreasing particle size.

14.4.4 Electrostatic Effects

Objects with opposite electrical charge are attracted to each other. This method is employed by electronic air cleaners to attract particles to the collecting plates. Under certain circumstances, electrostatic charges may be created within fibrous filter media and will usually assist dust collection.

14.5 CATEGORIES OF AIR FILTERS AND CLEANERS

Air filters and cleaners can be categorized as follows:

- Particulate Air Filtration
- Gas-Phase Air Filtration
- Pollutant Destruction
- Particulate Air Filtration
- Gas-Phase Air Filtration
- Pollutant Destruction

14.5.1 Particulate Air Filtration

Two types of air filtration devices that physically remove particulate pollutants from the air are as follows:

- Mechanical Air Filters
- Electronic Air Cleaners

14.5.1.1 Mechanical Air Filters

Mechanical air filters remove particles from the air stream by capturing them on filter materials. Mechanical air filters can capture dust, pollen, dust mites, and some molds. Mechanical filters are rated in accordance with the efficiency by which they remove particulate pollutants from the air stream that passes through it. Types of mechanical air filters encountered in HVAC systems include the following:

- Panel Air Filters
- Renewable Media Filters
- Washable Air Filters
- Extended Surface Filters
- High-Efficiency Particulate Filtration
- Ultra-Low Penetration Filtration
- Super Ultra-Low Penetration Filtration

The efficiency of a mechanical air filter is determined by the filter’s minimum efficiency reporting value (MERV). Table 14-1 provides a table showing MERV filter ratings, comparable dust spot efficiencies, arrestance, typical controlled contaminant, typical applications and limitations, and typical air filter or cleaner type. Flat or panel air filters with a MERV of 1 to 4 are usually used in residential furnaces and air conditioners. Medium efficiency filters that have a MERV of 5 to 13 are reasonably efficient at removing both small and large particulate pollutants.

Pleated or extended surface filters are typically used in commercial and institutional air distribution systems. Medium efficiency filters that have a MERV of between 7 and 13 are generally less expensive than high efficiency particulate air (HEPA) filters but can be nearly as efficient at removing particulate pollutants while reducing the expected pressure across a HEPA filter. Using a filter with a MERV rating between 7 and 13 in lieu of a HEPA filter should result in increased airflow and a resulting reduction in fan power and noise.
<table>
<thead>
<tr>
<th>Standard 52 Minimum Efficiency Reporting Value</th>
<th>Dust Spot Efficiency</th>
<th>Arrestance</th>
<th>Typical Controlled Containment</th>
<th>Typical Applications and Limitations</th>
<th>Typical Air Filter/ Cleaner Type</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>n/a</td>
<td>n/a</td>
<td>&lt;0.30 pm Particle Size</td>
<td>Cleanrooms</td>
<td>&gt;99.99% efficiency on 10-20 pm Particles</td>
</tr>
<tr>
<td>19</td>
<td>n/a</td>
<td>n/a</td>
<td>Virus (unattached)</td>
<td>Radioactive Materials</td>
<td></td>
</tr>
<tr>
<td>18</td>
<td>n/a</td>
<td>n/a</td>
<td>Carbon Dust</td>
<td>Pharmaceutical Materials</td>
<td>Particulates</td>
</tr>
<tr>
<td>17</td>
<td>n/a</td>
<td>n/a</td>
<td>All Combustion smoke</td>
<td>Carcinogenic Materials</td>
<td>&gt;99.97% efficiency on .3 pm Particles</td>
</tr>
<tr>
<td>16</td>
<td>n/a</td>
<td>n/a</td>
<td>.30 - 1.0 pm Particle Size</td>
<td>General Surgery</td>
<td></td>
</tr>
<tr>
<td>15</td>
<td>&gt;95%</td>
<td>n/a</td>
<td>All Bacteria</td>
<td>Hospital Impatient Care</td>
<td></td>
</tr>
<tr>
<td>14</td>
<td>90 - 95%</td>
<td>&gt;98%</td>
<td>Most Tobacco Smoke</td>
<td>Smoking Lungs</td>
<td></td>
</tr>
<tr>
<td>13</td>
<td>89 - 90%</td>
<td>&gt;98%</td>
<td>Proplet Nuclei (Sneeze)</td>
<td>Superior Commercial Buildings</td>
<td></td>
</tr>
<tr>
<td>12</td>
<td>70 - 75%</td>
<td>&gt;95%</td>
<td>1.0 - 3.0 pm Particle Size</td>
<td>Superior Residential</td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>60 - 65%</td>
<td>&gt;95%</td>
<td>Humidifier Dust Lead Dust</td>
<td>Better Commercial Buildings</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>50 - 55%</td>
<td>&gt;95%</td>
<td>Milled Flour</td>
<td>Hospital Laboratories</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>40 - 45%</td>
<td>&gt;90%</td>
<td>Welding Fumes</td>
<td>Commercial Buildings</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>30 - 35%</td>
<td>&gt;90%</td>
<td>3.0 - 10.0 pm Particle Size</td>
<td>Commercial Buildings</td>
<td>Pleated Filters - Disposable, extended surface area, thick with cotton-polyester blend media, cardboard frame.</td>
</tr>
<tr>
<td>7</td>
<td>25 - 30%</td>
<td>&gt;90%</td>
<td>Mold Spores Hair Spray</td>
<td>Better Residential</td>
<td>Cartridge filters - Graded density viscous coated cube or pocket filters, synthetic media.</td>
</tr>
<tr>
<td>6</td>
<td>&lt;20%</td>
<td>85 - 90%</td>
<td>Fabric Protector Dusting Aids</td>
<td>Industrial Workplace</td>
<td>Throwaway - Disposable synthetic panel filter.</td>
</tr>
<tr>
<td>5</td>
<td>&lt;20%</td>
<td>80 - 85%</td>
<td>Cement Dust Pudding Mix</td>
<td>Paint Booth Inlet</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>&lt;20%</td>
<td>75 - 80%</td>
<td>&gt; 10.0 pm Particle Size Pollen</td>
<td>Minimal filtration</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>&lt;20%</td>
<td>70 - 75%</td>
<td>Dust Mites Sanding Dust</td>
<td>Residential</td>
<td>Washable - Aluminum Mesh</td>
</tr>
<tr>
<td>2</td>
<td>&lt;20%</td>
<td>65 - 70%</td>
<td>Spray Paint Dust</td>
<td>Window A/C Units</td>
<td>Electrostatic - Self charging woven panel filter.</td>
</tr>
<tr>
<td>1</td>
<td>&lt;20%</td>
<td>&lt;65%</td>
<td>Textile Fibers Carpet Fibers</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 14–1 Mechanical Air Filter MERV Rating Information

Source: Air Filters, Inc. (www.airfilterusa.com)
Higher efficiency filters with a MERV of 14 to 16 are similar in physical appearance to HEPA filters. HEPA filters have a MERV rating from 17 to 20.

MERV ratings for mechanical filters are determined using the testing procedure specified in the American Society of Heating, Refrigerating, and Air Conditioning Engineers (ASHRAE) Standard 52.2 entitled Method of Testing General Ventilation Air-Cleaning Devices for Removal Efficiency by Particle Size. Using ASHRAE Standard 52.2, the MERV rating of a mechanical filter is determined by adding particles of varying sizes to a controlled airstream passing through the filter being tested. A laser particle counter samples the air as it enters and leaves the filter under test. The entering and exiting particle counts are then compared to calculate the particle size efficiency of the filter in accordance with ASHRAE Standard 52.2. From this calculated particle size efficiency, the MERV rating for the filter is determined in accordance with the standard.

14.5.1.2 Electronic Air Cleaners

Electronic air cleaners use electrostatic attraction to trap and remove particulate pollutants from the airstream. Typical electronic air cleaners used in HVAC systems include the following:

- Washable Electronic Air Cleaners
- Agglomerator Electronic Air Cleaners
- Self-Contained Electronic Air Cleaners

As air flows through the ionization section of an electronic air cleaner, particles receive an electrostatic charge. The charged particles are then attracted to and accumulate on a series of flat plates referred to as the collector and it carries an opposite charge from the ionization section. Electronic air cleaners that operate as described in this section are sometimes referred to as electrostatic precipitators.

Unlike mechanical air filters, there is no standard for measuring the effectiveness of an electronic air cleaner. Electronic air cleaners are very good at removing small particles from the airstream but not as efficient at removing larger particulates. Electronic air cleaners can also produce ozone which is a lung irritant.

14.5.2 Gas-Phase Air Filtration

Gas-phase air filtration removes gaseous pollutants from the air stream by using a material referred to as a sorbent. Typical types of gas-phase air filtration devices used in HVAC systems include the following:

- Activated-Carbon Air Filtration
- Chemically-Impregnated Adsorption Air Filtration
- Catalytic-Adsorption Air Filtration

A commonly used sorbent is activated carbon or charcoal that absorbs the gaseous pollutants as they pass through the gas-phase air filter. Sorbents used in gas-phase air filtration will only absorb certain pollutants and will not reduce the concentrations of pollutants that they are not designed to absorb. When considering the use of gas-phase air filtration, always be sure that the sorbent selected will absorb the anticipated contaminant and reduce the concentration to an acceptable level.

14.5.3 Pollutant Destruction

In the case of living particulates it is often desired to destroy the particulate rather than just capture and retain it in a filter. The most common method for destroying living particulates used in commercial and institutional buildings is ultraviolet germicidal irradiation (UVGI) which uses ultraviolet light. UVGI systems are available that will destroy many biological pollutants including viruses, bacteria, allergens, and molds that are airborne or found on HVAC cooling coils, drain pans, and ductwork. UVGI is not a substitute for mechanical air filters and UVGI should be used in conjunction and downstream from mechanical filters. UVGI is also used to destroy biological pollutants in domestic water and deionized water (DI) systems.

14.6 PANEL FILTERS

14.6.1 Viscous Impingement Filters

Viscous impingement filters are flat panel filters made up of coarse fibers with a high porosity. The filter media are coated with a viscous substance such as oil that acts as an adhesive on particles that impinge on the fibers. Design air velocity through the media is usually in the range of 200 to 800 fpm (1 to 4 m/s). These filters are characterized by low-pressure drop, low cost, and good efficiency on lint but low efficiency on normal atmospheric dust.

A number of different materials have been used as the filtering medium including coarse glass fibers, animal hair, vegetable fibers, synthetic fibers, metallic wools, expanded metals and foils, crimped screens, ran-
dom-matted wire, and synthetic open cell foams. Viscous impingement filters are commonly made ½ to 4 in. (13 to 102 mm) thick with 1 to 2 in. (25 to 50 mm) nominal thickness being most popular. In the thicker configurations, the dust-holding capacity can be substantial. Unit panels are available in both standard and special sizes up to about 24 by 24 in. (600 by 600 mm). This type of filter is often used as a prefilter to higher efficiency filters.

The loading rate of a filter depends on the type and concentration of the dirt in the air being handled and the operating cycle of the system. Manometers or draft gauges are often installed to measure the pressure drop across the filter bank and thereby indicate when the filter requires servicing. The allowable pressure drop varies from one installation to another but unit filters should be serviced when their operating resistance reaches 0.50 in. wg (125 Pa) or per manufacturer instruction. For viscous impingement filters the decline in filter efficiency that occurs when all the viscous coating has been absorbed by the collected dust may be the limiting factor in operating life rather than the increased resistance caused by dust load.

The manner of servicing unit filters depends on their construction. Disposable filters are constructed of inexpensive materials and are intended to be discarded after one period of use. Permanent unit filters are generally constructed of metal to withstand repeated handlings. Various cleaning methods have been recommended for filters of this design. The most widely used cleaning method for permanent unit filters involves washing the filter with steam, water, or detergent and then recoating the filter with its recommended adhesive by dipping or spraying.

To provide uniform operating resistance, especially where large banks are involved, it is sometimes advisable to use the rotation system of filter maintenance. In this procedure, a portion of the filter bank that is typically 20 to 25 percent is reconditioned on an established schedule. Unit viscous filters are also sometimes arranged for in-place washing. With in-place washing, cleaning is accomplished by hot water sprayed from a hose or the use of fixed or moving nozzles. At the completion of the washing cycle, the necessary adhesive is applied. Where filters are washed in place, provision should be made to collect and drain the water to prevent leakage from the filter housing.

14.6.2 Dry-Type Extended-Surface Filters

The media used in dry-type air filters are random fiber mats or blankets of varying thicknesses, fiber sizes, and densities. Media of bonded glass fiber, cellulose fibers, wool felt, synthetics, and other materials have been used commercially. The medium in filters of this class is frequently supported by a wire frame in the form of pockets or V-shaped pleats. In other designs, the medium may be self-supporting because of inherent rigidity or because airflow inflates it into extended form. Pleating of the medium provides a high ratio of medium area to face area that moderates pressure drop in spite of the medium's density and fineness. In some designs, the filter media are replaceable and are held in position in permanent metal cell sides. In other designs, the entire cell is disposed of after it has accumulated its dirt load.

The efficiency of dry-type air filters is usually higher than that of panel filters, and the variety of media available makes it possible to furnish almost any degree of cleaning efficiency desired. Dry-type filter media and filter configurations also have dust-holding capacities that are typically higher than panel filters. Coarse prefiltrers placed ahead of extended surface filters may be economically justified by the even longer life they add to the main filters by capturing larger particles and lint.

The initial resistance of an extended surface filter varies with the choice of media and the filter geometry. Commercial designs typically have an initial resistance from 0.5 to 1.00 in. wg (125 to 250 Pa). It is customary to replace the media when the resistance doubles. Dry media providing higher orders of cleaning efficiency will offer a higher resistance to airflow. The operating resistance of the fully dust loaded filter must be considered in the system design since that is the maximum resistance against which the fan will be called upon to operate. If the variation between initial and final resistance is sufficient to cause a problem in system design, the filters may be replaced on the rotation system discussed in the section on viscous-impingement unit filters.

Flat panel filters with media velocity equal to duct velocity are possible only in the lowest efficiency units of the dry type with open cell foams and textile denier non-woven media.

14.6.3 Intermediate Efficiency Filters

For dry filters with an intermediate efficiency range, the filter media area is much greater than the face area of the filter. Therefore, the velocity through the filter
media is substantially lower than the velocity approaching the filter face. Media velocities range from 6 to 90 fpm (0.030 to 0.46 m/s), although the approach velocities may be as high as 750 fpm (3.8 m/s). Depth in direction of airflow varies from 2 to 36 in. (50 to 914 mm).

Filter media used in the intermediate efficiency range include the following materials:

- Fine glass fibers.
- Thin non-woven mats of fine glass fibers, cellulose, or cotton wadding.
- Non-woven mats of comparatively large diameter fibers.

### 14.6.4 HEPA Filters

Dry filters of very high efficiency are referred to as high efficiency particulate air (HEPA) filters. These filters are made in an extended surface configuration of deep space folds of submicron glass fiber paper. HEPA filters operate at face velocities near 250 fpm (1.3 m/s) with a resistance rising from 0.50 to 1.00 in. wg (125 to 250 Pa) or more over their service life. These filters are the standard for clean room, nuclear, and toxic-particulate applications.

### 14.7 RENEWABLE MEDIA FILTERS

#### 14.7.1 Moving-Curtain Viscous Impingement Filters

Automatic moving-curtain viscous filters are available in two main types. In one type, random-fiber medium is furnished in roll form. Fresh medium is automatically fed across the face of the filter frame, while the dirty medium is rewound onto a roll at the bottom. When the roll is exhausted, the tail of the medium is wound onto the take-up roll and the entire roll is thrown away. A new roll is then installed and the cycle is repeated.

Moving-curtain filters have the medium automatically advanced by motor drives on command from a timer or pressure switch. A pressure switch control measures the pressure drop across the medium and switches on and off at chosen upper and lower set points. This control conserves the medium but care should be taken when locating static pressure tubes so that the true pressure drop is being sensed. In addition, modulating outside air and return air dampers can affect airflow patterns and total flow to such an extent that pressure drop control systems may not function properly. Timers help to avoid these problems and their duty cycles can be adjusted to provide satisfactory operation with acceptable media consumption.

With the second type of automatic viscous-impingement filter, linked media are installed on a traveling curtain that intermittently passes through an adhesive reservoir. In the adhesive reservoir the media drop load and takes on a new coating of adhesive. The media forms a continuous curtain that moves up one face and down the other face. A media curtain made of metal that is continually cleaned and renewed with fresh adhesive should last the life of the filter mechanism.

Periodically, the precipitated dirt must be removed from the adhesive reservoir. This is accomplished by scraping the dirt into a tray that can be conveniently suspended from the reservoir lip. Where it is desirable to eliminate the maintenance required to remove the precipitated dirt from the reservoir, the adhesive may be pumped through oil clarifiers or be allowed to circulate through settling tanks.

Most automatic viscous filters are equipped with a fractional horsepower motor operating the drive mechanism through a gear reducer. A timer control will make the operating period adjustable so that the media travel can be matched to dust concentration. The resistance of an automatic filter will remain approximately constant as long as proper operation is obtained. A resistance of 0.40 to 0.50 in. wg (100 to 125 Pa) at a face velocity of 500 fpm (2.5 m/s) is typical.

#### 14.7.2 Moving-Curtain Dry-Media Filters

Random-fiber or non-woven dry media of relatively high porosity are also used in moving-curtain or roll filters for general ventilation service. Operating face velocities are generally lower than for viscous-impingement filters at about 200 fpm (1.02 m/s). Special automatic dry filters are also available which are designed for the removal of lint in textile mills and dry cleaning establishments and the collection of lint and ink mist in pressrooms. The medium used is extremely thin and serves only as a base for the buildup of lint that also acts as a filter medium.

In the previous forms of renewable dry-media filters, the dirt-laden media are discarded when the supply roll is used up. Another form of filter designed specifically for dry lint removal consists of a moving curtain of wire screen that is automatically vacuum cleaned at a position out of the air stream. Recovery of the collected lint is sometimes possible with such a device.
14.7.3 **Performance Of Renewable Media Filters**

Dust-holding capacity and efficiency are equally important for renewable media filters as for unit filters. Test methods and determination of arrestance and efficiency are the same as for unit filters. However, the test procedure calls for fresh curtain or media to be advanced into the air stream when resistance reaches the manufacturer's design level, typically between 0.40 to 0.50 in. wg (100 to 125 Pa). Cycles of dust feeding and curtain advance are repeated until about two complete duct heights of curtain have been advanced. Arrestance and efficiency values are obtained at intervals during this process to establish steady-state values. Dust-holding capacity in this case is defined as steady-state arrestance multiplied by dust loading per unit area of media.

14.8 **ELECTRONIC AIR CLEANERS**

14.8.1 **Electronic Air Cleaner Operation**

Electronic air cleaners use electrostatic precipitation principles to collect particulate matter but they operate on lower voltages than the type commonly used on industrial stacks. The designation electronic air cleaner has been standardized to distinguish the class of electrostatic precipitator that alone is suitable for cleaning ventilating air. The following three types of electronic air cleaners are used in HVAC systems:

- Ionizing-Plate
- Charged-Media Nonionizing
- Charged-Media Ionizing

14.8.1.1 **Ionizing-Plate**

In this type of filter, positive ions are generated at the high potential ionizer wire. These positive ions flow across the air stream, striking and adhering to any dust particles carried by the air stream. These particles then pass into the system of charged and grounded plates. They are driven to the plates by the force exerted by the electric field on the charges they carry and are captured by the plates are removed from the air stream.

In a typical electronic air cleaner design, a direct current potential of 12,000 volts is used to create the ionizing field and 6000 volts is maintained between the plates where captured dust is precipitated. These voltages require some safety measures. A typical arrangement provides means for making the equipment inoperative when any door affording access to high-voltage parts is opened. Operation can be resumed only after all doors are closed. The voltages necessary for operation of the equipment are usually obtained by means of high-voltage, direct-current power packs operating from a standard 120-volt, single-phase branch circuit.

Air cleaners of this type, unless incorporating prefilters or after-filters, offer negligible resistance to airflow. Therefore, care must be exercised in arranging the duct approaches on the entering and leaving sides of the cleaners so that the airflow is distributed uniformly over the entire cross-sectional area. The efficiency of the electronic air cleaner is sensitive to air velocity, and the device itself has much less tendency to balance air stream velocity differences than do filters which present higher resistances.

In most systems, prefilters are used with electronic air cleaners. The resistance, including the filters generally ranges from 0.14 to 0.26 in. wg (35 to 63 Pa) at velocities of 300 to 500 fpm (1.5 to 2.5 m/s). Suitable screens of no coarser than 16 mesh should be installed across all outdoor air entrances to prevent insects, leaves, bits of paper, and similar material from entering the cleaner. Where lint is present in appreciable quantities such as in recirculated air, devices for lint removal should be installed ahead of the cleaner with provision for cleaning the lint collector periodically.

Electronic air cleaners of the ionizing type are efficient, low-pressure drop devices for removing fine dust and smoke particles. Collector plates are often coated with special oil as an adhesive. Cleaning is generally accomplished by washing the cells in place with hot water from a water hose or using a fixed or moving nozzle system. The bottom of the equipment is made watertight and is provided with a drain.

Electrical forces drive particles to the collecting surface. After a particle touches the collecting surface, primarily intermolecular adhesion forces may hold it. Therefore, it is very important with the washed type of electronic air cleaner to be sure that either the dust is naturally adherent or that the plates are always covered with adhesive.

Electronic air cleaners, however, are often used without any adhesive treatment on the plates. Under such conditions, the precipitator may form agglomerates that eventually blow off the plates. In these situations, a secondary filter or storage section downstream should follow the electronic air cleaner. The dry agglomerates produced in the precipitator are allowed to blow off and be caught by the downstream filter. The use of an automatic replaceable media filter...
for catching these agglomerates results is an overall combination which provides both a high degree of cleaning efficiency and the convenient maintenance associated with an automatic filter of this design.

Another type of electronic cleaner uses a high efficiency dry filter as the collector. In this version, the ability of the electronic air cleaner portion to agglomerate very small particles is utilized to increase the life of the high efficiency filter. This works because the weight of dust held by dry filters increases with increasing particle size of the collected dust. When handling certain types of atmospheric dust, the ionizing wires and struts may require periodic cleaning.

### 14.8.1.2 Charged Media Nonionizing

The charged media air cleaner combines certain characteristics of both dry filters and electronic air cleaners. It consists of a dielectric filtering medium that is usually arranged in pleats like a typical dry filter. No ionization means is employed. The dielectric filtering medium may consist of glass fiber mat, cellulose mat, or other similar material. The dielectric filtering medium is supported on or in contact with a grid work consisting of alternately grounded and charged members. Charged members are usually maintained at a direct current (DC) voltage of 12,000 volts. As a result, an intense and nonuniform electrostatic field is created through the dielectric medium. Airborne particles approaching the field are polarized and drawn to filaments of fibers of the media.

A cleaner of this type offers resistance to airflow when clean of about 0.10 in. wg (25 Pa) at 250 fpm (1.3 m/s). The resistance of the charged media electronic cleaner rises as dust is accumulated on the media. As a result of these characteristics, the filter tends to equalize the air distribution over its face. Like typical replaceable media mechanical filters, the charged media precipitator is serviced by replacing the filtering medium. The dielectric properties of the media are impaired when the relative humidity exceeds 70 percent.

### 14.8.1.3 Charged Media Ionizing

A charged media ionizing electronic air cleaner combines the effects used in the two types previously described. Dust is charged in a corona-discharge ionizer and then collected on a charged media filter mat. This arrangement provides higher efficiency than its charged media would without an ionizer. Such filters must, however, be carefully designed to minimize space-charge effects.

### 14.8.2 Space Charge

Dust which passes through an ionizer, that is charged but not removed, carries the electrical charge into the building’s space. If continued on a large scale, a space charge will be built up which tends to drive this charged dust to the walls. As a result, an electronic air cleaner which for any reason charges but does not remove the dust, can blacken walls faster than if no cleaning device were used.

### 14.8.3 Ozone

All high voltage devices are capable of producing ozone that is both toxic and corrosive. Properly designed and maintained, electronic air cleaners produce ozone concentrations that are only a fraction of the levels acceptable for continuous human exposure. Continuous arcing and brush discharge in an electronic air cleaner may yield ozone levels that are annoying or mildly toxic and will be indicated by a very strong ozone odor. Although the nose is very sensitive to ozone only actual measurement of the concentration can determine that a hazardous condition exists.

### 14.9 Air Cleaner and Filter Location

Many varieties of air cleaners and filters are available with different filtering efficiencies. Unless the filters are maintained regularly, system static resistance increases and airflow is diminished. The primary consideration in the selection and location of filters is that they should be readily accessible. In a built-up system, there should be a minimum of three feet between the upstream face of the filter bank and any obstruction.

Good mixing of outdoor and return air is also necessary for good filter performance. A sample of atmospheric dust gathered at any given point will generally contain materials common to that locality, together with other components that originated at a distance but were transported by air currents or diffusion. These components vary with the geography of the locality, the season of the year, the direction and strength of the wind, and proximity of dust sources.

A poorly placed outside air duct or duct connection to the mixing plenum can cause uneven loading of the filters and bad distribution of air through the coil section. As a result of the low resistance of the partially clean areas, the filter gage may not indicate that this condition exists.

Air filters are often supplied as part of packaged HVAC units. However, built-up systems often require
a thorough analysis of air filtration components and how the HVAC system is to be used. Due to energy conservation considerations, odor removal or control equipment is used where excessive dilution of the air by ventilation may be too costly.

14.10 FILTER INSTALLATION

The in-service efficiency of an air filter is sharply reduced if air leaks around it through either leaky bypass dampers or poorly designed or unsealed frames. The higher the efficiency of the filters, the more attention that must be paid to the rigidity and sealing effectiveness of the filter frame. In addition, high efficiency filters must be handled and installed with extreme care.

Air cleaners should be installed in the outside air intake ducts of buildings and residences and in the recirculation and bypass air ducts as well. Cleaners are logically placed ahead of heating or cooling coils and other air-conditioning equipment in the system to protect them from dust. The dust captured by the filters in an air intake duct is likely to be mostly larger particulate matter while lint may predominate within the building.

When HEPA filters are employed to protect critical areas such as clean rooms, it is important that the filters be installed as close to the room as possible to prevent the pickup of particles between the filters and the outlet. The ultimate in this trend is the so-called laminar flow room, in which the entire ceiling or one entire wall becomes the final filter bank.

The published performance data for all air filters are based on straight-through unrestricted airflow. Filters should be installed so that the face area is at right angles to the air flow. Eddy currents created by nearby fittings or obstacles in the airstream and dead air spaces should be avoided. Air should be distributed uniformly over the entire filter surface using baffles or diffusers if necessary. Filters can be damaged if higher than normal air velocities impinge directly on the face of the filter.

Failure of a well-designed air filter installations to give satisfactory results can, in most cases, be traced to faulty installation or improper maintenance or both. The most important requirements of a satisfactory and efficiently operating air filter installation are as follows:

- The filter must be of ample size for the amount of air and dust load it is expected to handle. An overload beyond design of 10 to 15 percent should be regarded as the maximum allowable. When air volume is subject to future increase, a larger filter should be installed.

- The filter must be suited to the operating conditions such as the degree of air cleanliness required, amount of dust in the entering air, type of duty, moisture content of the air, allowable pressure drop, operating temperatures, and maintenance facilities.

- The filter type should be the most economical for the specific application. The initial cost of the installation should be balanced against efficiency and depreciation as well as expense and convenience of maintenance.

The following recommendations apply to filters installed with central fan systems:

- Duct connections to and from the filter should change size or shape gradually to assure even air distribution over the entire filter area.

- Access doors of convenient size should be provided to the filter service areas.

- All access/inspection doors should be gasketed to prevent infiltration of unclean air. All connections and seams of the ducts on the clean air side (downstream of filters) should be sealed as airtight as possible. Any filter bank must be gasketed to prevent bypass of unfiltered air. This is increasingly important as the efficiency filters used increases.

- Electric lights should be installed in the chamber in front of and behind the air filter.

- Filters installed close to an outside air inlet should be protected from the weather by suitable louvers with a built-in, large-mesh wire screen.

- Filters, other than electronic air cleaners, should have permanent indicators to give a warning when the filter resistance reaches too
14.11 HVAC Systems Applications

• Electronic air cleaners should have an indicator or alarm system to indicate when the power supply is not on.

14.11 ODOR REMOVAL

14.11.1 Odor Removal Overview

When the concentration of odorous vapors is insufficient for perception, the air is said to be odor-free. Therefore, to abate odor, it is necessary to either:

• Remove or adequately reduce concentration of offending gases or vapors.
• Interfere with perception by odor modification.

There are many sources of discomfort in occupancy areas. Air from outdoors may contain automotive exhaust, furnace effluents, industrial effluents, or smog. Industrial spaces may have odors from chemical products such as printing ink, dyes, or synthetics and manufactured rubber products. Offices, arenas and other enclosed spaces may contain objectionable body odors.

Odors can also result from wetted air-conditioning coils that become dirty or metals and coatings used on coils that increase the possibility of objectionable odors. Linoleum, paint, upholstery, rugs, drapes, or other household furnishings may also cause odors. Food as well as cooking and putrefaction of animal and vegetable matter can produce objectionable odors.

14.11.2 Odor Removal Methods

Odor removal can be accomplished by physical or chemical means. Available methods include:

• Ventilation With Outdoor Air
• Washing and Scrubbing
• Condensation Absorption
• Chemical Oxidation
• Odor Modification
• Chemical Absorption
• Vapor Neutralization
• Combustion

Air washing or scrubbing and combustion are of considerable value to process industries. Washing or scrubbing, like filtering, can be applied to remove particulates and in some cases as a means to recover valuable product. Odors associated with the particulates are removed indirectly in such processes. Combustion of obnoxious gases is practicable and effective where economical. These methods alleviate the effects of harmful exhaust gases and particulates on people, vegetation, and property.

Ventilation, adsorption, chemical reaction, and odor modification are used effectively in HVAC systems to remove odor. Removal of odorous gases or vapors with water sprays in HVAC systems is most practicable for water-soluble odors such as ammonia but not for water-insoluble odors. However, disposal of certain types of collected odorants is becoming increasingly difficult.

14.11.2.1 Ventilation With Outdoor Air

When using outside air for ventilation, objectionable and gaseous odors, irritants, particulates are diluted or replaced by the outdoor air. Due to the fact that outdoor air adds substantially to the heating and cooling load, it is desirable to minimize ventilation requirements. However, increasing air pollution and overbuilding have reduced the availability of clean outdoor air to use for ventilation in many urban areas.

In planning the use of ventilation air for odor control, consideration should be given to the increase in heat load and the availability of adequate clean air. The required diluting volume of ventilation air for any given type of installation can be used as a comparative guide to other performance requirements as an alternative odor-controlling method.

14.11.2.2 Washing and Scrubbing

Where odorous vapors are soluble or emulsifiable in a liquid, with or without chemical reaction, odor removal by liquid absorbents may be suitable. Absorbents are water, modified water solutions, or oils. Odorous vapors odors can be removed by either spray washers, packed scrubbers, or wet cooling coils. However, the usual dry filter cannot remove odorous vapors unless the filter is made of special adsorbent materials.

Over time, the absorbing media becomes saturated with the contaminant. The system can be maintained by discarding the saturated liquid, by continuous regeneration of the liquid by heating and aeration with.
clean air, or by adding reactant or absorbent chemicals that regenerate the absorbing liquid as it accumulates odors.

14.11.2.3 Chemical Oxidation

Odors are generally destroyed by oxidation. While oxidizing gases such as ozone and chlorine can oxidize odors in water, concentrations required for air deodorization would be so high that they would be toxic to occupants. The major effect of ozone generators is to reduce sensitivity of the sense of smell rather than reduce actual odor concentration.

Two types of oxidizing systems are commercially available:

- Washer System
- Solid System

In the washer system, odorous air is scrubbed with aqueous solutions of potassium permanganate, chlorine, or metal ion salts. In the solid system, potassium permanganate is embodied in a pelletized activated alumina substrate and applied to the air stream similar to that used for activated carbon. Carbons impregnated with reactant chemicals are also available.

In the aqueous system, odors scrubbed from the air react with the oxidizing chemicals and may be converted to carbon dioxide and water or to other intermediate decomposition products that are less objectionable. The oxidizing chemicals are replaced periodically as they are consumed. In the solid oxidizing system, the activated substrate adsorbs odors and moisture from the air. The adsorbed moisture solubilizes the permanganate and activates the oxidizing system. As in the washer system, odors are converted to less objectionable compounds and generally fixed in the pellets. These pellets or impregnated carbons are replaced periodically.

Performance can be varied according to need by modifying the thickness, face air velocity, total porosity, and pore size distribution of the activated substrate. Carbons impregnated with reagent chemicals are available for specific contaminants. When the activated charcoal has adsorbed its full odor capacity, it is removed and replaced with a fresh bed of material. Some may be of a material and construction that can be regenerated for reuse.

14.11.2.4 Odor Modification

An alternative to removal or destruction of objectionable odorants is to introduce other chemicals that will either:

- Modify perceived odor quality to make it more acceptable.
- Reduce perceived odor intensity to an acceptable level.

The first alternative is accomplished by the well-known technique of odor quality masking. The second method is accomplished by a phenomenon called counteraction. A mixture of odorants comprised of a malodorant and counteracting agent generally smells less intense than the sum of the intensities of the unmixed components. Given the correct proportions, the mixture may smell less intense than the malodorant alone. The general guidelines for using counteractants to mitigate the impact of a malodorant are as follows:

- Counteractants should not be used as a substitute for ventilation.
- Counteractants will probably work best against weak malodors and should be used only after other countermeasures, such as ventilation, have reduced the odor to a low level.
- Counteractants are usually quite odorous themselves and are formulated to mask as well as to counteract.
- Insofar as quality masking is one of their modes of action, the perceived quality of the counteractant should be chosen with care.
- The counteractant should not be permitted to mask or otherwise interfere with warning odors such as the odor of leaking propane or natural gas.

14.11.2.5 Chemical Absorption

Adsorption is the physical condensation of a gas or vapor on an activated, solid substrate. While polar adsorbents such as activated alumina and silica gel have been used, relatively nonpolar activated charcoals or carbons are more common. These activated carbons are available from a number of manufacturers and are made from such diverse materials as coal, coconut shells, peat and petroleum residues that are produced with high internal porosity. The activated material is placed in filters through which the air passes.
15.1 INTRODUCTION

Hydronics refers to HVAC systems that use water as the heat transfer medium. Hydronic distribution systems distribute chilled water, hot water, or steam for the purpose of conditioning building zones. This chapter will focus on chilled and hot water distribution systems because these distribution systems are the most common in commercial and institutional buildings that have central heating and cooling plants. Steam distribution systems are still encountered in older buildings but for the most part steam distribution systems are used in multi-building complexes where a central heating plant serves multiple buildings or a building is connected to a municipal steam supply.

Hydronic distribution systems can be used to condition building zones in the following ways:

- Convection terminal units that are placed in the zone and heat or cool the air that comes in contact with them. An example of a convection terminal unit is a simple steam or hot water radiator.
- Radiant heating system that conditions a zone by heating or cooling the room surfaces that surround the zone.
- Water-to-air heat exchangers, which are often referred to as coils, condition a zone by heating or cooling the air delivered to the zone in central air handling units (AHUs) or air distribution equipment such as variable-air-volume (VAV) terminal units.
- Combination of these methods.

This chapter will cover convection terminal units because they are an integral part of the hydronic distribution system and heat or cool the space directly. Radiant heating systems often use hydronic piping embedded in room surfaces to condition building zones but other energy sources can also be used such as electric resistance heating to condition the space. Similarly, water-to-air heat exchangers are used throughout buildings to condition supply air and are addressed throughout his manual.

15.2 HOT WATER DISTRIBUTION SYSTEM OPERATION

Figure 15-1 shows a simple hot water distribution system. The boiler provides hot water that is pumped through the hot water supply piping system by the hot water pump, see Figure 15-1. The hot water enters the convection terminal unit that is essentially a water-to-air heat exchanger, passes through the heat exchanger and exits the convection terminal unit. Cool air enters the convection terminal unit and is warmed by absorbing heat energy from the hot water as the air passes through the water-to-air heat exchanger. The warm air then exits the convection terminal unit and mixes with the existing air in the zone and warms it. The hot water leaves the convection terminal unit at a lower temperature and returns to the boiler through the hot water return piping to be reheated and recirculated.

This chapter focuses on the hydronic distribution system that circulated the hot water from the central heating plant. Chapter 10 covers central heating plants and addresses boilers and associate equipment in detail.

15.3 CHILLED WATER DISTRIBUTION SYSTEM OPERATION

Figure 15-2 illustrates a simple chilled water distribution system that might be encountered in a commercial or institutional building. The chiller is the heart of the chilled water distribution system, see Figure 15-2. The purpose of a chiller is to extract heat from the building by supplying chilled water for circulation through water-to-air heat exchangers located throughout the building. The chilled water piping as well as the piping associated with the cooling tower including pumps and other equipment comprises the chilled water distribution system.

Chilled water in the chiller’s evaporator loop is pumped through the chilled water supply piping by the chilled water supply pump. The chilled water supply passes through the convection terminal unit that is essentially a water-to-air heat exchanger. As warm air passes through the convection terminal unit the chilled water absorbs heat from the air and cools it. The cool air supplied by the convection terminal unit then mixes with the existing air in the zone to cool the zone. The chilled water is supplied by the chiller and enters the convection terminal unit at approximately 45°F (7.2°C), absorbs heat from the warmer air passing through convection terminal unit, and exits the convection terminal unit typically 10°F (5.6°C) warmer or approximately 55°F (12.8°C). The water then returns to the chiller through the chilled water return piping to be chilled down again and recirculated.

There is a second hydronic-piping loop in Figure 15-2 that runs between the condenser and the cooling tower outside the building and those loops are covered in Chapter 9.
FIGURE 15-1 SIMPLE HOT WATER DISTRIBUTION SYSTEM
FIGURE 15−2 SIMPLE CHILLED WATER DISTRIBUTION SYSTEM
15.4 HYDRONIC DISTRIBUTION SYSTEM ADVANTAGES AND DISADVANTAGES

15.4.1 Hydronic Distribution System Advantages

A major advantage of using a hydronic distribution system with convection terminal units is the reduction in building space when compared to an air distribution system that requires fan rooms to house air handling units and duct to transport the conditioned air. A hydronic distribution system provides all of the benefits of a central cooling and heating plant while retaining the ability to positively shut off local terminals in unused building spaces. An all-hydraulic HVAC system using convection terminal units usually provides individual room control with no cross-contamination of recirculated air from one zone to another. Extra capacity for quick pull-down response may also be provided.

Since hydronic distribution systems are capable of heating with low temperature water they are particularly suitable for solar or heat recovery refrigeration equipment applications. For existing building retrofit, it is often easier to install a hydronic distribution system than the ductwork required for air distribution systems. Hot water used as a heat transfer medium is economical because water:

- Provides heat along entire outdoor exposures by using convection terminal units such as finned-tube radiation heaters.
- Responds quickly but uniformly to load changes using minimum pipe sizes.
- Permits piping to run at any pitch or level, up or down, to match building or site configuration.
- Provides thermal inertia that helps balance diverse system load requirements from all building zones with uniform input at fuel burners.

In addition, the ability to reset system supply water temperatures with respect to outdoor air temperature or the building load provides:

- An economical way to match system heat output to load requirements so as to minimize overheating, pipe losses and fuel waste.
- Increased comfort that results from uniform convection terminal unit surface temperature that is made possible by maintained water flow rates at varying temperatures.
- Improved automatic control valve operation by minimizing closed port positions.

15.4.2 Hydronic Distribution System Disadvantages

Hydronic distribution systems require much more maintenance than air distribution systems and this work typically must be done in the occupied spaces. Each unit that cools requires a condensate pan and drain system that must be periodically cleaned and flushed. Condensate disposal can be difficult and costly. It may also be difficult to clean the coils depending on the design and service access. Filters in convection terminal units are usually small, low in efficiency, and require frequent changing to maintain the airflow through the unit. However, in some HVAC system designs, condensate drain systems can be eliminated if dehumidification is provided by a central ventilation air system.

Space ventilation usually requires opening of windows unless outside wall apertures or air intakes are installed. Ventilation rates are affected by stack effect and wind direction and velocity. Summer room humidity levels tend to be relatively high, particularly if modulating chilled water control valves are used for room temperature control. Unacceptably low winter room humidity can occur in colder climates. Supplying properly dehumidified or humidified ventilation air through a separate central ventilation system can prevent these problems. However, this offsets one of the advantages of all-hydraulic systems of not requiring duct. Most systems today utilize an energy recovery unit or dedicated outdoor air system to provide ventilation air.

Often, two-pipe changeover systems are installed to obtain minimum initial cost. These systems may require frequent changeover during intermediate seasons and, even if zoned, present operating difficulties. A better, although not necessarily more economical, solution is using four-pipe or two-pipe systems with electric strip heat. If the cooling coil is used to handle solar cooling loads in the winter, chilled water will be required on a year-round basis.
15.5 HYDRONIC PIPING SYSTEM CLASSIFICATION

Hydronic piping systems can be classified in a variety of ways. Typically, hydronic distribution systems are classified by the following three characteristics:

- Operating Temperature
- System Flow
- Piping Arrangement

15.6 HYDRONIC DISTRIBUTION SYSTEM OPERATING TEMPERATURE CLASSIFICATIONS

Hydronic distribution systems can be classified by operating temperature as follows:

- Low Temperature Hot Water System
- Medium Temperature Hot Water System
- High Temperature Hot Water System
- Chilled Water System
- Dual Temperature Water System

15.6.1 Low Temperature Hot Water System

A low-temperature hot water (LTW) system is a hot water heating system operating within the pressure and temperature limits of the ASME boiler construction code for low pressure heating boilers. The maximum allowable working pressure for low pressure heating boilers is 160 psi (1100 kPa) with a maximum temperature limit of 250°F (121°C). The usual maximum working pressure for LTW system boilers is 30 psi (207 kPa). However, boilers specifically designed, tested, and stamped for higher pressures may be used at working pressures up to 160 psi (1100 kPa). Steam-to-water or water-to-water heat exchangers are also sometimes used to provide heat to low-temperature water systems.

15.6.2 Medium Temperature Hot Water System

A hot water heating system operating at temperatures of 350°F (177°C) or less with pressure not exceeding 150 psi (1034 kPa) is classified as a medium temperature hot water (MTW) system. The usual design supply temperature is approximately 250 to 325°F (121 to 163 °C) with a typical pressure rating of 150 psi (1034 kPa) for boilers and equipment.

15.6.2.1 High Temperature Hot Water System

High-temperature hot water (HTW) systems operate at temperatures over 350°F (177°C) with pressures of about 300 psi (2068 kPa). The maximum design supply water temperature is 400 to 450°F (204 to 232°C), with a pressure rating of about 3000 psi (20684 kPa) for boilers and equipment. When designing and installing high-temperature water systems it is important to check the pressure-temperature rating of each component against the design characteristics of the system to ensure that each component has a rating that is sufficient for use in the system.

15.6.3 Chilled Water System

A chilled water (CW) system operates with a usual design supply water temperature of 40 to 55°F (4 to 13 °C) and normally operates at or below a pressure of 125 psi (862 kPa). Antifreeze or brine solutions may be used for systems that require operating temperatures below 40°F (4°C) to protect against freezing. Chilled water systems that use well water systems may use supply temperatures of 50°F (10°C) or higher.

15.6.4 Dual Temperature Water System

A dual-temperature water (DTW) system is a hydronic system that circulates both hot and chilled water through a common piping system and terminal heat transfer equipment. These systems are usually operated within the pressure and temperature limits of LTW systems. Typical winter design hot water supply temperatures are about 100 to 150°F (38 to 66°C) and summer chilled water supply water temperatures are between 40 and 55°F (4 to 13°C).

15.7 HYDRONIC DISTRIBUTION SYSTEM FLOW

Hydronic distribution systems can be classified as either of the following two system flow methods:

- Gravity Flow
- Forced Flow

A gravity-flow hydronic distribution system uses the difference in weight between the supply and return water columns of any circuit or system to circulate water through the system. These systems require careful design and layout of the piping systems and are rarely used. Note: Gravity-flow hydronic was more widely...
used in the early years of central heating than today but energy efficiency concerns may influence greater use in the future.

A forced-flow hydronic distribution system uses pumps that are typically driven by an electric motor to maintain the necessary flow through the hydronic piping. With a forced-flow hydronic distribution system piping does not have to be run at a specific level or pitch to ensure flow. This allows for more flexibility in routing hydronic piping and the ability to work around architectural or structural requirements. The disadvantage of a forced-flow hydronic distribution system compared to a gravity-flow system is that it requires energy to generate flow using electrically driven pumps that result in additional operating costs of the over life of the system.

15.8 HYDRONIC DISTRIBUTION SYSTEM ARRANGEMENTS

Common hydronic distribution system arrangements are as follows:

- One Pipe
- Two Pipe
- Three Pipe
- Four Pipe
- Summer-Winter

15.8.1 One-Pipe

One-pipe hydronic distribution systems can be classified as either:

- Series Loop
- Diverting Valve

15.8.1.1 Series Loop

A series loop is a continuous run, single of pipe from the supply connection to the return connection. Convection terminal units are a part of the loop, see Figure 15-3. Loops may run directly to and from the boilers and chillers or multiple loops may connect directly to hot or chilled water mains. Water temperature will drop progressively as each convection terminal unit heats or cools the air passing through it. The amount of the temperature change in the water depends on the convection terminal unit heat transfer properties and rate of water flow.

A decrease in loop water flow rate will increase the water temperature drop across each convection terminal unit as well as across the entire loop. The water temperature will decrease or increase progressively from the first to the last convection terminal unit in the series loop depending on whether the system is in the heating or cooling mode. The effectiveness of each convection terminal unit gradually decreases from the first to the last unit in the loop. As a result, it is extremely difficult to maintain comfort in separate spaces when they are heated or cooled with a single series loop. Control of output from individual convection terminal units on a series loop is impractical except by controlling the airflow through them. Manual dampers can be used on natural convection units or an automatic fan or face-and-bypass damper control can be used on forced air units.

15.8.1.2 Diverting Valve

With a diverting valve, each convection terminal unit has a supply and a return tee installed on the main. One of the two tees is a special diverting tee that creates a pressure drop in the main flow to divert a portion of the main flow to the convection terminal unit. One diverting tee is usually sufficient for upstream convection terminal units. Both a supply and return tee is usually required for downstream units to overcome thermal head. These special tees are proprietary and the manufacturer's literature should be consulted for flow rates and pressure drop data.

One-pipe circuits allow manual or automatic control of flow to individual connected heating units. Simple on-off control rather than flow modulation control is advisable for diverting valve one-pipe hydronic distribution systems because of the relatively low pressure and flow diverted. The loop length and the thermal load imposed on a one-pipe circuit are usually small because of the limitations listed.

15.8.2 Two-Pipe

Two-pipe hydronic distribution systems can be classified as one of the following two system types:

- Direct Return
- Reverse Return

15.8.2.1 Direct Return

Figure 15-4 provides a diagram of a two-pipe, direct return hydronic distribution system. With a direct re-
FIGURE 15–3  ONE-PIPE HYDRONIC DISTRIBUTION SYSTEM
FIGURE 15–4 TWO-PIPE HYDRONIC DISTRIBUTION SYSTEM (DIRECT RETURN)
turn, two-pipe hydronic distribution system the return main flow direction is opposite of the supply main flow, see Figure 15-4. This results in the return water from each unit taking the shortest path back to the boiler or chiller. The direct-return system is popular because less main pipe length is required. However, circuit-balancing valves are usually required on units or sub-circuits must be used in order to balance water flow. Depending on system design, if the required shut off head for the control valves is within their capabilities, the system can be somewhat self-balancing when the system has equal friction loss.

15.8.2.2 Reverse Return

Figure 15-5 provides a diagram of a two-pipe reverse return hydronic distribution system. With a reverse-return two-pipe hydronic distribution system the return main flow is in the same direction as supply flow. After the last unit is fed, the return main returns all water to the boiler or chiller. Since water flow distance from and to the boiler or chiller is virtually the same through every unit on a reverse-return system, the adjustment of balancing valves can be better accomplished. These systems are also somewhat self balancing when the system has equal friction loss depending on the shut off capabilities of the control valves.

15.8.3 Three-Pipe

Figure 15-6 provides a diagram of a three-pipe hydronic distribution system. The three-pipe system satisfies the variations in load in each zone by providing independent sources of heating and cooling to each zone. This is accomplished by providing access to both hot and chilled water through the supply piping. Each convection terminal unit contains a single coil that acts as an air-water heat exchanger. A three-way valve at the inlet of the coil controls the amount of hot or chilled water supplied to the convection terminal unit coil. The three-way valve admits the water from either the hot or cold water supply depending on whether the zone served needs to be heated or cooled. The water leaving the coil is carried by a common pipe back to the central heating and cooling plant as shown in the diagram.

Zone control for three-pipe systems is typically achieved by using a special three-way modulating valve that modulates either hot or chilled water supply but does not mix the two. The modulating three-way valve at the inlet to the convection terminal unit is a special design that admits either hot water or cold water to the coil. The hot port on these special three-way valves gradually moves from open to fully closed and the cold port gradually moves from fully closed to open. The valves are constructed so that at mid-range there is an interval in which both ports are completely closed allowing neither hot nor chilled water to flow through the coil.

During the period between heating and cooling seasons, if both hot and chilled water is available, any convection terminal unit can operate anywhere within its range from maximum heating to maximum cooling. With a three-pipe hydronic distribution system, each convection terminal unit operates independently of the other units being served. Due to the operating cost penalty that results from the simultaneous supply of heating and cooling loads and the energy wasted by mixing the hot and chilled water back into the common return piping, the use of three-pipe hydronic distribution systems under these conditions is not recommended.

15.8.4 Four-Pipe

Figure 15-7 provides a diagram of a four-pipe, single-coil hydronic distribution system. Four-pipe hydronic distribution systems derive their name from the fact that there are four pipes that serve each convection terminal unit. These four pipes provide the following (Figure 15-7):

- Chilled Water Supply (CWS)
- Chilled Water Return (CWR)
- Hot Water Supply (HWS)
- Hot Water Return (HWR)

A four-pipe hydronic distribution system can supply single or dual coil terminal units and can address heating and cooling load variations in a zone by using a constant temperature air supply conditioned by hot water or chilled water supplied by the hydronic distribution system.

15.8.4.1 Single Coil

Figure 15-7 illustrates the four-pipe hydronic distribution system that uses one coil convection terminal units. The three-way mixing valves located at the inlet of the convection terminal unit coil admit water from either the hot or cold water supply as required. At the outlet of the coil, a diverting valve directs the returning water to the appropriate system depending on the setting of the mixing valve. This arrangement requires a special three-way mixing valve that was originally developed for three-pipe hydronic systems as described in Section 15.8.3. The mixing valve at the inlet of the single-coil convection terminal unit modulates the hot
FIGURE 15–5 TWO-PIPE HYDROSTATIC DISTRIBUTION SYSTEM (REVERSE RETURN)
FIGURE 15−6 THREE-PIPE HYDRONIC DISTRIBUTION SYSTEM

- System Pump
- Boiler
- Chiller
- Common Return Piping
- Direction of Water Flow (Typical)
- Control Valve (Typical 3 Places)
- Connection (Typical)
- CWS Piping
- Convection Terminal Unit (Typical 3 Places)
FIGURE 15–7 FOUR-PIPE HYDRONIC DISTRIBUTION SYSTEM (SINGLE COIL CONVECTION TERMINAL UNITS)
and chilled water flow but is designed and constructed to be shut off at midrange and never allow the hot and chilled water supplies to mix in the coil. The valve at the convection terminal unit outlet is a two-position diverting valve that allows either the hot or chilled water leaving the coil to return through the appropriate piping system.

Comparing Figures 15-6 and 15-7, it can be seen that a four-pipe hydronic distribution system using single coil convection terminal units is very similar to a three-pipe hydronic distribution system. The main difference between the two systems is the ability in the four-pipe hydronic distribution system to separate the returning hot or chilled water. If the various zones that are served by a four-pipe hydronic distribution system can be providing both heating and cooling simultaneously, a four-pipe hydronic distribution system can provide energy savings over a three-pipe system that mixes hot and chilled return water in a single return pipe. Under these conditions, the decision to select a three-pipe or four-pipe for a particular building should be based on a life cycle cost analysis that considers both the first cost of the system and the system operating cost.

15.8.4.2 Dual Coil

Figure 15-8 illustrates the four-pipe hydronic distribution system that uses dual coil convection terminal units. The use of dual coil convection terminal units eliminates the need for the special purpose mixing valve and diverting valve needed when a single-coil convection terminal unit, see Figure 15-8. Instead, a control valve is inserted in both the hot and chilled water supply piping on the input to the dual coil convection terminal unit and no diverting valve is required at the outlet.

During peak cooling and heating, the four-pipe system operates like the two-pipe hydronic distribution system discussed in Section 15.8.2 with only the heating or cooling coil in the convection terminal unit operating. During the period between the heating and cooling seasons, any dual coil convection terminal unit on the four-pipe hydronic distribution system can be operated at any capacity level from maximum cooling to maximum heating if both hot and chilled water are available. In addition, each convection terminal unit can be operated independently of any other unit on the system and respond effectively to the space conditioning needs of the zone that it serves.

15.8.5 Summer-Winter

A summer-winter hydronic distribution system is a variation of the two-pipe hydronic distribution systems, see Figures 15-4 and 15-5. With a summer-winter hydronic distribution system, the hydronic distribution system is supplied by either the boiler or chiller depending on whether it is heating or cooling season. Hot water is supplied to convection terminal units for heating in the winter and chilled water is supplied to the convection terminal units for cooling in the summer from the building’s central heating and cooling plant. Manual or automatic changeover valves that are typically located in the building’s central heating and cooling plant can be used to switch between the hot and chilled water supply input to the hydronic distribution system. Therefore, Figures 15-4 and 15-5 would be modified by showing a separate boiler or chiller that are both capable of supplying the required hot or chilled water to the system through a common three-way diverting valve. This three-way diverting valve would only allow the hydronic distribution system to be supplied from either the boiler or chiller at any time.

The advantage of a summer-winter hydronic distribution system is that either the boiler or chiller is operating when needed but they are never both operating simultaneously. This results in more economical system operation but may not be acceptable in geographic areas where the need for heating to cooling in the spring or fall is often unpredictable or where these transition periods are long. Summer-winter hydronic distribution systems do not offer the flexibility of other hydronic distribution systems that allow the simultaneous supply of both hot and chilled water. Summer-winter hydronic distribution systems are an economical and widely used piping arrangement utilized in hotels, garden apartments, and other installations where seasonal control is satisfactory.

Some of the disadvantages of a summer-winter hydronic distribution system are as follows:

- Possible thermal shock of equipment if changeover is done incorrectly.
- Requirement for nearly identical loads in all spaces.
- Face and bypass dampers may be required on individual coils if humidity is a problem.
- Possible requirement for outside air reset on both hot and chilled water.
- Possible need for a supplemental cooling DX unit if the building contains zones with dissimilar loads.
FIGURE 15–8 FOUR-PIPE HYDRONIC DISTRIBUTION SYSTEM
(DUAL COIL CONVECTION TERMINAL UNITS)
• Time required making the changeover each season.

• Changeover is difficult for smaller facilities without personnel knowledgeable with system operation.

15.9 MULTI-LOOP HYDRONIC DISTRIBUTION SYSTEMS

Figure 15-9 provides a diagram of a multi-loop hydronic distribution system which is also known as a primary-secondary hydronic distribution system. There is a primary system, or loop as it is sometimes called, and multiple secondary systems or loops. The primary system maintains a constant or required minimum flow through the boiler or chiller, see Figure 15-9. The secondary hydronic distribution systems in Figure 15-9 are connected directly to the primary hydronic distribution systems. All of the issues with controlling the flow in convection terminal units in the one-pipe hydronic distribution system are present in the primary loop of a multi-loop hydronic distribution system. Flow through secondary loops of multi-loop hydronic distribution systems can be controlled with secondary loop pumps and valves, unlike one-pipe hydronic distribution systems.

A variation on the secondary system loop being directly connected to the primary system is keeping the water in the two systems completely separated through the use of a water-to-water heat exchanger. Under this arrangement, the hot or chilled water in the primary system is hydraulically separated from the hot or chilled water in the secondary systems or loops. This arrangement makes it impossible for either the primary or secondary systems to be affected by the operation of any other secondary system. In large buildings where several secondary systems are served by a single primary system, there may be advantages to separating the water circuits. In high-rise buildings, the secondary water system should be divided horizontally into two or more zones in order to limit system pressures caused by height. In addition, the efficiency of heat exchangers must be considered due to the possible additional pressure drop required.

15.10 CONTROLLING HYDRONIC DISTRIBUTION SYSTEM FLOW

15.10.1 Constant Volume Flow Systems

A constant volume or constant flow hydronic distribution system is a piping system where the required system flow rates are fixed and do not change during normal system operations. Constant volume systems can be achieved by using either straight through piping or three-way valves.

15.10.1.1 Straight-Through Piping System

A straight-through piping system does not contain any automatic temperature control valves in the piping circuit. Flow through the source, piping system, and convection terminal units remain constant. Temperature control is accomplished by the modulating the system water temperature or by using dampers at convection terminal units. An application of a straight through piping system would be to provide constant volume water flow to a convection terminal unit with system water temperature reset using an outside air temperature controller. The system water temperature would increase with a decrease in outside air temperature and vice versa to condition the zone that is served by the convection terminal unit.

15.10.1.2 Three-Way Valves

Constant system flow may also be achieved using three-way automatic temperature control valves at convection terminal units. Three-way valves are sometimes referred to as bypass valves that allow circulation around a terminal unit when heat is not required. System water temperatures remain more constant with systems utilizing three-way control valves that permit closer temperature control than is can be achieved with a straight-through piping system. There are two types of three-way valves that are used in constant volume hydronic distribution systems:

• Diverting Valves
• Mixing Valves

Figure 15-10 provides a diagram of a constant volume hydronic distribution system using three-way diverting valves at the input of the convection terminal units. A diverting valve is primarily used in two position control applications where the flow is required to be diverted from one pipe to another. A diverting valve has one inlet and two outlets, see Figure 15-11. The position of the valve determines whether the fluid entering the valve at input AB exits through outlet A or B of the valve. In the case of convection terminal units,
FIGURE 15-9 MULTI-LOOP HYDRONIC DISTRIBUTION SYSTEM

- Boiler or Chiller
- Primary System Pump
- Primary Supply Piping
- Primary Return Piping
- Secondary System Pump
- Secondary Supply Piping
- Secondary Return Piping
- Convection Terminal Unit (Typical 3 Places)
- Control Valve (Typical 3 Places)
- Direction of Water Flow (Typical)
- Secondary System

To Other Secondary Systems
FIGURE 15–10 CONSTANT VOLUME HYDRONIC DISTRIBUTION SYSTEM (DIVERTING VALVE)
FIGURE 15–11 DIVERTING VALVE
the diverting valve determines whether supply water passes through the coil or completely bypasses it, effectively shutting down the unit.

Figure 15-12 illustrates a constant volume hydronic distribution system using a three-way mixing valve. A mixing valve is similar to a diverting valve in appearance but is built and operates completely differently. Like a diverting valve, a mixing valve has three ports but, as its name suggests, a mixing valve allows the two inputs to be mixed and discharge through a single port. Mixing valves are primarily used where proportional, or modulating control, is required. A mixing valve contains two inlets and one outlet that allows the mixture of two fluids, see Figure 15-13. In the case of convection terminal units, the valve allows the mixing of bypass and coil return flow to the system return piping.

When using modulating three-way valves, the system is considered constant but flow will vary due to the fact that the liquid flow capacity coefficient \( C_v \) rating of the valves is with either port fully open. \( C_v \) is defined as the flow rate in gallons per minute (GPM) (l/s) through the valve at a pressure drop across the valve of 1 pound per square inch (psi) (Pa). Intermediate positioning of the valve results in a varying \( C_v \) that can sometimes be higher or lower depending on the manufacturer of the control valve. Also, there should be a throttling valve in the bypass around the convection terminal unit that can result in additional balancing being required.

Mixing valves may be required for two-position control applications similar to a diverting valve. However, this application does not constitute nor require a diverting valve. Mixing and diverting valves have been designed and subsequently named for their internal construction and flow handling capabilities.

15.10.2 Variable Volume Systems

A variable volume or variable flow hydronic distribution system is a system where the required system flow rate varies during normal system operation. Variable volume systems can be installed using two-way automatic temperature control valves at the convection terminal units.

15.10.2.1 Two-Way Automatic Temperature Control Valves

Two-way automatic temperature control valves can either operate with two-position or proportional control. Both the two-position and proportional control method reduces the flow rate through a convection terminal unit when heat transfer is not required. Figures 15-14 and 15-15 show the construction of typical two-way control valves. The use of the two-way valves at system terminal units can result in varied system flow rates and pressures during normal operation. When a majority of convection terminal units on a piping loop are isolated by two-way control valves there is the possibility of very low flow rates through pumps and primary heat exchange equipment that can result in high system differential pressures causing leakage or the failure of automatic control valves. The following two methods are commonly used to maintain minimum flow rates and pressures that are needed to maintain satisfactory hydronic system operation:

- Differential Pressure Control Valves
- Variable Speed Pumping

15.10.2.1.1 Differential Pressure Control Valve

As the two-way control valves at terminal units close, system differential pressure increases and is monitored by a differential pressure controller. The controller modulates the differential pressure control valve (DPCV) to maintain a system differential pressure that is satisfactory for the pump and the primary heat exchange equipment operation. A DPCV is normally found in smaller, variable volume systems.

15.10.2.1.2 Variable Speed Pumping

Similar in control as described for a DPCV, the differential pressure controller resets the pump speed to reduce system differential pressure. Variable speed pumping using variable frequency drives (VFDs) is addressed in detail in Chapter 17.

15.11 HYDRONIC DISTRIBUTION SYSTEM COMPONENTS

In addition to the pumps and piping associated with the hydronic distribution system, there are a number of other components that need to be installed as part of the system. These additional system components include the following:

- Air Vents
- Drains and Shutoffs
- Balance Fittings
- Expansion Tanks and Separators
FIGURE 15–12 CONSTANT VOLUME HYDRONIC DISTRIBUTION SYSTEM (MIXING VALVE)
FIGURE 15–13 MIXING VALVE
FIGURE 15−14 SINGLE-SEATED TWO-WAY VALVE

FIGURE 15−15 DOUBLE-SEATED TWO-WAY VALVE
• Strainers
• Thermometers
• Flexible Connectors
• Gages

15.11.1 Air Vents

If air and other gases are not eliminated from the flow circuit, they may cause air binding in the terminal heat transfer elements and noise in the piping circuit. High points in piping systems and terminals units should be vented with manual or automatic air vents. However, the fluid velocity in the system at these points must not be excessive. As automatic air vents may malfunction, valves should be provided at each vent to permit service without draining the system. The discharge of each vent should be piped to a point where water can be wasted into a drain or container. If a plain expansion tank is used, free air contained in the circulating water should be removed from the piping circuit and trapped in the expansion tank by a boiler dip tube or other air separation device. If a diaphragm-type tank is used, all air should be vented from the system.

15.11.2 Drains and Shutoffs

All low points should be equipped with drains. Provisions should be made for separate shutoff and drain of individual equipment and circuits so that the entire system does not have to be drained for service of a particular component.

15.11.3 Balance Fittings

Balance fittings should be installed as needed to permit the balancing of individual convection terminal units and major subcircuits. Such fittings should be placed at the circuit return, whenever possible. Balance valves should be sized for minimal pressure drop and not just pipe size. Manufacturer’s data should be used to select and size balance fittings.

15.11.4 Expansion Tanks and Air Separators

Expansion tanks and air separators are also included as part of the hydronic distribution system.

Expansion tanks provide a volumetric “cushion” in hydronic systems that accommodates the expansion and contraction on the contained fluid. There are three basic types of expansion tanks:

• The air and fluid are not separated and air is vented via float-controlled devices or manually and these are typically located at system high points in open systems.
• The air and fluid are not separated within the tank and air is generally vented manually using an immersed tube to maintain a “designed” amount of air in the expansion tanks.
• Bladder tanks are the more often used expansion tank in newer systems since an integral bladder separates the fluid from the air which removes the possibility that dissolved oxygen will enter the fluid at the expansion tank.

Air separators remove entrained air from the circulating fluid. Air is a “bad actor” in hydronic systems and can create air pockets which will interfere with or may actually block fluid circulation through that section, cause flow noise in piping and at piping accessories and cavitation at the pump. In severe cases cavitation can erode some types of pump housings and impellers. Entrained air can dissolve in the fluid and accelerate corrosion of system components and shorten the life of antifreeze additives. Air separators are required hydronic system components because even in a closed system, air enters the system through valve packings, pump flanges, and via oxygen diffusion through the walls of polymer tubing.

15.11.5 Strainers

Strainers should be used where necessary to protect the hydronic distribution system equipment. Strainers placed in the pump suction need to be analyzed carefully to avoid cavitation. Large separating chambers are available that can serve as main air venting points and dirt strainers ahead of pumps. Automatic control valves or spray nozzles operating with small clearances require protection from pipe scale, gravel, welding slag, and other water borne debris that can pass through the pump and its protective separator. Individual fine mesh strainers may be required to be installed ahead of each control valve and ahead of individual coils where they will protect both the coil and control valve. Condenser water systems without water regulating valves do not necessarily require a strainer. If a cooling tower is used and a strainer is provided in the tower basin it will be usually adequate for condenser water but, typically, strainers are normally relegated to the pump section.
15.11.6 Thermometers

To assist the system operator and the Testing, Adjusting, and Balancing (TAB) technician in troubleshooting the hydronic distribution system thermometers or thermometer wells should be installed. Permanent thermometers with correct scale range and separable sockets should be used at all points where temperature readings are regularly needed. Thermometer wells should be installed where readings will be needed during start-up and balancing as well as where readings will be needed for operating and maintaining the system.

15.11.7 Flexible Connectors

Flexible pipe connectors should be installed at pumps and other vibrating equipment to reduce the transmission of pipe vibration. Vibrations are transmitted through the water column across a flexible connection and reduce the effectiveness at the connector. Flexible connectors also serve to prevent damage that is sometimes caused by misalignment of the piping to the equipment that it serves.

15.11.8 Gages

Gage cocks should be installed at points where pressure readings will be required during startup, balancing, and troubleshooting. Gages permanently installed in the system will deteriorate due to vibration and pulsation and may not provide long term reliability. However, this problem can be overcome by using snubbers at the gage cock or oil-filled gages.

15.12 HYDRONIC SYSTEM HEAT TRANSFER MODE

15.12.1 Convection Defined

Convection is one of the three modes of heat transfer along with conduction and radiation. Convection is the transfer of thermal energy between a surface and a fluid passing over it when the two are at different temperatures. In the case of convection terminal units, the surface is the water-to-air heat exchanger in the unit, often simply referred to as a coil, and the air that passes through or over the coil. If chilled water is flowing through the coil and warm air passes over the coil, heat energy from the air will be transferred to the chilled water and the air will be cooled. Similarly, if cool air is passed through the coil and hot water is flowing in the coil, heat energy from the hot water will be transferred to the air and the air will be warmed. The resulting warmed or cooled air will then mix with the air in the zone being served by the convection terminal unit and the temperature will increase or decrease, respectively.

15.12.2 Natural Convection

Natural convection is also referred to as free or heat-driven convection. Natural convection relies on changes in the air density to generate airflow through the convection terminal unit. Warm air is less dense than cool air and is pushed upward toward the ceiling by its natural buoyancy, carrying the heat energy with it. When this air cools, its density increases and it sinks to the floor. This phenomenon is the basis for the operation of non-fan powered convection terminal units such as a cast-iron radiator. Cool air, being denser that warm air, gathers at the floor level in a space and as it is heated by the radiator it expands and it rises to heat the space. The warm radiator results in a constant air flow through it providing both airflow and conditioned air to the space.

15.12.3 Forced Convection

Forced convection is when air is forced to move across the surface of a water-to-air heat exchanger and this is typically accomplished by the use of a fan in convection terminal units. An example of when a fan is needed in a convection terminal unit is when the unit is used for cooling. Since cool air is denser than warm air, buoyancy would not result in warm air flowing naturally through a cooling coil in a convection terminal unit located at floor level. As a result, forced convection would be required to move the warm air through the water-to-air heat exchanger to cool the air for reintroduction back into the space.

15.13 CONVECTION TERMINAL UNITS

15.13.1 Types

There are many types of terminal units that can be used with hydronic distribution systems. These terminal units are water-to-air heat exchangers that heat and cool the zone they serve through natural or forced convection. Convection terminal units used in commercial and institutional HVAC systems include the following:

- Air-Coil Units
- Fan-Coil Units
- Induction Units
- Unit Ventilators
15.13.1.1 Air-Coil Units

Air-coil units are simply water-to-air heat exchangers that are usually incorporated into air-handling units, air terminal units such as VAV boxes, or other air distribution equipment. Air-coil units can be served by either a two-pipe or three-pipe hydronic distribution system. With a two-pipe hydronic distribution system, the air-coil unit will usually be either a dedicated heating or cooling coil unless it is a summer-winter changeover system. With a three-pipe hydronic system, the air-coil unit can be either a heating or cooling coil depending on whether hot or chilled water is flowing through it. If both hot and chilled water are available, then the air-coil can switch from being a heating coil to being a cooling coil depending on the needs of the zones that it serves.

15.13.1.2 Fan-Coil Units

Fan-coil units are self-contained units that include a fan and either one or two water-to-air heat exchangers referred to as coils. The number of coils included in the fan-coil unit will depend on whether it is served by a three- or four-pipe water distribution system. When there are two coils, one is used for heating and the other coil is used for cooling. This type of fan-coil unit is sometimes referred to as a four-pipe, fan-coil unit. For fan coil units with one coil, that coil is used for both heating and cooling and is often referred to as a three-pipe fan-coil unit. Fan coil units can be mounted either on the wall or the ceiling of the zone that they serve. Instead of a hydronic heating coil, the fan-coil unit could also have an electric heating coil that provides the heating function.

Fan coil units operate by pulling air from the space and conditioning the air by passing it through the coil or coils in the unit and then returning the conditioned air to the space. The air is heated when hot water is being circulated through the operating coil and cooled when chilled water is being circulated through the operating coil. Hot water ratings are usually based on flow rates or temperature drops at various entering water and air temperatures. The air returned to the zone and mixes with the existing air to maintain the desired temperature.

A local thermostat usually regulates the zone temperature by controlling both the fan and the amount of hot or chilled water that circulates through the coil. The amount of hot or chilled water circulating through the coil is modulated by a control valve that is integral to the fan-coil unit. Fan control by the thermostat is either on or off but fan-coil units generally have manual fan speed controls that allow the occupant to select between two or more fan speeds.

Fan-coil units can also include an outside air intake and damper that allows the introduction of outside air into the space for indoor air quality. Outside air intakes are common in wall-mounted fan-coil units that serve a zone in an enclosed space that has no other effective means of introducing outside air into the space. Ceiling-mounted and interior wall-mounted fan-coil units can also have ducted outside air supplied to them.

15.13.1.3 Induction Units

An induction unit is essentially a fan-coil unit that pulls air from the space into the unit using velocity nozzles instead of a fan. The high-velocity air in turn induces a secondary air flow through the unit which is where the unit gets its name. The induced air is then conditioned by flowing through the coil or coils as it does in the fan-coil unit. Other than the high-velocity air supply through nozzles, induction units operate the same as fan-coil units.

15.13.1.4 Unit Ventilators

Unit ventilators, originally developed for application in school classrooms, are used today in a wider range of applications. Unit ventilators consist of a forced convection heating or cooling unit with dampers to allow introduction of controlled amounts of outdoor air to provide a complete cycle of heating, ventilating, ventilation cooling or mechanical cooling, as required. Condensation may be a problem during summer operation unless chilled water flow is stopped when fans are not operating. Condensate drains may be required.

These units often contribute to humidity and indoor air quality problems in schools and other commercial and institutional buildings. Normally these systems include both a preheat and a cooling coil. Often sensible space temperature is satisfied with a cooling discharge temperature between high to provide dehumidifica-
tion or latent heat control. It installed with a cooling or reheat coil, winter conditions may require an antifreeze solution in the chilled water system or the draining of all coils. Also, the return and outside air dampers need to be set for the minimum outside air supply requirement.

15.13.1.5 Radiators

Radiators are simple, water-to-air heat exchangers that are used exclusively for space heating. Radiators rely solely on natural convection for heating exterior perimeter wall surfaces. Radiators are generally rated at various average water temperatures and at one or more water velocities. Velocity corrections provided by the manufacturer can be used to adjust for varying water temperatures.

15.13.1.6 Finned-Tube Radiation Heaters

Finned-tube radiation heaters rely on natural convection, just like traditional radiators, but are more efficient because they have a greater surface area for heat transfer.

15.13.1.7 Valance Units

Valance units are finned-tube, water-to-air heat exchangers that are either installed high on a wall for cooling or low on the wall for heating. Valance units normally do not include a fan and rely on the natural convection of air to move them through the unit so that it can be conditioned when passing over the finned-tube heat exchanger. Chilled or heated water is circulated through the unit, which consists of a finned coil mounted in an insulated sheet metal enclosure and positioned so that openings permit air to pass over the coil by convection. As a result, valance units only have one coil and are either dedicated to heating or cooling. Valance units are used in building renovation where there is limited space for equipment in the zone requiring additional heating and cooling.

Valance units are most commonly used for heating and are sometimes referred to as baseboard radiators. Valance units resemble electric resistance baseboard heaters. Located close to the floor, these units use natural convection to draw cool air from the space at floor level, condition it as it moves through the finned-tube radiator, and then supplies the conditioned air to the space through the top of the unit. Supplemental heating may be required under windows in cold climates. Ventilation must be provided separately.

Valence units used for cooling and dehumidifying are usually hung at the junction of the wall and ceiling. When cooling and dehumidifying, the cool air at the coil drops, drawing warm air through the coil. Additionally, if units reduce the wet-bulb temperature of the air for dehumidification, then condensate will result and will need to be removed.

The advantages of valance units include quiet operation, reasonably good cooling and dehumidification and low installation cost. Valance unit disadvantages include minimal air distribution, slow response on the cooling cycle, difficulty in cleaning, and lack of winter humidification.

15.13.1.8 Convector

A convector is a high-capacity water-to-air heat exchanger that includes one or more finned-tube heat exchangers enclosed in housing that allows air from the space to be drawn in from the bottom, heated by passing through the finned-tube heat exchangers, and then reintroduced to the space through the top of the unit. Convector sometimes include fans but normally rely on natural airflow through the unit by induced natural convection to condition the space.

Cabinet convectors were widely used in low temperature water systems. Convector are used extensively in areas where high output is needed and limited space is available, and where linear heat distribution is not desired. Typical areas where convectors are used to provide heat include corridors, entries, restrooms, storage areas, workrooms, kitchens, and similar locations.

15.13.1.9 Unit Heaters

Unit heaters are available in the following configurations:

- Horizontal Propeller Fan
- Down Blow
- Cabinet

Unit heaters consist of a fan and coil packaged in a single housing that is served by the hot water distribution system. Unit heaters are used in locations where there is a need for high heat output in a small space where no cooling is required. Cabinet units are frequently applied in corridors and at entrances to blanket doors that are frequently opened. Unit heaters only recirculate existing air and do not provide outside ventilation air.
Unit heaters are normally used in unfinished spaces such as truck docks, warehouses, and other similar locations to provide heat. Unit heaters can also channel their heat output directly downward that allows them to be used in small spaces needing a large amount of heat such as building entrances where doors are opened frequently.

15.13.1.10 Chilled Beams

15.13.1.10.1 Operation

A chilled beam is a convection terminal unit that is similar to a radiator or valence unit. Like a radiator or valence unit, a chilled beam relies on convection to cool the zone served as well as keep the air circulating in the zone. Unlike a radiator or valence unit, a chilled beam is either installed in a ceiling or suspended from a ceiling instead of being floor or wall mounted.

Chilled beams have an air-to-water heat exchanger also referred to as a cooling coil that is connected to the building’s chilled water loop. As warm air rises in the zone served by the chilled beam, it passes through the air-to-water heat exchanger and is cooled by the chilled water. The cool air leaving the chilled beam is heavier than the warm air below and the cool air displaces the warm air causing it to flow upward toward the ceiling. At the ceiling, the warm air comes in contact with the air-to-water heat exchanger in the chilled beam and the cycle begins again.

15.13.1.10.2 Types

There are two basic types of chilled beams and they are as follows:

- Passive Chilled Beams
- Active Chilled Beams

Figure 15-16 shows a passive chilled beam that is suspended from the ceiling or structural overhead of the space it serves. A passive chilled beam consists of a cooling coil installed in an enclosure that allows air to pass through the enclosure and its cooling coil from top to bottom, see Figure 15-16. A passive chilled beam only remove sensible heat from the zone served and does not remove latent heat nor does it have the capability of providing fresh air to the space. As a result, a separate ventilation system may be needed to deliver outdoor air to the zone as well as remove latent heat.

Figure 15-17 illustrates an active chilled beam that is installed in the ceiling of the zone served. Active chilled beams are also sometimes referred to as induction diffusers. Comparing Figure 15-16 to Figure 15-17 it can be seen that the difference between a passive and active chilled beam is that an active chilled beam has ventilation air supplied to it that mixes with the warm air returning from the space. As a result, active chilled beams remove both sensible and latent heat from the zone served. In addition it includes an integrated means of providing outdoor air requirements to the zone.

15.13.1.10.3 Chilled Beams Versus VAV HVAC Systems

In general, chilled beams are more efficient than VAV HVAC systems because the temperature of the chilled water supply required by a chilled beam is usually higher than the chilled water supply required by a VAV HVAC system. In addition, chilled beams do not require as much fan energy as a VAV HVAC system because chilled beam systems only need to move supply and return air to and from the space and they utilize natural convection forces to move air in the space. Besides energy savings, chilled beams are also quieter than VAV HVAC systems and normally considered to provide increased occupant comfort.

A disadvantage of chilled beams is that they usually require more ceiling area than a VAV HVAC system. This additional ceiling area can make the layout and coordination of other ceiling equipment such as luminaires, sprinklers, and speakers more difficult. In addition, when mounted in the ceiling cavity the height of this cavity may need to be greater to accommodate a chilled beam unit than needed for a VAV air terminal unit to allow air to circulate through the unit. Care must also be taken to make sure that the chilled beam temperature does not drop below the dewpoint of the surrounding air or condensation will result.

15.13.1.11 Radiant Panels

Radiant can be used to either heat or cool a building space. Radiant panels use both radiant and convective heat transfer to condition a space. Radiant panels can be installed on the floor, on walls, or on the ceiling. Radiant panels are typically manufactured and installed as a unit similar to other convection terminal units but they can also be field fabricated with either hydronic piping or electric heating elements installed in a building surface such as a floor slab. Hot water or an electric resistance heating element is usually used for heating and chilled water is typically used for cooling in radiant panels.
FIGURE 15–16 PASSIVE CHILLED BEAM


FIGURE 15–17 ACTIVE CHILLED BEAM

15.13.2 Selecting Convection Terminal Units

Convection terminal units must be selected for sufficient capacity to match both the calculated sensible and latent heating and cooling loads. Manufacturers' catalog ratings should be used for actual operating conditions. Ratings indicate water temperature, temperature drop or rise, entering air temperature, water velocity, and airflow. Ratings are usually provided for standard test conditions with correction factors. Alternatively, curves and rating tables covering a range of operating conditions are provided by the product manufacturers.

In any single circuit having similar loads and a single control point, the terminal units should be of similar response types. Cast iron radiators should not be installed in the same controlled circuit as baseboard or convector units. Use caution when including fan-operated units with natural convection units on the same circuit.

15.14 THERMAL FLUIDS

15.14.1 Glycol Solutions

Glycol solutions are commonly used in hot water heating systems and chilled water cooling systems when there is a danger of any portion of the system freezing. The use of glycol and water mixture as an antifreeze solution may be needed in the following situations:

- Systems supplying heating coils subjected to 100 percent outdoor air.
- Isolated parts or zones of a heating system where intermittent operation or long runs of exposed piping increase the danger of freezing.
- Process cooling applications requiring temperatures below 40°F (4°C).
- Snow-melting applications.

While using ethylene glycol is comparatively expensive and tends to create corrosion problems unless suitable inhibitors are used, it may be the only practical solution in many cases. Solutions of propylene glycol and triethylene glycol as well as certain other heat transfer fluids may also be used. Ethylene glycol is commonly used in hydronic systems. However, propylene, which is a food-grade substance, is used in lieu of ethylene in many applications. Many facilities, including pharmaceutical and food processing, do not permit the use of ethylene in their facilities.

15.14.2 Effects Of Glycol Solutions

The use of glycol solutions can affect the performance of heating and chiller plant equipment such as boilers and chillers as well as the hydronic distribution system. Table 15-1 provides the physical and heat transfer characteristics of aqueous ethylene and propylene glycol solutions and how the use of an aqueous glycol solution in a hydronic distribution system could impact the systems operation.

As the concentration of both ethylene and propylene glycol increases between 30 and 60 percent by volume of aqueous solution, the freezing point decreases from 2°F (-16°C) and 7°F (-14°C), respectively at 30 percent concentration to 70°F (-55°C) and -55°F (-48°C), respectively at 60 percent concentration, see Table 15-1. While increasing the concentration of glycol has the desirable effect of decreasing the freezing point of the aqueous solution, it adversely impacts the heat transfer capability of the system due to the solution's decreasing specific heat and system's pumping capability because the solution's specific gravity increases.

The specific heat in Table 15-1 compares the heat transfer capability of the aqueous glycol solution to that of water. The specific heat for both glycols at 30 percent concentration results in a specific heat of the aqueous glycol solution that is about 90 percent that of water, see Table 15-1. As the concentration of glycol increases to 60 percent, the specific heat of the aqueous glycol solution drops further to about 80 percent of water. This means that to get the same heat transfer from the circulating aqueous glycol solution as water, it will require an increase in flow rate.

Similarly, the specific gravity columns in the table compare the weight per gallon of aqueous glycol solution to that of water at concentrations between 30 and 60 percent. The specific gravity of the aqueous ethylene glycol solution increases from about 1.05 to 1.09 and aqueous propylene glycol increases from about 1.03 to 1.05 from 30 to 60 percent concentration, respectively, see Table 15-1. This increase in weight per unit volume compared to water may not impact pump performance at the design flow rate for water. However, the increased weight per unit volume of the aqueous glycol solution coupled with the required increased flow rate for heat transfer as discussed in the previous paragraph could result in pumps being undersized for the application if the system was designed for water as the heat transfer medium.
### Table 15-1 Physical and Heat Transfer Characteristics

<table>
<thead>
<tr>
<th>Concentration (Pct By Volume)</th>
<th>Freezing Point (°F/°C)</th>
<th>Specific Heat (80°F/26.7°C)</th>
<th>Specific Gravity (80°F/26.7°C)</th>
<th>Freezing Point (°F/°C)</th>
<th>Specific Heat (80°F/26.7°C)</th>
<th>Specific Gravity (80°F/26.7°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>30</td>
<td>-13°/25°</td>
<td>0.860</td>
<td>1.060</td>
<td>-8°/22°</td>
<td>0.895</td>
<td>1.034</td>
</tr>
<tr>
<td>40</td>
<td>-13°/25°</td>
<td>0.860</td>
<td>1.060</td>
<td>-8°/22°</td>
<td>0.895</td>
<td>1.034</td>
</tr>
<tr>
<td>50</td>
<td>-36°/37°</td>
<td>0.815</td>
<td>1.077</td>
<td>-29°/34°</td>
<td>0.850</td>
<td>1.041</td>
</tr>
<tr>
<td>60</td>
<td>-70°/55°</td>
<td>0.768</td>
<td>1.090</td>
<td>-55°/48°</td>
<td>0.805</td>
<td>1.046</td>
</tr>
</tbody>
</table>

**NOTE:** The information contained in this table is general in nature and provided for illustrative purposes only. The information in the table does not represent the actual properties of any manufacturer’s product. In all cases, the equipment manufacturer and glycol manufacturer’s instructions and recommendations should be followed when using any antifreeze solution in the hydronic distribution system.

#### 15.14.3 Installation and Maintenance

The smallest possible concentrations of glycol should be used to produce the desired anti-freeze properties to minimize pumping losses, reduction of heat transfer losses and the cost of the glycol—which must be treated as a service item with periodic replenishment/replacement. The total water content of the system should be calculated carefully to determine the volume of glycol required. The solution can be mixed outside the system in drums or barrels and then pumped into the hydronic distribution system. When pumping in the glycol solution, air vents should be watched to prevent loss of solution. The hydronic system and the cold water supply should not be permanently connected and automatic fill valves should not be used.

Glycol normally includes an inhibitor to help prevent corrosion. Solutions should be checked annually to ensure the solution still retains the needed antifreeze and corrosion inhibiting properties. The glycol manufacturer’s testing procedures and recommendations should be followed.

Certain precautions, such as those listed below, regarding the use of inhibited glycol solutions should be taken to extend the solution’s life and to protect equipment:

- Thoroughly clean and flush the system before installing the glycol solution.
- Use water that is classified as soft and low in chloride and sulfate ions to prepare the glycol solution whenever possible.
- Limit the maximum operating temperature in a closed hydronic system to that recommended by the manufacturer. In heat exchangers, limit temperatures in accordance with manufacturer recommendations to prevent deterioration of the solution.
- Check the concentration of glycol and corrosion inhibitor on a regular basis as recommended by the glycol manufacturer.
- Know and follow any disposal requirements imposed by the Environmental Protection Agency and local authorities regarding the disposal of glycol solutions.

#### 15.15 SIZING CENTRAL EQUIPMENT FOR ALL-WATER SYSTEMS

Central equipment for an all-water system should be sized based on the block load of the entire building at the time of building peak. Central equipment for an all-water system should not be sized based on the sum of individual convection terminal unit peak loads because these loads are often not coincident and will result in larger central equipment than necessary. When estimating the cooling load, it is important to include appropriate diversity factors for both lighting and people loads. The heating load should be based on maintaining the unoccupied building at design temperature plus an additional allowance for pickup capacity so the building temperature can be set back at night. Over sizing the central equipment for an all-water system will result in increased system first costs, inefficient system operation over the life of the installation, and increased maintenance costs.

If water supply temperatures or quantities are to be reset at times other than at peak load, the adjusted settings must be adequate for the most heavily loaded space in the building. An analysis of individual zone load variations should be performed. Also, if the Southern exposure or interior zone loads require chilled water in cold weather the use of the cooling tower with a water-to-water interchanger should be considered. Under these conditions, the cooling load
will be very low when only a few units are operating in cold weather and it must be determined that the chiller can operate safely at this reduced load.

15.16 HYDRONIC DISTRIBUTION SYSTEM DESIGN

15.16.1 Approaches to Energy Efficient Design

Four approaches are possible for the design of energy efficient large heating systems:

- Higher supply temperatures.
- Primary-secondary pumping.
- Terminal equipment designed for smaller flow rates.
- Variable pumping through the chiller and boiler to maintain minimum flow.

All four of these approaches can be used either independently or in combination with one another to achieve the most efficient system design.

Using higher supply water temperatures is an obvious way to achieve higher temperature drops and lower flow rates. Terminal units with a reduced heating surface can be used. These smaller terminals are not necessarily less expensive because their required operating temperatures and pressures may increase manufacturing cost as well as problems with pressurization, corrosion, expansion, and control. The use of higher supply water temperatures will probably result in an increased first cost of the system due to the need to insulate the piping more and ongoing higher operating costs due to heat loss.

Primary-secondary pumping reduces the size and cost of the distribution system and also uses larger flows and lower temperatures in the convection terminal units or secondary loops. A primary pump circulates water in the primary distribution system while one or more secondary pumps circulate water to the convection terminal circuits, see Figure 15-9. The connection between primary and secondary provides complete hydraulic isolation of both circuits and permits a controlled interchange of water between the two. Thus, a high supply water temperature can be used in the primary at a low flow rate and high temperature drop, while lower temperature and conventional temperature drops can be used in the secondary. However, there can be problems with primary-secondary chilled water systems when the secondary load does not match the primary loop’s capabilities which results in a low temperature differential and adversely impacts chiller efficiency.

For example, a system could be designed with primary-secondary pumping in which the supply temperature from the boiler was 300°F (149°C), the supply temperature in the secondary was 200°F (93°C), and the return temperature was 180°F (82°C). This would result in a conventional 20°F (11°C) temperature drop in the secondary zones, but would permit the primary circuit to be sized on the basis of a 120°F (67°C) drop. The primary-secondary pumping arrangement is most advantageous with terminal units such as convectors and finned radiators that are generally unsuited for small flow rates.

The relative insensitivity of heat transfer to flow change is an important reason for not oversizing the pump. For example, an increase in flow rate to 150 percent of design will increase heat transfer only a few percent. However, the increase in pumping capacity will increase energy use and aggravate control problems.

There is less latitude in selecting supply water temperatures for cooling applications than heating applications because there is a narrow range of water temperatures low enough to provide adequate dehumidification and high enough to avoid chiller freeze up. Selecting proper air quantities and heat transfer surface at the terminals can substantially reduce circulated water quantities. Using terminals suited for a 12°F (8°C) rise, rather than an 8°F (4°C) rise, reduces circulated water quantity requirements and pump power by one-third as well as increases chiller efficiency.

15.16.2 Design Flow Rates

15.16.2.1 Required Flow Rates

The required water flow rate is the total of the flow rates required by the individual terminal units to provide the rated heating or cooling capacity at design conditions. Flow rates in system design are used in the following two ways:

- Assigning an arbitrary system temperature difference.
- Finding a minimum system flow rate by summing the flow rates to individual convection terminal units or zones.
Using an arbitrary system temperature difference is common practice and works well for small system design. Using a minimum system flow rate is recommended for large systems because it usually results in the least overall cost and best performing system. In large systems, the design temperature difference should be the result and not the starting point of the design.

### 15.16.2.2 Small Buildings

In small buildings, flow rates to all convection terminal units are frequently based on an assumed overall system temperature drop. For example, that overall system temperature drop could be 20°F (5°C) for heating and 8°F (11°C) for cooling. This method has, historically, provided sound economy and satisfactory performance in small systems.

### 15.16.2.3 Large Buildings

For large systems, a minimum system flow rate should be determined to provide the lowest cost and maximum control. Several methods are used to determine minimum system flow rate to the individual convection terminal units and, by summation, the total system flow rate. Assumptions and data that can be used for determining minimum system flow rates in large buildings are as follows:

- Various basic terminal units are capable of producing full capacity at specific temperature drops. For example, baseboard and finned-tube units are selected frequently for a 20 to 50°F (11 to 28°C) drop, unit heaters for a 50°F (28°C) drop, and extended surface coils in air systems for a 100°F (55°C) drop.

- Manufacturer catalog data provide capacity ratings at various flow rates and entering temperatures.

- Convection terminal units can be selected based on minimum flow rate or maximum temperature difference to produce desired capacities.

- A constant temperature approach can be used for each class of equipment and can be assumed based on leaving water temperature minus final heated medium temperature. For example, this approach might be 20°F (1°C). For an air handler with 120°F (49°C) final air temperature, the leaving water would be at 140°F (60°C). Similarly, a domestic water heater might have a final temperature of 180°F (82°C) that would require a leaving system water of 200°F (93°C). Assuming a constant supply water temperature, the temperature difference and flow rate for each convection terminal unit can be determined. Flow rates determined in this manner are usually stated in gallons per minute (gpm) (L/s) or mass flow in pounds per hour (lb/h) (kg/h). Mass flow rates are established by dividing capacity in thousands of Btu per hour (MBh) by the temperature difference.

### 15.16.3 Pressure Drops

#### 15.16.3.1 General Design Range

The general range of pipe friction loss used for design of hydronic systems is between 1 and 4 ft/100 ft (0.31 to 1.22 m/30 m). A value of 2.5 ft/100 ft (0.76/30 m) represents the mean to which most systems are designed. Wider ranges may be used in specific designs if the following are considered:

- Piping Noise
- Air Separation
- Valve and Fitting Pressure Drop

#### 15.16.3.1.1 Piping Noise

Velocity should be considered in terms of piping noise. Closed loop hydronic piping systems are generally sized below certain arbitrary upper limits such as the following:

- Velocity limit of 4 fps (1.22 m/s) for 2-inch (50 mm) pipe and under.

- Pressure drop limit of 4 ft/100 ft (1.22 m/30 m) for piping over 2-inch (50 mm) diameter.

Velocities in excess of 4 fps (1.22 m/s) can be used with larger sized piping. This limitation is generally accepted, although it is based on relatively inconclusive experience with noise in piping. It is apparent that water velocity noise is caused not by water but by free air, sharp pressure drops, turbulence, or a combination of these. These in turn cause cavitation or flashing of water into steam. Therefore, it seems practical to use higher velocities if proper precautions are taken to eliminate air and turbulence.

#### 15.16.3.1.2 Air Separation

Piping noise can be caused by free air. The hydronic distribution system should be equipped with air separ-
ation devices to minimize the amount of entrained air in the piping circuit. Air should be vented at the highest point of the system as discussed in Section 15.11.1. In the absence of venting, air can be entrained in the water and carried to separation units at flow velocities of 1.5 to 2 fps (0.46 to 0.61 m/s) or more in pipe sizes 2 in. (50 mm) and under. Minimum velocities of 2 fps (0.61 m/s) are therefore recommended. For pipe sizes 2 in. (50 mm) and over, minimum velocities corresponding to a head loss of 0.75 ft/100 ft (0.23 m/30 m) are normally used. Particular attention should be paid to maintenance of minimum velocities in the upper floors of high-rise buildings when the air tends to come out of solution because of the reduced pressures. Higher velocities should be used in downflow return mains feeding into air separation units located in the basement.

15.16.3.2 Valve and Fitting Pressure Drop

Pressure drops associated with valves and fittings should be accounted for in the hydronic piping system design. Consult manufacturer literature for pressure drop information associated with a particular valve or fitting under the anticipated conditions of use.

15.16.4 Design Procedure

15.16.4.1 Design Requirements

Regardless of the method used to determine the flow through each convection terminal unit, the desired result should be listed in terms of mass flow in lb/h (kg/h) on the preliminary plans or in a schedule of flow rates for the piping system. Note that in the design of small systems or in chilled water systems, the determination may be made in terms of gpm (l/s) instead of pounds per hour. In an equipment schedule or on the plans, starting from the most remote terminal and working back towards the pump, progressively list the cumulative flow in each of the mains and branch circuits in the entire distribution system.

15.16.4.2 Preliminary Pipe Sizing

For each portion of the piping circuit, a tentative pipe size is selected from the unified flow charts using a value of pipe friction loss ranging from 0.75 to 4 ft/100 ft (0.23 to 1.22/30 m). Residential piping size is often based on pump selection using pipe-sizing tables that are available from piping manufacturers.

15.16.3 Preliminary Pressure Drop

Using the preliminary pipe sizing, calculate the pressure drop through each portion of the piping. Find the total pressure drop in several of the longest circuits to determine what maximum pressure drop through the piping, including the convection terminal units and control valves, which must be offset in the form of pump head.

15.16.4.4 Preliminary Pump Selection

The preliminary pump selection should be based on the ability of the pump to fulfill the capacity requirements as determined.

15.16.4.5 Final Piping Layout

An overall examination of the piping layout should be made at this point to determine any adjustments in piping sizes needed to be made. The pressure drop in the primary loops should be about equal so that excessive heads are not required to serve any one part of the building.

In determining the final system friction loss, both the first cost of the hydronic distribution system and the operating costs of the pump should be considered. In general, lower heads and larger pipe sizes become more economical when longer amortization periods are considered on larger systems. On the other hand, in small systems it may be more economical to select the pump first and design the piping system to meet the available head. Adjustments should be made in the piping system design and in the pump selection to achieve an economical system based on life-cycle cost.

When the final piping layout has been established, the friction loss for each section of the piping system can be determined by reading directly from the pressure drop charts for the mass flow rate in each portion of the piping system. After the friction loss at design flow for all sections of the piping system, all fittings, convection terminal units, and control valves have been calculated, they should then be summarized for several of the longest piping circuits, to determine the precise head against which the pump must operate at design flow.

15.16.4.6 Final Pump Selection

After the final pressure drop calculations have been completed, a final selection of the pump can be made.
16.1 INTRODUCTION

Hydronic pumps are an integral part of HVAC systems that include both central cooling and heating plants discussed in Chapters 9 and 10, respectively. Hydronic pumps are used in HVAC applications to move hot, chilled, supply, and makeup water through the hydronic piping systems associated with each of these systems. The following specific-purpose pumps are used in central heating and cooling plants:

- Chilled Water Pumps
- Hot Water Pumps
- Condenser Water Pumps
- Condensate Water Pumps
- Boiler Feed Water Pumps

16.2 PUMPS

16.2.1 Purpose

Pumps in HVAC systems are hydronic or liquid pumps that have the specific purpose of producing sufficient pressure to overcome piping system resistance and move fluids through the piping system at the required flow. Although pump laws are essentially the same as the fan laws described in Chapter 13, there are differences between pumps and fans due to the difference in characteristics between water moved by hydronic pumps and air moved by fans.

The major difference between moving water and air is that water is essentially an incompressible fluid whereas air is a compressible fluid. Water is an incompressible fluid because it acts as a solid and its volume is not reduced at the pressures at which hydronic HVAC systems operate. Air, on the other hand, is a gaseous vapor comprised of standard dry air, water vapor, and other constituents as discussed in Chapter 2. Air can be compressed at the pressures that HVAC air distribution systems operate. As a result, the selection and operation of hydronic pumps is different for air from that of fans. Additionally, the weight of the air is ignored in fan applications but the weight of water is an important factor in selecting and sizing pumps for hydronic distribution systems.

16.2.2 Types

There are two basic types of pumps used in HVAC applications, which are as follows:

- Positive Displacement Pumps
- Centrifugal Pumps

16.2.2.1 Positive Displacement Pumps

The typical types of positive displacement pumps used in HVAC systems are as follows:

- Reciprocating (Piston) Pumps
- Rotary Pumps
- Screw Pumps

One characteristic that is common to all positive displacement pumps is the ability to overcome excessive pressures. These pumps are called positive displacement pumps because they take a discreet quantity of fluid at a low pressure and transform it into the same quantity of fluid at a higher pressure. This implies very little slippage within the pump due to very little leakage around the impeller and high pressures at the pump outlet.

Positive displacement pumps operating in an open system will not be able to lift water on their suction side higher than the equivalent of 34 ft of water (10 m of water) or equivalent atmospheric pressure. (Note: Pumps operating in closed systems “balance” the pressure on the intake and discharge side because the height of the fluid column are equivalent on both.) Since the specific gravity of oil is less than water, the suction height would be higher if oil was being pumped. Similarly, the suction height of a liquid with a higher specific gravity than water would be less than 34 ft (10 m). This is because all pumps operating in an open system depend on the atmospheric pressure to push the fluid into the pump’s suction side. Positive displacement pumps are typically not used to move water in HVAC applications. Positive displacement pumps are typically used to move liquids other than water in HVAC applications. For example, large oil transfer pumps are generally rotary, positive displacement pumps in HVAC systems.

16.2.2.2 Centrifugal Pumps

The most common type of hydronic pump used in HVAC applications is the centrifugal pump. Centrifugal pumps are constructed with less stringent tolerances than positive displacement pumps. As a result, centrifugal pumps have higher fluid slippage than positive displacement pumps and as the pressures increase more fluid slips past the impeller and less fluid is delivered to the outlet of the pump. Figure 16-1 shows the pump curve for a centrifugal pump. The impellers of
FIGURE 16–1 TYPICAL CENTRIFUGAL PUMP PERFORMANCE CURVES
the centrifugal pumps are classified according to the configuration of the impeller’s vanes similar to fans. Since centrifugal pumps are used almost exclusively as hydronic pumps in HVAC systems, this chapter will focus on centrifugal pumps.

16.2.3 Typical Centrifugal Pump Curve

Figure 16-1 provides a pump curve for a typical centrifugal pump. The pump curve is also referred to as the head-capacity curve because it graphically illustrates the performance of the pump by relating pump total head with pump flow rate. The pump curve is the upper curve in Figure 16-1 that is labeled “HD.” In addition to the pump curve, manufacturers also provide other information such as pump efficiency which is the curve labeled “efficiency” and pump brake (bHP) horsepower which is labeled “H.P.”, in Figure 16-1. Both pump efficiency and brake HP are shown as a function of fluid flow rate like pump total head.

Figure 16-1 shows the fluid flow rate on the horizontal axis. It is common for hydronic pump flow rate to be illustrated in U.S. gallons per minute (gpm) or liters per second (l/s) for HVAC Applications. However, some hydronic pump curves provide mass-based flow rates in pounds per minute or pounds per hour. These flow rates can be converted into the more common units of gpm using the specific weight of the fluid being pumped in pounds per U.S. gallon. However, care must be taken when using flow rates other than gpm because the specific weight of water changes with temperature. Also, mixing water with additives like glycol will change its characteristics, including specific weight.

The left vertical axis of Figure 16-1 shows the pump total head in feet. Pump manufacturers nearly always express pump total head in feet because this makes the pump curve applicable to any liquid with any specific gravity including water and water mixtures. If the vertical axis units are something other than total head in feet such as pounds per square inch (psi) then pressure and density corrections may necessary to convert from the units given to head in feet. Additionally, even if the vertical axis is given in “feet” that pressure may actually be total feet, dynamic feet, or may have no identification at all other than “feet.” Pump differential pressure requirements are determined by a combination of elevation, friction, and velocity heads. Therefore, total head is the most useful reference unit for HVAC pump applications.

Figure 16-1 illustrates the typical centrifugal pump curve for a pump operating at a constant speed with a specific impeller diameter. Pump manufacturers develop pump curves by testing the pump under specified test conditions. The pump curve shown in Figure 16-1 illustrates the operational characteristics of the pump from a flow rate of zero gallons per minute when the pump is “dead headed” to a maximum flow rate of 300 gpm (19 l/s). As noted, the pump curve labeled “HD” in Figure 16-1 shows the total pump head in feet as a function of flow rate in gallons per minute. For example, it can be seen that for a fluid flow rate of 150 gpm (9.5 l/s) the pump’s total head will be 75 feet.

Figure 16-1 shows the pump efficiency as a function of fluid flow rate on the left vertical axis. As can be seen from the efficiency curve the maximum pump efficiency of about 60 percent is achieved for a fluid flow rate around 220 gpm (14 l/s). Similarly, input power expressed in HP is shown on the right vertical axis as a function of fluid flow rate. At a fluid flow rate of 220 gpm (14 l/s), the pump would require an input power of about 6 brake HP.

16.3 CENTRIFUGAL PUMP OPERATION

16.3.1 Pump Pressures or Heads

The purpose of a pump used for HVAC applications is to achieve the required fluid flow rate by producing enough pressure to overcome the system resistance. The pump impeller produces a positive pressure on the discharge side and a lower or relatively negative pressure on the suction or intake side. It is important to note the use of the term "relative" in a discussion of the pressures that pumps produce. The system elevation static pressure is a major factor in the operation of a hydronic pump in an open system. When the pump is running there is a redistribution of pressures in the system because of the combination of the elevation pressures and the pressures produced by the pump. However, when the pump is shut down the system pressures return to the same values of elevation static established before the pump was started. The pressures produced by the pump add to or subtract from the initial shut-down static pressures.

Since the operating discharge pressure produced by the pump is an increase, this value is added to the shut-down pressure in the discharge piping. The operating suction pressure produced by the pump is a decrease and this value is subtracted from the shut-down pressure in the suction piping. If the system piping and equipment losses were taken into account and the pump was sized to overcome those losses and to withstand the system static and dynamic pressures, the pump should produce the required fluid flow rate.
However, the pump may not be able to achieve the required fluid flow rate due to its sensitivity to the pressure conditions at its inlet. Centrifugal pump operating characteristics are generally not as sensitive to discharge conditions as they are to inlet conditions.

In an open system, a pump does not suck water into its inlet connection. Lift occurs because the pump produces a pressure at its suction that is less than atmospheric pressure and atmospheric pressure then pushes the liquid up and into the pump. The greatest lift possible would occur if there were a perfect vacuum at the pump suction. This would be a vacuum of 29.92 in. Hg (760 mm Hg) which is equivalent to 14.7 psi (101 kPa) or 34 ft of water (10 m) at sea level. Therefore, with a perfect vacuum at the pump suction, the maximum lift would be a distance of 34 ft (10 m) if the liquid is water in a normal temperature range. However, hydronic pumps used in HVAC systems are not able to produce a perfect vacuum so the maximum lift would be less than 34 ft (10 m).

To illustrate, assume that the maximum lift that a hydronic pump can produce is 22 ft (6.7 m). Some of this maximum lift will need to overcome friction loss in the suction pipe. If the friction through an intake strainer and the pipe and fittings totals 5 ft (1.5 m) then the maximum possible lift becomes a distance of 22 ft (6.7 m) maximum possible lift less the 5 ft (1.5 m) of friction losses on the intake or 17 ft (5.2 m). Even after the liquid reaches the pump inlet flange there is a further reduction of pressure between the inlet flange and the eye of the impeller. Such a reduction must exist to overcome internal losses due to friction and turbulence in the pump inlet while still providing a pressure differential between inlet flange and impeller eye that will cause flow into the impeller.

Due to the physical weight of the water being circulated by a hydronic pump, elevation within the system is a major factor in the selection of the pump. Insufficient positive pressure on the suction of a pump can result in the pump failing to operate. In addition, the lowest pressure in the system exists within the pump. The significance of this lowest pressure is that it must not be less than the vapor pressure of the liquid being pumped. If it is less, the result will be a reduction in the quality of liquid pumped and cavitation will result. For a hydronic pump, cavitation sounds like the pump is full of “marbles.” When the pumps are to handle hot liquids or have high inlet pressure drops, care must be taken to see that the required net positive suction head (NPSH) does not exceed the NPSH available (NPSHAR) at the pump or cavitation may result.

### 16.3.2 Practical Pump Curves

Figure 16-1 shows performance curves for a typical centrifugal pump. Figure 16-2 provides a set of performance curves that would be provided by a pump manufacturer for a centrifugal pump. In both figures, the key information conveyed is the relationship between the fluid flow rate of capacity shown on the horizontal axis that the pump will deliver against a certain resistance or head on the left vertical axis.

To illustrate using Figure 16-2, if the system has a total resistance of 240 ft (73 m) and the pump is running at rated speed with a 7.75 in. (200 mm) impeller and 15 HP motor then the pump will be operating at Point A designated the design operating point. Assuming that the net positive suction head requirements are met, the pump will deliver 125 gpm (8 l/s) under these conditions.

The efficiency curve is a measure of the actual pump output versus the output of a theoretically perfect pump that converts all of the input energy from the motor into pumping action. In Figure 16.1, pump efficiency was shown as a single curve that was a function of fluid flow rate for a particular pump. In Figure 16.2, a family of pump curves is provided for varying impeller sizes. As a result, pump efficiency in Figure 16-2 is illustrated by single-value efficiency contour lines. The efficiency of the pump with a 7.75 in. (200 mm) impeller operating at Point A would be between 55 and 58 percent or about 57 percent.

In Figure 16-2, the pump horsepower curves are shown as downward sloping diagonal curves for standard sized motors of 7.5, 10, 15, 20, and 25 HP. Due to the slope of the horsepower curves, the pump with a 7 in. (178 mm) impeller never exceeds 10 HP even though it is close to it at its highest fluid flow rate of 155 gpm (9.8 l/s). When an 8.5 in. (216 mm) impeller is used, both the 15 and the 20 HP curves are both crossed by the pump characteristic curve. At low flow rates, a 15 HP motor would be adequate to drive the pump with an 8.5 in. (216 mm) impeller. However, above a fluid flow rate of about 90 gpm (5.6 l/s) a 20 HP motor would be required to drive the pump with an 8.5 in. (216 mm) impeller. For this reason care should be taken when selecting a pump and motor combination because increasing the fluid flow rate of the pump could exceed the capability of the pump motor. For the 7.75 in. (200 mm) impeller pump used to illustrate pump operation and efficiency, a 15 HP motor would be adequate to drive this pump over its fluid flow rate range.

Pumps for HVAC applications are often specified to be “non-overloading” at any point on the pump curve.
FIGURE 16−2 TYPICAL CENTRIFUGAL PUMP PERFORMANCE CURVES SUPPLIED BY PUMP MANUFACTURERS
A non-overloading centrifugal pump is one that will not overload its associated motor. In the previous example, if the pump with a 7.75 in. (200 mm) impeller pump were paired with a 15 HP motor this pump-motor combination would be non-overloading because the pump performance curve does not cross the pump horsepower curve at any point along its capacity range.

Figure 16-2 also shows the net positive suction head (NPSH) curves which relate to cavitation.

16.3.3 Pump Impeller Staging

Staging of the impellers is one of the ways in which centrifugal pumps overcome their inability of pump against high heads. Pumping through two or more stages multiplies the head capacity. For example, most water well pumps are constructed with multiple stages in order to achieve sufficient head to supply water from wells with a standing water column in excess of 34 ft (10 m) from the surface since well water pumps are open systems and single-stage pump lift would be limited to atmosphere pressure.

16.3.4 Parallel and Series Pump Operation

Due to variation in either pump flow rate or pump total head or requirements in an HVAC system, the use of more than one pump in parallel or series to achieve the needed range in pump flow rate or pump head may be required. Multiple pumps can be used in parallel to increase pump flow rate and multiple pumps in series can be used to increase pump head. Using pumps in series to increase total pump head is not often done in HVAC systems.

Paralleling centrifugal pumps uses a number of pumps in parallel to produce efficient operation over a wide range of flow rates from minimum to maximum flow rate. Typically pumps of equal size are used in parallel operation. Paralleling multiple pumps requires that the number of pumps running is the most efficient for a particular combination of system head and flow. Therefore, care should be taken in staging parallel pumps to ensure that the number operating at any time is the most efficient combination.

16.3.5 Pump Redundancy

It is common practice to provide identical pumps that may or may not be able to operate in parallel. In most cases, one of the two pumps is a spare that is used in the event of the failure of the operating pump. On some systems the chilled water and condenser water system characteristics allow the use of a single spare pump for both systems with valved connections that permit the spare pump to be used on either system.

16.4 HYDRONIC SYSTEM OPERATION

16.4.1 Hydronic System Types

Hydronic system can be classified as either of the following two system types:

- Closed Systems
- Open Systems

16.4.1.1 Closed Systems

In a closed system, the system pressure can be regulated or limited by the pressure relief valve sometimes referred to as a safety valve and the automatic water makeup valve sometimes referred to as the line pressure regulator valve. The pressure in various parts of the system will vary from top where there is less pressure to the bottom where there is more pressure due to the static head whether the pump is running or not. If the pump is located at the top of the system rather than at the bottom, the suction pressure of the pump will be lower.

When the pump is not running, the discharge pressure gage and the suction pressure gauge will have similar if not equal readings. When the pump is started, the gage pressures on the suction and discharge of the pump will change. The discharge pressure will be higher than the suction pressure when the pump is running.

The closed system has resistance to flow and as the fluid flow rate increases so does the total head of the system. Figure 16-3 illustrates a system curve for a hydronic distribution system. The hydronic distribution system curve is similar to the air distribution system curve covered in Chapter 13. This relationship between the fluid flow rate and system total head can be calculated by the following equation:

\[
\frac{H_2}{H_1} = (\frac{\text{gpm}_2}{\text{gpm}_1})^2
\]

Where:

- \(H\) = Total Head
- \(\text{gpm}\) = Fluid Flow Rate

By choosing an system operating point such as the design conditions of 58 ft total head (17.7 m) and 200
Move to left by increasing resistance (throttling values) or reducing impeller size
Move to right by decreasing resistance (install larger piping or equipment), increase impeller diameter, increase pump speed.
The design conditions represented by Point 2 on Figure 16-3 is the intersection of the pump head capacity curve for a fixed impeller size and the design system curve. The pump head capacity curve would be determined from the pump performance curves provided by the pump manufacturer. For example, the pump head capacity curve in Figure 16-3 could be one of the pump operating curves for different impeller sizes shown in Figure 16-2.

When most hydronic systems are put into operation for the first time, the actual system curve is similar to the one designated “piping system oversized” in Figure 16-3 that intersects the pump head capacity curve at Point 6. In order to move the system operating point back toward the desired design conditions, the usual approach is to partially close the discharge valve in order to increase system head and reduce the fluid flow rate. Shifting the system operating point from Point 6 to Point 2 by closing a valve and introducing more resistance into the system is similar to “riding the fan curve” for an air distribution system.

Like air distribution systems, hydronic distribution systems are dynamic and the system operating point is continually shifting due to zones needing more or less chilled or hot water flow. The use of variable frequency drives (VFD) and changing pump speed is a more energy efficient approach to accommodating varying system loads and operating on the design system curve.

16.4.1.2 Open Systems

This same dynamic fluctuations are continually taking place in open systems. For example, where the fill valve is regulating the water level in a sump or basin that is connected directly to the suction side of the pump when the pump draws down the water level on start-up, the make-up water flows until the original water level is achieved. When the pump stops, the excess water drains back into the sump and raises the level on the suction side above its neutral position. As a result, there are not many differences between open and closed hydronic system characteristics. However, lift in an open system includes the elevation weight of the fluid on both sides of the pump instead of just on the discharge side of the pump as it is for a closed system.
FIGURE 16-4 CORRECT PUMP CONNECTION TO EXPANSION TANK

FIGURE 16-5 INCORRECT PUMP CONNECTION TO EXPANSION TANK
temperature is stored in the expansion tank during periods of high operating temperatures and is returned to the system when the system water temperature is lower. The expansion tank must be able to store the required volume of water during maximum design operating temperatures without exceeding the maximum allowable operating pressure and to maintain the required minimum pressure when the system is cold. An automatic fill valve is typically used to maintain the minimum system pressure by supplying water to make up for leakage. Other systems supply additional system water by using a manually operated valve.

The location of the expansion or compression tank connection in relation to the hydronic system pump is important. A properly located pump will pump away from the junction of the tank with the system piping as in Figure 16-4. This location ensures an increase in system operating pressure over and above that set by the tank and the pressure-reducing valve in the make-up water line. However, if the pump discharges into the tank junction with the system piping, there will be a decrease in system pressure during pump operation, see Figure 16-5. When there is a high pump head, the system pressure can be less than atmospheric pressure causing air to be induced into the system. This could lead to air-bound terminal units, reduced flow, pump damage, and increased system deterioration due to new oxygen being brought into the system.
CHAPTER 17

MOTORS AND VARIABLE FREQUENCY DRIVES
17.1 INTRODUCTION

Electric motors are used almost exclusively to drive HVAC equipment in commercial and institutional facilities. An electric motor is an energy conversion device that converts incoming electrical energy to mechanical energy to drive fans, pumps, compressors, and other rotating or reciprocating HVAC equipment. Motors used in buildings for HVAC applications use a significant amount of energy and need to be properly selected and sized. They must also meet minimum efficiency requirements imposed by government regulations, industry standards, and energy codes.

HVAC systems are dynamic and are required to continuously adapt to changing cooling and heating load conditions throughout the building. The most efficient method for motor-driven HVAC equipment to respond to these load changes is by varying the output of the HVAC equipment to match the load requirements. For fans, pumps, and other rotating HVAC equipment, this can be efficiently accomplished by changing the speed of the motor driving the equipment. Historically, electric motors have been single-speed devices with their speed determined by their internal construction and the frequency of their supply voltage. Now, variable frequency drives (VFDs) are being used to control the speed of the motor driving HVAC equipment in order to reduce energy usage, improve performance, and increase overall HVAC system effectiveness and efficiency.

17.2 ELECTRIC MOTOR TYPES

There are three types of electric motors used in commercial and institutional buildings.

- Direct Current Motors
- Synchronous Motors
- Induction Motors

17.2.1 Direct Current Motors

Direct current (DC) motors require a DC power supply and are used when alternate speed control and performance is required over the entire speed range of the motor. In addition to requiring a DC power supply, typical DC motors also require brushes on the motor shaft in order to transfer electric energy to the rotor winding. Variations on this requirement include permanent magnet DC motors that are usually restricted to smaller fractional horsepower motors and brushless DC motors. DC motors are rarely used to drive HVAC equipment. The precise speed and torque control offered by a DC motor is seldom required by HVAC equipment because it can tolerate speed and torque variances and still perform satisfactorily. DC motors also require an inverter to convert the building’s alternating current (AC) power supply to DC, require more maintenance due to rotor brushes and rings, and typically cost more than a comparably sized induction motor. DC motors are used extensively in manufacturing equipment and machine tools where the ability to closely control speed and torque are needed. The main application for DC motors in commercial and industrial buildings is in elevator systems.

17.2.2 Synchronous Motors

Synchronous motors are AC machines although the synchronous motor itself requires both a three-phase AC supply to its stator winding and a DC supply to its rotor winding similar to a DC motor. Synchronous motors will either have an electronic inverter that converts the building’s AC supply to DC for power supply to the rotor or have a mechanical DC generator that is driven by the motor’s shaft that generates DC power for the motor’s rotor winding. When a shaft-driven DC generator is used to supply DC to the rotor winding, the synchronous motor is started as an induction motor. Once it reaches near-rated speed it automatically transitions to operation as a synchronous motor.

Synchronous motors operate at synchronous speed that is determined by the frequency of the AC supply voltage at the motor’s terminals and the number of magnetic poles comprising the motor’s stator. In other words, the speed of a synchronous motor is fixed by the supply voltage frequency and internal motor construction and is constant from no-load to full-load operation. Synchronous motors are rarely used to drive HVAC equipment because of their complexity and the fact that near constant speed from no-load to full-load is not required for most HVAC equipment. Synchronous motors are usually only used to drive HVAC equipment when the horsepower rating of the motor required exceeds that available for induction motors. In large industrial and process plants synchronous motors are used to drive large fans, pumps, and other mechanical equipment due to the required horsepower. In commercial and institutional facilities, synchronous motors are typically only used to drive large chillers that exceed the horsepower rating of commercially available induction motors.

By adjusting the rotor field voltage, the power factor of a synchronous motor can be adjusted to supply reactive power rather than absorbing reactive power. Synchronous motors can be used to improve a commercial or institutional facility’s power factor by over-
17.2 HVAC Systems Applications

17.2.3 Induction Motors

17.2.3.1 Characteristics

The third major type of electric motor is the induction motor. Induction motors comprise approximately 98 percent of the motors used in the United States because of their simple, rugged construction. Figure 17-1 shows a cutaway view of an integral horsepower squirrel-cage induction motor with a totally enclosed fan cooled (TEFC) enclosure. Induction motors use the AC power supplied by the building’s distribution system and are manufactured as either single- or three-phase units. Induction motors have no direct mechanical connection between the stator and rotor. Voltage is induced in the rotor by a rotating magnetic field in the stator winding which produces shaft torque and drives the mechanical load. The induction motor gets its name from the fact that it uses induction rather than brushes and slip rings to induce voltage in the rotor winding. As a result, induction motors are less expensive than DC or synchronous motors and require minimal maintenance over their lifetime.

17.2.3.2 Types

The two types of induction motors are as follows:

- Wound-Rotor Induction Motors
- Squirrel-Cage Induction Motors

Wound-rotor induction motors get their name because their rotors consist of coils of wire similar to DC and synchronous motors. Wound-rotor motors are more expensive than squirrel cage induction motors but have the advantage that inserting a variable resistance in the rotor coil circuit can vary the motor’s torque and speed operating characteristics. This additional control of a wound-rotor induction motor requires a mechanical connection to the rotor windings that involves brushes and slip rings that increase maintenance over the life of the motor. Wound-rotor induction motors are typically used in manufacturing and machine tool applications where the precise control of torque and speed is required. Wound rotor induction motors are not typically used to drive HVAC equipment because the precise control of torque and speed is not required.

Squirrel-cage motors differ from wound-rotor induction motors in the construction of its rotor. Squirrel-cage induction motor rotors are constructed of parallel bars embedded in a solid cylindrical rotor and connected at both ends by solid end rings, see Figure 17-2. Unlike wound-rotor induction motors, there are no rotor windings and no external rotor connections through mechanical brushes and slip rings. As a result, squirrel-cage induction motors are inexpensive in comparison to other motor types. They are extremely reliable, rugged, and require little if any maintenance over their life. Their operating characteristics are well suited for driving HVAC equipment and are used almost exclusively to drive pumps, fans, and other constant load HVAC equipment. Squirrel-cage induction motors are available in horsepower ratings that range from ½ to over 500 horsepower. As a result, even large chillers that once required a synchronous motor drive now use squirrel-cage induction motors.

A squirrel-cage induction motor is essentially a single-speed machine even though its speed varies slightly with load. HVAC equipment including fans and pumps operate more efficiently when their output can be matched to the load that they are serving by varying the speed at which they are operated. The speed of squirrel-cage induction motors can be controlled using variable frequency drives which in turn can control the output of the driven HVAC equipment.

17.3 Induction Motor Size Classification

Induction motors are typically classified as either fractional horsepower motors (FHP) or integral horsepower motors (IHP). As the name implies, FHP induction motors are motors that have a rated horsepower that is less than 1 horsepower. IHP induction motors are motors that have a rated horsepower that is a whole number and 1 horsepower or greater. Three-phase IHP induction motors are used almost exclusively to drive HVAC fans, pumps, and other equipment. Single-phase, IHP induction motors are also available in sizes up to approximately 25 horsepower but are typically only used when three-phase power is not available. Smaller single-phase FHP motors are normally used to drive small fans.

17.4 Induction Motor Purpose Classification

Motor manufactures classify IHP squirrel-cage induction motors by purpose in the following three ways:

- General Purpose
**FIGURE 17-1 TYPICAL INTEGRAL HORSEPOWER SQUIRREL CAGE INDUCTION MOTOR**

Source: www.geocities.com/vijaykumar77/inductionmotor/inductionmotor.html

**FIGURE 17-2 TYPICAL SQUIRREL CAGE INDUCTION MOTOR ROTOR**

Source: www.engineersedge.com/motors/induction_ac_motor.htm
General-purpose motors are motors that have standard ratings, operating characteristics, and mechanical construction. General-purpose motors are usually used to drive HVAC equipment in commercial and institutional buildings. Definite purpose motors are motors that are produced in volume for a specific purpose application in accordance with National Electrical Manufacturer Association (NEMA) standards. An example of a definite purpose motor commonly used in HVAC systems would be motors used in hermetically-sealed motor-compressor refrigeration units. These motors operate in the atmosphere of the refrigerant rather than the normal machine room atmosphere. Special-purpose motors are motors that have operating characteristics or physical construction that make them a one-of-a-kind, custom motor that is designed and built for a specific application and may or may not conform to NEMA standards. Special-purpose motors are rarely used to drive HVAC equipment.

17.5 INDUCTION MOTOR

17.5.1 Operation

Induction motors are comprised of a stator winding and a rotor winding, see Figure 17-1. The stator winding is stationary and located on the frame of the motor and the rotor winding is located on the rotor or drive shaft that rotates and drives the load. When a three-phase voltage is connected to the motor’s stator windings, current flows in the stator winding and a rotating magnetic field is produced. This rotating magnetic field induces a current in the rotor winding that generates a magnetic field in the rotor and rotational torque is developed as the rotor magnetic field attempts to align itself with the rotating stator’s magnetic field. The result is a spinning motor shaft that can be used to drive a fan, pump, or other motor-driven HVAC equipment.

One of the advantages of an induction motor over DC and AC synchronous motors is there is no direct electrical connection between the stator and rotor. Both DC and AC synchronous motors require a mechanical connection, such as slip rings, between the stator and rotor in order to power the rotor. With an induction motor, current is induced in the rotor winding across the air gap between the stator and rotor by the stator’s rotating magnetic field. The motor’s shaft torque and rotation is the result of the interaction between the stator and rotor magnetic fields. Induction motors get their name from the fact that they rely on magnetic induction for operation.

17.5.2 Advantages

Induction motors have a number of advantages over DC motors and AC synchronous motors due to their construction and operation including the following:

- Simple and Rugged Construction
- Minimal Maintenance
- No Brushes or Slip Rings to Maintain for Squirrel-Cage Motors
- High Reliability
- High Efficiency
- Low Cost

However, induction motors have one major disadvantage when compared with DC motors; an induction motor is essentially a single-speed motor. Due to the way an induction motor is constructed, it can only run at one speed which is why variable frequency drives are used to vary the speed of fans, pumps, and other HVAC equipment for more efficient, economical operation.

17.5.3 Synchronous Speed

The speed of an induction motor is primarily determined by the frequency of the supply voltage and the number of poles in its stator winding. The following equation demonstrates the relationship between induction motor shaft speed to the frequency of the supply voltage and the number of poles in the stator winding:

\[
ns = \frac{120 \cdot f}{P}
\]

Equation 17-1

Where:

- \(ns\) = Shaft Synchronous Speed (rpm)
- \(f\) = Frequency of Supply Voltage (Hz)
- \(P\) = Number of Poles in the Stator

This equation provides the synchronous speed or theoretical maximum speed of an induction motor that corresponds to the speed of the stator’s rotating magnetic field. For example, the synchronous speed of a four-pole induction motor that is powered by 60-Hz power system would be as follows:
17.5 HVAC Systems Applications

17.5.4 Full-Load Speed

An induction motor’s full load speed is less than its synchronous speed, see Figure 17-3. This will always be the case because the induction motor’s rotor must rotate at speed less than synchronous speed to develop shaft torque to drive a mechanical load. If the rotor rotated at synchronous speed, then no torque would be developed to drive a mechanical load due to the electromagnetic interacting between the induction motor’s stator and rotor windings. Even at no load, an induction motor’s rotor will rotate at less than synchronous speed because the motor needs to develop enough torque to overcome bearing friction, the resistance of air passing over the rotating rotor, motor cooling fan power, and other mechanical losses.

The difference between actual induction motor speed and synchronous speed is usually referred to as slip. Slip is the difference between synchronous speed and actual speed expressed as a percent of synchronous speed. Slip is calculated as follows:

\[
\text{Percent Slip} = \frac{\text{Synchronous Speed} - \text{Actual Speed}}{\text{Synchronous Speed}} \times 100
\]

To illustrate, consider the four-pole induction motor that had a synchronous speed of 1800 rpm. If the actual speed of this motor at full load provided by the motor manufacturer is 1725 rpm then this induction motor’s full load slip is as follows:

\[
\text{Percent Slip at Full-Load} = \left( \frac{1800 \text{ rpm} - 1725 \text{ rpm}}{1800 \text{ rpm}} \right) \times 100 = 4.2\%
\]

Figure 17-3 illustrates graphically the change in motor speed from no-load to full-load by the dashed vertical lines at the right side of the diagram labeled slip.

17.5.5 Induction Motor Operating Speed

Induction motors are manufactured in standard horsepower ratings and typically do not operate at their rated horsepower on a continuous basis. Therefore, the actual speed of the induction motor is a function of the mechanical load imposed on its shaft by the fan, pump, or other HVAC equipment being driving. The induction motor will operate at a speed between full-load and no-load depending on the load or, said another way, between full-load and no-load percent slip, see Figure 17-3. As the mechanical load on the shaft of the motor changes, so does the motor speed. Further, as the mechanical load on the motor increases, the motor speed also decreases.

17.6 INDUCTION MOTOR SPEED-TORQUE RELATIONSHIP

17.6.1 Induction Motor Speed-Torque Curve

Figure 17-3 illustrates the speed-torque relationship for a typical induction motor that is used to drive HVAC fans and pumps in commercial and institutional HVAC systems. The graphical relationship between motor speed expressed as percent synchronous speed on the horizontal axis and motor percent full-load torque on the vertical axis is commonly referred to as the motor’s torque-speed curve. The speed-torque curve in Figure 17-3 is for a typical NEMA Design B induction motor. In addition to motor speed and torque, Figure 17-3 also illustrates the relationship between both percent synchronous speed on the horizontal axis and percent full-load current on the vertical axis.

17.6.2 Relationship Between Induction Motor Speed And Torque

The typical induction motor’s speed starts at zero percent synchronous speed or 0 rpm and increases to a maximum of just under 100 percent synchronous speed as its maximum no-load speed, see Figure 17-3. Zero percent synchronous speed occurs when the motor’s rotor is at standstill which occurs either at startup or when something is blocking shaft movement and the rotor is unable to turn. When the motor’s rotor is at standstill or blocked and cannot rotate this condition is referred to as “locked rotor”.

Even at no-load, the induction motor’s speed will be less than synchronous speed and there will be some slip. This slip will be less than the motor’s full-load slip but is necessary to develop enough torque to overcome bearing friction, air resistance, cooling fan operation, and other mechanical losses resulting from the motor’s rotation. The induction motor cannot develop any mechanical torque unless the rotor is rotating at a speed less than synchronous speed due to the electromagnetic interaction between the motor’s stator and rotor.

From the typical induction motor torque-speed characteristic in Figure 17-3, it can be seen that a typical induction motor used in HVAC applications passes
through the following four torque-speed points going from startup to full-load:

- Locked Rotor Torque
- Pull-Up Torque
- Breakdown Torque
- Full-Load Torque

### 17.6.2.1 Locked-Rotor Torque

Locked-rotor torque is the torque that the induction motor will develop when its rotor is at rest and rated voltage at rated frequency is applied at its terminals. Locked-rotor torque is also sometimes referred to as starting torque and must be greater than the inertia of the mechanical load that it is driving in order to get the fan, pump, or other HVAC load running and accelerating to full-load speed.

### 17.6.2.2 Pull-Up Torque

Pull-up torque is the minimum torque that the induction motor will develop when accelerating from rest to breakdown torque, see Figure 17-3.

### 17.6.2.3 Breakdown Torque

After passing through pull-up torque, the induction motor continues to accelerate until it reaches breakdown torque. Breakdown torque is the maximum torque that the motor will develop when rated voltage at rated frequency is applied to the motor’s terminals, see Figure 17-3.

### 17.6.2.4 Full-Load Torque

After passing through the breakdown torque-speed point, the induction motor continues to accelerate until it reaches its full-load, torque-speed point. Full-load torque represents the torque necessary to produce the induction motor’s rated horsepower at the induction motor’s full-load speed.

### 17.6.3 Relationship Between Induction Motor Speed and Current

Figure 17-3 shows the induction motor’s current as a function of motor speed. The induction motor draws a very high current at starting or when the rotor is blocked and at standstill, see Figure 17-3. This high starting current approximately six times full-load current for a typical induction motor driving a fan, pump, or other HVAC equipment and is referred to as locked-rotor current. In HVAC equipment manufacturer literature, maximum locked-rotor current is often given in amperes and designated locked-rotor amperes (LRA).

Locked-rotor current is important because it can cause nuisance tripping of HVAC equipment overcurrent devices on startup. In addition, starting large HVAC equipment can result in very large transient currents that can cause voltage problems for other equipment served by the building’s distribution system as well as increased utility demand. The possible impacts of starting HVAC equipment such as chillers with large induction motors can be mitigated using soft motor-starting techniques or variable frequency drives, discussed later in this chapter.

As the induction motor accelerates, the current drawn by the induction motor decreases until the motor reaches its operating point which is determined by its mechanical load somewhere between full-load and no-load torque. Figure 17-3 shows the full-load current and is the current drawn by the induction motor at its full-load torque or horsepower rating when rated voltage at rated frequency is supplied at the motors terminals. Full-load current is also referred to as the motor’s nameplate current rating and is often given in HVAC manufacturer’s literature as full-load amperes (FLA).

Even under no load, an induction motor’s current will not be zero because the induction motor has mechanical losses and magnetic losses that need to be overcome to keep the rotor running.

### 17.7 INDUCTION MOTOR CHARACTERISTICS

#### 17.7.1 Motor-Driven HVAC Equipment

HVAC equipment is normally supplied with the motor or motors required to drive the equipment at its rated output. The characteristics of an induction motor supplied with a piece of motor-driven HVAC equipment are typically determined by the HVAC equipment manufacturer based on mechanical load characteristics and intended operation of the fan, pump, or other equipment. However, the testing, adjusting, and balancing (TAB), commissioning, operation, and maintenance of installed HVAC equipment often requires the understanding of motor operating characteristics along with being able to determine when a motor should be replaced.
17.7.2 Induction Motor Operating Characteristics

There is a set of characteristics that describes the operation of an induction motor. Most of this information can be obtained from the nameplate of the motor. However, even though most of this information is required to be provided on the nameplate of the motor, there is no required format for presenting the information and different manufacturers present the information differently. If the information is not presented in a usable format or is shown on the motor nameplate, the needed information can be obtained directly from the motor manufacturer. The following lists common motor induction motor characteristics that are generally provided in manufacturer catalogs and shop drawings, required in HVAC equipment specifications, and listed on the HVAC equipment’s motor nameplate:

- Horsepower
- Service Factor
- Rated Voltage Magnitude
- Rated Voltage Frequency
- Number of Phases
- Full-Load Current
- Power Factor
- Full-Load Efficiency
- Full-Load Speed
- NEMA Design Letter
- Code Letter
- Duty Cycle
- Ambient Temperature
- Altitude
- Frame Size
- Enclosure Type

For additional information about induction motor operating characteristics as well as other information about design, construction, and application of electric motors, see the National Electrical Manufacturers Association (NEMA) Standard MG-1 entitled Motors and Generators or the condensed Information Guide for General Purpose Industrial AC Small and Medium Squirrel-Cage Induction Motor Standards.

17.7.2.1 Horsepower

Horsepower (hp) is the English unit for mechanical power that is also defined as energy expended per unit time. Motors are rated in horsepower and in particular brake horsepower (bhp). Brake horsepower is the horsepower developed at the motor’s shaft without any of the losses associated with the fan, pump, or other HVAC equipment connected to the motor’s shaft.

17.7.2.2 Service Factor

The service factor of a motor indicates how much horsepower in excess of its rating a motor can develop at its shaft continuously without damaging the motor. Service factor is expressed as a decimal multiplier that can be thought of as a safety factor. For example, a 20 hp motor with a typical 1.15 service factor would be capable of driving a 23 hp mechanical load without overloading or damaging the motor. An induction motor’s service factor can be used by HVAC equipment manufacturers to handle expected temporary or emergency overloads without going to the next higher standard motor size which in this case would be a 25 hp motor. Typical service factors for induction motors are 1.00 which allows no overload, 1.15, and 1.25. Most HVAC equipment use induction motors with a 1.15 service factor.

17.7.2.3 Rated Voltage Magnitude

The rated or nameplate voltage is the nominal alternating current (AC) effective or root-mean-square (rms) voltage on which an induction motor is designed to operate. However, like HVAC systems, electric power distribution systems are dynamic and the voltage at the terminals of motors is always changing due to fluctuations in the voltage supplied by the utility or changes internally in the building’s electrical load.

When electric motors were first used to drive mechanical loads in industrial plants problems were encountered because the motor’s rated mechanical output was determined at the nominal system voltage at which it was going to be used. Early electric power distribution systems were often subject to considerable voltage drop and lower than rated voltage delivered to the motor terminals often resulted in motors failing to start and being damaged due to overheating when driving its rated mechanical load. To compensate for this, the voltage at which motor manufacturers rated motors was less than nominal voltage of the electric distribution system on which they were installed. Today, elec-
tric power distribution systems in commercial and institutional buildings are much better and the voltage delivered to the terminals of a motor is typically closer to nominal system voltage but the rated voltage that induction motors are designed, tested, and rated for is less than nominal system voltage, in most cases. Table 17-1 provides standard electric power distribution system voltages and the corresponding rated motor voltages.

<table>
<thead>
<tr>
<th>Nominal Distribution System Voltage (V rms)</th>
<th>Rated Or Nameplate Motor Voltage (V rms)</th>
</tr>
</thead>
<tbody>
<tr>
<td>120</td>
<td>115</td>
</tr>
<tr>
<td>208</td>
<td>200</td>
</tr>
<tr>
<td>208</td>
<td>208</td>
</tr>
<tr>
<td>240</td>
<td>230</td>
</tr>
<tr>
<td>480</td>
<td>460</td>
</tr>
</tbody>
</table>

Table 17-1 System and Motor Voltages

17.7.2.3.1 Dual Voltage Rated Motors

Motors can have a dual voltage rating where standard rated or nameplate voltages are integer multiples of one another. The two rated motor voltages are generally shown in manufacturer literature, shop drawings, and on motor and HVAC equipment nameplates with the lower rated voltage and the higher rated voltage separated by a slash mark (/). Common examples of dual-voltage rated motors encountered in HVAC equipment are:
- 115/230 V
- 230/460 V

Dual-rated motors can be connected and operated at either of the listed voltages depending on the nominal voltage of the system that the HVAC equipment or motor will be connected to. A wiring diagram showing how to connect the motor for either the lower or higher voltage is provided in the HVAC equipment installation instructions, shop drawings, motor nameplate, or in all three places.

Dual rated motors will also have their rated current shown for each voltage in the same way. For instance, for a three-phase induction motor with a dual voltage rating of 230/460 volts might have its rated current shown as 54/27 amperes. When connected for service at 230 volts, the induction motor would have a rated, full-load current of 54 amperes. When connected for service at 460 volts, the induction motor would have a rated, full-load current of 27 amperes.

17.7.2.3.2 Rated Voltage Tolerance

The actual voltage delivered to the terminals of a motor will normally not be either the nominal distribution system voltage or the motor’s rated or nameplate voltage. As a result, motors are designed to operate by motor manufacturers at their rated horsepower for a +/-10 percent range of voltage around the motor’s rated or nameplate voltage per NEMA MG-1. Therefore, an induction motor driving a fan, pump, or other HVAC equipment should operate satisfactorily within the voltage ranges, see Table 17-2.

Even though an induction motor will operate satisfactorily over a range of +/-10 percent of its rated or nameplate voltage, the voltage variation will impact important motor operating parameters that in turn can impact the operation of motor-driven HVAC equipment. The general impact of voltage variations are described in the Institute of Electrical and Electronics Engineers’ (IEEE) Standard 141 entitled IEEE Recommended Practice for Electric Power Distribution for Industrial Plants which is also known as the IEEE Red Book. Table 17-3 provides the typical impact of applied voltage that is at 90 and 110 percent of the motors rated or nameplate voltage on an induction motor’s operation.

<table>
<thead>
<tr>
<th>Nominal Distribution System Voltage (V rms)</th>
<th>Rated Or Nameplate Motor Voltage (V rms)</th>
<th>Rated Voltage Tolerance</th>
</tr>
</thead>
<tbody>
<tr>
<td>120</td>
<td>115</td>
<td>Minimum Voltage 90% Rated (V rms)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Maximum Voltage 110% Rated (V rms)</td>
</tr>
<tr>
<td>208</td>
<td>200</td>
<td>180</td>
</tr>
<tr>
<td>208</td>
<td>208</td>
<td>188</td>
</tr>
<tr>
<td>240</td>
<td>230</td>
<td>207</td>
</tr>
<tr>
<td>480</td>
<td>460</td>
<td>414</td>
</tr>
</tbody>
</table>

Table 17-2 Voltage Tolerance Ranges
### Table 17–3 Voltage Impact on Induction Motor’s Operation

<table>
<thead>
<tr>
<th>Motor Operating Characteristic</th>
<th>Typical Motor Voltage Variation Impact</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>90% Rated Voltage</td>
</tr>
<tr>
<td>Starting Torque</td>
<td>-19%</td>
</tr>
<tr>
<td>Maximum Running Torque</td>
<td>-19%</td>
</tr>
<tr>
<td>Percent Slip</td>
<td>+22%</td>
</tr>
<tr>
<td>Full-Load Slip</td>
<td>-0.2% to -1.0%</td>
</tr>
<tr>
<td>Starting Current</td>
<td>-10%</td>
</tr>
<tr>
<td>Full-Load Current</td>
<td>+5% to +10%</td>
</tr>
<tr>
<td>No-Load Current</td>
<td>-10% to -30%</td>
</tr>
<tr>
<td>Temperature Rise</td>
<td>+10% to +15%</td>
</tr>
<tr>
<td>Full-Load Efficiency</td>
<td>-1% to -3%</td>
</tr>
<tr>
<td>Full-Load Power Factor</td>
<td>+3% to +7%</td>
</tr>
<tr>
<td>Magnetic Noise</td>
<td>Slight Decrease</td>
</tr>
</tbody>
</table>

### 17.7.2.3.3 Voltage Unbalance

In addition to the impact of variations in voltage magnitude, three phase induction motors can also be impacted by an unbalance in the magnitude of voltage between the three phases. Per NEMA Standard MG-1, induction motors should operate satisfactorily up to a maximum voltage unbalance of 5 percent. However, the cause of any significant voltage unbalance should be investigated and corrected because unbalanced phase voltages at the terminals of a three-phase induction motor can cause a corresponding percent current unbalance in the motor’s stator windings that is six to ten times the percent supply voltage unbalance. This unbalanced current can result in increased motor operating temperatures, increased motor losses, and decreased operating efficiency as well as shortened motor life.

### 17.7.2.4 Rated Voltage Frequency

The nominal rated voltage power system frequency throughout the United States is almost exclusively 60 Hertz (Hz) which is 60 cycles per second. There are some isolated areas in the United States where the power system frequency may be different but these locations are rare and often in remote areas with their own dedicated generation. Induction motor characteristics including speed depend on the frequency of the voltage to the motor. Frequency is rarely a problem because U.S. utilities are required to maintain a very tight frequency tolerance of between 59.9 to 60.1 Hz.

However, most of the world (including Europe) has standardized on a nominal rated power system frequency of 50 Hz and motor-driven HVAC equipment produced for operation on these systems may not operate properly in the U.S. When purchasing or installing motor-driven HVAC equipment that was produced outside of North America always check that the HVAC equipment and its motors are rated for operation at 60 Hz.

### 17.7.2.5 Number Of Phases

Induction motors are also designed for either single-phase or three-phase operation. Single-phase motors are usually smaller horsepower motors and typically have a rated voltage of 115, 200, 208, or 230 volts and a maximum horsepower rating of 10. Three-phase induction motors are usually larger and have a rated voltage of 200, 208, 460, or higher volts and can have a horsepower rating in the hundreds. The availability of three-phase power as well as the size of the HVAC equipment or motor will determine whether the HVAC equipment should be rated for single- or three-phase.
17.7.2.6 Full-Load Current

Full-load current is the rated current that a motor will draw when it is driving its rated mechanical load and has rated voltage at its terminals. The motor’s full-load current is when the motor is operating at steady state and may be significantly different when the motor is started or during transient load conditions. When an induction motor is started from rest, it can draw six to ten times its rated full-load current as it overcomes the driven fan or pump inertia and accelerates to full speed. Measuring the actual current drawn by a motor during operation and comparing the measured operating current to the rated full-load current is a quick method for determining if a motor is overloaded mechanically or other problems.

17.7.2.7 Power Factor

Power factor is the ratio of the induction motor’s full-load real power input expressed in watts (W) to its full-load apparent input power expressed in voltamperes (VA). It is typically expressed as a percentage. It should be noted that the power factor for an integral horsepower induction motor is not constant but varies from no load to full load. At no load or startup the motor’s power factor will be at its minimum and it will increase with load until it reaches its maximum at or near full mechanical load.

17.7.2.8 Full-Load Efficiency

Full-load efficiency is a measure of how efficiently the induction motor converts real electrical power input at the motor terminals to mechanical output at the motor shaft. Motor efficiency is the ratio of the motor’s input real electrical power to its output mechanical power expressed as a percent. Motor losses include resistive losses in the windings, magnetic losses in the rotor and stator, friction losses in bearings, air resistance to the rotating rotor, ancillary equipment losses such as cooling fans on the motor shaft, and others. In general, large motors are more efficient than smaller motors and can have efficiencies greater than 95 percent.

The efficiency of IHP squirrel-cage induction motors continues to increase in response to the demand for increased building efficiency by building owners as well as minimum efficiencies established by federal and state legislation, local building and energy codes, and industry standards. When specifying, procuring, or replacing a motor or piece of motor driven HVAC equipment, care must be taken to ensure that the efficiency of the motor or motor(s) supplied meets the required minimum efficiency. In addition, since HVAC system motors are a significant portion of a commercial or institutional building’s energy use, an economic analysis should be performed to evaluate if better than minimum efficiency motors are economically justified when specifying, purchasing, or repairing motor-driven HVAC equipment.

17.7.2.9 Full-Load Speed

Full-load speed is the speed of the induction motor expressed in revolutions per minute (rpm) when its shaft is loaded at its rated mechanical load. Induction motors never operate at synchronous speed due to slip. The rated full-load speed provided in the motor manufacturer’s catalog, shop drawing, or motor nameplate will always be less than synchronous speed. Table 17–4 provides some typical rated full-load integral horsepower squirrel-cage induction motor speeds for a 60 Hz system and given number of poles.

17.7.2.10 NEMA Design Letter

NEMA Standard MG-1 assigns design letters to three-phase squirrel-cage induction motors. The purpose of these design letters is to provide a method for classifying motors based on their torque-speed characteristics. NEMA MG-1 classifies integral horsepower squirrel-cage induction motors based on their torque-speed curves as either Design A, B, C, or D. Figure 17–4 illustrates the torque-speed characteristics for each of these design letters. It is important that the torque-speed characteristics of the HVAC equipment being driven by an integral horsepower squirrel-cage induction motor be matched to the motor torque-speed characteristics as classified by NEMA design letters.

<table>
<thead>
<tr>
<th>Number Of Poles</th>
<th>Synchronous Speed (rpm)</th>
<th>Rated Full Load Speed (rpm)</th>
<th>Percent Slip (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2</td>
<td>3600</td>
<td>3450</td>
<td>4.2</td>
</tr>
<tr>
<td>4</td>
<td>1800</td>
<td>1725</td>
<td>4.2</td>
</tr>
<tr>
<td>6</td>
<td>1200</td>
<td>1160</td>
<td>3.3</td>
</tr>
<tr>
<td>8</td>
<td>900</td>
<td>870</td>
<td>3.3</td>
</tr>
</tbody>
</table>

Table 17–4 Induction Motor Speeds for Specified Poles
17.7.2.10.1 NEMA Design A Motor

Figure 17-4 shows the torque-speed curve for a Design A motor. Design A motors are general-purpose motors that are characterized by having high locked rotor or starting currents and average locked rotor torques. The slip associated with Design A motors is typically less than 5 percent. Design A motors are suitable for applications where the driven mechanical load has low inertia and starts are infrequent. Typical applications for Design A motors include fans, blowers, pumps, and machine tools.

17.7.2.10.2 NEMA Design B Motor

Design B motors are the most common NEMA design classification used in commercial and institutional building applications including HVAC fans and pumps. Like Design A motors, Design B motors are general-purpose motors that have average starting torques and currents along with relatively high breakdown torque, see Figure 17-4. Design B motor pull-up torques allow for rapid acceleration to full load speed. Slip is limited to 5 percent or less at full load for Design B motors.

17.7.2.10.3 NEMA Design C Motor

A Design C motor has high starting torque and average starting current, see Figure 17-4. Breakdown torque for a Design C motor is slightly less than that of a Design B motor but slip is usually slightly more than a Design B motor but typically less than 5 percent. Design C motors are usually used in applications where the driven mechanical equipment breakaway and starting torques are higher than Design B motors. The efficiency of a Design C motor is usually less than a Design B motor. Due to their high starting torque, Design C motors are often used to drive mechanical loads with high starting inertia such as large centrifugal blowers and loaded starts such as piston pumps and compressors.

17.7.2.10.4 NEMA Design D Motor

Design D motors have high starting torques, moderate starting currents, and high breakdown torques, see Figure 17-4. Slip for a Design D motor often exceeds 5 percent at full load. Design D motors are typically used in applications where the driven load is heavy, applied suddenly, and removed at frequent intervals. Design D motors have very high starting torques which makes them suitable for driving mechanical loads with very high inertia and loaded starts such as punch presses, shears, and forming machines. Design D motors are seldom used to drive HVAC equipment.

17.7.2.10.5 NEMA Design E Motor

Design E motors are no longer addressed in NEMA Standard MG-1 or produced. Design E motors were introduced in the mid-1990s in response to the motor efficiency requirements mandated by the 1992 Energy Policy Act (EPACT) that set minimum efficiency requirements for both standard general purpose Design A and B motors. Design E motors were also close to the operating characteristics of the International Electrotechnical Commission (IEC) standard metric motors used in Europe and other parts of the world. Design E motor characteristics were close to those of Design A and B motors and the need for Design E motors was eliminated by energy-efficient Design A and B motors.

17.7.2.11 Code Letter

Locked rotor indicating code letters associated with integral horsepower squirrel-cage induction motors are provided in Table 430.7(B) of the National Electrical Code (NEC) that is published by the National Fire Protection Association (NFPA) as NFPA 70. Code letters range from A through V and define the allowable inrush current for the motor by a range of values given in kilovoltamperes (kVA) per horsepower. For example, a Code Letter H motor’s locked rotor or starting current will range from 6.3 kVA/hp to 7.09 kVA/hp. For a 20-hp, 460-volt integral horsepower squirrel cage induction motor this would result in an inrush current of between 158 and 178 amperes starting which would be between about 5.9 and 6.6 times the motor’s rated full-load current of 27 amperes.
FIGURE 17-4 NEMA DESIGN LETTER TOQUE-SPEED CURVES
17.7.2.12 Duty Cycle

The duty cycle for a driven mechanical load is a measure of the operating characteristics of that load with respect to time. The most common duty cycle for HVAC fans and pumps is continuous which equates to a near constant mechanical load running for an indefinitely long period of time or continuously. Intermittent duty mechanical loads operate for alternate intervals of load and no load with each interval of load or no load being for a definite time period. Standard duty cycles range from five minutes to one hour. Intermittent duty mechanical loads are often machine tools and other industrial fabrication and process equipment. Generally, all motors used to drive HVAC equipment in commercial and institutional buildings should be rated for a continuous duty cycle.

17.7.2.13 Ambient Temperature

Induction motors are typically designed by the motor manufacturer to operate continuously at full-load in an ambient temperature of 40°C (104°F). A major cause of induction motor failure is insulation deterioration and breakdown due to higher than rated ambient temperature. To avoid premature failure, motors operating in higher than rated ambient temperatures must be derated in accordance with the manufacturer’s recommendations or increased to the next higher standard size.

17.7.2.14 Altitude

Induction motors are also typically designed to operate continuously at sea level in a maximum 40°C (104°F) ambient temperature. At higher altitudes the air is thinner and less capable of cooling the motor. If an induction motor will be installed at high altitudes, above 3300 ft (1000 m), then the motor supplied with the HVAC equipment should be sized in accordance with the manufacturer’s recommendations.

17.7.2.15 Frame Size

Motor frame size needs to be considered when replacing an induction motor in an existing piece of HVAC equipment. Standard motor frame sizes are defined in NEMA Standard MG-1 and the frame size designation which typically consists of three numbers and a letter such as 364T determines a number of key motor installation dimensions including the motor mounting plate bolt hole pattern, shaft height, and shaft diameter.

17.7.2.16 Enclosure Type

Induction motors are also classified by the type of enclosure. NEMA Standard MG-1 classifies motor enclosure types based on the degree to which the enclosure protects the motor from its environment and the motor’s method of cooling. Motor enclosures range from a common open drip-proof (ODP) motor enclosure to an explosion-proof motor enclosure for use in explosive atmospheres. The most common induction motor enclosures used in HVAC equipment are the following:

- Open Drip-Proof
- Totally Enclosed Fan Cooled

ODP induction motors are typically used in clean indoor environments and allow ambient air in the space where the motor is located to circulate through the motor in order to cool it. ODP enclosures are designed and tested to protect the motor from liquid drops or solid particles that strike or enter the motor enclosure at 0 to 15 degrees downward from the vertical.

Totally enclosed fan cooled (TEFC) induction motors are also commonly used both inside and outside to drive HVAC equipment. Figure 17-1 illustrates a TEFC induction motor. A TEFC induction motor is totally enclosed and does not allow air to enter or circulate through the motor to cool it, see Figure 17-1. Instead, the fan used to cool the motor is located outside the enclosure and draws air across the outside of the enclosure to cool the motor. While TEFC enclosures fully enclose the motor, they are not intended to be fully air or watertight.

17.8 MOTOR STARTING

Most integral horsepower induction motors used to drive HVAC fans, pumps, and other equipment are started by connecting the motor directly to the full-voltage electrical supply using a contactor in a standard full-voltage, non-reversing (FVNR) motor starter, see Figure 17-5. This method of starting motors is also referred to as across-the-line starting. Full-voltage starting results in starting or inrush current of between four to ten times rated motor full-load current with the typical induction motor used to drive HVAC equipment drawing about six times full-load current at startup. For most HVAC equipment installed in commercial and institutional buildings, this inrush impacts other equipment or the facility’s utility demand charge. As a result across-the-line starting is commonly used because it is simple and does not require special internal motor wiring or controls.
FIGURE 17–5 TYPICAL FULL-VOLTAGE MOTOR STARTER

THREE-PHASE POWER SUPPLY

CONTROLLER & MOTOR DISCONNECT WITH MOTOR BRANCH CIRCUIT SHORT CIRCUIT PROTECTION

MOTOR CONTROLLER WITH MOTOR BRANCH CIRCUIT OVERLOAD PROTECTION

THREE-PHASE INTEGRAL HORSEPOWER INDUCTION MOTOR

M
For larger induction motors that drive chillers and other large equipment where full-voltage starting currents would result in voltage dips on the facility’s distribution system or increased demand charges, soft start methods should be considered when specifying or procuring HVAC equipment. These soft start methods include the following:

- Part Winding Starting (PWS)
- Autotransformer Stating
- Wye Start/Delta Run

It should be noted that variable frequency drives that will be covered in the next section also reduce inrush current for integral horsepower motors at startup.

### 17.9 VARIABLE FREQUENCY DRIVES

#### 17.9.1 Speed Control For Efficient HVAC System Operation

Integral horsepower induction motors used to drive fans, pumps, and other HVAC equipment in commercial and institutional buildings are single-speed electromechanical energy conversion devices. Most HVAC systems and the equipment that comprise them are designed to meet building cooling requirements on the hottest days of the year and meet building heating requirements on the coldest days of the year. As a result, the HVAC system and its associated equipment require full capacity only a fraction of the time each year. Instead, the HVAC system operates at reduced capacity most of the time.

A VAV HVAC system reacts to reduce cooling load in zones served by adjusting the position of VAV air terminal dampers to reduce the amount of conditioned air supplied to the zone. This reduction in conditioned airflow to the zone results in the desired temperature adjustment in the zone. However, without a corresponding reduction in fan speed the fan will continue to operate at its rated output which will result in an increase in fan static pressure and a corresponding increase in required fan horsepower, see Figure 13-13. In other words, operating the VAV HVAC system at reduced airflow results in the fan “riding the fan curve” with a corresponding increase in energy use for a reduced cooling load.

As discussed in Chapter 13, reducing the fan speed reduces airflow, static pressure, and brake horsepower in accordance with the fan laws. Figure 13-14 illustrates that reducing fan speed by 10 percent results in a corresponding reduction of 27 percent in the required fan brake horsepower instead of actually increasing fan horsepower for the same airflow when maintaining constant fan speed and “riding the fan curve”, see Figure 13-13. Reduction in fan brake horsepower results in a corresponding reduction in the electric energy use by the induction motor driving the fan.

The increase in efficiency and the corresponding energy savings resulting from the ability to adjust the speed of the IHP squirrel-cage induction motor was illustrated using the example of a supply fan in a VAV HVAC system. Similar efficiency increases and energy savings can be realized by varying the speed of fans, pumps, and other HVAC equipment under partial load conditions throughout the HVAC system, see Figure 17-6. Variable frequency drives (VFDs) can be used to vary the speed of IHP squirrel-cage induction motors in commercial and institutional HVAC systems to increase HVAC system efficiency where required by energy codes and standards or where the additional investment in a VFD can be economically justified based on a life cycle cost analysis.

#### 17.9.2 Variable Frequency Drive Operation

A VFD is a solid-state electronic power conversion device that converts a sinusoidal input voltage of constant frequency and magnitude into a variable frequency and magnitude voltage at its output. A VFD is installed between the building distribution system that is supplying 60-Hertz electric power to the motor and the IHP squirrel-cage induction motor that drives the fan or other HVAC equipment, see Figure 17-7.

The VFD controls the speed and torque characteristics of an IHP squirrel-cage induction motor that is used to drive a fan, pump, or other HVAC equipment. The purpose of a VFD is to drive the HVAC equipment at the lowest speed required to deliver the necessary system capacity required under partial load conditions. A VFD controls the speed, torque, and resulting horsepower output of the IHP squirrel-cage induction motor that it drives.

In the United States, the frequency of the VFD input voltage is 60 Hz and the nominal three-phase voltages typically available in commercial and institutional facilities are 208, 240, and 480 volts. A VFD is a multistage device that takes its three-phase input voltage and converts it to DC by rectifying and filtering it in the first stage. The second stage of a VFD is referred to as an inverter because it takes the DC output from the first stage and converts it to a variable frequency and magnitude output voltage. Depending on the design of the VFD, the output voltage might be a quasi sinusoidal voltage waveform consisting of fixed duration pulses of varying polarities and magnitudes.
FIGURE 17–6 HVAC SYSTEM VFD USE

Source: Schneider Electric (Square D) Data Bulletin 8800DB0601 08/2006

HVAC Systems Applications • Second Edition 17.17
FIGURE 17-7 VFD DRIVE SYSTEM FUNCTIONAL DIAGRAM

- THREE PHASE POWER SUPPLY
- VFD
- VARIABLE FREQUENCY DRIVE
- INDUCTION MOTOR
- MOTOR-FAN COUPLING
- VFD OUTPUT
- SUPPLY FAN
- SUPPLY AIR
17.9.3 Variable Frequency Drive Benefits

The benefits of using a VFD to drive variable load motor-driven HVAC equipment include:

- Improved Motor Power Factor
- Reduced Motor Starting Current
- Reduced Thermal and Mechanical Stresses on Motors and Coupling Devices During Startup
- Energy Savings

The energy savings that result from the use of a VFD to drive fans and pumps comes from the affinity laws that relate flow, pressure, and power for fans and pumps. These laws were covered in Chapter 13 for fans in Chapter 16 for hydronic pumps. The ratio of flow and speed for HVAC fans and pumps is expressed by the following relationship:

\[
\frac{\text{flow}_1}{\text{flow}_2} = \left(\frac{\text{rpm}_1}{\text{rpm}_2}\right)^3
\]

Where flow is the volumetric rate of supply of air for a fan or water for a hydronic pump.

Similarly, brake horsepower required for a given fan or pump speed is governed by the following ratios for both HVAC fans and pumps:

\[
\frac{\text{Bhp}_1}{\text{Bhp}_2} = \left(\frac{\text{rpm}_1}{\text{rpm}_2}\right)^3
\]

Combining these two affinity laws yields the relationship between the ratio of brake horsepower and the ratio of flow rate for HVAC fans and pumps:

\[
\frac{\text{Bhp}_1}{\text{Bhp}_2} = \left(\frac{\text{flow}_1}{\text{flow}_2}\right)^3
\]

Rewriting this equation illustrates the impact on brake horsepower resulting from changing the volumetric rate of air or water flow through the fan or pump, respectively, by changing fan or pump speed.

\[
\text{Bhp}_2 = \left(\frac{\text{flow}_2}{\text{flow}_1}\right)^3 \cdot \text{Bhp}_1
\]

To illustrate the relationship between required brake horsepower and airflow, consider the amount of power required by a fan or pump when the respective air or water volumetric flow rate is 70 percent of the HVAC equipment’s rated capacity by reducing the drive motor’s speed. The required brake horsepower required to be delivered by the drive motor to deliver 70 percent of the fan or pump its rated capacity is as follows:

\[
\text{Bhp} @ \text{reduced speed} = \left(\frac{70\% \text{ flow} @ \text{reduced speed}}{100\% \text{ flow} @ \text{rated speed}}\right)^3 \times \left(\frac{100\% \text{ Bhp} @ \text{rated speed}}{100\% \text{ flow} @ \text{rated speed}}\right)
\]

Therefore, when a fan or pump is required to supply 70 percent of its rated output at reduced speed it only requires 34 percent of its rated input brake horsepower. This magnitude reduced brake horsepower input to the fan or pump is not realized when the fan or pump volumetric flow rate is throttled back using dampers or valves, respectively. Figure 17-8 illustrates the relationship of percent flow, pressure, and brake horsepower to speed for typical HVAC fans and pumps.

17.9.4 Variable Frequency Drive Installation Requirements

Installation of a VFD should be in accordance with both VFD and motor manufacturer recommendations as well as Article 430/Motors, Motor Circuits, and Controllers of the National Electric Code (NEC). The requirements of NEC Article 430 apply to VFD installation except as specifically supplemented or modified by Part X/Adjustable Speed Drive Systems which includes conductor sizing, motor overload and overtemperature protection, disconnect requirements, and other VFD-specific installation requirements.

Most VFDs incorporate the necessary motor branch circuit short circuit and overload protection as well as the controller disconnect required by the NEC. Therefore, separate motor branch circuit short circuit and overload protection as well as a controller disconnect, typically required for full-voltage IHP squirrel-cage induction motor starting may not be required if these components are supplied by the VFD or HVAC equipment manufacturer as an integral part of the VFD or HVAC equipment, see Figure 17-3. When specifying or purchasing a VFD or HVAC equipment with an integral VFD, it should be verified with the manufacturer whether of not the NEC-required controller disconnect as well as motor branch circuit short circuit and overload protection is supplied as part of the VFD.
FIGURE 17-8 FAN AND PUMP OPERATION AS A FUNCTION OF SPEED

A = % FLOW  
B = % PRESSURE  
C = POWER REQUIRED
CHAPTER 18
HVAC SYSTEM CONTROL
18.1 INTRODUCTION

HVAC systems seldom operate at their maximum design point and both the external and internal thermal loads are constantly changing. The purpose of HVAC system control is to ensure that the HVAC system can effectively and efficiently adapt to changing outdoor conditions as well as changing internal occupancy and activities.

18.2 CONTROL SYSTEM BASICS

18.2.1 Control System Elements

An HVAC control system is comprised of the following seven elements:

- Controlled Process
- Controlled Variable
- Setpoint
- Sensor
- Controller
- Controlled Agent
- Controlled Device

Figure 18-1 provides a generic block diagram of an HVAC control system that will be used to illustrate the function of each of these seven elements. A geometric shape in Figure 18-1 represents each of the above elements. Rectangles represent processes where an action is taken, rectangles with rounded ends represent sensors that measure the controlled variable, circles represent the point where a measured variable is monitored, and parallelograms represent outside input to the control process. Solid lines linking the geometric shapes represent physical relationships between the control elements and dashed lines represent signal paths where information is passed between the control elements.

18.2.1.1 Controlled Process

A process can be defined as a systematic operation that transforms an input to an output for a definite purpose. The quality of a process is the degree to which the output of the process meets the requirements of the people or follow-on processes using the output. The controlled process is the overall HVAC system, an HVAC subsystem, a piece of HVAC equipment, or some other HVAC system device or function that is controlled by the control system, see Figure 18-1. Figure 18-1 shows that the controlled process is the focus of the control system.

To illustrate a controlled process in the context of an HVAC system, consider the simple block diagram of a multizone VAV HVAC system serving a zone through a VAV terminal unit, see Figure 18-2. The controlled process in this case is the VAV terminal unit with the input being the constant temperature conditioned airflow supplied by the upstream air-handling unit and the output being the thermal comfort experienced by occupants in the zone served.

18.2.1.2 Controlled Variable

The controlled variable is the condition that the control system measures and controls. In Figure 18-1, the controlled variable is represented by the circle at the output of the controlled function. The controlled variable is the characteristic of the controlled process output that is measured and controlled as an indicator of the quality of the process. Essentially any physical attribute of the output stream that can be measured could be the controlled variable. If the controlled process is a zone in the VAV HVAC system discussed above, then controlled variable might be the measured temperature in the zone. The quality of the system output in this case might be judged based solely on the ambient temperature experienced by occupants in this zone.

Figure 18-1 is a simple block diagram that is being used to define each of the seven elements of an HVAC control system as well as illustrate the interrelationship between the elements. For simplicity, the controlled variable is shown as a single variable measured and controlled at the output of the controlled process. In practical HVAC control schemes there may be multiple controlled variables for a single controlled process. For example, as was discussed in Section 1.4.1, there are actually four variables that determine human thermal comfort which are temperature, humidity, air movement, and air quality. In addition to temperature, any or all of the remaining three variables could also be measured and controlled to improve the quality of the controlled process, which in this case, is conditioning a zone served by a VAV HVAC system.

Besides the number of controlled variables, the location of the controlled variable in relation to the controlled process can also vary. Normally, the controlled variable will be on the output of the controlled process but it could also be at the input of the controlled process or within the controlled process itself. Using the VAV HVAC system serving a building zone as an example, the airflow through the VAV terminal unit serving the zone is determined both by the position of
FIGURE 18–1 GENERIC HVAC CONTROL SYSTEM BLOCK DIAGRAM
FIGURE 18-2 VAV TERMINAL UNIT CONTROL BLOCK DIAGRAM
the internal damper and the static pressure at the input of the VAV terminal unit. As discussed in Section 3.1, most VAV terminal units are pressure independent and adjust the VAV terminal unit damper in response to not only the temperature in the zone served but also the static pressure at the inlet of the VAV terminal unit. The VAV terminal unit is the controlled process and the controlled variables are the temperature of the zone which represents its output and the pressure at the inlet of the VAV terminal unit, see Figure 18-2.

18.2.1.3 Setpoint

Setpoint is the desired condition of the controlled variable. The setpoint sets the benchmark for the controlled variable and it is what the measured value of the variable is compared to determine if action is needed to achieve the desired output from the controlled process. The setpoint is usually determined during the design process for controlled variables that are a function of system or equipment capabilities, mandated codes and standards, or good design practice. An example of a controlled variable setpoint determined during the design, installation, or commissioning process might be carbon dioxide concentration in the supply air to a zone. Similarly, the setpoint could be based on output requirements during operation such as the temperature setting of the thermostat in a space. Setpoints that can be adjusted during system operation such as the temperature setting of the thermostat in a space. Setpoints that can be adjusted during system operation such as the temperature setting of a thermostat can be adjusted manually or automatically based on time of day, sensed occupancy, or other factors. Similarly, commercial and institutional building control systems typically allow setpoints to be changed remotely via a building automation system (BAS).

The VAV terminal unit has a setpoint for both the zone temperature and the VAV terminal unit inlet pressure, see Figure 18-2. The VAV terminal unit inlet pressure setpoint for this controlled process was probably set during the commissioning process and the temperature setpoint was either set manually or automatically during operation based on conditions in the zone.

18.2.1.4 Sensor

The sensor is the device that sends a signal to the controller reporting the value or state of the controlled variable. Sensors can be provided for any measurable controlled variable. Among many other possibilities, sensors incorporated into HVAC control systems include the following:

- Temperature
- Humidity
- Flow
- Pressure
- Carbon Dioxide
- Carbon Monoxide

There are two sensors in Figure 18-2 reflecting the fact that there are two controlled variables for the VAV terminal unit which is the controlled process in this example. At the input of the VAV terminal unit there is a pressure sensor. This pressure sensor measures the static pressure at the inlet of the VAV terminal unit and reports that pressure to the VAV controller. In addition, there is a temperature sensor in the form of a thermostat located in the zone served by the VAV terminal unit that monitors the ambient temperature in the zone and sends a signal to the VAV controller reporting the measured temperature.

18.2.1.5 Controller

The controller is the processor that compares the setpoint of the controlled variable to the measured value of the controlled variable that is reported by the sensor. The controller compares these two values, decides what action to take based on the “rules” hardwired or programmed into it, and then initiates the required action by sending the appropriate signal to the controlled device.

In the VAV terminal unit example, the VAV controller receives the data sent by both the pressure and temperature sensors and processes it with the pressure and temperature setpoints to determine the action that needs to be taken by the VAV damper controller mechanism to achieve the desired zone thermal conditions that is the output of the controlled process. Once the controller determines the appropriate action that needs to be taken based on the rules hardwired or programmed into it, a signal is transmitted by the controller to the damper controller to initiate the desired action.

18.2.1.6 Controlled Device

The controlled device is the component that reacts to the output signal of the controller. The controlled device could be any number of devices such as the following:

- Actuators
- Automatic Dampers
• Damper Operators
• Automatic Valves
• Valve Operators
• Contactors and Relays
• Motor Starters
• Variable Frequency Drives

In the case of the VAV terminal unit, the controlled device would be the damper control mechanism. The signal from the controller will cause the VAV terminal unit to open or close to cause increased or decreased airflow to the zone, respectively.

18.2.1.7 Controlled Agent

The controlled agent is the physical parameter that is manipulated by the controlled device. Manipulating the controlled agent will in turn impact the output of the controlled process to achieve the desired process outcome. In the case of the VAV terminal unit example, see Figure 18-2, the controlled agent is the airflow through the VAV terminal unit into the zone. Varying the airflow of the constant temperature air supply will maintain the desired temperature in the zone. Other controlled agents might be the volumetric flow rate of chilled or hot water circulated through a coil in an air-handling unit, the airflow through a damper, among other HVAC system physical parameters.

18.3 CONTROL LOOPS

18.3.1 Control Loop Defined

The following four elements make up a physical control loop, see Figure 18-1:

• Sensor
• Controller
• Controlled Device
• Controlled Agent

The sensor is the device that monitors the controlled variable and transmits the value or state of the controlled variable back to the controller. The controller compares the value or state of the controlled variable to the setpoint and, when there is a difference, determines the action to be taken based on a set of rules that are hardwired or programmed into it. The controller then initiates the action to be taken by sending a signal to the controlled device. The controlled device then changes its state in response to the controller signal and causes a change to the controlled agent. The change in the controlled agent will then impact the output of the controlled process that in turn will be monitored by the sensor and the whole cycle or loop continues. Figure 18-3 illustrates this control loop schematically.

18.3.2 Types Of Control Loops

There are two types of control loops used in HVAC systems.

• Closed Loop Control
• Open Loop Control

18.3.2.1 Closed Loop Control Systems

The control system block diagrams in both Figures 18-1 and 18-2 are closed loop control systems. In a closed loop control system, the controller measures actual changes in the controlled variable via the sensor and initiates action through the controlled device to bring the actual system output in line with the desired system output as defined by the controlled variable setpoint. The corrective action is a continuous process that continues until the controlled variable is brought to the desired value within the design limitations of the controller, see Figure 18-3. With a closed loop control system, the results of the action taken by the controller are continually monitored by the sensor and fed back to the controller for further action which is referred to as feedback.

18.3.2.2 Open Loop Control Systems

An open loop control system differs from a closed loop control systems in that an open loop control system usually takes corrective action to offset the impact of an external change to a controlled variable. Open loop control systems are also referred to as feed-forward control systems because there is no feedback mechanism like a closed loop control system. An example of an open loop or feed-forward control system would be the use of an outside thermostat that measures outdoor temperatures to adjust the operation of the building’s HVAC system. In this case, the outside thermostat is the sensor controlling the HVAC system and the inside temperature of the building is not part of the control loop. There are instances where open loop control systems are used in HVAC systems but to be effective these open loop control systems are integrated with closed loop control systems to ensure that changes made by the controlled agent actually provide the desired output from the controlled process.
18.3.3 Closed Loop HVAC Control System Example

Figure 18-4 provides an example of a closed loop HVAC control system involving the control of the supply air temperature through a cooling coil in an air-handling unit. The temperature probe in Figure 18-4 is the control system sensor. The temperature probe senses the temperature of the supply air passing through the cooling coil and transmits the temperature to the controller. The controller compares the actual supply air temperature with the desired supply air temperature which is the setpoint input to the controller. The controller sends a control signal to the valve which is the controlled device and instructs it to either open or close based on whether the sensed temperature in the air stream is higher or lower than the setpoint. In this example, the controlled agent is the chilled water where increasing the flow of chilled water through the cooling coil will reduce the temperature of the supply air and decreasing the flow of chilled water through the coil will increase the temperature of the supply air.

18.4 THERMOSTAT: SIMPLE CONTROL SYSTEM

Up until now, the sensor, controller, and controlled device were shown as separate devices. In some cases two or more of these devices will be integrated together as a single unit. An example when all three of these devices are integrated together into a single unit is a simple residential or light commercial thermostat that controls a unitary HVAC system directly by simply turning it on or off. A simple wall-mounted thermostat contains a temperature sensor which has provisions for setting the thermostat at the desired temperature which is the setpoint. It also contains the controller which compares the setpoint and measured temperature and initiates the needed corrective action through the controlled device which is simply a set of contacts in the thermostat. Closing those contacts causes the air conditioner or furnace to start and supply the needed cooling or heating to bring the ambient temperature in the space in line with the desired temperature.

18.5 TYPES OF CONTROL SYSTEMS

The dashed lines connecting the sensor to the controller and the controller to the controlled device in Figures 18-1 and 18-2 represent control signal paths that allow information to be passed between devices. The value or state of the control variable is sensed by the sensor and transmitted to the controller. The controller processes the information sent by the sensor with the controlled variable setpoint and transmits a signal indicating the appropriate action to the controlled
FIGURE 18–4 EXAMPLE HVAC CLOSED LOOP CONTROL SYSTEM
device. The method used to transmit information and control signals in HVAC control systems often is used to describe the type of control system. Over time, the methods used to transmit control information and signals have changed with advancing technology.

The four primary methods that have been used to transmit information and control signals in HVAC systems are as follows:

- Pneumatic Control
- Analog Electric Control
- Direct Digital Control

Direct digital control is used almost exclusively in today’s new HVAC systems in commercial and institutional buildings. However, existing HVAC systems still use pneumatic, analog electric, and analog electronic control systems.

18.5.1 Pneumatic Control

18.5.1.1 Pneumatic Control Overview

Pneumatic controls were among the first automatic HVAC control systems. In these controls systems compressed air is used to transmit control signals. Pneumatic control systems used compressed air to directly initiate a change in the controlled device and did not require a transducer to convert the digital signal into an electromechanical action. Pneumatic controls are capable of measuring changes in the controlled variable using pneumatic sensors and transmitting the measured changes to the pneumatic controller. Pneumatic logic within the controller could be used to process the inputs received from sensors and produce an output that would signal the action that the controlled device should take. Pneumatic HVAC controls were reliable but were difficult to calibrate, were not as accurate as direct digital controls, and required regular maintenance.

18.5.1.2 Pneumatic System Components

Pneumatic control systems are made up of the following components:

- A source of clean, dry, compressed air to provide the operating energy.
- Airlines (or mains) that are usually either plastic or metal tubing that connect the air supply to the controlling and controlled devices.
- Sensing devices such as thermostats and humidistats that detect and measure a change in the controlled variable.
- Regulating devices or controllers such as thermostats and humidistats.
- Branch air lines leading from the controller to the controlled devices.
- Controlled device or operators such as valves or damper motors.
- Devices positioned by the operator such as a valve or damper.

Figure 18-5 shows the elements of a typical pneumatic control system.

18.5.1.3 Pneumatic Control System Operation

In the basic pneumatic control system, after the air is filtered by the filter and the pressure is reduced to a suitable value by the pressure-reducing valve, it flows through the air main to the controller, which in this example, is a thermostat along with other controllers, relays, or operators that form part of the control system, see Figure 18-5. The function of the controller is to regulate the positioning of the controlled device or operator. It does this by taking air from the air main at a constant pressure and delivering it through the branch line to the controller at a pressure that is varied according to the change in the controlled condition.

In Figure 18-5, the controlled device is a valve or motor. When a change in air pressure is transmitted to the valve it causes the valve to move further toward open or closed. How much it moves and in which direction depends on the magnitude of the change at the thermostat, the direction of the change, and the design and construction of the valve.

18.5.1.4 Operators and Control Elements

A pneumatic operator is essentially no more than a pressure-actuated device. A pneumatic operator functions to provide changes in position corresponding to changes in the branch-line air pressure. Consequently, it is built around a diaphragm that moves in response to changes in air pressure. Pneumatic operators are available in various forms for positioning dampers and valves. Damper operators are usually mounted on the ductwork or the damper frame and utilize a push rod
and crank-arm to drive the damper blades. Valve operators mount directly on the valve bonnet and directly positions the valve stem.

### 18.5.1.4.1 Valve Assemblies

Numerous valve assemblies are available depending on the control action required, the piping requirements, the space limitations, and the type of controlled medium, see Figure 18-6. Figures 18-6A and 18-6B illustrate two types of single-seated valve assemblies. Single-seated valves are used where tight seal-off is required. Figure 18-6C illustrates a double-seated valve. With a double-seated valve, the control agent is between the two seats which balance the inlet pressures between the two discs of the plug assembly. As a result, the force required by the operator to position the plug assembly is greatly reduced. Figure 18-6D illustrates a three-way mixing valve. This valve has two inlets and a common outlet. Figure 18-6E illustrates a three-way diverting valve that has a common inlet and two outlets.

For straight-through valve, the term “normally open” indicates that the valve returns to the open position when there is zero air pressure in the diaphragm chamber, see Figure 18-6A. For the valve in Figure 18-6B, the term “normally closed” indicates that the valve returns to a closed position when there is zero air pressure in the diaphragm chamber. For three-way assemblies, the valve has a normally closed position, see Figure 18-6D or a normally open position, see Figure 18-6E with respect to straight-through flow only. For three-way valves, constant total flow is always maintained through the common inlet or output port.

Referring to Figure 18-6A for a normally open valve, the position maintained by the valve stem depends on the balance of forces on it. These include:

- The force from the air pressure over the diaphragm.
- The opposing force from the operator spring.
- Any other forces acting on the valve plug.

On an increase in controller branch-line air pressure to the valve operator or diaphragm chamber, the diaphragm will tend to move down, moving the valve toward the closed position until the tension of the spring is increased sufficiently to balance the increased force on the diaphragm. Conversely, a decrease in controller branch line air pressure will result in the spring moving upward and moving the valve stem toward the open position.

Referring to Figure 18-6B for a normally closed valve, the construction of the operator is similar to the operat-
18.5.1.4.2 Dampers

A number of dampers are available to control the flow of air through HVAC equipment and the air distribution system. The most common type of dampers are multiblade louver dampers. Figures 18­7A and 18­7B provide diagrams of typical automatic multiblade dampers.

Figure 18-7A illustrates the principle of a normally open damper. Similar to the discussion for valves, the term normally open indicates that the damper will return to the open position with zero air pressure in the operator diaphragm chamber. Referring to Figure 18-7A, an increase in air pressure in the diaphragm chamber will force the rolling diaphragm piston against the tension of the spring. A decrease in air pressure in the diaphragm chamber will allow the spring to force the piston and diaphragm back to the normal or relaxed position.

Figure 18-7B shows the principle for a normally closed damper. The term normally closed indicates that the damper will return to the closed position for zero air pressure in the operator diaphragm chamber.

18.5.1.5 Controllers

18.5.1.5.1 Function Of A Controller

The function of the controller is to regulate the positioning of the controlled device. This is accomplished in a pneumatic system by taking air from the supply main at a constant pressure and delivering it through the branch line to the controlled device at a pressure that is proportional to the measured condition. The various conditions commonly measured and controlled include temperature, humidity, and pressure.

18.5.1.5.2 Controller Types

Controllers differ fundamentally as to:

- Direction of change in branch pressure on a given change in the measured conditions.
FIGURE 18–7 AUTOMATIC MULTIBLADE DAMPERS

DAMPER

AIR FLOW

SPRING

ROLLING DIAPHRAGM

OPERATOR PISTON

0 PSIG

(A)

DAMPER

AIR FLOW

0 PSIG

(B)
• The method of varying or controlling the branch-line air pressures.
• The type of change produced in the branch-line pressure.

With respect to the direction of pressure change in the branch-line, a controller may be direct acting if the branch-line pressure increases in response to an increase in the measured condition or reverse acting if the branch-line pressure decreases. A controller may be one of the following types:

1. Bleed type if control air is continually exhausted from the branch-line in relation to a fixed rate of flow into the branch-line from the main.
2. Non-bleed type if control air is exhausted from the branch-line only while reducing the branch-line pressure.
3. Pilot-bleed type which consists of:
   a. A pilot-bleed assembly which continually exhausts a very small amount of air in relation to the amount used by the bleed type.
   b. A non-bleed valve unit or relay which is controlled by the pilot-bleed and exhausts air only while reducing the branch-line pressure.

A controller may be classified either as proportional if the branch-line pressure is changed gradually or two-position if the branch-line pressure changes rapidly from zero to maximum fixed pressure. Controllers can also be classified as either single- or two-pressure. A single-pressure controller uses a constant pressure from the supply main. Two-pressure devices use a main that is alternately switched from a lower to a higher fixed pressure and back. For example, day-night thermostats automatically switch from a day setting at a fixed higher pressure to a fixed lower pressure night setting. Summer-winter thermostats may switch from reverse acting at a lower fixed pressure for cooling in the summer to direct acting at a higher fixed pressure for heating in the winter.

18.5.1.5.3 Throttling Range

For the basic proportional pneumatic thermostat, the throttling range is defined as that change in the measured medium required to change the branch-line pressure from low to high pressure such as 3 to 13 psig (21 to 90 kPa). For example, suppose that the thermostat is direct acting with a throttling range of 3°F (2°C) and that it is set at 72°F (22°C). When the temperature is 73.5°F (23°C) which is half the throttling range above the setpoint, the branch-line pressure would be 70.5°F (21°C) which is half the throttling range below the setpoint and the branch-line pressure is 3 psig (21 kPa). Between these two levels, pressures and temperatures are in proportion.

If the thermostat were reverse acting, the branch-line pressure would vary inversely with the temperature. If the throttling range and settings were the same as for the direct-acting controller where lower fixed 3 psig (21 kPag) pressure would correspond to the fixed higher temperature 73.5°F (23°C) and the higher fixed 13 psig (90 kPa) pressure would correspond to the fixed lower 70°F (22°C) temperature.

18.5.1.5.4 Proportional Band

The term proportional band is used with the pneumatic sensor-controller systems. Proportional band is defined as the change in the measured medium in terms of a percent of the primary sensor span that is required to change the controller branch-line pressure from the lower to the higher pressure. This term is synonymous with throttling range except that the adjustment scale is calibrated in percent rather than units of measured temperature, pressure, or other specific units.

18.5.1.5.5 Differential

This is a term applied to two-position controllers. For a two-position thermostat, differential is defined as the change in the measured medium required to rapidly change the branch-line pressure no pressure to full main pressure.

18.5.1.5.6 Bleed-Type Controller

Figure 18-8A provides a diagram of a typical direct-acting, bleed-type thermostat. Except for the construction and material of the sensing element, other bleed controllers, such as for humidity and pressure, are constructed and operate similar to the typical direct-acting, bleed-type thermostat illustrated in Figure 18-8A. Changes in branch-line pressure are obtained by bleeding varying amounts of air to the atmosphere.

Changes in the controlled condition cause a vane or flapper to move toward or away from the nozzle through which the air bleeds. In this way, the flapper regulates the rate at which the air is bled from the branch-line. The restriction, which is the other determining factor, must be large enough so that it can furnish sufficient air to move the operator to the advanced position in the desired time when the flapper completely closes the nozzle. It must also be small enough to pre-
vent air from entering the branch line so rapidly that when the nozzle is wide open it is not able to bleed the branch-line to zero pressure in the desired time. A simple change in the relative position of the bimetal or in a lever-system arrangement, depending on the particular controller provides reverse action, see Figure 18-8B.

18.5.1.5.7 Thermostat Operation

The flapper-nozzle operation in Figure 18-9 is basically the same for all thermostats. The thermostat provides a branch line airflow or pressure that is a function of the ambient temperature in the room or controlled space. The force of the temperature sensing bimetal, acting on the flapper is balanced by the feedback force of the pilot pressure that is acting on the opposite side of the flapper through the nozzle. When the bimetal force changes as the result of either a temperature or setpoint change, the position of the flapper changes over the nozzle and a new pilot chamber pressure is created. This pilot pressure feeds into the valve unit flow amplifier that converts the low capacity pilot pressure to a high capacity branch line change. In this thermostat, a feedback feature at the nozzle provides a pressure regulating effect that negates the affect of normal air supply functions on the branch line. The throttling or proportioning range adjustments are made by changing the flapper lever ratio. Setpoint changes for the thermostat are made with a cam that changes the bimetal operating force.

18.5.2 Analog Electric Or Electronic Control

The difference between analog electric and electronic control is that the signal sent by analog electric controls usually has sufficient power to cause a change in the controlled device’s state, directly. Analog electronic controls, like direct digital controls, usually required a transducer to convert the control signal to a pneumatic or electric signal with enough power to change the state of the controlled device.

18.5.3 Direct Digital Control

Direct digital control (DDC) control systems have almost totally replaced pneumatic, analog electric, and analog electronic control systems in commercial and institutional buildings. Direct digital control systems are microprocessor-based systems that use digital logic to transmit control system information and signals. Direct digital control systems always include a transducer to convert the digital control signal sent by the controller to affect the change in the controlled device. In most cases, the transducer is built into the controlled device.
One of the advantages of direct digital control is that these systems are both low power and low voltage allowing control cabling to be run throughout the building without being enclosed in raceway. However, when installed in plenums the cable must be plenum rated. The cable used today to interconnect devices is often an unshielded twisted pair (UTP) copper cable that is also used for other building structured cabling systems such as local area networks (LANs), life safety and security systems, and other voice/data/video applications. Hardwired connections using UTP are most common however some systems will allow the use of wireless communications between devices using WiFi, Bluetooth, and ZigBee technologies, power over Ethernet (POE), optical fiber cable, or powerline carrier.

18.6 SENSING ELEMENTS

18.6.1 Sensing Element Types

A sensing element is a device which measures changes in the controlled variable and produces a proportional effect or signal for use by the controller. Sensing elements include the following:

- Temperature
- Rod and Tube
- Sealed Bellows
- Electrical

18.6.1.1 Temperature Sensing Elements

Temperature sensing elements usually consist of one of the four following types of sensors:

- Bimetal Strips
- Rod and Tube
- Sealed Bellows
- Electrical

18.6.1.1.1 Bimetal Strips

A bimetal element is composed of two thin strips of dissimilar metals fused together. As a result of these two metals having different coefficients of thermal expansion, the element bends as the temperature varies and produces a change in position.

18.6.1.2 Rod and Tube

Like bimetal strips, rod-and-tube elements also consist of two dissimilar metals. The tube consists of a high
expansion metal tube inside of which is a low expansion rod with one end attached to the rear of the tube. The tube changes length with changes in temperature which causes the free end of the rod to move.

**18.6.1.1.3 Sealed Bellows**

A sealed bellows element with or without a remote bulb can be either vapor-, gas-, or liquid-filled. Changes of temperature cause changes in pressure or volume of the gas or liquid which result in a change in force or movement. A remote bulb element is a sealed bellows or diaphragm to which a bulb or capsule is attached by means of a capillary tube and the entire system is filled with vapor, gas, or liquid. Changes of temperature at the bulb result in changes of pressure or volume which are communicated to the bellows or diaphragm through the capillary tube.

**18.6.1.1.4 Electrical**

A resistance element is made of wire with electrical resistance that changes with temperature changes. A thermistor is a special kind of semiconductor in which electrical resistance changes with temperature changes. A thermocouple is a union of two dissimilar metals joined at the ends that have a generated voltage that varies as a function of temperature change.

**18.6.2 Humidity**

Humidity sensing devices usually are either hygroscopic or electrical. Hygroscopic devices result in a change in size or form to cause a mechanical deflection using organic materials such as a paper or manufactured material such as nylon. Electrical devices cause a change in electrical characteristics such as resistance or capacitance due to the hygroscopic nature of the element.

**18.6.3 Water Flow**

Water flow sensing elements use a variety of basic sensing principles and devices to sense and measure water flow. Typical water flow sensing devices include:

- Orifice Plates
- Pitot Tube
- Venturi
- Flow Nozzles
- Turbine Meter
- Magnetic Flow Meter
- Vortex Shedding Flow Meter

Each of these has characteristics of rangeability, accuracy, and complexity or cost that make it suitable for different situations. In general, the pressure differential types of devices which include orifice plates, Pitot tubes, venturis, and flow nozzles are simple and inexpensive but have limited range, so their accuracy depends on how they are applied and used. The more sophisticated flow sensing means, such as turbine, magnetic, or vortex shedding flow meters are more expensive but have good range and, therefore, have good accuracies over the full range of flows they can measure.

**18.6.4 Pressure**

Pressure sensing elements can be divided into two general classes depending on pressure range. For pressures or vacuums measured in pounds per square inch or inches of mercury (kPa), the element is usually a bellows, diaphragm, or Bourdon tube. One side of the element may be open to the atmosphere in which case the element responds to pressures above or below atmospheric. A differential-pressure element has connections to both sides so that it will respond to the difference between two pressures. For low ranges of pressure or vacuum that are usually measured in inches of water, such as the static pressure in an air duct, the measuring element may be an inverted bell immersed in oil, a large slack diaphragm, or a large flexible metal bellows. The element usually is one of the differential types when employed in connection with orifices. Pitot tubes and similar accessories may be used to measure flow, velocity, or liquid level as well as static pressure.

**18.6.5 Other Sensing Elements**

Sensing elements for other purposes such as for flame detection or for measuring smoke density, specific gravity, current, carbon dioxide (CO₂), carbon monoxide (CO), or other physical parameters are often necessary for the complete control of a heating, ventilating, or air-conditioned system.

**18.7 CONTROL SIGNAL TRANSMISSION**

**18.7.1 Controller Action**

Controllers are devices that take the sensor measurement, compare it with the desired controlled condition or setpoint, and then provide an output signal that causes a control action to take place. Controllers used in HVAC systems...
systems are often categorized by their control mechanism which can be pneumatic, analog electric or electronic, or digital. Digital controllers are used almost exclusively in HVAC systems today however older systems in existing buildings may still use pneumatic or analog electrical or electronic controllers.

18.7.2 Control Signal Transmission

Pneumatic controllers are actuated by changes in air pressure through pneumatic tubing and analog electric and electronic controllers that respond to changes in voltage or current levels in wired circuits. The remainder of this section will focus on digital control signal transmission which can be accomplished by using either wired or wireless control signal transmission.

18.7.3 Wired Digital Control Signal Transmission

Wired digital control signal transmission is usually accomplished using multiconductor cable designed for the application. These multiconductor cables can be manufactured specifically for HVAC system use, as in the case of thermostat cable, or could be standard low-voltage cable such as unshielded twisted pair (UTP) cable or optical fiber cable. The selection and installation of cabling for digital signal transmission must be in accordance with control system manufacturer requirements and local codes.

18.7.4 Wireless Digital Control Signal Transmission

18.7.4.1 WiFi

WiFi is based on IEEE Standard 802.11b and allows wireless broadband networking. WiFi has a bandwidth or data throughput of 11 megabits per second (Mbps) or 11 million bits of information per second and operates in the ISM Band like ZigBee™. A faster version of the standard 802.11b WiFi is IEEE Standard 802.11g which has a bandwidth of 54 Mbps and is backward compatible with IEEE 802.11b devices. WiFi has an effective operating range of about 300 feet. WiFi is not typically used for wireless digital control signal transmission between controllers and controlled devices because it bandwidth and speed of signal transmission is not required and its power consumption is high. However, WiFi is can be used for wireless programming and monitoring of HVAC control systems.

18.7.4.2 Bluetooth

Bluetooth has become a common technology for networking computer devices such as computer peripherals and wireless cell phone headsets. Bluetooth is also used in building control systems as the wireless control link between a programming device such as a standard personal computer (PC) or handheld personal computer (HPC) and a programmable controller. Bluetooth operates at 2.4 GHz but has a much lower bandwidth and effective operating range than WiFi. Bluetooth has a data throughput of 1 Mbps and an operating range of about 30 ft (9 m) which is sufficient for data exchange between nearby Bluetooth enabled devices.

18.7.4.3 ZigBee

ZigBee™ is the name given to an open-architecture wireless personal area network (WPAN) technology that is designed specifically for linking devices that perform a monitoring and control function. For buildings, these devices can include anything from a simple thermostat to more complex HVAC, lighting, and other control devices. Since ZigBee™ compliant products are based on an industry standard and not any one company’s proprietary system, any building monitoring and control device that is ZigBee™ compliant can be used. If widely used and adopted, ZigBee™ could become an important factor in the future of building automation.

ZigBee™ is based on IEEE Standard 802.15.4 that was ratified in 2003 and provides a standard for low-powered, digital radio frequency (RF) transceivers that operate primarily in the unregulated Industrial, Scientific, and Medical (ISM) Band at 2.4 GHz. The ZigBee™ Alliance (Alliance), which is now comprised of more than 200 firms, including Honeywell, Eaton, and other manufacturers, subsequently developed a networking specification that includes network protocols and security based on this IEEE standard.

ZigBee™ is both a self-organizing and self-healing network. Being a self-organizing network means that any ZigBee™ compliant device that is introduced into a ZigBee™ environment will be automatically incorporated into the network as a node. The self-healing network capability means that if a node is removed or fails, the network will automatically detect the loss of the node and work around it. ZigBee™ devices can organize themselves into a variety of network structures including a tree structure that allows large area coverage and essentially no limit on the number of nodes or devices connected to the network. However, in a
simple star or radial configuration the effective distance and size of a ZigBee™ network is limited to about 100 feet and 255 devices.

One of ZigBee™'s most important features is its ultra-low power requirement. ZigBee™ devices have just two states: active and sleep. In the active state these devices process, receive, and transmit data. The time in the active state is very short because little information needs to be transferred to fulfill their monitoring and control function. In the sleep state, the device is dormant using very little power until it needs to go active again. As a result, ZigBee™ devices can be battery powered and the batteries in these devices can last months or years before they need to be replaced. Therefore, ZigBee™ monitoring and control devices do not need to be connected to a utility branch circuit for power.

ZigBee™ has advantages over both WiFi and Bluetooth in building automation applications. It is much simpler than either WiFi or Bluetooth making it physically smaller, cheaper, and easier to integrate into building products. ZigBee™ was designed for this specific application so it is better matched to building automation requirements, uses less power, and more economical than either WiFi or Bluetooth. For these reasons, ZigBee™ may be poised as the dominant wireless networking technology for building automation in the future.

18.8 BUILDING AUTOMATION AND CONTROL SYSTEMS

Direct digital control has led to the development and widespread use of building automation and control systems (BAS). These computer-based control systems control the building's HVAC system from a central location and are increasingly integrated with the building lighting control system, life safety and security system, voice/data/video systems, and other related building systems. Building automation and control systems are networked systems which has a central station that monitors and controls the overall system from one location as well as stores data on system operation. Networked with the central station, are intelligent remote building automation, control panels and equipment that are capable of operating independently of the central station. In addition, many of these building automation and control systems also have the capability of being remotely monitored and controlled over the Internet allowing centralized monitoring and control of HVAC, refrigeration, lighting, life safety and security, and other building systems by owners with multiple properties around the city, region, county, or world. In addition to simply monitoring and controlling the HVAC system, these building automation and control systems allow the owner to change setpoints and setup schedules of operation, trend logs, alarms, and other reporting features, as required.

18.9 REMOTE HVAC SYSTEM MONITORING AND CONTROL

Advances in technology have made the remote monitoring of commercial building systems economical. It wasn't long ago that system control and data acquisition (SCADA) systems were only used by utilities and heavy industry because they were typically custom systems that were expensive to implement and maintain. Today, similar systems for commercial buildings are available “off the shelf” making the remote monitoring of a building next door or one on the other side of the world practical. Remote monitoring does not need to be restricted to HVAC systems. Any system of which the operation is important to the building owner or tenant can be monitored including refrigeration, plumbing, power, and other systems. Remote monitoring provides advantages beyond detecting an imminent system failure and then taking prompt corrective action. Remote monitoring can also provide information that can be used to detect long-term trends in system operation that may lead to future problems.

18.10 OPEN-ARCHITECTURE CONTROL SYSTEMS

18.10.1 Open-Architecture Control System Defined

An open-architecture control system is one where the hardware and software specifications are in the public domain and available to anyone wanting to design and manufacture hardware components or develop software tools that will operate with the system. This is in contrast to closed-architecture control systems where the original system developer maintains control of the system specifications and is the only entity that can design and manufacture components or develop software tools for the system. Proprietary or closed-architecture control systems have been the norm in the building industry with each building system being treated as an independent system and having its own standalone control system. Integration and interoperability between proprietary system-specific building control systems is difficult.

The trend today is to view buildings not as a group of independent systems that need to be individually optimized but a collection of interdependent subsystems that need to be optimized collectively in order to
provide an efficient building that addresses the needs of its occupants. An open-architecture system is best suited to provide the flexibility and interoperability that building owners and occupants prefer.

18.10.2 Advantages Of Open-Architecture Control Systems

In addition to more readily allowing various building systems to be integrated and the building operation optimized as a whole, an open-architecture control system offers building owners a number of other advantages beyond operational efficiency including the following:

- Allows for a multi-vendor system that permits system integrators to select control devices that are best for each application.
- Allows incorporation of any system into the overall open-architecture control system including lighting, life safety and security, and other systems that may not be possible with proprietary systems.
- Allows for competitive bidding not only for the initial installation but also, more importantly for service, over the life of the system. Future system expansions and upgrades as well as everyday moves, add-ons, and changes can be negotiated or competitively bid because the owner is no longer locked into a single, proprietary system.
- Allows for system service to be performed by any qualified firm and the owner is no longer locked into the supplier of a proprietary system for service over the life of the system.
- Allows the system to be upgraded using the latest technology in control devices from any vendor whose devices are compatible with the open-architecture control system so the owner is not locked into the pace at which a proprietary control system supplier adopts new technology.
- Allows legacy proprietary control systems and components in existing buildings to be integrated into open-architecture control systems through system gateways and individual device interfaces.

18.10.3 Open-Architecture Control Standards

There are two dominant open-architecture control standards used in North America:

- BACnet
- LonWorks

18.10.3.1 BACnet Control Standard

The BACnet standard was developed by the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) to provide an open-architecture control standard for building mechanical systems and equipment. BACnet stands for “building automation and control network” and is an American National Standards Institute (ANSI) approved standard. The BACnet standard is designated ANSI/ASHRAE 135-2004 and entitled BACnet – A Data Communication Protocol for Building Automation and Control Networks. Even though BACnet could be used for other systems, its use has been limited to mainly mechanical systems.

18.10.3.2 LonWorks Control Standard

LonWorks-compatible control devices communicate with each other using the ANSI approved standard that was originally developed by Echelon Corporation and adopted by the Electronics Industry Association (EIA) and Consumer Electronics Association (CEA) as EIA/CEA 709.1-B-2002 entitled Control Network Protocol Specification. LonTalk is the communications protocol based on EIA/CIA 709.1-B-2002 that allows LonWorks-compatible control devices to communicate based on the Open Systems Interconnection (OSI) seven-layer reference model for peer-to-peer network communications.

The heart of the LonWorks system is the Neuron chip that is produced by Echelon and implements the lower six layers of the OSI reference models. This leaves only the seventh layer, which is the application protocol, which needs to be dealt with when implementing a LonWorks control system. Each Neuron chip is assigned a unique, permanent, 48-bit code for addressing that is referred to as the Neuron ID. Manufacturers of LonWorks compatible devices then embed neuron chips along with input/output (I/O) devices and application programming to run the chip so each controlled device has a unique system address.

18.11 CONTROL SYSTEM APPLICATIONS

Applications of controls to systems or apparatus of a general nature will be found in this section. Many con-
control combinations can be used with central fan systems and the following subsections deal with only a few of the typical combinations for controlling outdoor and return air dampers, preheat coils, heating coils, cooling coils, and humidity control. The complete control system is obtained by combining the selected method for each item into an integrated set of controls. Consideration must be given to interrelation of the several parts as well as the sequence of operation for a fully integrated HVAC system.

18.11 Outdoor Air

18.11.1 Basic Outdoor Air Control

Outdoor air to meet minimum ventilation or natural cooling requirements should be available whenever the fan is running and is usually provided in one of the following ways:

- By means of an outdoor air damper which opens to a minimum position when the fan is started and may be further opened by automatic or manual controls.
- By means of a minimum outdoor air damper which opens fully when the fan is started. The maximum outdoor air damper may then provide natural cooling.

Recirculated air and exhaust air or relief air dampers must be synchronized to operate with the outdoor dampers. Outdoor air and exhaust air dampers should close when the fan stops.

18.11.2 100 Percent Outdoor Air

100 percent outdoor air can be used in buildings with large exhaust air requirements. This often occurs in laboratory buildings. Control consists of a two-position outdoor air damper interlocked to run when the supply air fan runs. Interlocks are also provided between the supply and exhaust fans. Heat recovery should be considered for use with any 100 percent outdoor air system.

18.11.3 Static Pressure Control

When exhaust air quantities are large but variable, static pressure control of outdoor air can be used, see Figure 18-10. A pressure sensor detects pressure changes caused by an increase or decrease in exhaust air and the controller adjusts the outdoor air quantity to match. This system is not common. It is more typical to control of the supply fan on VAV HVAC systems as discussed in Chapter 3.

18.11.4 Economizer Cycle

Economizer cycle controls are commonly used to improve the energy efficiency of the HVAC system. Figure 18-11 shows the traditional economizer cycle. When the supply fan is started, the outside air damper opens to the minimum position to meet the ventilation air requirement of the space. When the outside air temperature is above a high limit, the outdoor air damper stays in the minimum position. When the outdoor air temperature is below the high limit, the outdoor, return, and relief air dampers are modulated to maintain a fixed air temperature not less than the low limit supply air set point which is typically between 55°F to 60°F (12°C to 16°C), see Figure 18-11.

In practice, a variety of control configurations can achieve this sequence. Figure 18-11 shows only the basic elements required. Considerable energy can be saved by adding reset of the low limit set point in response to cooling load. The set point is reset upward as the cooling load decreases to eliminate the need for reheat. Most HVAC systems include the reset and it is often a simple retrofit that can be added to older systems. In single zone units, mixed air is often controlled by the space thermostat.

18.11.5 Enthalpy Control

Enthalpy control of outdoor air is an economizer cycle that senses both the temperature and humidity of the return and outdoor air, see Figure 18-12. Sensing both the temperature and humidity of the return and outdoor air results in total heat being the system control variable. The low limit mixed air temperature control is still used as it is with the basic economizer cycle. Except in very humid climates, enthalpy control results in only small energy savings over a conventional economizer cycle that uses dry-bulb temperature as the system control variable. In addition, frequent and regular maintenance is required to ensure that the accuracy of the wet-bulb and dew-point sensors are maintained.

18.11.6 System Shutdown

The HVAC system must be started before the building is occupied and the warm-up/cool-down control cycle is used when starting up the air handling system after night, weekend, or holiday shutdown. This cycle keeps the outdoor air damper closed until the building temperature is within the design heating or cooling tem-
FIGURE 18–10 STATIC PRESSURE CONTROL OF OUTDOOR AIR

FIGURE 18–11 BASIC ECONOMY CYCLE FOR CONTROL OF OUTDOOR AIR
18.11.2 Preheat Coil Control

The function of a preheat coil is to temper the outdoor air to prevent the possibility of freezing beyond the preheat coil. The proper selection, installation, and sizing of steam traps are very important to ensure rapid elimination of condensate to prevent freezing. Other important features are vacuum breakers to ensure condensate removal, provisions for rapid air elimination from the coil, and the use of vertical rather than horizontal tubes, wherever possible.

Figure 18-13 illustrates preheat coil control using an outdoor thermostat that is usually set a few degrees above freezing and controls a two-position preheat coil valve. This valve provides full flow of the heating medium which is typically steam or hot water and reduces the possibility that the preheat coil will freeze. Figure 18-13 illustrates control of an electric preheat coil.

Figure 18-14 shows a hot water preheat control system providing both protection against freezing and accurate control of downstream temperatures. The circulating pump provides a constant water flow through the coil at a minimum velocity which is sufficient to prevent freezing. The purpose of the three-way valve in Figure 18-14 is to admit enough hot water to provide downstream air temperature control. This system is especially useful with systems that admit 100 percent outdoor air or in systems where stratification of mixed air cannot be avoided. A flow switch should be added to stop the fan and close the outdoor air damper in case of pump failure. The pump may be optionally started and stopped by an outdoor air thermostat and must automatically restart after power failure.

Figure 18-15 shows an alternative pump arrangement using primary secondary pumping and a two-way valve. The check valve will permit hot water circulation if the pump fails during a call for heating from the discharge thermostat.

18.11.3 Heating Coil Control

18.11.3.1 Hydronic Coils

The control systems can be used for heating and reheat coil control, see Figures 18-14 and 18-15. In particular, where very close control of downstream temperature is required, circulating pump systems should be used. However, Figure 18-16 shows a typical control...
FIGURE 18−13 OUTDOOR AIR CONTROL OF PREHEAT COIL

FIGURE 18−14 PREHEAT SECONDARY PUMP AND THREE-WAY VALVE
FIGURE 18–15 PREHEAT SECONDARY PUMP AND TWO-WAY VALVE

FIGURE 18–16 HEATING COIL AND TWO-WAY VALVE
Electric heating coils can be controlled in either a two position or modulating mode. Two-position operation must use power relays with contacts sized to handle the power required by the heating coil. A timed, two-position control requires a timer and contactors. The timer can be electromechanical, but is typically solid state and provides a time base of one to five minutes. The percentage of "on" time is determined by thermostat demand. Since rapid cycling of mechanical contactors can lead to maintenance problems solid-state controllers are preferred. Solid-state controllers make cycling so rapid that control is effectively proportional. For safety and often code requirements an electric heater must have a minimum airflow switch as well as high-temperature limit sensors that include one with manual reset and one with automatic reset. As a result, face and bypass dampers are not used in this arrangement. Figure 18-17 shows the control system with a solid-state controller and safety controls.

18.11.3.2 Electric Coils

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18.11.4 Cooling and Humidity Control

18.11.4.1 Cooling and Humidity Control Overview

Since most cooling operations also involve a change in humidity these processes will be covered together. Cooling coils in central air handling units can use chilled water, refrigerant, or other fluids as the cooling medium. Many cooling processes also involve dehumidification. The amount of dehumidification incorporated into the cooling process depends on the effective coil surface temperature and is limited by the freezing point of water. If the water condensing out of the air stream is allowed to freeze on the coil surface airflow will be restricted and, in severe cases, could essentially block airflow. The lowest practical limit is about 40°F (5°C) dew point on the coil surface. This will result in a relative humidity of about 30 percent at a space temperature of 75°F (24°C), see Figure 18-17. When lower humidities are required, chemical dehumidifiers may be necessary.

18.11.4.2 Chilled Water Coil Control

The simplest cooling control system involves a coil with a two- or three-way valve controlling the flow of chilled water or a solenoid valve controlling the flow of refrigerant in response to a room or duct thermostat. A circulating pump may be used to further improve control response, similar to that used with the preheat coils in Figures 18-14 and 18-15. However, this control system does not provide humidity control especially at light sensible loads.

Figure 18-19 illustrates a cooling and dehumidification system that also includes reheat. The chilled water valve serving the cooling coil is normally closed whereas the valve for the hot water serving the heating coil is normally open, see Figure 18-19. The reason for using a normally closed valve in the chilled water circuit is to prevent cooling during a control power failure. The chilled water valve is typically modulated in response to coil air discharge temperature or space temperature. When a maximum relative humidity limit is required, a space or return air humidistat should be used with the space thermostat, see Figure 18-19. A relay selects the higher of the two output signals and controls the cooling coil valve accordingly.

18.11.4.3 Direct Expansion Coil Control

Direct expansion (DX) cooling coils are usually controlled by solenoid valves in the refrigerant liquid line which is an example of two-positional control. Control can be improved by using two or more stages of control with the solenoid valves controlled in a sequence having a differential of one or two degrees between stages. The first stage could be the first coil row on the entering airside with the following rows forming the second and succeeding stages.

18.11.4.4 Evaporative Cooling Control

Sensible cooling can also be provided by the evaporative cooling effect using standard evaporative coolers.
FIGURE 18-17 ELECTRIC COIL WITH SOLID-STATE CONTROLLER

FIGURE 18-18 COOLING AND DEHUMIDIFICATION: PRACTICAL LOW LIMIT
or air washers. Evaporative cooling is a constant wet-bulb process, see Figure 18-21. The process efficiency is described by the ratio of dry-bulb temperature difference between inlet and outlet divided by the difference between inlet dry-bulb and inlet wet-bulb temperatures. Air washers are usually 90 to 95 percent efficient with evaporative coolers varying between 50 to 90 percent. The spray pump is controlled by space temperature, see Figure 18-20. This will always result in high relative humidity in the space but humidity will not be controlled since it is primarily a function of outdoor wet-bulb temperature.

Figure 18-22 shows an evaporative pan humidifier with a steam heater. The steam supply is controlled by a space or duct humidistat. An interlock is provided to shut off the heating when the fan is stopped. Hot water or electrical heating systems may also be used in place of the steam heater. Figure 18-21 shows how steam jet humidifiers are controlled.

### 18.11.5 Static Pressure Control

Since fans are selected to produce the desired results for a certain system characteristic, any change in the system characteristic will change the output of the fan. For example, these changes occur on single-duct systems in which the volume of air delivered is varied by a VAV air terminal unit. System characteristics can also change when the relative quantities of outdoor air and recirculated air are varied, when exhaust fans are run intermittently, and under other specific conditions. Such changes result in a change of static pressure in the system, a change of noise level, and a change of velocities at air outlets that upset the airflow patterns.

Static pressure control of some form is usually required in most HVAC systems and is accomplished by a static pressure controller that measures the pressure at a selected point in the system with respect to an atmospheric reference point or some other point inside the building. Static pressure control can be used to actuate a damper operator, control fan speed through a VFD, or initiate other control functions. For VAV HVAC systems, static pressure control is achieved by measuring the static pressure downstream from the fan unit and using the measured static pressure to control the fan speed through the VFD as discussed in Chapter 3.

Static pressure controls can also be used to pressurize a building or space relative to adjacent spaces or outdoors. Typical applications include clean rooms that use positive pressure to prevent infiltration as discussed in Chapter 20, laboratories requiring either positive or negative pressure depending on their use as
FIGURE 18-20 EVAPORATIVE COOLING PROCESS

FIGURE 18-21 EVAPORATIVE COOLING WITH AN AIR WASHER
discussed in Chapter 21, as well as various manufacturing processes such as paint spray rooms.

18.11.6 Zone Control

18.11.6.1 Zone Control Overview

Most zone control systems are designed to supply heat to a zone at a rate equal to the heat loss. This rate of heat input may be established by either room thermostats or by controllers responsive to outdoor conditions, or by a combination of both. The outdoor controller responds to temperature and may respond to solar radiation and wind velocity and direction. Features that may be added to the basic zone control system include:

- Means for maintaining a lowered economy temperature or completely shutting off the heat at night or at other times when the zone is unoccupied. This may be accomplished automatically with an occupancy sensor or building automation system or manually.

- An arrangement providing rapid warm-up following a period at the non-occupancy temperature. During this warm-up period, full or higher than normal heat may be admitted to the system. The warm-up cycle may be initiated automatically by an occupancy sensor or building automation system or manually with a switch. A warm-up thermostat located within the zone may be arranged to terminate the warm-up period independently of the building automation systems time function or manual switch when the zone temperature approaches the desired occupancy temperature.

- A manual switch to provide full heat or no heat independently of other controls.

- Means for shutting off the heat completely when the outdoor temperature reaches a point where heat is no longer needed.

- A high-limit thermostat in the zone to shut off the heat completely when the zone becomes overheated.

18.11.6.2 Zone Heating-Cooling Coordination

Spaces or areas where heating and cooling are separately controlled occur frequently such as VAV cooling systems coupled with a perimeter heating system. Coordinated controls are needed to measure the zone heating and cooling requirements and then reset supply temperatures of both heating and cooling sources.
to meet those requirements. Figure 18-23 shows an example of mixed zones with both VAV cooling and perimeter heating with load demand reset to both supply temperatures.

### 18.11.6.3 Steam Distribution

In continuous flow steam systems, the quantity of steam supplied to the zone is varied in accordance with the demands of the control system. To obtain equalized distribution of the steam, metering orifices are usually required on the inlets to all radiators or convectors. Supplementary controls or low heat output are often desirable. The following two arrangements are common:

- Varying the difference in pressure between the supply and return that results in partially filling the heating units in accordance with demand. This method can be used on atmospheric or vacuum return systems.

- Varying the pressure in the system while maintaining a constant differential pressure between the supply and return. This results in the heating units being filled with steam at a temperature that is proportional to demand.

In intermittent systems, steam at full pressure is supplied intermittently to the zone. The length of the on and off periods when full pressure steam is supplied is varied in accordance with the demands of the control system. Two common arrangements are:

- A timing device controlled by an outdoor thermostatic element can be used to vary the on-off periods of boiler operation or the opening and closing of a steam valve as a function of outdoor temperature.

- A control responding to an outdoor element and an element attached to a radiator or convector. This control varies the length and frequency of the on-off intervals so that the radiator or convector temperature is varied in accordance with outdoor temperature.

For proper operation of zone control equipment, the heating system must be carefully designed and installed as well as maintained. Vents, traps, vacuum pumps, and valves must be carefully inspected and repaired or replaced when required. Piping must be of adequate size and return piping must be vented.

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**FIGURE 18–23 MIXED LOADS WITH DEMAND RESET**
FIGURE 18–24 COIL CONTROL USING A THREE-WAY VALVE

Notes:
Constant chilled water flow at all conditions
Must use all pumps regardless of load
Difficult to keep chillers on time at light loads
18.11.6.4 Hydronic Distribution

Proper control of hot and chilled water flow rates in hydronic systems must consider both the hydraulic and thermal requirements of the distribution system. Failing to account for hydraulic principles in the piping network design and the pump and control valve selection can make the hydronic system uncontrollable. Proper location and use of the compression tank, as well as dynamic separation of primary and secondary pump circuits, are important for stable system control.

Of the load control methods used in larger systems, valve control for variable flow in the load circuit is the most commonly used. Capacity of the heating or cooling coil in air systems can be controlled using either a three-way or two-way throttling valve. The three-way valve is usually a mixing valve located in the outlet piping of the coil. As the valve mixes the two flow streams, the total flow rate through the branch piping is essentially constant. Figure 18-24 is an example of three-way valve control for a central chilled water plant system.

Using a two-way throttling valve reduces the flow rate and increases pressure differentials across the pump and load circuits. This requires more attention to valve and pump selection because the pump operating point will move as the valves reduce system flow rate, see Figure 18-25. Due to the more exacting design requirements of the two-way valve control at terminal units, the three-way valve has been widely used. This does waste energy because there is a constant flow rate and constant pump horsepower regardless of cooling load.

To solve the problem of variable water flow to terminal units and constant flow through the chiller while operating the chiller plant more efficiently, the scheme in Figure 18-26 has been used. Figure 18-26 illustrates this scheme with two chillers but any number of chillers can actually be used. At full load, both chillers are on line and full flow goes to the terminal units. As the terminal unit valves modulate from decreased load, the flow will decrease and the pressure drop from supply to return mains will increase. The pressure differential controller will sense this and partially open the bypass valve to compensate. The bypass valve is sized to match the flow through one chiller which enables one chiller and pump to shut down when the bypass is fully open. As the load increases to a point where the bypass is closed completely, the second chiller and pump can be started.

Figure 18-27 shows a hot water generator and chiller connected by changeover valves to the distribution piping, with the primary control as the means of changeover from the heating to cooling source.
Notes:
System chilled water flow decreases load
Bypass valve modulates open to reduce "P" increases
Chiller #2 and pump shut down when bypass valve is nearly open

FIGURE 18–26 TWO-WAY VALVE WITH PUMP BYPASS
Changeover can be initiated either manually or automatically by a solar compensated outdoor thermostat. The two-pipe central plant changeover sequence from heating to cooling must assure that the temperature of the return water to the chiller does not cause excessively high evaporative pressures. If the water temperature to the chiller exceeds about 75°F (24°C), a reciprocating compressor chiller could be overloaded by the resulting high suction pressure. To avoid this, a return water bypass loop around the chiller can be used to reduce the entering water temperature until the system return temperature has cooled to an acceptable level. In addition, reset of the hot water temperature with outdoor temperature can assure that the water temperature entering the chiller is much lower at changeover than at design heating conditions.

The changeover from cooling to heating should not allow cold water to enter a hot boiler, as thermal shock to the heat exchanger could occur. A pipe-mounted aquastat can help avoid this by working as a low-limit control of the boiler three-way bypass valve. Although it is simpler and safer to use shell and tube heat exchangers to isolate the boiler and chiller from the distribution piping, the use of shell and tube heat exchangers will result in a higher initial installed cost and lower the operational efficiency.

Hydronic or hot water heating systems permit use of relatively simple automatic temperature control systems. For installations that do not require different supply water temperatures simultaneously and where the boiler is not used to heat domestic hot water, outdoor reset control of the supply water temperature can be accomplished by direct control of the burner from a thermostat that is located in the boiler water and is reset in accordance with outdoor temperatures. Figure 18-28 illustrates this system. The water temperature controller should not be reset below 120°F (52°C) to prevent condensation of flue gases and oxygen reabsorption into the boiler water.

In systems using converters or heat exchangers for hot water supply, a controlling thermostat actuated by the leaving water temperature modulates the heat source to the converter. The set point of this supply water thermostat can be readjusted in accordance with outdoor temperature. In some systems, a boiler or converter supplies water at a constant temperature and three-way mixing valves are used to supply water at different temperatures for various uses or to multiple zones.
In this case, a three-way mixing valve under control of a thermostat in the valve discharge blends boiler water with return water to provide supply water at the desired temperature. The control setting of this thermostat in the valve discharge can be reset in accordance with outdoor temperatures. Since low flow through the boiler can cause damage, a primary-secondary pumping system should be used.

On systems using outdoor-reset supply water and lowered night or weekend temperatures, provisions should be made for warm-up on restoration to day or normal operation. This can be accomplished by the building automation system, a warm-up space thermostat, or some similar means of providing higher than normal water temperatures during the warm-up period. Precautions should be taken to prevent the freezing of water in a hot water heating system whenever design outdoor temperatures are extremely low particularly if the system is to be operated on an intermittent basis. In selecting a means of freeze protection, consideration should be given to the type of system as well as outdoor temperature. The possibility of water freezing in the mains can be minimized by using an outdoor thermostat to provide continuous circulation of the water whenever the outdoor temperature drops below freezing. Another arrangement uses an interval timer and outdoor thermostat to provide intermittent cycling of the circulating pump whenever the outdoor temperature drops below some predetermined temperature.

**18.11.7.2 Load Control**

Figure 18-29 shows a control system for load control of a gas- or oil-fired boiler. The boiler controls are usually supplied with the equipment and include combustion controls including flame failure, high temperature and other safety cutouts as well as capacity controls. Boiler capacity is usually controlled by intermittent burner firing although fuel input modulation is common in larger systems. In most cases the boiler is controlled to maintain a constant water temperature although the temperatures may be reset from an outdoor air thermostat when the boiler is not used for domestic hot water heating. In Figure 18-29, the supply water temperature is reset by a master-submaster arrangement with the outdoor master thermostat resetting the submaster thermostat according to the reset schedule shown. The water temperature should not be reset below 120°F (49°C) to avoid flue gas condensation and possible boiler damage.
18.11.7.3 Zone Control

Zone control usually consists of maintaining a constant supply temperature from the boiler and varying the flow rate to the zone. A room thermostat is shown controlling a three-way valve that varies the preheat coil capacity. Larger systems with sufficiently high pumping requirements use VFDs, pump discharge valves with minimum-flow bypass valves, and two-speed drives to reduce excessive secondary pumping capacity.

18.11.7.4 Heat Exchangers

Hot water heat exchangers or steam-to-water converters are sometimes used instead of boilers as hot water generators. The control scheme in Figure 18-29 can be used with either low-pressure steam or boiler water ranging from 200 to 260°F (93 to 127°C) as the heating source. The supply water thermostat controls a modulating two-way valve in the steam or hot water supply line. An outdoor thermostat is usually used to reset the supply water temperature downward as the load decreases. A flow switch interlock can be used to close the two-way valve when the hot water pump is off.

18.11.8 Electric Heat Control

18.11.8.1 Electric Heat Control Overview

Typical methods used to control heat output of electric heating equipment are typically either solid-state control or some form of on-off control.

Electrical resistance heat transfer devices differ from steam or hot water heat exchangers in that steam or hot water heat exchangers tend to be somewhat self-regulating. For example, if airflow through a steam or hot water coil is stopped, the coil surfaces will approach the temperature of the entering steam or hot water but cannot exceed this temperature. The only heat removed from the steam or hot water will be the losses by convection and radiation to the surrounding area. No damage to the coil generally results under conditions of no airflow.

This is not true of electric coils or heaters. If the means of heat removal is blocked, such as stopping airflow through an electric coil, but power is still applied, the heater will continue to generate rated heat output. This will result in high element temperatures and unsafe conditions. Therefore, it is imperative that control and power circuits be interlocked with heat transfer devices so that the power to the heater will be shut off upon shutdown of the device. Where air or water flow is provided mechanically by a fan or pump, use of flow or differential pressure switches for interlocking is recommended in addition to electrical interlocks through auxiliary contacts in motor starters or VFDs. Limit thermostats should also be provided to de-energize heaters if safe operating temperatures are exceeded.

In addition to the airflow interlock device, an automatic reset high limit thermostat and a manual reset backup high limit device are usually applied to duct heaters, see Figure 18-31. The auto-reset high limit normally de-energizes the control circuit. However, if the control circuit has an inherent time delay or uses solid-state switching devices a separate safety contact or may be desirable. The manual reset backup limit is generally arranged to interrupt all current to the heater independently, should all other control devices fail.

18.11.8.2 Basic Control Modes

The basic control modes for electric heat are as follows:

- Two-Position Action
- Incremental Action
- Proportional Action

18.11.8.2.1 Two Position

Two-position action may be one of the following:

- Direct Mode
- Indirect Mode
- Timed Mode

Direct mode, on-off control systems provide an economical means of controlling temperatures where the load is small enough to be switched directly by the room or remote thermostat. Mild temperature fluctuations and varying degrees of offset can be expected depending on sensing element location, system time constant, and operating characteristics of the thermostat.

Indirect or pilot mode systems are similar to direct on-off control except that a separate contact device is added to handle the load current. Three systems are commonly used:

- A pilot-duty electric thermostat controlling a contractor.
FIGURE 18–29 HYDROIC SYSTEM LOAD AND ZONE CONTROL

FIGURE 18–30 STEM-TO-HOT WATER HEAT EXCHANGE CONTROL
A pneumatic thermostat piloting a pressure-electric contact.

An electronic thermostat controlling a relay.

Solid-state switching to control heat input in proportion to load.

Timed two-position action systems are similar to indirect ones except that they include heaters or other means of introducing controlled heat anticipation to increase cycling rate and reduce temperature overshoot.

18.11.8.2.2 Incremental

Incremental action may be any one of the following:

- Sequence Control
- Time-Proportional
- Vernier Control Mode

The sequence control mode uses motor-driven switches or several pressure-contact switches to provide incremental control of electric heaters by actuating switches in sequence to energize elements until the heating demand is satisfied. The number of steps used depends on several factors such as allowable temperature rise per step, which is a function of accuracy desired, number of circuits available in the heater, and capacity or load rating of switching device contacts. Contactors can be used to increase load rating per step of the switching device.

The time-proportional mode approaches full proportional control of heater output by establishing a fixed time base and allowing current to flow in the heater a percentage of the fixed time base. The percentage on time varies with heating demand as sensed by proportional output thermostat.

The vernier control mode combines sequence control with solid-state power control to provide full modulation of larger heaters. The heating coil is divided into several sections with one section being modulated by a solid-state controller and the rest being stepped in sequence by a controller. On a demand for heat, control is on the modulated section until it reaches 100 percent output at which time the next stage is stepped in. As the increased heat supply is sensed by the discharge thermostat, power to the modulated section is cut back as required to meet demand. This arrangement allows the modulated section to act as a vernier on each of the other sections as they are stepped in or out. The modulated section capacity should be 25 to 50 percent larger than
the stepped sections to prevent cycling between stages.

18.11.8.2.3 Proportional

In proportional action, the controlled device is positioned proportionally in response to changes in the controlled variable.

The output of the controller is proportional to the difference between the sensed valve and its set point. The controlled device is normally adjusted to be in the middle of its control range at set point, by using an offset adjustment.

18.11.9 Gas-Fired Heat Control

Gas-fired air handling units are used for makeup air, heating, heating ventilating, and heating ventilating-cooling applications where gas is readily available. Custom-built or packaged units are available for either indoor or outdoor installation.

Gas-fired heating equipment is typically located on the roof to achieve lower installed costs and smaller air distribution systems. For these installations, care should be taken in the selection and location of controls in view of the effects of wide ambient temperature ranges on both the controls and the devices controlled. Weatherproofed and heated compartments or walk in areas will minimize the potential for these types of problems.

The gas-fired air-handling unit may be either direct- or indirect-fired. Direct-fired units have no combustion chamber. The products of combustion are burned and mixed directly with the makeup air stream for combined discharge into building space. Applications are limited to makeup air heaters to replace exhaust air with tempered 100 percent outdoor air alleviating negative pressure problems. A direct fired make up air heater also produces a substantial amount of moisture that is introduced directly into the space being heated. For example, 1,000,000 btu's (29.3 kWs) of natural gas will produce 18 lbs (8.2 kg) of water per hour. This water is introduced into the airstream as vapor. This addition of water vapor must be considered in the overall moisture load of the building.

Indirect-fired heaters vent products of combustion to the outside atmosphere and require a combustion chamber and heat exchanger. Applications are generally unlimited for heating, ventilating, and cooling units including gas-fired single-zone and multizone unit applications.

Controls for gas-fired units are divided into burner and safety, limit, and flame control devices. Temperature control consists of those remaining controls used to measure and control space or discharge temperatures as well as ventilation. Temperature controls vary the burner-firing rate by controlling solenoid, motorized, diaphragm or butterfly gas valves. Bypass airflow around a duct furnace should be carefully limited.

18.11.10 Refrigeration System Control

Control of refrigeration equipment varies considerably depending on the manufacturer. Smaller sizes of reciprocating equipment can be controlled directly by the space temperature and larger equipment will "pump down" when the space temperature closes the solenoid valve(s) on the cooling coil(s) or chiller. When pumping down, the low-pressure control stops the compressor but unloading equipment can allow the compressor to meet a varying load before this happens.

On direct-expansion systems, the compressor should not be allowed to operate unless the fan of the conditioning unit is in operation. Likewise, in a chilled water coil installation, the compressor should not operate unless the water-circulating pump is operating. Proper wiring between a fan or pump starter and compressor starter will provide this feature.

The control methods used for centrifugal refrigeration equipment must be verified with the manufacturer because most have built-in modulation controls. This is required because the typical air-conditioning system only requires maximum cooling for a few consecutive hours during a cooling season. Other load conditions for typical air conditioning systems are generally one of the following:

- Long periods of almost constant partial load.
- A load varying from minimum to maximum in a relatively short time.

In any capacity modulating system it is desirable for the horsepower required to drive the compressor to be reduced at the same rate as the decrease in load. In other words, it would be desirable for the power required to be 80 percent of full load power when the machine is operating at 80 percent capacity. Various methods of accomplishing capacity control have been used. There are considerable variations in their effectiveness and the resulting efficiency at part load. Some methods of capacity control are as follows:

- Off-And-On Control
- Cylinder Unloading
CONTROL SYSTEM LAYOUT AND OPERATIONAL CONSIDERATIONS

18.12.1 Size of Controlled Area

Individually controlled areas should not be excessively large because the difficulties of obtaining good distribution and of finding a representative location for the space controls become greater as the area increases. Each individually controlled area must have similar load characteristics throughout and should otherwise conform to the recommendations given in the section on zone control.

For uniform conditions throughout the area, equitable distribution must be provided by good design, careful sizing of equipment, and proper balancing of the system. The device monitoring a system condition can measure conditions only at the point at which it is located and it cannot compensate for variable conditions throughout the area caused by improper distribution or inadequate design. Areas or rooms having dissimilar load characteristics or different conditions to be maintained should be individually controlled. The smaller the controlled area, the better the control will be and the closer the system will more nearly approach the desired performance objectives.

18.12.2 Location of Space Controllers

Space controllers must be located where they will sense the variables they are intended to control and where the condition is representative of the entire area or zone served by the controller. In large open areas that have more than one zone, thermostats should be in the middle of the zone to assure that they are not affected by conditions in the adjacent, surrounding zones. It is common to select three different locations in a large space to install space temperature controllers.

18.12.2.1 Wall-Mounted Thermostats

Wall-mounted thermostats usually are placed on inside walls or columns at a representative height in the conditioned space. Outside wall locations should be avoided. Thermostats should not be mounted where they will be affected by heat from other sources such as direct rays of the sun, pipes or ducts in the wall, convectors, direct air currents from diffusers, and other similar conditions. The location should provide ample air circulation and airflow should be unimpeded by furniture or other physical obstructions. Wall-mounted thermostats should never be placed in spaces such as corridors, lobbies, foyers, or other building circulation areas unless used for control of these areas only.

18.12.2.2 Return-Air Thermostats

Return air thermostats can be used to control floor-mounted unitary HVAC systems such as induction or fan-coil units. On induction and fan-coil units, the sensing element will be located behind the return air grille. However, on classroom unit ventilators that use up to 100 percent outdoor air for natural cooling a forced flow-sampling chamber will normally be provided for the sensing element.

If return air sensing is to be used with central fan systems, the sensing element should be located as near as possible to the space being controlled. This location will tend to eliminate influence from other spaces and the effect of any heat gain or loss in the duct. Where combination supply/return air light fixtures are used to return air to a ceiling plenum, the return air opening of a light fixture can be used as a location for a return air-sensing element.

18.12.2.3 Diffuser-Mounted Thermostats

Diffuser-mounted thermostats usually have sensing elements mounted on ceiling supply diffusers of the circular or square type and depend on aspiration of room air into the supply air stream. They should be used only on high aspiration diffusers that are adjusted for a horizontal air pattern. The diffuser in which the element is mounted should be in the center of the occupied area of the zone being controlled.

18.12.3 Seasonal Changeover

Where changeover from summer to winter will be accomplished automatically, there should be a period of inoperation of both the heating and cooling systems. Otherwise, cycling from one function to the other could occur. This period of inoperation can be accomplished by a two-position controller with an adjustable differential sensing outdoor temperature or by a controller sensing the temperature at a representative conditioned space.

18.12.4 Unoccupied Space Temperatures

When temperatures employed during unoccupied periods are lower than normally maintained during
periods of occupancy, the proper day and night temperature time cycle is often established by the building automation system or an automatic timer. Sufficient time should be allowed in the morning to pick up the conditioning load well before there is any heavy load increase in the conditioned spaces.

Automatic temperature setback controls are essentially a combination of two thermostats and a time clock. One thermostat is set for a daytime temperature and the other thermostat is set for a nighttime temperature. The time clock switches control from one thermostat to the other at a preset time. Modern electronic thermostats “internalize” these functions without the need for separate devices or timers but still may require periodic checking for time accuracy.

18.12.5 Multiple Thermostats Per Zone

Buildings or zones laid out in a modular concept may be designed for subdividing to meet occupants’ needs. Until the actual subdividing is done, operating inefficiencies can occur if more than one thermostat is located in a zone. If the system is such that one thermostat can activate the heating system and another thermostat in the same zone can simultaneously activate the cooling system, then that building area should be controlled from a single thermostat until actual subdividing of the space is accomplished.

18.12.6 Dead Band Control Systems

A dead band control system is a system where there is a temperature range such as 70 to 76°F (21 to 24°C) where there is only ventilation but no heating or cooling provided. Figure 18-32 illustrates the operation of a dead band control system. In most existing HVAC systems, the throttling range between “no heat” and “no cool” is usually small and about 2°F (1.5°C). By extending the throttling range, the thermostat gradually calls for more heating or cooling as the temperature floats farther away from the dead band, see Figure 18-32. At the limits of the throttling range, 68°F and 78°F (20°C and 26°C) for example, the HVAC system is working “full throttle” to return the space to a more comfortable temperature.
19.1 INTRODUCTION

This chapter addresses smoke control systems. Smoke is recognized as the major cause of death in fire situations. Smoke often migrates to building locations remote from the actual fire, threatening life and damaging property. Stairwells and elevator shafts frequently fill with smoke during a fire and can block occupant evacuation and inhibit fire-fighting operations. Smoke control systems control smoke movement through the use of fans to control airflow and pressure differences within the building. The purpose of a smoke control system is to reduce occupant deaths and injuries from smoke and facilitate fire-fighting operations. Another important objective of smoke removal and control systems is to reduce property damage that results from smoke.

19.2 SMOKE CONTROL SYSTEMS

A smoke control system can be designed to provide a safe escape route, a safe refuge area, or both. A smoke control system can meet its objectives even if a small amount of smoke infiltrates protected areas. However, smoke control systems are typically designed assuming that no smoke infiltration will occur. Figure 19-1 provides factors that affect the effectiveness of a smoke control system.

Automatic fire suppression systems are a key part of many fire protection systems and the effectiveness of these systems is an important factor in controlling and extinguishing building fires. However, it is important to recognize that the function of fire suppression and smoke control systems are very different and cannot be substituted for each other. Automatic fire suppression systems limit the growth rate and the size of the fire but do not necessarily reduce or eliminate the movement of smoke. On the other hand, smoke control systems can maintain tolerable conditions along critical egress routes and have little effect on the fire. Sprinkler systems are generally designed to suppress fires not necessarily extinguish them.

The smoke control system must be designed to override the local controls in an HVAC system so that the air supply necessary to pressurize non-fire spaces is supplied. Automatic activation of a smoke control system should be considered with the primary means of activation being an alarm from a smoke detector system located in the building areas. A smoke control system should be equipped with a remote control center from which the smoke control system can be manually overridden and to which there is easy access for the fire department officials.

19.3 SMOKE MOVEMENT

19.3.1 Smoke Movement Major Driving Forces

The major driving forces causing smoke movement in a building are:

- Stack Effect
- Buoyancy
- Expansion
- Wind
- HVAC Systems

During a fire, smoke movement within the building is usually caused by a combination of these factors.

19.3.2 Stack Effect

When it is cold outside, there often is an upward movement of air within vertical building shafts, such as stairwells, elevator shafts, dumbwaiter shafts, mechanical shafts, or mail chutes. This phenomenon is referred to as stack effect. The air in the building has a buoyant force because it is warmer and less dense than the outside air. This buoyant force causes air to rise within the shafts of buildings. The significance of normal stack effect is greater for low outside temperatures and for tall buildings. However, stack effect can occur even in a one-story building. Conversely, when the outside air is warmer than the inside building air, there will be downward airflow through vertical building shafts. This downward airflow is referred to as reverse stack effect.

Smoke movement within a building fire can be dominated by stack effect. For example, in a building with normal stack effect the existing air currents can move smoke considerable distances from the fire origin, see Figure 19-2. If the fire is below the neutral plane, smoke moves with the building air into and up the shafts. This upward smoke flow is enhanced by any buoyancy forces on the smoke due to its temperature. Once above the neutral plane, the smoke flows out of the shafts into the upper floors of the building. If the leakage between floors is negligible, the floors below the neutral plane, except the fire floor, will be smoke free.

Smoke from a fire located above the neutral plane is carried by the building airflow to the outside through openings in the exterior of the building assuming leakage between floors is negligible. All floors other than
- Occupancy Type And Characteristics
- Evacuation Plan
- Refuge Areas
- Distribution Of Occupant Density
- Human Life Support Requirements
- Form Of Detection And Alarm
- Fire Service Response To Alarm Characteristics
- Fire Suppression System Characteristics
- Type Of Building, Heating, Ventilation And Conditioning Systems
- Energy Management System
- Building Security Provisions
- Controls
- Status Of Doors During Potential Fire Condition
- Potential Fire Threats
- Internal Compartmentation And Architectural Characteristics
- Building Characteristics
- Building Leakage Paths
- Exterior Temperatures
- Wind Velocity
- Arson

FIGURE 19-1 SMOKE CONTROL SYSTEM DESIGN FACTORS
19.3HVAC Systems Applications
• Second Edition

FIGURE 19-2 AIR MOVEMENT DUE TO NORMAL AND REVERSE STACK EFFECT

Note: Arrows Indicate Direction of Air Movement

the fire floor will remain smoke-free. When the leakage between floors is considerable, there is an upward smoke movement to the floor above the fire floor.

Figure 19-2 shows the air currents caused by reverse stack effect. These forces tend to affect the movement of relatively cool smoke in the reverse of the normal stack effect. In the case of hot smoke, buoyancy forces can be so great that smoke can flow upward even when conditions favor reverse stack effect.

19.3.3 Buoyancy

High temperature smoke from a fire has a buoyancy force due to its reduced density. In a building with leakage paths in the ceiling of the fire room, this buoyancy-induced pressure forces smoke movement to the floor above the fire floor. In addition, this pressure causes smoke to move through any leakage paths in the walls or around the doors of the fire compartment. As smoke travels away from the fire, its temperature drops due to heat transfer and dilution. As a result, the effect of buoyancy generally decreases as distance from the fire increases.

19.3.4 Expansion

In addition to buoyancy, the energy released by a fire can cause smoke movement due to expansion. In a fire compartment with only one opening to the building, building air will flow into the fire compartment and hot smoke will flow out of the fire compartment.

19.3.5 Wind

Wind can have a pronounced effect on smoke movement within a building. Frequently in fire situations, a window breaks in the fire compartment. If the window is on the leeward side of the building, the negative pressure caused by the wind vents the smoke from the fire compartment. This can greatly reduce smoke movement throughout the building. However, if the broken window is on the windward side, the wind forces the smoke throughout the fire floor and even to other floors. This both endangers the lives of building occupants and hampers fire fighting. Pressures induced by the wind in this type of situation can be relatively large and can easily dominate smoke movement throughout the building.

19.3.6 HVAC System

The HVAC system frequently transports smoke during building fires. In the early stages of a fire, the HVAC
system can serve as an aid to fire detection. When a fire starts in an unoccupied portion of a building, the HVAC system can transport the smoke to a space where people can smell the smoke and be alerted to the fire. However, as the fire progresses, the HVAC system will transport smoke to every zone that it serves and will endanger the life of people occupying those zones. The HVAC system also supplies air to the fire space that can aid combustion and hinder control of fire growth. For these reasons, HVAC systems traditionally have been shut down when fires have been detected in the building or zones served. Although shutting down the HVAC system prevents it from supplying air to the fire, it does not prevent smoke movement through the supply and return air ducts, air shafts, and other building openings due to stack effect, buoyancy, or wind.

19.4 SMOKE MANAGEMENT

The term “smoke management” includes all methods that can be used singly or in combination to modify smoke movement for the benefit of occupants and fire fighters and for the reduction of property damage. The use of barriers, smoke vents, automatic fire/smoke dampers, smoke dampers, and smoke shafts are traditional methods of smoke management.

The effectiveness of a barrier in limiting smoke movement depends on the leakage paths in the barrier and on the pressure difference across the barrier. Holes where pipes penetrate walls or floors, cracks where walls meet floors, and cracks around doors are a few possible leakage paths. The pressure difference across these barriers depends on stack effect, buoyancy, wind, and the HVAC system.

The effectiveness of smoke vents and smoke shafts depends on their proximity to the fire, the buoyancy of the smoke, and the presence of other driving forces. In addition, when smoke is cooled by sprinklers, the effectiveness of smoke vents and smoke shafts is greatly reduced. Fan systems are sometimes installed to overcome this limitation. These fan systems are referred to as “smoke control systems” and they create pressure differences and airflows to limit smoke movement within the building and facilitate its exhaust.

19.5 SMOKE CONTROL

Smoke control uses the barriers such as walls, floors, and doors that are used in traditional smoke management in conjunction with airflows and pressure differences generated by mechanical fans.

Figure 19-3 illustrates a pressure difference across a barrier acting to control smoke movement. Within the barrier is a door. The high-pressure side of the door can be either a refuge area or an escape route. The low-pressure side is exposed to smoke from fire. Airflow through the cracks around the door and through other construction cracks prevents smoke infiltration to the high-pressure side.

When the door of the barrier is opened, airflow through the open door results. When the air velocity is low, smoke can flow against the airflow into the refuge area or egress route, see Figure 19-4. This smoke backflow can be prevented if the air velocity is sufficiently large, see Figure 19-5. The magnitude of the velocity necessary to prevent backflow depends on the energy release rate of the fire.

The two basic principles of smoke control can be stated as follows:

- Airflow by itself can control smoke movement if the average air velocity is of sufficient magnitude.
- Air pressure differences across barriers can act to control smoke movement.

The use of air pressure differences across barriers to control smoke is referred to as pressurization. Pressurization results in airflows of high velocity in the small gaps around closed doors and other smoke barrier penetrations. This high-velocity airflow from a high-pressure to a low-pressure area prevents smoke backflow through these openings. However, when there are only small cracks as there are around closed doors, designing to and measuring air velocities is impractical. In this case, the appropriate physical quantity is pressure difference. Consideration of the two principles as separate has the added advantage that it emphasizes the different considerations that need to be given for open and closed doors.

Due to the fact that smoke control relies on air velocities and pressure differences produced by mechanical fans, smoke control has the following three advantages in comparison to the traditional methods of smoke management:

- Smoke control is less dependent on tight barriers. Allowance can be made in the design for reasonable leakage through barriers.
- Stack effect, buoyancy, and wind are less likely to overcome a smoke control system than a passive smoke management system. In the absence of smoke control, these driving
FIGURE 19–3 SMOKE CONTROL SYSTEM USING PRESSURE DIFFERENTIAL ACROSS A SMOKE BARRIER TO PREVENT SMOKE MIGRATION FROM THE LOW- TO THE HIGH-PRESSURE SIDE

FIGURE 19–4 SMOKE BACKFLOW AGAINST LOW AIR VELOCITY THROUGH AN OPEN DOORWAY
forces cause smoke movement to the extent that leakage paths allow. However, pressure differences and airflows of a smoke control system will act to oppose these driving forces.

Smoke control systems can be designed to prevent smoke flow through an open doorway in a barrier by the use of airflow. Doors in barriers are open during evacuation and are sometimes accidentally left open or propped open during fires. In the absence of smoke control, smoke flow through these doors is common.

Smoke control systems should provide a path for smoke movement that allows it to be exhausted to the outside the building. The dilution of smoke in the fire space is not a means of achieving smoke control. In other words, smoke movement cannot be controlled by simply supplying and exhausting large quantities of air from the space or zone in which the fire is located. This supplying and exhausting of air is sometimes referred to as purging the smoke. Due to the large quantities of smoke produced in a fire, purging cannot ensure breathable air in the fire space. In addition, purging in itself cannot control smoke movement because it does not provide the needed airflows at open doors and the pressure differences across barriers. However, for spaces separated from the fire space by smoke barriers, purging can significantly limit the level of smoke.

19.5.1 Airflow

Airflow can also be used to stop smoke movement through a building zone or space. However, the two places where air velocity is most commonly used to control smoke movement are open doorways and corridors. The required high air velocities that are needed for smoke control through open doorways and corridors are both expensive and difficult to achieve. The use of airflow is most important in preventing smoke backflow through an open doorway that serves as a boundary of a smoke control system.

Even though airflow can be used to control smoke movement, it should not be the primary method because the quantities of air required are so large. The primary means of controlling smoke movement should be the establishment of air pressure differences across partitions, doors, and other building dividers.

19.5.2 Pressurization

To control smoke movement, the pressure differences produced by a smoke control system must be sufficiently large that they are not overcome by pressure
fluctuations, stack effect, smoke buoyancy, and the wind. However, the pressure difference produced by a smoke control system should not be so large that doors become hard or impossible to open. Codes and standards should be referred to for allowable pressures and forces.

19.5.3 Purging Smoke

In general the systems discussed in this section are based on the two basic principles of smoke control. However, it is not always possible to maintain sufficiently large airflows through open doors to prevent smoke from infiltrating a space that is intended to be protected. Ideally, such occurrences of open doors will only happen for short periods of time during evacuation. Smoke that has entered such a space can be purged or at least diluted by supplying outside air to the space.

In reality, it is impossible to ensure that the concentration of the contaminant is uniform throughout the compartment. As a result of buoyancy, it is likely that higher concentrations of contaminant will tend to be near the ceiling. Therefore, an exhaust inlet located near the ceiling and a supply outlet located near the floor would probably purge the smoke quickly. Caution should be exercised in the location of the supply and exhaust points to prevent the supply air from blowing into the exhaust inlet and thus short circuiting the purging operation.

19.6 SIMPLE STAIRWELL PRESSURIZATION

Many pressurized stairwells are designed and built with the goal of providing a smoke-free escape route in the event of a building fire. A secondary objective is to provide a smoke-free staging area for firefighters. On the fire floor, a pressurized stairwell must maintain a pressure difference across a closed stairwell door so that smoke infiltration is prevented. Codes and standards should be referred to for allowable pressures and forces.

19.6.1 Single Injection Systems

A single injection system is one that has pressurization air supplied to the stairwell at one location. The most common injection point is at the top, see Figure 19-6. With this system, there is the potential for smoke feedback into the pressurized stairwell via smoke entering the stairwell through the pressurization fan intake. As a result of this possibility, including a control system that can automatically shutdown the stairwell pressurization system in the event of smoke feedback should be considered.

For tall stairwells, the effectiveness of a single injection system can be compromised when a few doors are open near the air supply injection point. All of the pressurization air can be lost through these open doors and the system will not be able to maintain positive pressure across doors further from the injection point. This situation is especially likely with bottom injection systems when a ground level stairwell door is left open. To prevent this, the building height often limits the use of single injection systems. As a result, care should be taken when using top injection stairwells for taller buildings and for all bottom injection stairwells. Airflow measurement of the single injection fan systems is limited and minimal if no ductwork is available for airflow measurement.

19.6.2 Multiple Injection Systems

Figures 19-7 and 19-8 are two examples of many possible multiple injection systems that can be used to overcome the limitations of single injection systems. The pressurization fans can be located at ground level, roof level, or at any location in between. In all cases, the supply air intake should be separated from exhaust points, outlets from smoke shafts and roof smoke and heat vents, open vents from elevator shafts, or other building openings that might expel smoke from the building in a fire situation. Ideally, this separation should be as great as practical. As a result of hot smoke rising, consideration should be given to locating supply air intakes below such critical openings. However, outdoor smoke movement that might result in smoke feedback depends on location of fire, location of points of smoke leakage from the building, wind speed and direction, and on the temperature difference between the smoke and the outside air.

Figures 19-7 and 19-8 shows the supply duct in a separate shaft. However, systems have been built that have eliminated the expense of a separate duct shaft by locating the supply duct in the stairwell itself. If the duct is located inside the stairwell, care must be taken that the duct does not become an obstruction to orderly building evacuation.

Many multiple injection systems have been built with supply air injection points on each floor. These represent the ultimate in preventing loss of pressurization air through a few open doors. However, injection points at each floor may not be necessary to overcome the limitations of a single injection system. Relief dampers should be provided if there is a possibility of overpressure.
FIGURE 19−6 TOP INJECTION STAIRWELL PRESSURIZATION

CAUTION: THIS SYSTEM SHOULD NOT BE USED FOR TALL STAIRWELLS

FIGURE 19−7 MULTIPLE INJECTION WITH GROUND LEVEL FAN
19.6.3 System Limitations

A simple stairwell pressurization system has a supply air system that injects air continuously to the stairwell in the event of a building fire. Both the single and multiple injection systems are simple systems. The supply air fan can be either a centrifugal or a propeller fan. When a propeller fan is used, it should be roof mounted in the horizontal plane with a wind shield. Wall mounted propeller fans without wind shields are not recommended because the flow rates of these fans are highly dependent on the wind.

When all the stairwell doors are closed, the system must maintain satisfactory pressurization. Simple pressurization systems have two limitations:

- They generally are not capable of producing the airflows through open stairwell doorways necessary to prevent smoke backflow when an outside stairwell door is also open.

- When stairwell doors are open, the pressure difference across closed stairwell doors can drop to low levels.

These limitations obviously restrict applications that are appropriate for simple stairwell pressurization systems.

19.7 COMPLEX STAIRWELL PRESSURIZATION

19.7.1 Categories Of Complex Stairwell Pressurization

More complex systems can be grouped into three categories:

- Overpressure relief systems that vent or relieve part of the supply air when all the stairwell doors are closed.

- Supply air fan bypass systems that bypass excess supply air back to the fan inlet.

- Combination stairwell pressurization and fire floor smoke venting systems.

The discussion of simple single and multiple injection systems in the previous sections apply to the above complex stairwell pressurization systems.

19.7.2 Overpressure Relief

The total airflow rate is selected to provide at least the minimum air velocity when a specific number of doors
are open. When all the doors are closed, part of this air is relieved through a vent in order to prevent excessive pressure buildup that could otherwise result in excessive door-opening forces. This excess air can be vented either to the building or to the outside. Exterior vents can be subject to adverse effects of the wind so wind shields are recommended.

Barometric dampers that close can be used to minimize the air losses through the vent when doors are open and the pressure drops below a specified value. Figure 19-9 illustrates a pressurized building stairwell with overpressure relief vents at each floor. In systems built with vents between the stairwell and the building, the vents typically have one or more fire dampers in series with the barometric damper.

An exhaust air duct can be used in a pressurized stairwell as a means of overpressure relief. The purpose of the exhaust air duct is to maintain the required pressure differences using the normal resistance of non-powered exhaust duct.

Exhaust air fans can also be used to prevent excessive pressures when all stairwell doors are closed. A differential pressure sensor should control the exhaust air fan to ensure that the exhaust air fan does not operate when the pressure difference between the stairwell and the building falls below a specific level. This should prevent the exhaust air fan from pulling smoke into the stairwell when a number of open doors have reduced stairwell pressurization. Such an exhaust air fan should be specifically sized so that the pressurization system will perform as required. A wind shield for the exhaust air fan is recommended because the exhaust air fan performance can be affected by the wind.

An alternate method of venting a stairwell is through an automatically opening stairwell door to the outside at ground level. Under normal conditions, this door would be closed and in most cases locked for security reasons. Provisions need to be made so that this lock does not interfere with the automatic operation of the system.

Possible adverse wind effects are also a concern with a system that uses an open outside door as a vent. Occasionally, high local wind velocities develop near the exterior stairwell door and are difficult to estimate in the vicinity of new buildings. Local objects on a wall can act as wind breaks or wind shields.

**FIGURE 19–9  STAIRWELL PRESSURIZATION WITH AIR SUPPLY AT EACH FLOOR**

Notes:

1. Vents to the building have a barometric damper and one or two fire dampers in series

2. A roof mounted supply fan is shown, however the fan may be located at any level

3. A manually operated damper may be located at the stairwell top for smoke purging by the fire department.
19.7.3 Supply Air Fan Bypass

In this system, the capacity of the supply air fan is sized to provide at least the minimum air quantity through open doors when a specified number of doors are open. Figure 19-10 illustrates a supply air fan bypass system. Modulating bypass dampers that are controlled by one or more static pressure sensors vary the rate of airflow into the stairwell. These static pressure sensors sense the pressure difference between the stairwell and the building. When all the stairwell doors are closed, the pressure difference increases and the bypass damper opens to increase the bypass air and decrease the flow of supply air to the stairwell. In this way, excessive stairwell pressures and excessive pressure differences between the stairwell and the building are prevented.

19.7.4 Smoke Venting

Smoke venting of the fire floor can be used to improve the performance of the stairwell pressurization. This smoke removal may or may not be part of a zoned smoke control system. Three different types of smoke removal can be considered:

- Exterior Wall Vents
- Smoke Shafts

Besides providing a path for smoke removal, exterior wall vents allow an increased pressure difference across a closed fire floor stairwell door and allow increased air velocity through an open fire floor stairwell door. Venting the fire floor is also a way to reduce the potential hazards resulting from a broken fire floor window.

Smoke shafts are similar to external wall vents except that smoke from the fire floor is vented through a shaft. Venting is aided by the buoyancy forces of hot smoke. Smoke shafts should be constructed in accordance with local codes.

The effect of a fan powered smoke exhaust system on the performance of a pressurized stairwell is similar to that of exterior wall vents. The exhaust fans can be individually located on each floor or can be used in combination with a smoke exhaust shaft.

19.8 ZONED SMOKE CONTROL

Pressurized stairwells are intended to control smoke to the extent that they inhibit smoke infiltration to the stairwell. However, in a building with just a pressurized stairwell, smoke can flow through cracks in floors and partitions and through shafts to damage property.
and threaten life at locations remote from the fire. The concept of zoned smoke control is intended to limit this type of smoke movement within a building.

A building is divided into a number of smoke control zones, each zone separated from the others by partitions, floors, and doors that can be closed to inhibit the movement of smoke. In the event of fire, pressure differences and airflows produced by mechanical fans are used to limit the smoke spread to the zone in which the fire initiated. The concentration of smoke in this smoke zone goes unchecked and accordingly, in zoned smoke control systems, it is intended that building occupants evacuate the smoke zone as soon as possible after fire detection.

Frequently, each floor of a building is chosen to be a separate smoke control zone. However, a smoke control zone can consist of more than one floor, or a floor can consist of more than one smoke control zone. Figure 19-11 illustrates some typical arrangements of smoke control zones.

In Figure 19-11, the smoke zones are indicated by a minus sign and pressurized spaces are indicated by a plus sign. Each floor can be a smoke control zone, see Figures 19-11(a) and 19-11(b) or a smoke zone can consist of multiple floors, see Figures 19-11(c) and 19-11(d). All of the non-smoke zones in a building can be pressurized, see Figures 19-11(a) and 19-11(c) or only non-smoke zones adjacent to the smoke zone may be pressurized see Figures 19-11(b) and 19-11(d). Similarly, a smoke zone can also be limited to a part of a floor, see Figure 19-11(e).

When a fire occurs, all of the non-smoke zones in the building or only zones adjacent to the smoke zone may be pressurized, see Figures 19-11 (b) and 19-11(d). The latter system is frequently referred to as a "pressure sandwich" and this concept design has the drawback that it is dependent upon proper construction of shafts. It is possible to have smoke flow through shafts past the pressurized zone and into unpressurized spaces. However, pressurizing all non-smoke zones reduces this possibility. The importance of properly locating supply air inlets of pressurized stairwells also applies to the supply air inlets for non-smoke zones.

19.8.1 Smoke Zone Venting

Venting of smoke from a smoke zone is important because it prevents significant overpressures due to thermal expansion of gases as a result of the fire. In addition, venting results in some reduction of smoke concentration in the smoke zone. Venting can be accomplished in the following three ways:

- Exterior Wall Vents
- Smoke Shafts
- Mechanical Venting (Or Exhaust) Systems

Smoke purging that consists of equal air supply and exhaust rates is not considered here for smoke control. The reason that this type of smoke purging is not considered is because it typically does not produce pressure differences or airflows that can control smoke movement. Smoke purging at the airflows available with HVAC systems usually cannot significantly reduce smoke concentrations in the smoke zone resulting from a large fire.

19.8.2 Exterior Wall Vents

Exterior wall vents can consist of windows or wall panels that open automatically when the smoke control system is activated. In order to minimize adverse effects of wind, the area of wall vents should be evenly distributed among all the exterior walls. For a building that is much longer than it is wide, the vents can be evenly divided between the two long sides. Exterior wall venting is most appropriate for buildings with open floor plans and least suitable when the floor plan is divided into many compartments. Precautions should be taken in the design of exterior walls to minimize the possibility of exterior fire spread to floors above the vent because the flow of hot gases through a wall vent can be substantial.

19.8.3 Smoke Shafts

A smoke shaft is a vertical shaft extending from the bottom to the top of a building with openings at the top to the outside and openings to building spaces at each floor. These openings are fitted with normally-closed dampers. Only the damper on the fire floor and the top outside damper should open to vent smoke to the outside in a fire situation. Smoke shafts should be constructed in accordance with local code requirements.

Smoke shafts are not a means of smoke control and by themselves cannot prevent smoke spread throughout the fire floor and to other floors. However, smoke shafts in conjunction with pressurization of non-smoke zones can provide pressure differences to control smoke movement. In order to ensure airflow through open stairwell doors sufficient to prevent smoke backflow, it may be necessary to pressurize stairwells as well.
FIGURE 19–11 TYPICAL SMOKE CONTROL ZONE ARRANGEMENTS
Smoke shafts lend themselves to use in buildings with open floor plans. The air movement caused by smoke shafts operating under normal stack effect conditions tends to pull smoke toward the smoke shaft inlet on the fire floor. It is recommended that smoke shafts be located as far as possible from exit stairwells so that smoke in the vicinity of the shaft inlet does not pose an increased hazard during evacuation or fire fighting. Hot smoke often stratifies near the ceiling so it is recommended that smoke shaft inlets be located in or near the ceiling.

19.8.4 Mechanical Exhaust

A dedicated exhaust system or the exhaust air fans of the HVAC system can be used to provide mechanical exhaust for the smoke zone. Mechanical exhaust of the smoke zone is generally used in conjunction with pressurization of non-smoke zones. In this type of system, stairwell pressurization can greatly reduce the chance of smoke backflow into stairwells.

Mechanical exhaust by itself can result in sufficient pressure differences for smoke control. However, in the event of window breakage or a large opening to the outside from the smoke zone, mechanical exhaust may no longer be able to maintain favorable pressure differences. For this reason, mechanical exhaust alone does not constitute an adequate smoke control system when there is a significant probability of window breakage or an opening from the smoke zone to the outside.

Exhaust gases from the fire are often at high temperatures and have the potential to cause fan failure. The temperature of the smoke as it goes through the exhaust system decreases due to heat transfer and dilution. When the HVAC system is being used for smoke exhaust, the recommended approach is to choose a large enough smoke control zone so that dilution alone will lower the temperature of the gases at the fan to the point where an ordinary fan can be used safely. This approach depends on the fire being limited to only a portion of the smoke zone such as a single room. Then, even for a fire in the compartment nearest the fan, the temperature of the gases reaching the fan will be sufficiently diluted by cooler air being drawn into the fan from other compartments.

In cases where the exhaust gas temperature cannot be reduced through dilution, heat-rated fans can be used. In addition to the fan, the ductwork in this system needs to be able to withstand the exhaust gas temperature to which the ductwork will be subjected. Low temperature smoke from a fire space with sprinklers should not pose temperature problems for either fans or ductwork.

In the smoke zone, the location of the inlets of the mechanical exhaust is an important consideration. These inlets should be located away from exit stairs so that smoke in the vicinity of the inlet does not pose an increased hazard during evacuation or fire fighting. As a result of hot smoke frequently stratifying near the ceiling, it is recommended that exhaust inlets be located in or near the ceiling.

19.8.5 Refuge Areas

A refuge area is a place within a building where building occupants are protected for a period of time from the heat and smoke produced by a building fire. In a limited sense, any non-smoke zone of a zoned smoke control system is a refuge area intended to protect occupants for the period of time needed for evacuation.

19.8.6 Smoke Dampers

A smoke damper is a device, installed in an air distribution system that is designed to resist the movement of air or smoke in the event of a building fire. A smoke damper can be used for either traditional smoke management or smoke control. For additional information on the selection, installation, and testing of smoke dampers refer to the Fire, Smoke and Radiation Damper Installation Guide for HVAC Systems published by SMACNA.

19.8.7 HVAC System Considerations

Most zoned smoke control systems use the fans of the HVAC system to control smoke movements as follows:

- In the smoke zone 100 percent of the return air is exhausted to the outside and supply air to the smoke zone is shut off. Alternately, exterior wall vents or smoke shafts may be used.
- In non-smoke zones, supply air is 100 percent outside air and the exhaust air is shut off.
- In stairwells, zoned smoke control systems may include stairwell pressurization to further protect escape routes.

Opening a stairwell door on a floor of a non-smoke zone increases the pressure difference across the closed stairwell door on the smoke zone. Opening doors in a stairwell on both a non-smoke zone floor and the smoke zone floor results in considerable airflow from the stairwell to the smoke zone that is accompan-
ied by reduced pressure difference across the boundary of the smoke zone.

Opening only a stairwell door in the smoke zone will result in some airflow from the stairwell to the smoke zone. This flow rate depends on the level of pressurization of the non-smoke zones and the leakage characteristics of the stairwell. It is possible that this flow rate may be insufficient to prevent smoke backflow into the stairwell. For this reason, stairwell pressurization should be considered to provide an additional level of protection.

The pressure differences across the boundary of the smoke control zone depend on the supply and exhaust rates and the leakage characteristics of the building. The supply and exhaust rates of zoned smoke control systems have in the past been determined by the capacity of the HVAC system. By proper selection of the smoke zone, hot buoyant smoke can be controlled even when stairwell doors are open. If the smoke zone consists of the fire floor and the adjacent floors, then the smoke that migrates to the adjacent floors will be cooled due to dilution and heat transfer.
20.1 INTRODUCTION

Cleanrooms are used extensively in medical, research, and manufacturing facilities including aerospace, bioscience, pharmaceuticals, medicine, computing, and food processing. A cleanroom is a specially constructed and enclosed area that is environmentally controlled with respect to airborne particulates, temperature, humidity, air flow patterns, air motion, and lighting. In order to achieve all of the necessary conditions, environmental systems are specifically designed to address all of these parameters. The airborne particulate elements necessitate special filtration while temperature and humidity requirements are invariably achieved with humidifiers, reheat coils, cooling coils, and integrated control systems to efficiently control these elements.

20.2 CLASSES OF CLEANROOMS

20.2.1 Federal Standard 209

Air cleanliness levels classify cleanrooms. Federal Standard (FS) 209 was developed by the U.S. government to classify cleanrooms constructed and used by federal agencies. In December 1963 the federal government adopted FS 209 which was entitled Airborne Classes in Cleanliness and Clean Zones. FS 209 quickly became the accepted standard for cleanrooms used throughout the United States in private industry and served as the basis for the development of cleanroom standards in other countries.

Table 20-1 provides a summary of FS 209C classification requirements that remained in effect between 1963 and 1992. Table 20-1 shows the classification of cleanrooms and clean zones using FS 209 through 1992 corresponded to the maximum concentration of particles in the 0.5-micrometer column. The maximum concentration of particles in the 0.5-micrometer column corresponded to the maximum number of particles equal to or larger than 0.5 micrometer per cubic foot of air in the cleanroom or clean zone. For example, the maximum concentration of particles in the air that is equal to or greater than 0.2 micrometer for a Class 100 cleanroom is given in the 0.2-micrometer column as 750 particles per cubic foot.

Between the original adoption of FS 209 in 1963 and its discontinuation by the federal government in 2001, FS 209 underwent a number of revisions, see Table 20-2. Major changes to FS 209 occurred with FS 209C and FS 209E. FS 209C added statistical sampling of particles to determine cleanroom and clean zone classification, certification, and ongoing verification October 1987. In September 1992, FS 209E added metric (SI) particulate concentration limits specified in particles per cubic meter, changed the classification names to conform with the new metric particulate concentration limits, expanded the number of cleanroom or clean zone classifications from the original 6, see Table 20-1 to 13, see Table 20-2. Table 20-3 provides cleanroom and clean zone classification under FS 209E.

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Table 20–1 FS 209(D) Cleanroom Classifications

Source: FS 209: Airborne Particulate Cleanliness Classes in Clean Rooms and Clean Zones (Through Revision D).

¹FS 209 (D) maximum concentration of particles in air expressed in particles per cubic foot. Particle size is in micrometers and is equal to or greater than the specified size. For a Class 100 clean room, the maximum concentration of particles in the air that are equal to or greater than 0.2 micrometer in size is given in the 0.2-micrometer column as 750 particles per cubic foot.
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**Table 20-2 History of FS 209: Airborne Particulate Cleanliness Classes in Clean Rooms and Clean Zones**

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**Table 20-3 FS 209(E) Cleanroom Classifications**

Source: FS 209(E): *Airborne Particulate Cleanliness Classes in Clean Rooms and Clean Zones*

¹FS 209 (D) maximum concentration of particles in air expressed in particles per cubic meter (m³) and cubic foot (ft³). Particle size is in micrometers (µ) and is equal to or greater than the specified size. For a Class 3.5 class clean room, the maximum concentration of particles in the air that are equal to or greater than 0.2 micron in size is given in the 0.2-micrometer column as 26,500 particles per cubic meter or 750 particles per cubic foot.

²Cleanroom classifications designated as English are equivalent to the FS 209 (D) cleanroom classifications, see Table 20-1.
Table 20-3 shows that the six original FS 209 cleanroom and clean zone classifications were retained and designated as M X.5 in FS 209(E). For example, a Class 100 clean room under FS 209(D) corresponded to an M 3.5 class clean room under FS 209(E). Other whole number cleanroom classifications were designated as M X or for instance M 3. Under FS 209(E), the X in the designation corresponded approximately to the base ten logarithm of the maximum allowable number of particles in the air per cubic meter that are equal to or greater than 0.5 micrometer in size. These maximum particle concentrations are given in the 0.5-micrometer cubic meter column in Table 20-1. For example, for a concentration of 3,530 particles per cubic meter corresponds to an FS 209(E) cleanroom classification of 3.5 when rounded to the nearest tenth as follows:

FS 209 (E) Cleanroom Class = \log_{10}(3530) = 3.5

On November 29, 2001 the federal government canceled and discontinued use of FS 209E and replaced it with the International Organization for Standardization (ISO) for cleanrooms and associated controlled environments.

20.2.2 ISO 14644 Family Of Standards

ISO 14644 is entitled *Cleanrooms and Associated Controlled Environments* and is the standard currently used by the federal government and private industry internationally to classify cleanrooms and similar environments. Unlike FS 209, ISO 14644 is not a single standard but a family of standards that addresses cleanroom specification, design, equipment, construction, and operation. ISO 14644 consists of the following nine individual parts, see Table 20-4.

ISO 14644-1 classifies cleanrooms and associated controlled environments based on the following equation:

\[
C_N = 10^{N(0.1)^{2.08}}
\]

Where:

- \(C_N\) = Maximum permitted number of particles per cubic meter equal to or greater than the specified particle size, rounded to the nearest whole number.
- \(N\) = ISO class number that must be a multiple of 0.1 and 9 or less.
- \(D\) = Particle size in micrometers.

Table 20-5 provides a table showing the nine ISO cleanroom and associated controlled environment classifications based on the above formula. Table 20-3 shows the class names from the discontinued FS 209E in both English and metric (SI) units that closely approximate the ISO 14644-1 classifications.

20.3 CLEANROOM HVAC SYSTEMS

To eliminate or alleviate the likeliness that particulates may settle on surfaces within a cleanroom, air motion and airflow patterns become a design consideration. Depending on the degree of cleanliness required, it is common to have systems delivering considerably more air for this purpose than is needed to maintain temperature and humidity. The nature of cleanrooms also indicates the presence of:

- High Variable Internal Heat Gains
- Considerable Amounts of Exhaust Air
- Requirement to Maintain the Space at a Positive Pressure

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</table>

Table 20-4 ISO 14644 Family of Standards
While the heating, ventilating and air conditioning system design is critical, the HVAC system alone does not guarantee cleanroom conditions. The following practices are normally employed in addition to the HVAC system to control particulate contamination from sources other than supply air:

- Construction Finishes
- Particulate Production Control
- Personnel and Garments
- Materials and Equipment
- Personnel Entry and Egress Control

### 20.3.1 Construction Finishes

Strategies involving construction finishes that are used to maintain cleanroom conditions include the following:

- In general, all surfaces consist of smooth, monolithic, cleanable, and chip-resistant, with minimum seams, joints using inorganic sealants and no crevices or moldings to minimize off gassing.
- Floors finishes consisting of sheet vinyl, epoxy, or polyester coating with carried-up wall base.
- Walls covered with plastic, epoxy, baked enamel, stainless steel, or polyester with minimum projections.
- Ceilings consisting of plaster are covered with plastic, epoxy, or polyester coating or with plastic-finished acoustical tiles.
- Lights that are flush mounted, sealed with the tube removal preferably from within room.

### 20.3.2 Particulate Production Control

Strategies involving particulate production control that are used to maintain cleanroom conditions include the following:

- Grinding, welding, and soldering operations are shielded and exhausted.

<table>
<thead>
<tr>
<th>ISO Class</th>
<th>0.1 μm</th>
<th>0.2 μm</th>
<th>0.3 μm</th>
<th>0.5 μm</th>
<th>1.0 μm</th>
<th>5.0 μm</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>10</td>
<td>2</td>
<td>—</td>
<td>—</td>
<td>—</td>
<td>—</td>
</tr>
<tr>
<td>2</td>
<td>100</td>
<td>24</td>
<td>10</td>
<td>4</td>
<td>—</td>
<td>—</td>
</tr>
<tr>
<td>3</td>
<td>1000</td>
<td>237</td>
<td>102</td>
<td>35</td>
<td>8</td>
<td>—</td>
</tr>
<tr>
<td>4</td>
<td>10000</td>
<td>2370</td>
<td>1020</td>
<td>352</td>
<td>83</td>
<td>—</td>
</tr>
<tr>
<td>5</td>
<td>100,000</td>
<td>23,700</td>
<td>10,200</td>
<td>3520</td>
<td>832</td>
<td>29</td>
</tr>
<tr>
<td>6</td>
<td>1,000,000</td>
<td>237,000</td>
<td>102,000</td>
<td>35,200</td>
<td>8,320</td>
<td>293</td>
</tr>
<tr>
<td>7</td>
<td>—</td>
<td>—</td>
<td>—</td>
<td>352,000</td>
<td>83,200</td>
<td>2930</td>
</tr>
<tr>
<td>8</td>
<td>—</td>
<td>—</td>
<td>—</td>
<td>3,520,000</td>
<td>832,000</td>
<td>29,300</td>
</tr>
<tr>
<td>9</td>
<td>—</td>
<td>—</td>
<td>—</td>
<td>35,200,000</td>
<td>8,320,000</td>
<td>293,000</td>
</tr>
</tbody>
</table>

**Table 20−5 ISO Cleanroom and Other Associated Controlled Environments Classifications**

Source: ISO 14644-1 Classification of Air Cleanliness

1ISO 14644-1 maximum concentration of particles in air expressed in particles per cubic meter. Particle size is in micrometers and is equal to or greater than the specified size. For an ISO Class 3 clean room, the maximum concentration of particles in the air that are equal to or greater than 0.2 micrometer in size is given in the 0.2-micrometer column as 237 particles per cubic meter determined in accordance with the testing procedures specified in ISO 14644-2.

2FS 209E cleanroom classifications that closely approximate ISO 14644-1 cleanroom classifications. FS 209E classifications provided for reference and comparison only. There is no direct correlation between ISO 14644-1 and FS 209E cleanroom classifications. See Table 20-3 for FS 209E maximum concentration of particles in air by size that correspond to these classifications in both English ad metric units.
20.3.3 Personnel and Garments

Strategies involving personnel and garments that are used to maintain cleanroom conditions include the following:

- Hands and face are cleaned before entering area.
- Lotions and soap containing lanolin are used to tighten skin particles.
- Solvent contact with the skin is conducive to skin peeling and flaking, and is to be avoided.
- Wearing cosmetics and skin medications are not permitted.
- Smoking and eating are not permitted.
- Lint-free smocks, coveralls, gloves, and head and shoe covers are worn.
- Wearing of jewelry is not permitted.

20.3.4 Materials and Equipment

Strategies involving materials and equipment that are used to maintain cleanroom conditions include the following:

- Equipment and materials are cleaned before entry.
- Non-shedding paper and ballpoint pens are used. Pencils and erasers are not permitted.
- Work parts are handled with gloved hands, finger cots, tweezers, and other methods to avoid transfer of skin oils and particles.
- Static charge is often controlled at the workstation by introducing positive ions.

20.3.5 Personnel Entry and Egress Control

Strategies involving personnel entry and egress that are used to maintain cleanroom conditions include the following:

- Air locks are used to maintain pressure differentials.
- Air showers are used to remove particulates from personnel before entry into clean areas.
- Sticky mats on the floor at entrances.

20.4 CLEANROOM AIRFLOW PARAMETERS

The types of cleanroom systems can be described in terms of combinations of the following parameters:

- Airflow Patterns
- Filtration
- Bypass Or Excess Airflow

20.4.1 Airflow Patterns

The airflow patterns are determined by examining the class and whether the contaminants being controlled are toxic, explosive, odorous or harmful to the product being protected. The conventional flow system is the one used in the least critical of spaces. A conventional flow system is characterized by the use of non-aspirating ceiling outlets to prevent the collection of particles on the ceiling. The return grilles are mounted low along long sidewalls in wall plenums or return ducts, see Figure 20-1. This system does not afford any protection to operators through contamination.

A cross-flow laminar flow cleanroom distributes clean air from one side of the room to the direct opposite wall, see Figure 20-2. The air velocities across the room perpendicular to the airflow should be 50 feet per minute (fpm). The other airflow patterns are the cross flow and down flow laminar flow. Air velocities of 90 fpm are required with the air distribution through terminal High Efficiency Particulate Air (HEPA) Filters.

This continuous flushing of contaminants in one direction requires considerable planning from the operations side so that contaminants are not being flushed where they may be unwanted. Obviously, the down flow room would not have that problem where the ceiling supply and floor return would create a protective washing of occupants, contents, and product, see Figure 20-3.

All laminar flow arrangements are applicable to work stations as well as entire rooms. In combination with conventional flow systems, they can provide small areas with very high degrees of contamination control. Where extremes of control are required for particular processes such as an ISO Class 4 cleanroom with localized ISO Class 2 laminar flow benches, the laminar workstation is often the only practical means for meet-
FIGURE 20–1 CONVENTIONAL FLOW CLEANROOM

FIGURE 20–2 CROSS FLOW LAMINAR FLOW CLEANROOM
ing the operational criteria. Figure 20-4 illustrates a laminar flow workstation arrangement.

20.4.2 Filtration

The methods of filtration include the use of different types of filtration depending on the activity and required maximum particulate level in the cleanroom.

20.4.3 Bypass Or Excess Airflow

When the airflow requirement exceeds the conditioned air requirement, the need for bypass air becomes necessary. Providing for bypass airflow can be achieved in one of the following ways:

- The central or primary air supply fan circulates air through the filtration system. A secondary HVAC unit for the heating, air conditioning, and humidification of the space draws air from the return air system and outside air make up. This system then supplies conditioned air to the suction side of central fan, see Figure 20-5.

- The primary HVAC unit supplies the required airflow for conditioning the space from the return air plenums plus outside air make up. A secondary bypass fan draws the additional amount of air required for the cleanroom from the return air plenums. Both airstreams join on the supply airside at a point upstream from the HEPA filtration system, see Figure 20-6.

A series of packaged units consisting of fans and HEPA filters circulate the required cleanroom air quantities. A secondary HVAC unit supplies the conditioning and makeup outside air, see Figure 20-7.
FIGURE 20–6 CONVENTIONAL CLEANROOM WITH BYPASS FAN

FIGURE 20–7 CONVENTIONAL CLEANROOM WITH PACKAGED FAN/HEPA UNITS
### Table 20–6 Air Pressure Relationship

<table>
<thead>
<tr>
<th>Application</th>
<th>Pressure Differential</th>
</tr>
</thead>
<tbody>
<tr>
<td>General</td>
<td>0.05 in. of water higher than surroundings</td>
</tr>
<tr>
<td>Between clean room and uncontaminated section</td>
<td>0.05 in. of water, minimum</td>
</tr>
<tr>
<td>Between uncontaminated and semicontaminated section</td>
<td>0.05 in. of water</td>
</tr>
<tr>
<td>Between semicontaminated section and locker area</td>
<td>0.01 in. of water</td>
</tr>
</tbody>
</table>

### 20.5 DESIGN AND PERFORMANCE CONSIDERATIONS FOR CLEANROOMS

In the design and construction of cleanrooms, it is important that an integrated project design approach be taken. This approach involves both the design team and the construction team including key specialty contractors such as the HVAC contractor. In cleanroom design, walls sometimes become ducts, rooms become pressure vessels, and ceilings often become diffusers. Figure 20–8 provides a list of HVAC design considerations and should be reviewed when designing a cleanroom.

### 20.6 CONTROL SYSTEMS

#### 20.6.1 Pressurization

A cleanroom facility may consist of multiple rooms with different requirements for contamination control. All rooms in a clean facility should be maintained at static pressures sufficiently higher than atmospheric to prevent infiltration by wind or other effects. Differential pressures should be maintained between the rooms sufficient to assure airflow outward progressively from the cleanest spaces to the least clean during normal operation and during periods of temporary upsets in the air balance, as when a door connecting two rooms is suddenly opened per Table 20–6.

Static pressure regulators can maintain desired room pressures by operating dampers, fan speed via a variable frequency drive, or a combination of these methods to vary the ratio of supply to return or exhaust air. To provide control over room pressures, airflow variations should be minimized. Exhaust airflow from rooms through hoods should be maintained constant by continuous hood operation or appropriate bypasses. In many systems, door openings to the outside are protected by air locks, and provision is made for offsetting the pressure loss variations across filters as the dust loading increases. Use of sliding doors instead of hinged provides smoother pressure fluctuations and less turbulence when entering or leaving the room.

#### 20.6.2 Temperature

Temperature control is required to provide stable conditions for materials, instruments and for personnel comfort. Heat loads from lighting are stable and personnel loads vary. The heat generated by process operations such as soldering, welding, heat-treating, and heated pressure vessels can be high and variable.

However, the large air quantities required for contamination control can offset internal heat gains at an acceptable rate of response at the thermostat. In areas where heat producing equipment is concentrated, supply air patterns should be analyzed to determine the acceptability of resulting temperature gradients, see Table 20–7. The dynamics of high volume airflow can impact temperature control.

<table>
<thead>
<tr>
<th>Capacity Range</th>
<th>Temperature</th>
<th>Humidity, Percent</th>
</tr>
</thead>
<tbody>
<tr>
<td>Control Point</td>
<td>67°F to 77°F</td>
<td>40 to 55%</td>
</tr>
<tr>
<td>Control Tolerance</td>
<td>±2°F</td>
<td>±5 %</td>
</tr>
<tr>
<td>General Applications</td>
<td>±0.5°F</td>
<td>±2 %</td>
</tr>
<tr>
<td>Critical Applications</td>
<td>2.5°F to 4°F</td>
<td></td>
</tr>
<tr>
<td>Response Rate</td>
<td>change per hour</td>
<td></td>
</tr>
</tbody>
</table>

### Table 20–7 Cleanroom Temperature and Humidity
• Class Of Room

• Temperature And Humidity Requirements

• Availability Of Electrical Utilities

• Availability Of 24 Hour, Year Round Heating Source

• Availability Of 24 Hour, Year Round Cooling Source

• Amount Of Required Exhaust

• Architectural Integrity Of Structure For Pressure Maintenance

• Space Availability Within Architectural Boundaries For Possible Wall And/Or Ceiling Plenums, Location Of Mechanical Equipment And Clearance For Major Duct Systems

• Ability Of Ceiling To Support Additional Loading

• Careful Analysis Of Possible Static Pressures Duct System May Experience

• Review Of Insulation Requirements

• Analysis Of Fan Performance Versus Variable System Static Pressures

• Possible Static Pressure Control Requirements

• Providing For Adequate Means For Air Balancing

• Need For Backup Equipment

FIGURE 20–8 CLEANROOM MECHANICAL DESIGN CONSIDERATIONS
20.6.3 Humidity

Humidity control is necessary to maintain proper conditions for certain production processes:

- Prevent Corrosion
- Prevent Condensation On Work Surfaces
- Eliminate Static Electricity
- Provide Personnel Comfort
- Provide Stable Environment For Work Processes

Table 20-7 gives recommended humidity conditions.

Corrosion of precisely manufactured surfaces including bearings, electrical contact surfaces, ball bearing raceways, and miniature-gear trains can occur when the relative humidity exceeds 50 percent. At a relative humidity below 40 percent, static charges can form that attract dust particles and can become airborne in objectionable concentrations.

In cleanrooms, humidity control is affected more by external influences such as weather changes than by variations in moisture generated within the space. If processes involving evaporation must take place within the cleanroom, they should be confined within ventilated enclosures. Some precision manufacturing processes such as vacuum tube manufacture require the relative humidity to be below 35 percent. Under these low humidity conditions, precautions must be taken to control and protect the process from the effects of static electricity.

20.6.4 Makeup Air

Ventilation and makeup air must also be considered for both room and unitary HVAC equipment application. In the use of suction exhaust benches, where the exhaust air is discharged out of the area through ductwork, the makeup air may be supplied from within the area or may be ducted in. If this supply air is ducted from an external source, it should be prefiltered and conditioned. If the air is supplied from within the area, it should be verified that the space air conditions will not be adversely affected.

20.6.5 Exhaust Air

Cleanroom activities often require the use of exhaust air equipment. Workstations emitting toxic fumes, ovens emitting heated fumes, small machinery operations, and similar operations need exhaust air capability. Workstations often use the cleanroom air as their source of makeup air. As a result, the cleanroom makeup air volume must be increased to equal the volume of exhaust air. The exhaust air ducts for the equipment should be carefully routed to maintain laminar airflow within the cleanroom.

20.6.6 Dilution Air

Many cleanroom activities require the use of solvents, vapors, gases, and the like, which are normally harmless to the operating personnel if concentrations are maintained below certain levels. For example, solvents and solder flux often used in manufacturing electronic components are not toxic in low concentrations; therefore, it is not necessary to completely isolate these substances from the cleanroom atmosphere.

For many airborne substances, requirements to avoid excessive worker exposure are generally established by the American Conference of Governmental Industrial Hygienists (ACGIH). The U.S. Occupational Safety and Health Administration (OSHA) set specific standards for the allowable concentration of airborne substances. These limits are based on working experience, laboratory research, and medical data, and are subject to constant revision. Reference to the latest standards available should be made when evaluating a cleanroom exposure.

20.6.7 Noise and Vibration

Noise is one of the most difficult variables to control within contamination control equipment, due to the high volumes of air necessary to provide the cleanliness levels required. For normal applications of laminar flow equipment, the noise level is designed to be below 65 decibels (0.0002 microbar) as measured on the A Scale of calibrated noise-level meter. In applications where extreme quiet is of utmost importance, the noise levels of this equipment may be reduced to about 50 decibels (A Scale) (NC).

In normal applications of contamination control equipment, vibration displacement levels do not need to be dampened below 0.5 μm (20 micro inches) in the 1 to 50 Hz range. Where electronic microscopes or similar ultra-sensitive instrumentation are to be used, smaller deflections or attention to different frequency ranges might be required.

20.7 HVAC DUCTWORK

If air-conditioning equipment can be located in the space above the recirculating plenum, conditioned air
duct lengths are minimized. Conditioned air supply ducts should enter the plenum at a minimum of 10 feet upstream of the primary blowers to assure efficient blower operation. A manually adjustable, opposed-blade damper should be installed at each discharge outlet. This method of discharging conditioned air allows each primary blower to handle an equal volume of conditioned and primary air, thus assuring a more uniform room temperature and humidity gradient. The supply duct may be installed within the plenum, or if insufficient plenum height is available, the duct may be routed above the plenum with each discharge outlet entering the plenum. In this case, the discharge duct should turn 90 degrees after entering the plenum for best air distribution.

20.8 TYPICAL CLEANROOM SYSTEMS

20.8.1 ISO Class 4 Cleanrooms

The system in Figure 20-9 is used for ISO Class 4 cleanrooms that are used for manufacturing processes such as the manufacture of very large scale integrated (VLSI) circuits and very high speed integrated (VHSI) circuits. An ISO Class 4 cleanroom environment has a 0.5 micrometers and greater about one-tenth that permitted in an ISO Class 5 cleanroom, see Table 20-5.

The primary air is supplied through a pressurized plenum and filtered through ultra low penetration air filters. A raised floor or grating provides laminar airflow within the cleanroom area.

The primary air is supplied to the pressurized plenum by a centrifugal fan driven by a variable frequency drive. The large volume of primary air (540 air changes per hour for a cleanroom of 10 ft (3 m) height and 90 f/m (27 m/s) air velocity through the ULPA filter) requires the use of vane axial fans that have efficiencies exceeding 80 percent.

The secondary air is provided by an air handling unit consisting of a mixing box, coil section with preheating coil and face and by-pass dampers and a non by-pass cooling coil.

The preheating coil and the cooling coil in the make-up air-handling unit maintains a fixed leaving air temperature year-round. The pre-conditioned outside air and the return air from the cleanroom to the secondary air-handling unit are mixed in the mixing box of the secondary air-handling unit.

The cooling coil in the secondary air-handling unit maintains the design temperature of the cleanroom. If the room relative humidity increases above the setpoint, the cooling coil provides more dehumidification, and simultaneously the heating coil will be in the reheat mode to maintain design temperature. When the relative humidity decreases below the setpoint, a humidifier installed in the secondary air handling unit discharge duct provides humidification.

The make-up air-handling unit provides room pressurization. A static pressure sensor installed in the room modulates the supply fan speed in order to maintain the required positive pressure. The primary airflow is reduced to conserve energy when the cleanroom is unoccupied.

The HVAC equipment room must be isolated from the cleanroom by an expansion joint separation for the maximum reduction of vibrations that are critical for the process equipment performance.

20.8.2 ISO Class 5 Cleanrooms

The system in Figure 20-10 is used for ISO Class 5 clean rooms. The primary air is provided by a primary air-handling unit and distributed through medium pressure ductwork. The primary air is supplied to the room by individually ducted final HEPA filters. The entire cleanroom ceiling area is covered with final HEPA filters installed in the ceiling system.

The primary air-handling unit consists of a mixing box, filter section with rigid filters, and supply fan section with a centrifugal fan.

If applicable, another make-up air-handling unit is used to supply both outside air required for room pressurization and makeup air for any process equipment exhaust. This make-up air-handling unit typically consists of the following components:

- An outside air intake, filter section with two-inch prefilters and 95 percent NBS bag filters.
- A coil section with preheating coil with face and by-pass dampers and a non by-pass cooling coil.
- A supply fan section with a centrifugal fan driven by a variable frequency drive.

The make-up air-handling unit provides room pressurization. A static pressure sensor installed in the room modulates the supply fan speed in order to maintain the required positive pressure. The primary airflow is reduced to conserve energy when the cleanroom is unoccupied.

The HVAC equipment room must be isolated from the cleanroom by an expansion joint separation for the maximum reduction of vibrations that are critical for the process equipment performance.
FIGURE 20-9 TYPICAL ISO CLASS 4 CLEANROOM
FIGURE 20–10 TYPICAL ISO CLASS 5 CLEANROOM
The air flows through a chase area ceiling into the plenum and is ducted back to the mixing boxes of the primary and the secondary air-handling units. The chase areas are used for returning the air. Branch exhaust ducts for workstation exhaust are located in these chases and connected to the main exhaust ductwork located in the basement area. Since the return air chases are used for process equipment connections and services, the return air is pre-filtered.

Either the cooling or the heating coil in the secondary air-handling unit maintains a fixed leaving air temperature on the discharge side of the primary air supply fan. An electric or hot water reheat coil that is mounted in the supply ductwork that serves the zone maintains the design temperature in each room. If the relative humidity in any zone increases above the setpoint, the secondary air-handling unit cooling coil provides more dehumidification while the heating coil is simultaneously in the reheat mode to maintain design temperature. When the relative humidity decreases below the setpoint, a duct-mounted humidifier provides humidification. To conserve energy, the supply airflow to the rooms is reduced by adjusting the speed of the fan when the cleanrooms it serves are unoccupied.

### 20.8.3 ISO Class 7 Cleanrooms

The system in Figure 20-11 is used for ISO Class 7 cleanrooms with local ISO Class 5 areas. The air-handling unit consists of the following components:

- A mixing box.
- Filter section with two-inch prefilters and 95 percent NBS bag filters. The two-inch prefilters are used only for the start-up of the system.
- Coil section with cooling coil and an opposed blade damper installed above the cooling heating coils.
- A supply fan section with internally isolated airfoil centrifugal fan driven by a variable frequency drive.

The air is distributed through medium pressure ductwork and supplied to the space by individually ducted final HEPA filters. The air from the room is transferred through low sidewall registers into return air chases. From the ceiling return plenum the air is ducted to the mixing box of the air-handling unit. The corridor adjacent to the cleanroom has ISO Class 7 cleanliness level. The air is supplied through a duct-mounted in-line HEPA filter and ceiling diffusers.

Outside air is provided by a central make-up air-handling unit that is connected to several air-handling units. The pre-conditioned outside air and the return air from the cleanrooms are mixed in the air-handling unit mixing box. A part of the mixed air is cooled and dehumidified by the cooling coil and the other part is unaltered, passing through the space above the cooling coil. The quantity of the air passing through the cooling coil is constant and depends on the room sensible and latent loads.

### 20.9 CLEANROOM TESTING

Two basic types of testing used to evaluate cleanroom system performance are as follows:

- Initial Performance Testing
- Ongoing Operational Monitoring

The initial performance testing is normally conducted before occupancy of the facility. However, a cleanroom facility cannot be fully evaluated until it has performed under full occupancy and the process to be performed within it is operational. Therefore, the methods for conducting initial performance tests and ongoing operational monitoring are similar. All testing should be carried out in accordance with the standards to which the clean room was designed.

Sources for contamination can be both external and internal. For both conventional flow and laminar flow cleanrooms, the major source for external contamination is through the primary air loop. Therefore, similar leak testing of the HEPA filter banks may be conducted. HEPA filters should be tested both before installation and while in place. Tests for pinhole leaks should be performed in accordance with manufacturer recommendations and applicable standards.

Once it has been determined that the filter bank is properly sealed, other sources of external contamination should be tested. In laminar flow facilities, this testing consists of testing physical barriers such as walls, partitions, windows, doors, and the like. In addition to testing physical barriers, all ductwork downstream of the filter bank should be tested in conventional flow rooms. Again, all testing should be conducted in accordance with applicable standards and manufacturer recommendations.

The processes that are carried out inside the cleanroom may generate internal sources of contamination. To properly evaluate the performance of the cleanroom in
FIGURE 20–11 TYPICAL ISO CLASS 7 CLEANROOM
controlling internal contamination, tests must be conducted at critical areas within the cleanroom. The contamination level within a conventional flow cleanroom reaches a plateau and the level of contamination tends to equalize throughout while in a laminar flow cleanroom the contamination typically stratifies. It is therefore critical to properly sample work areas within a laminar flow facility. This sampling should be done in accordance with applicable standards and manufacturer recommendations.

If the air volume and room pressures are critical, make sure that there is an accurate means available to measure airflow. The preferred method is a duct traverse but ducts are not always accessible and outlet selection can be critical to allow for proper airflow measurement. Sometimes the high velocities and air volumes do not allow for accurate airflow measurement. Also, access to the HVAC equipment is critical for testing, adjusting, and balancing (TAB) and maintenance work.

Other testing such as temperature and humidity gradients, lighting levels, sound and vibration levels, are also often required in evaluating the performance of a cleanroom. These tests are similar to those conducted for any laboratory facility and are not covered in this chapter.
21.1 INTRODUCTION

This chapter addresses exhaust and make up air systems for research laboratories. Research laboratories are spaces arranged, equipped, and environmentally controlled for experimental work. These facilities may consist of one room, several rooms, or an entire building devoted to particular research, development, or testing activities. Such facilities are employed by industrial, educational, and governmental agencies for basic and applied research.

21.2 LABORATORY ENVIRONMENTAL REQUIREMENTS

Environmental requirements for a research laboratory include the following:

- Regulation of Temperature
- Humidity
- Pressure
- Air Motion
- Air Cleanliness
- Light, Sound, And Vibration

21.3 SUPPLY AIR SYSTEMS

21.3.1 Supply Air System Arrangement

The minimum unit served by the air supply system is the laboratory module. It is the smallest repetitive unit in which research is performed. Consequently, air supply systems are designed with the laboratory module as the basic unit of space for which air is provided and temperature and humidity control are maintained. An air supply system for a laboratory may use one system or a combination of systems to satisfy the laboratory thermal requirements, directional airflow, and exhaust air requirements. The factors of room size, physical arrangement, occupancy, type of exhaust system, and economic constraints each contribute to determining the type of supply air system.

A variety of air supply system types and arrangements may be utilized for laboratory conditioning. Systems may use high-, medium-, or low-pressure air distribution. Single-duct, dual-duct, and terminal reheat systems may be selected. Air supply may be introduced into the laboratory through ceiling diffusers, sidewall grilles, perforated ceiling panels, and outlets under the windows. The important factor is whether the system will satisfy the safe operating conditions of a laboratory.

Due to equipment heat loads, laboratory cooling and heating loads are characteristically highly variable. For this reason, individual laboratory rooms should always have separate thermostats. Additionally, laboratories that contain harmful substances should be balanced to assure directional airflow into the laboratory from adjacent clean spaces and corridors.

Exhaust requirements for laboratory facilities for the conveyance of contaminants to the atmosphere require the conditioning of large quantities of outdoor air. Selection of outdoor design conditions, therefore, materially affects the size and cost of refrigeration and heating facilities. Serious consideration should be given to using energy recovery devices.

21.3.2 Filtration

The filtration necessary for supply air depends on the type of activity being conducted in the laboratory. High Efficiency Particulate Air (HEPA) filters should be provided for special spaces where research materials or animals are particularly susceptible to contamination from external sources. In many instances, these critical applications can be accomplished within biological safety cabinets that are HEPA filtered rather than using HEPA filtration for the entire room.

21.3.3 Air Distribution

Air supply to a laboratory space must be arranged to minimize temperature gradients and air turbulence, especially near the face of the laboratory fume hoods and biological safety cabinets. It is very important that supply air outlets not impact the face velocity of fume hoods. The large quantities of supply air can best be introduced through perforated plate air outlets or diffusers that are specially designed for large air quantities. Supply air should not be introduced in the immediate vicinity of a laboratory fume hood or safety cabinet in order to avoid affecting the performance of these units. It should be emphasized that cross currents can negatively impact containment. Also, the air supply to a laboratory space should not discharge on a fire detector since this will retard the activation of the system.

Sidewall grilles are applicable to laboratory air supply if their terminal velocity, location, and air distribution patterns are analyzed and are consistent with the ceiling system’s space requirements. Special care is required to avoid discharging air across the face of a laboratory fume hood or biological safety cabinet.
Air outlets discharging through the windowsill may be either at the terminal of a supply duct or at the discharge of a terminal reheat induction unit. In either case, the air should be discharged up along the window. Supply grilles in the floor should be avoided because air discharge from these grilles entrain dirt from the floor and distribute it into the room. In some specialized applications, they can be used if the air must be provided in the immediate vicinity of equipment located in the center of an open space. Provisions for cleanout and modification to elevate the grille face about the floor are necessary.

Some general air distribution guidelines are as follows:

- The terminal velocity of supply air jets that are near hoods is at least as important as hood face velocity in the range of 50 to 150 fpm (0.25 to 0.76 m/s) face velocity.
- The terminal throw velocity of supply air jets (near hoods) should be less than the hood face velocity, preferably no more than one-half to two-thirds of the face velocity. Such terminal throw velocities are less than conventional practice for room air supply.
- Perforated ceiling panels provide a better supply system than grilles or ceiling diffusers. It is because the system design criteria is simpler, easier to apply and precise adjustment of fixtures is not required. Ceiling panels also permit a greater concentration of hoods than do wall grilles or ceiling diffusers.
- Wall grilles or registers should have double deflection louvers set for maximum deflection. The terminal velocity near hoods should be less than one-half of the face velocity of the hood.
- If the wall grilles are located on the wall adjacent to the hood, the supply air jet should be above the top of the hood face opening. If this can be done, grilles on the adjacent wall result in less spillage than grilles located on the opposite wall for equal terminal throw velocities.
- The terminal throw velocity from ceiling diffusers at the hood face should be less than the hood face velocity.
- Diffusers should be located away from the immediate front of the hood face. A larger number of smaller diffusers would be an advantage if the necessary low terminal velocity can be maintained.
- Blocking the quadrant of the ceiling diffuser blowing toward the hood face would result in less spillage.
- Perforated ceiling panels should be sized so that the panel velocity is less than the hood face velocity, preferably no more than two-thirds of the hood face velocity.
- Perforated ceiling panels should be located so that approximately one-third or more of the panel area is remote, that is more than 4 ft (1.22 m), from the hood.

21.3.4 Unitary Systems

The most adaptable form of air supply system consists of a separate air-handling unit for each laboratory space. Each unit is made up of a fan and air treatment apparatus with a capacity equal to that required to maintain space temperature and to balance the exhaust air requirements. The unitary system typically contains a cooling coil, heating coil, humidifier, and filter, and is serviced with electricity, chilled water, and steam or hot water.

In biomedical laboratories, the installation of unitary systems is discouraged where the cooling coil with condensate drip pan and roughing filter would be located within the laboratory. The moisture associated with the coil and drip pan and the dust collecting areas of the unit contribute to harborage, growth, and dissemination of molds and other organisms commonly found in the environment. These organisms can serve as undesirable contaminants to experimental cultures, specimens, and biological products.

21.3.5 Central Systems

The simplest form of a central system that can be used in a laboratory that is subject to variations in internal heat gain is a constant volume, terminal reheat system where the supply air is conditioned as follows:

- Dry-bulb temperature that will satisfy the maximum sensible heat load in any space.
- Dewpoint satisfactory to maintain the room humidity within an acceptable range.

Reheat coils in the branch duct serving each space can thermostatically control variations in heat gain in individual laboratory modules. This system is economical based on the following factors:
21.3 HVAC Systems Applications

- Close humidity control is not critical.
- Internal heat gains are moderate and fairly constant within a space and do not vary greatly between spaces.
- The exhaust air quantities are constant and in balance with the supply air necessary to maintain space conditions.

Hours of occupancy and operating conditions for each laboratory space should be approximately the same, since the entire central system must be in operation even if any only one space is being used. Many laboratories operate 24 hours a day or on a similar work schedule so this requirement does not always impose a hardship. Central systems with supplementary conditioning or central systems with auxiliary air supply should be considered when both of the following conditions are present:

- Heat gains are high and subject to variation.
- Exhaust air quantities are greater than supply air requirements for cooling.

21.4 EXHAUST AIR SYSTEMS

21.4.1 Exhaust Air System Classifications

The basis for classifying laboratory exhaust systems are:

- Laboratory fume hood and biological safety cabinet characteristics.
- Method of system operation and control

The classifications of lab exhaust systems are:

- Constant Air Volume
- Variable Air Volume

These classifications can be further broken down into the following two types of systems based on the arrangement of the major system components such as the fans, plenums, or duct mains and branches:

- Individual
- Central

21.4.1.1 Constant Air Volume Systems

A constant air volume system exhausts a fixed air quantity from each fume hood, safety cabinet, or room module. Since a constant air volume system will handle the same exhaust air quantity for any condition, the total number of laboratory fume hoods and safety cabinets to be installed throughout the facility must be limited by the capacities of the exhaust air and supply air systems. This system is very dependable and often selected for biological hazard (biohazard) contaminant laboratories.

The constant air volume system is flexible with respect to location of hoods, but may incur high ownership and operating costs if energy recovery is not utilized because of the large air volumes handled. These costs may impose a practical limitation on the total number of hoods and safety cabinets that can be installed in the building.

21.4.1.2 Variable Air Volume Systems

In many laboratories, all of the hoods and safety cabinets are seldom needed at the same time. Therefore, a laboratory operation that permits the application of a diversity factor for usage will allow the sizing of the exhaust air system at a capacity less than that required for the full operation of all units. Operating economies can be achieved by reducing the airflow during periods when some of the hoods and safety cabinets are not in use and the exhaust air system is operated at less than full capacity.

A reduction in exhaust air volume coupled with constant hood face velocity when the hood face opening is partially closed may be achieved by using a velocity-controlled hood. A sensing element responds to changes in hood face velocity and operates a motorized damper in the exhaust duct to maintain the face velocity within the desired range. When an individual exhaust air fan serves the hood, the operation of the damper will usually be sufficient to reduce the fan capacity but a static pressure regulator in the duct may supplement it. In large central systems, a modulating branch damper affords a satisfactory means for branch ducts but static pressure regulators in the exhaust plenums that control fan inlet vanes or discharge dampers must supplement system volume regulation.

Air valves that provide proper directional airflow while maintaining the required pressurization of spaces can also be used in laboratory VAV systems.

Variable air volume systems can provide initial and operating economies by enabling the shutdown of inactive fume hoods and safety cabinets. Complete exhaust system shutdown in a multi-hood facility is not desirable since cold air and contaminants can enter the laboratory through the exhaust system. More freedom
in the installation of the hoods and safety cabinets is possible since the total number of units that may be connected is not entirely dependent on the capacity of the exhaust system.

A particular operating problem with the variable air volume system is the regulation of the total simultaneous operating usage to match design usage factors. If the collective area of operating hood and safety cabinet openings at any one time exceeds design opening diversity values, the proper face velocity requirements may not be achieved and laboratory personnel could be endangered. All laboratory fume hoods and safety cabinets on a variable air volume system should be equipped with visual and audible alarms to warn the laboratory workers of unsafe airflows.

If in case the total usage is less than design values then bypass devices will probably be required on hoods in order to maintain supply air rates, provide adequate thermal capacity, and ensure proper air balance and flow patterns. The decision to select a variable volume exhaust system should not be made without the understanding and approval of the research staff and local safety officials.

21.4.1.3 Individual Exhaust Air Systems

Individual exhaust air systems utilize a separate exhaust connection, exhaust air fan, and discharge duct for each laboratory fume hood or biological safety cabinet. This arrangement is extremely flexible because the exhaust for the laboratory fume hood or biological safety cabinet served by the individual exhaust system does not directly affect the operation of any other area of the building. The individual system permits selective operation of individual hoods and safety cabinets merely by starting or stopping the fan motor.

The recommended operation is for exhaust air fans to be on at all times and to be electrically interlocked so that if any critical exhaust air fan is shut down the supply air fans will also shut down. Although more fans are used than for central systems, the overall space requirements are usually less for individual systems because of the small direct duct connection. However, the use of more fans will usually increase both the initial cost as well as ongoing operation and maintenance (O&M) costs.

Many research laboratories require directional airflow from the corridor into the laboratory for hazard containment. The shutdown of individual exhaust air systems will upset the proper directional airflow and may cause potentially hazardous contaminants and odors to flow into the corridor and adjacent rooms. If such a system is considered, appropriate precautions to reverse airflow, such as air locks, should be provided.

21.4.1.4 Central Exhaust Air Systems

Central exhaust air systems consist of one fan, a common suction plenum, and branch connections to multiple exhaust terminals. Central exhaust air systems generally have a lower initial cost than individual exhaust air systems, cost less to operate and maintain, permit low cost standby exhaust air fan provisions, and are applicable to remote high stack discharge requirements. Central systems are more difficult to balance and may have difficulties with parallel fan operation. The effects of the mixing of effluents from different research operations must be evaluated when central exhaust systems are considered.

A central exhaust air system works best when exhausting similar types of units such as laboratory fume hoods. The exhausting of laboratory fume hoods, biological safety cabinets, and special filtered units with one central system is difficult because pressure losses in various equipments differ.

21.4.2 Exhaust Air Fans and Ductwork

Exhaust air fans that handle contaminants should always be located outside of occupied building areas and be located as close as possible to the point of discharge to the atmosphere. The ductwork connected to the discharge side of the fan should be airtight and routed outside the building. It is possible to contaminate a building by pressurized exhaust air duct leakage. The fan discharge should be directly connected to the vertical discharge stack without cross connections to other exhaust systems. An exhaust fan should be supplied with a vertical stack that extends above any obstructions on the roof and should provide a high terminal velocity to discharge the exhaust air upward well above the building envelope. Care must also be taken to discharge the exhaust air away from any present or future air intakes.

21.4.3 Materials and Construction

The selection of materials and the construction of exhaust ductwork and fans depend on the following:

- Nature of the hood effluents.
- Ambient temperature.
- Lengths and arrangement of duct runs.
- Method of hood fan operation.
Flame and smoke spread rating.

Duct velocities and pressures.

Both present and future laboratory effluents should be considered when selecting duct materials and construction. Laboratory fume hood effluents can vary in temperature and are often classified generically as organic or inorganic chemical gases, vapors, fumes, or smokes, and qualitatively as acids, alkalis, solvents, and oils. Exhaust system ducts, fans, and coatings are subject to attack from such effluents that can cause:

- Corrosion and the destruction of unprotected metal by chemical or electrochemical.
- Dissolving of coatings and plastics often caused by solvent and oil effluents.
- Melting which can occur with certain plastics and coatings at elevated hood operating temperatures.

Ambient temperature of the space in which ductwork and fans are located affects the condensation of vapors in the exhaust system. Condensation can also cause the corrosion of metals with or without the presence of chemicals.

Ductwork is less subject to attack when the runs are short and direct and when the air is maintained at reasonably high velocities. The longer the duct, the longer the time of exposure to effluents and the greater the degree of condensation. Horizontal runs provide surfaces where moisture can remain longer than on vertical surfaces. If condensation is likely, sloped ductwork and condensate drains should be provided. However, it should be noted that the condensate drains can accumulate hazardous materials and must be dealt with accordingly.

Fan operation may be continuous or intermittent. Intermittent fan operation allows longer periods of wetness because of condensation than continuous fan operation would allow. Additionally, flame and smoke spread rating requirements as specified by codes, standards, and insurance requirements need to be considered and addressed.

Procedures and recommendations for the selection of materials and construction methods are as follows:

1. Determine the types of effluents and possible combinations of effluents that will be generated in the hood and handled by the exhaust system. Consider both present and future operations.

2. Classify the effluents as either organic or inorganic and determine whether they occur in gaseous, vapor, or particulate form. Also, classify any decontamination materials that may be used.

3. Determine the concentration of the reagents that will be used and the temperature of the effluents at the hood exhaust port. In research laboratories, this determination may be difficult.

4. Estimate the highest probable dewpoint of the effluents.

5. Determine the ambient temperature of the spaces in which the duct and exhaust blowers will be located. Consider if surface temperatures that are in the direct air stream from air outlets should be lower.

6. Consider the length and arrangement of duct runs and determine how they may affect the periods of exposure to fumes and the degree of condensation that may occur.

7. Consider the effects of intermittent versus continuous fan operation. If intermittent operation is desired, provide a time delay (about one hour) for the drying of wet surfaces before fan shutdown. Intermittent operation can easily unbalance airflows in the laboratory and cause unsafe conditions; continuous operation during working hours is the most prudent mode of operation.

8. Determine whether insulation, watertight construction, slope, and condensate drains will be required.

9. Materials and construction methods that are suitable for the application considering resistance to attack, weight, flame and smoke spread rating, and cost. Information on material properties is available in standard references and from manufacturers. Typical materials used in chemical fume exhaust duct systems and their characteristics are as follows:

a. **Glazed Tile.** Resistant to practically all corrosive agents except hydrofluoric acid. Heavy in weight, limited to round sections, joint sealants subject to attack, and considerable space required for directional changes.

b. **Cementatious Material.** Highly resistant porous surface requires an internal impervious coating to prevent retention of potentially flammable materials. They are limited to being used in round sections because of the difficulty in sealing joints, constructing directional changes or transitions, bracing and
supporting rectangular sections. Joint sealants are subject to attack.

c. **Galvanized Iron.** Subject to acid and alkali attack with cut edges under wet conditions. Galvanized iron is easily formed.

d. **Stainless Steel.** Subject to acid and chloride compound attack varying with the chromium and nickel content of the alloy. The higher the content, the higher the resistance. Stainless steels range from the commercial 200 to the 400 alloy series with ascending chromium and nickel content to proprietary alloys with custom chromium and nickel composition.

e. **Asphaltum-Coated Steel.** Resistant to acids but subject to solvent and oil attack. High flame and smoke spread rating. Base metal is vulnerable when exposed by coating imperfections and cut edges.

f. **Epoxy-Coated Steel.** Use of epoxy phenolic resin coatings on mild black steel can be selected for particular characteristics and applications. These coatings have been successfully applied for both specific and general usage but no one compound is inert or resistive to all effluents. Requires sandblast surface preparation for shop applied coating and field touchup of coating imperfections.

g. **Fibrous Glass.** Can be used when additional glaze coats are provided particularly for acid applications including hydrofluoric.

h. **Plastic Materials.** Have particular resistance properties to specific corrosive effluents. Their limitations are in physical strength, flame spread rating, and heat distortion.

10. Select fans constructed of the same materials as the ductwork or fans constructed of mild steel and with a suitable coating.

11. Provide outboard bearings, shaft seals, access doors, and multiple 200% rated belts for hood exhaust fans.

12. Design the ductwork and select the fan with consideration of potential fire and explosive hazard.

13. For some systems such as perchloric acid hoods, provide wash down facilities.

14. Develop duct layouts so that ducts may be easily inspected, decontaminated, and replaced, if necessary.

### 21.4.4 Exhaust Air Filtration

Depending on the hazard level associated with the laboratory operation and the degree of physical containment desired, filtration facilities for exhaust systems may be required. The hazardous or obnoxious pollutants to be removed from the exhaust air may be particulate, gaseous, or a combination of both.

The filter assembly may include a prefilter for coarse particle separation and a filter enclosure arranged for ready access and easy transfer of the contaminated filter to a disposal enclosure. Manufactured filter enclosures which feature bag-in/bag-out filter changing should be considered for hazardous exhaust situations. A procedure for the testing of the filter system integrity and suitable test openings are also necessary. A damper is often added for balancing airflow as the HEPA filter's resistance changes.

For convenient handling, replacement, and disposal with minimum hazard to personnel, the filter should be:

- Located immediately outside the laboratory area unless it is an integral part of a safety cabinet or hood.
- Ahead of the exhaust fan.
- Installed in adequate space that provides free, unobstructed access.
- Positioned at a convenient working height.

Some installations will require shutoff dampers and hardware for filter decontamination in the ductwork. The filter should be located on the suction side of the exhaust fan and as close as practical to the laboratory to minimize the length of contaminated duct.

Wet collectors or adsorption systems such as activated charcoal are often satisfactory for the removal of gas-phase toxic or odorous pollutants from exhaust air. In designing the installation of these devices, consideration must be given to the safety of maintenance and service personnel, the potential concentrations of exhaust materials, and the frequency of filter replacement.

### 21.5 LABORATORY FUME HOODS

#### 21.5.1 Laboratory Fume Hood Description

A laboratory fume hood is a ventilated enclosed workspace intended to capture, contain, and exhaust fumes, vapors, and particulate matter generated inside the en-
closure. It consists basically of side, back, and top enclosure panels, a work surface or counter top, an access opening called the face, a sash, and an exhaust plenum equipped with a baffle system for the regulation of airflow distribution.

The work opening is equipped with operable glass sash or sashes for observation and shielding purposes. A sash may be:

- Vertically Operable
- Horizontally Operable
- Vertically And Horizontally Operable

The latter provides maximum access for setting up apparatus, allows energy conservation due to the smaller face opening, and provides movable, protective shielding. The horizontally operable sash-type fume hood provides similar energy savings. Figure 21-1 illustrates the basic elements of a general purpose bench-type fume hood. Figures 21-2 and 21-3 show the basic features of a bypass-type fume hood and the auxiliary air fume hood, respectively.

21.5.2 Hood Performance

Containment of contaminants is based on the principle that a flow of air entering at the face, passing through the enclosure, and exiting at the exhaust port will prevent the escape of airborne contaminants from the hood into the room. The degree to which this is accomplished depends on the design of the hood, its installation, and its operation.

Air currents external to a hood can disturb its air pattern and cause a flow of contaminants into the breathing zone of the researcher. Cross currents are generated by movements of the researcher, people walking past the hood, thermal convection, supply air movement, and rapid operation of room doors and windows. Terminal supply air velocity in the vicinity of the hood should be limited to avoid disturbing the operation of the hood. It is important to avoid the location of hoods near doors and active aisles.

21.5.3 Performance Criteria

Performance criteria for fume hoods are flow control, spillage, and face velocity control. Flow control is the regulation of flow over the face opening of a hood. Flow control is obtained by adjusting the horizontal slots in the back baffle. One is at the bottom of the back baffle to draw air across the working surface, another is at the top to exhaust the canopy, and a third is frequently midway on the baffle. These adjustable openings permit regulation of exhaust distribution for specific operations.

Spillage or leakage outward through the face opening of contaminants from hoods into the laboratory can be caused by:

- Drafts in the room.
- Eddy currents generated at hood opening edges, surface projections or depressions.
- Thermal heads.
- High turbulence operations (blenders, mixers) within the hood.

Corner and intermediate posts, deep-deck lip depressions, sinks, and projecting service fittings near the face produce air turbulence and potential spillage conditions. Plain entrance edges can sometimes impact airflow velocity and turbulence like a fluid passing through a small orifice within 1 in. (25 mm) of the surface and to a depth of 6 in. (150 mm). Fumes generated in this area will be disturbed and possibly escape the hood enclosure. Airfoil shapes at the entry edges correct this condition. Correcting this one feature on existing hoods has been the key to making satisfactory hoods out of some units that have previously performed unacceptably.

Face velocity control is affected by variations in the resistance of a hood exhaust air system. Two common causes are:

- Variations in the face opening.
- Buildup of exhaust air filter resistance.

Most laboratory fume hoods do not require exhaust air filters. However, when filters are required it should be noted that increases in fume hood exhaust air system pressures due to filter loading can range from 50 to 100 percent of the clean filter condition when high-efficiency filters are used. Constant pressure regulation may be achieved in one of the following three ways:

- An automatic pressure controlled damper in the duct system.
- A manually adjusted damper.
- An exhaust fan vortex control.

It is good practice to equip laboratory fume hoods with alarm devices to detect failure of exhaust airflow. Devices that monitor the rotation of the fan shaft are
FIGURE 21-1 TYPICAL PROCESS FUME HOOD
FIGURE 21-2 TYPICAL BYPASS FUME HOOD WITH VERTICAL SASH AND BYPASS AIR INLET
FIGURE 21–3 TYPICAL AUXILIARY FUME HOOD

- AUXILIARY AIR INLET DUCT
- AUXILIARY AIR FLOW
- AUXILIARY AIR SUPPLY DUCT
- FULL WIDTH SUPPLY PLENUM
- EXHAUST DUCT
- SUPPLY SLOT
- SIDE BAFFLES (OPTIONAL)
also recommended. An alarm that is visual, audible, or both should be extended to all hoods served by an exhaust air fan.

21.5.4 Auxiliary Air Hoods

Auxiliary air hoods are designed for direct supply air connections to an auxiliary air supply system. These hoods are intended to reduce the volume of conditioned room air exhausted and thereby reduce the overall cooling load. Auxiliary air should not be introduced within the hood because less air is drawn through the hood face and the face velocity is lowered correspondingly. When the auxiliary air is introduced across or in front of the opening, the flow pattern of the auxiliary air stream is critical to hood performance. When auxiliary air is dispersed into the laboratory, it often causes undesirable effects on room temperature and humidity. Additionally, condensation on cold surfaces may also result. Turbulent air motion can occur if the air stream impinges on personnel working at the hood or on any hood surfaces.

The application of auxiliary air hoods should be based on the performance characteristics of the specific model selected. The performance should be determined by test and the most extreme toxicity level that may occur within the hood should be the basis for evaluation and application.

The air balance of auxiliary air fume hoods must be carefully maintained. If the exhaust airflow decreases while the auxiliary airflow remains constant, control effectiveness may be lost and the laboratory room airflow may be upset. This could cause air to flow out of the laboratory and into adjacent spaces including corridors rather than from the adjacent spaces into the laboratory as planned. For proper hood performance, these air balances must be carefully maintained.

21.5.5 Special Laboratory Hoods and Exhausts

Perchloric acid fume hoods are required for research operations that use perchloric acid. These hoods require exhaust ducts of smooth, impervious, and cleanable materials that are resistant to acid attack. Stainless steels with high chromium and nickel content or other nonmetallic materials are recommended. Ductwork should be short, direct, and vertical to the terminal discharge point. Internal water spray systems for periodic washing of the duct surfaces are mandatory for perchloric acid exhaust systems. Perchloric acid deposits in ductwork can become a major explosion hazard. Since perchloric acid is an extremely active oxidizing agent, organic materials should not be used in the exhaust system in such places as joint gaskets. Joint construction should be welded and ground smooth. A perchloric acid hood should not be used for work other than that involving the use of perchloric acid.

For high-level radioactive hood exhaust systems where ducts must be dismantled for decontamination, flanged neoprene-gasketed joints with quick disconnect fasteners will provide minimum time exposure to decontamination personnel.

21.6 BIOLOGICAL SAFETY CABINETS

21.6.1 Categories Of Biological Safety Cabinets

Biological safety cabinets (BCS) are sometimes referred to as safety cabinets or ventilated safety cabinets. These cabinets are categorized as follows:

- Class I Cabinet
- Class II – Type A Cabinet
- Class II – Type B Cabinet
- Class III Cabinet

Table 21-1 describes some of the applications and airflow data for biological safety cabinets. Biological safety cabinets are typically available in two standard sizes. These two standard sizes correspond to the length of the cabinet and the standard sizes are 4-foot and 6-foot in length. The 6 ft (1.8 m) length size is the common size used.

21.6.1.1 Class I Cabinets

The Class I cabinet is a partial containment cabinet and very useful for containment of mixers, blenders, and other equipment. Figure 21-4 provides a diagram of a typical Class I biological safety cabinet. Room air flows through a fixed front opening and prevents microbial aerosols that may be released within the cabinet enclosure from escaping into the room, see Figure 21-4. The exhaust air passing through the biological safety cabinet removes entrained particles. The exhaust air may pass through a HEPA filter before being discharged from the cabinet to the exhaust system.
<table>
<thead>
<tr>
<th>Cabinet</th>
<th>Research Applications</th>
<th>Normal Exhaust* (from 5-foot Unit)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Class I</td>
<td>Partial containment (protects employee but not the material). Exhaust 100%. Suitable for flammable substances. Recommended for animal autopsy work and low risk biological agents.</td>
<td>300 cfm</td>
</tr>
<tr>
<td>Class II, Type A</td>
<td>Protects employee and research material. Recirculates 70%; exhausts 30%. Suitable for biological agents of low and moderate risk. Not suitable for chemical carcinogens, toxic, explosive, or flammable substances as it recirculates these back within the cabinet. Cabinet's exhaust may be discharged back into room or to outdoors via general exhaust system.</td>
<td>400 cfm</td>
</tr>
<tr>
<td>Class II, Type B</td>
<td>Protects employee and research material. Recirculates 30%; exhausts 70%. Suitable for low-level volatile materials and for trace levels of chemical carcinogens in tissue culture. Requires external exhaust fan. Recommended selection for most biomedical research laboratories.</td>
<td>360 cfm, 0.75 to 1.5 in. wg over service file of HEPA filters</td>
</tr>
<tr>
<td>Class III</td>
<td>Very special cabinet. Custom-designed units used for high-risk biological and chemical research only. Seldom selected; used for control laboratories at nuclear facilities.</td>
<td>(Variable determined by filter selection.)</td>
</tr>
</tbody>
</table>

*Capacity of exhaust duct system at the point of connection to the biological safety cabinet.

**Table 21–1 Typical Characteristics Of Biological Safety Cabinets**

**FIGURE 21–4 TYPICAL CLASS I BIOLOGICAL SAFETY CABINET**
The HEPA exhaust air filter is optional and depends on the cabinet usage. This type of cabinet will not prevent contact exposure. The Class I cabinet is designed for general research operations with low- and moderate-risk etiological agents such as bacteria or viruses. It can be modified to contain chemical carcinogens by adding an appropriate exhaust air treatment system and by increasing the velocity through the front opening.

### 21.6.1.2 Class II Cabinets

Class II biological safety cabinets are also devices for partial containment, ventilated design for protection of personnel and research materials for ubiquitous contaminants in room air. The cabinet features an open front with inward airflow for personnel protection and HEPA-filtered, recirculation mass airflow for protection of research materials. The cabinet exhaust is filtered through a HEPA filter. Two models are available that are designated Type A and Type B or may also be referred to as Type 1 and Type 2.

Figure 21-5 shows a diagram of a typical Class II Type A biological safety cabinet. A Class II Type A biological safety cabinet has a fixed work opening with a minimum inflow velocity of 75 fpm (0.38 m/s). The average minimum downward air velocity is 75 fpm (0.38 m/s). This design provides for recirculation of approximately 70 percent of the total cabinet air. As a result, the Type A cabinet should not be used with flammable solvents, toxic agents, or radioactive materials. Although the Class II, Type A cabinet can be installed to discharge exhaust air to the room, it is preferable to have the cabinet discharge the exhaust air to the laboratory exhaust system via a special field fabricated duct connection referred to as a thimble unit.

The Class II Type B biological safety cabinet has a vertical sliding sash, see Figure 21-6. A Class II Type B cabinet is designed to maintain an inward airflow of 100 fpm at a work opening of 8 inches and have an average downward vertical air velocity of 50 fpm. This design requires the separate exhaust of approximately 70 percent of the air flowing through the work area. As a result, a Class II Type B biological safety cabinet may be used with a wider range of chemicals. However, this type of cabinet is not recommended for use with explosive vapors. Class II Type B cabinets can be used for aerosol generating processes such as sonicating and blending with low- and moderate-risk etiological agents.
21.6.1.3 Class III Cabinets

The Class III biological safety cabinet is a gastight, negative pressure containment system that provides a physical barrier between the agent and the worker. Figure 21-7 illustrates a Class III biological safety cabinet. Of all biological safety cabinets, this one provides the highest degree of personnel protection. Work is performed through arm-length rubber gloves attached to a sealed front panel. Room air is drawn into the cabinet through HEPA filters. Particulate materials entrained in the exhaust air are removed by HEPA filtration before discharge to the atmosphere.

A Class III system may be constructed to enclose and isolate incubators, refrigerators, freezers, centrifuges, and other research equipment. Double-door autoclaves, liquid disinfectant dunk tanks, and pass boxes are used to transfer materials into and out of the cabinet. Class III systems are used to contain highly infectious materials and radioactive contaminants. Although there are operational inconveniences with these cabinets, they are the equipment of choice when the highest degree of personnel protection is required. The use of Class III cabinets for research involving volatile substances has resulted in explosions.

21.6.2 Laminar Flow Hoods

Horizontal cross flow and vertical down flow laminar flow clean benches, which blow air out of the front opening and into the room, should not be used in a biomedical laboratory without risk assessment. They provide product protection, but are a hazard to the worker because they blow air across the research material and then directly into the face of the worker. Use of a laminar flow clean bench in a biomedical laboratory may subject the worker to potentially hazardous or allergenic substances.

21.7 BIOMEDICAL LABORATORIES AND ANIMAL RESEARCH FACILITIES

21.7.1 NIH Design Requirements Manual

The National Institutes of Health (NIH) publishes the NIH Design Requirements Manual for Biomedical Laboratories and Animal Research (DRM). The NIH DRM provides detailed requirements for the design and performance of NIH owned and leased facilities that are either new or renovated. The objective of the NIH DRM is to ensure that biomedical laboratories and animal research facilities can properly support the biomedical research that will be conducted in them. The complete NIH DRM is available from the NIH’s Office.
of Research Facilities (ORF) through the NIH’s website at www.nih.gov.

The NIH DRM was published in August 2008 and replaced the former 2003 edition of the NIH’s Design Policies and Guidelines. The NIH DRM is divided into eleven chapters and eight appendices, see Figure 21-7. Chapter 6 and Appendix E of the NIH DRM covers HVAC system design and performance requirements for HVAC systems in biomedical laboratories and animal research facilities. However, the requirements of the other chapters and appendices could also impact the design, installation, and operation of HVAC systems in biomedical facilities depending on scope of the facility construction or renovation and should be reviewed for applicability. To state a few, Chapter 2 addresses general design consideration for biomedical laboratories and research facilities, Chapter 3 includes Section 3-2 that addresses site utilities, Chapter 7 covers building automation systems (BAS), Chapter 8 addresses plumbing requirements, and Chapter 9 addresses fire protection requirements. Lastly, Appendix D provides additional information regarding BAS.

21.7.2 Standard Biosafety Levels

The NIH DRM recognizes the following four levels of biosafety (BSL) for biomedical laboratories and animal research facilities:

- Biosafety Level 1 (BSL-1)
- Biosafety Level 2 (BSL-2)
- Biosafety Level 3 (BSL-3)
- Biosafety Level 4 (BSL-4)

These biosafety levels are determined by a combination of laboratory practices and techniques, safety equipment, and the laboratory facility’s research activities. Each level addresses the operations performed for the documented or suspected route of transmission of potentially infectious agents. These levels are defined and discussed in detail in the joint Center for Disease Control (CDC) and NIH document entitled Biosafety in Microbiological and Biomedical Laboratories (BMBL) that is available on the CDC website at www.cdc.gov. The current edition—on which elements of the following text is based—of the BMBL is the 5th Edition issued February 2007. The BMBL is an advisory document that recommends best practices for the safe conduct of work in biomedical and clinical laboratories from a biosafety perspective.

21.7.2.1 Biosafety Level 1

Biosafety Level 1 (BSL-1) is the basic level of protection and is for agents that are not known to cause disease in humans that are normal and healthy. Biosafety Level 1 practices, safety equipment, design and construction are appropriate for undergraduate or secondary educational training and teaching laboratories. Other laboratories that do work with defined and characterized strains of viable microorganisms that are not known to consistently cause disease in healthy adult humans would also be Biosafety Level 1. Biosafety Level 1 laboratories rely on standard microbiological practices with no special primary or secondary barriers other than a sink for hand washing.

21.7.2.2 Biosafety Level 2

Biosafety Level 2 (BSL-2) is for handling moderate-risk agents that cause disease in humans that is of varying severity by ingestion, procedures where access to inner organs or other tissue is done via needle-puncture of the skin, or mucous membrane exposure. Practices, safety equipment, facility design and construction for Biosafety Level 2 are applicable to clinical, diagnostic, teaching, and other similarly purposed laboratories. Biosafety Level 2 laboratory work can typically be conducted on the open bench where the potential for producing splashes and spills is low. Work with aerosol or high splash potential items that could result in the risk of personnel exposure should be conducted in primary containment equipment or other acceptable devices. Secondary barriers such as hand washing sinks and waste decontamination facilities must be available in Biosafety Level 2 laboratories.

21.7.2.3 Biosafety Level 3

Biosafety Level 3 (BSL-3) is for agents with a known potential for aerosol transmission and for agents that may cause serious or potentially lethal infections. The agents addressed by Biosafety Level 3 may be either indigenous or exotic in origin. Biosafety Level 3 practices, safety equipment, facility design and construction practices are applicable to clinical, diagnostic, teaching, research, or production laboratories dealing with these types of agents. The primary hazards to personnel working with Biosafety Level 3 agents are ingestion, secondary infection originating from an infection already present in the body, and exposure to infectious aerosols. For Biosafety Level 3, emphasis is placed on both primary and secondary barriers to protect personnel in nearby areas of the facility, the community, and the environment from exposure to potentially infectious aerosols.
Chapter 1  Administration
Chapter 2  Design Considerations
Chapter 3  Civil Engineering and Site Development
Chapter 4  Architecture
Chapter 5  Structural
Chapter 6  HVAC
Chapter 7  Building Automation Systems
Chapter 8  Plumbing
Chapter 9  Fire Protection
Chapter 10  Electrical
Chapter 11  Telecommunications
Appendix A  References, Design and Safety Guidelines, Health and Safety Regulations, Codes and Standards
Appendix B  Architect-Engineer (A/E) Checklist of Services
Appendix C  Room Data Matrix
Appendix D  Building Automation Systems
Appendix E  Heating Ventilating and Air Conditioning
Appendix F  Links to References on the World Wide Web
Appendix G  Lease Facilities Design Requirements Manual Checklist
Appendix H  Downdraft Table Particle Capture Efficiency Calculation

FIGURE 21–7 NIH DESIGN REQUIREMENTS MANUAL CONTENTS
21.7.2.4 Biosecurity Level 4

Biosecurity Level 4 (BSL-4) is for exotic agents that pose a high individual risk of life-threatening disease by infectious aerosols and for which there is no available vaccine or therapy. Biosecurity Level 4 laboratories are high containment laboratories where the primary hazard to personnel is respiratory exposure to infectious aerosols, mucous membrane or broken skin exposure to infectious droplets, and secondary infection originating from an infection already present in the body. Personnel working with these unknown agents are required to be completely isolated from the Biosecurity Level 4 agents by working in a Class III BSC or in a full-body, air-supplied, positive-pressure personnel suit. Typically a Biosecurity Level 4 facility will be a separate, standalone building or completely isolated zone within a facility. A Biosecurity Level 4 facility has specialized ventilation requirements and waste management systems to prevent the release of viable agents to the environment.

21.7.3 Other Specialized Biosecurity Levels

In addition to the four standard biosecurity levels, there are also some specialized biosecurity levels that address experimentation with live animals. The four standard biosecurity levels are modified to address the practices, safety equipment, and facilities for activities involving infectious disease work with commonly used experimental animals. These specialized biosecurity levels are designated as follows:

- Animal Biosecurity Level 1 (ABSL-1)
- Animal Biosecurity Level 2 (ABSL-2)
- Animal Biosecurity Level 3 (ABSL-3)
- Animal Biosecurity Level 4 (ABSL-4)

Each of these Animal Biosecurity Levels provides increasing levels of protection for laboratory personnel, the community, and environment just as do the standard biosecurity levels.

Similarly, there is an additional specialized biosecurity level designated Biosecurity Level 3 Agriculture (BSL-3-Agriculture or BSL-3-Ag). This specialized biosecurity level addresses infectious biological agent experimentation activities involving large or loose-housed animals. In this case, the laboratory facility itself serves as the primary barrier to the release of infectious agents into the environment.

21.7.4 Biosecurity Equipment And Systems

When standard laboratory practices are insufficient to control the hazards associated with a particular agent or procedure, then additional biosecurity equipment and systems may be required. These additional biosecurity measures were referred to in the previous section on biosecurity levels and are grouped into the following two categories by the CDC and NIH:

- Primary Barriers
- Secondary Barriers

21.7.4.1 Primary Barriers

Primary barriers are intended to be the first line of protection for personnel to shield them from direct contact with infectious agents. Primary barriers include both personal protective equipment and protective laboratory equipment. Personal protective equipment includes gloves, coats, face shields, and other safety equipment specifically intended to individually protect people in the work area. Protective laboratory equipment includes BSCs, enclosed containers, and other engineering controls designed to remove and minimize exposure to hazardous agents. The BSC is the most common safety equipment used to protect personnel by containing hazardous slushes or aerosols that could result from laboratory procedures. Other safety equipment includes a safety centrifuge cup that prevents aerosols from being released when a hazardous material is centrifuged.

21.7.4.2 Secondary Barriers

Secondary barriers refer to the design and construction of the facility to protect not only laboratory personnel but also the surrounding community and environment from dangerous biological agents. The recommended secondary barriers for a laboratory involved in infectious agent research will depend on the risk of transmission of the biological agents with which the lab is working. Where the risk is possible direct contact with the infectious agents as it is in Biosecurity Levels 1 and 2, secondary barriers in these laboratories may include the physical separation of the laboratory work area from other areas in the facility, restricting public access to laboratory work areas, providing a decontamination facility, or hand washing facilities.

When there is the risk of infection by airborne biological agents as in Biosecurity Levels 3 and 4 laboratories, higher levels of primary barriers as well as multiple secondary barriers may be necessary to ensure the safety of laboratory personnel, the community, and en-
environment. Secondary barriers to protect against the release of infectious biological aerosols can include

- Ventilation systems that ensure directional airflow
- Air treatment systems to decontaminate or remove dangerous biological agents from the exhaust air
- Controlled access including airlocks at laboratory doors
- Separate buildings or isolated areas within buildings

21.7.5 Specific Ventilation Recommendations

For specific ventilation recommendations for biomedical laboratories and animal research facilities, the BMBL refers to the ASHRAE Laboratory Design Guide published by the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE).

21.7.6 HVAC Systems For Biosafety Containment Laboratories

The requirements for HVAC systems in biomedical laboratories and animal research facilities are provided in Chapter 2/Design Considerations and Chapter 6/HVAC of the NIH DRM. However, the design and installation requirements of other NIH DRM chapters could also impact the HVAC system design and installation depending on the scope of the project. Specifically, Chapter 6/HVAC contains the following five general sections that apply to all standard and animal biosafety levels:

6-1 HVAC Design Considerations
6-2 Air-Handling Systems
6-3 Piping Systems
6-4 Thermal Insulation Systems
6-5 Noise and Vibration

Section 6-1 covers specific HVAC design requirements which include detailed requirements for various occupancies, systems, and operations including specific design conditions, mechanical equipment location and access, room air distribution and pressurization, system installation, and documentation requirements, among others. Detailed requirements for air-handling systems are provided in Section 6-2 that include air distribution systems that meet SMAC-NA’s Duct Construction Standards’ requirements, location of outdoor air intakes and exhaust air discharge, detailed specification of equipment and components. Piping system requirements are covered in Section 6-3 that include hydronic and steam system layout along with the detailed requirements for piping system equipment and materials as well as piping system installation.

Thermal insulation requirements for ducts and piping are provided in Section 6-4. Section 6-5 covers HVAC equipment noise and vibration requirements including maximum noise levels.

Appendix E of the NIH DRM supplements the information contained in Chapter 6 by providing additional information the following topics:

- Calculating the ventilation rate for the removal of contaminants from biomedical laboratories
- Calculating the minimum separation distance between air intakes and exhausts
- Fume hood testing and alarm systems
- Calculation protocols for canopy hoods over autoclaves
- Selecting and specifying variable frequency drives for HVAC systems

21.7.7 Biosafety Level 3 and 4 Containment Laboratories

Biosafety Level 3 containment laboratories and Biosafety Level 4 high containment laboratories both address the protection of laboratory personnel, the community, and the environment from airborne infectious biological agents through both primary and secondary barriers. Section 6-6 specifically addresses the HVAC requirements for both standard (BSL-3) and animal (ABSL-3) Biosafety Level 3 containment laboratories. The requirements contained in Section 6-6 are in addition to the general requirements provided in Sections 6-1 through 6-5.

There are no specific requirements for either Biosafety Level 4 (BSL-4) biomedical laboratories or Animal Biosafety Level 4 (ABSL-4) animal research facility in either Chapter 2 or Chapter 6 of the NIH DRM.

Designers or contractors who are considering work in any facilities that are subject to Section 21.7 should verify that the most up to date reference documents are being used.
A.1 DISPLACEMENT VENTILATION

OVERVIEW

Displacement ventilation (DV) is an air distribution system that delivers conditioned supply air directly to occupants in the zone that it serves, much like any other ducted HVAC system. However, the difference comes in the lower air delivery velocity as well as the point of distribution, which is at a low point in the space to be conditioned. DV systems have been used in commercial and institutional buildings since the early 1970s. DV systems can provide increased indoor environmental quality (IEQ) and energy savings in the right application when compared with conventional mixed ventilation systems such as variable-air-volume (VAV) systems. As a result of these advantages, DV systems are being considered for and designed into sustainable buildings.

A.2 DV SYSTEM OPERATION

A DV system supplies conditioned air to the zone served at or near the floor to the "breathing zone" of occupants. DV systems are most often used in zones that do not require heating but, as discussed later in this appendix, DV systems can also be used for heating when the heating load is small. A DV system that only provides cooling or a combination system operating in the cooling mode supplies conditioned cold air from diffusers located at or near the zone's floor. The density of the conditioned cold air that is introduced into the space is greater than the existing warmer air in the space. As a result, the cooler conditioned air migrates along the floor through the space and displaces the lighter warmer existing air that rises due to natural buoyancy. This results in the thermal stratification of the air in the room. In effect, the cooler conditioned air introduced to the zone near the floor has the effect of displacing the warmer existing air and pushes it toward the ceiling. The warmest air in the zone exists at or near the ceiling where it is captured and exhausted from the room by exhaust vents. Many contaminants, such as VOCs become entrained more readily and are exhausted along with the ambient air.

As the cool supply air absorbs heat from solar radiation in the zone or by convection through contact with warm building surfaces, occupants, animals, and heat producing equipment, it becomes less dense and rises toward the ceiling. This rising warm air stream results in thermal stratification. This thermal stratification is typically limited to about 5°F in the occupied zone to ensure occupant comfort in accordance with ASHRAE 55 ASHRAE Standard 55 entitled *Thermal Environmental Conditions for Human Occupancy*. For a DV system, the occupied zone usually extends from the floor to a height of about 6 feet.

A.3 DV SYSTEM THERMAL PLUME

If the zone served by a DV system did not have any occupants or other heat sources, the cool supply air would eventually absorb heat from solar radiation and room surfaces and rise to the ceiling where it would be exhausted. However, where there are heat sources in the room such as people, animals, and electronic equipment, the cool air supplied by the DV system absorbs heat from these heat sources by convection. As heat is transferred from these heat sources to the cooler air coming in contact with the heat source, the warmer, less dense air begins to rise around the heat source. The warm air rising around a heat source creates a plume that surrounds the heat source from its base to the ceiling.

This vertical airflow around occupants results in the person being surrounded by a plume of upwardly moving fresh supply air. This thermal plume provides both thermal comfort and fresh air to the occupant resulting in increased IEQ. In addition, because each occupant is enveloped in his or her own thermal plume from floor to ceiling there is less cross contamination between occupants or other sources of odors and contaminants in the zone. As long as the pollutant is lighter than air or lighter than air when warmed, it will move upward with the air stream and be exhausted from the room.

With a conventional mixed ventilation system, fresh air is typically introduced at or near the ceiling through supply diffusers and then travels horizontally across the room where it is captured and exhausted by return vents. As a result, contaminants such as germs are spread from upstream sources to downstream occupants by the horizontal air stream traversing the room. The vertical plume surrounding occupants in a zone with DV reduces the chance of cross contamination as long as strong drafts due not occur. Drafts in zones utilizing DV can cause unwanted mixing of air and cross contamination between occupants and contamination sources.
A.4 DV SYSTEM ENERGY CONSIDERATIONS

In addition to improved IEQ, DV systems can also provide increased HVAC system energy efficiency when compared to conventional mixed air systems. Due to the fact that a DV system introduces cool supply air at or near the floor that comes in direct contact with building occupants, the conditioned air can be supplied to the zone at a higher temperature than with a conventional mixed air system. With a mixed air system, the supply air mixes with the existing air in the zone before it reaches the occupants. As a result, the conditioned supply air must be provided at a lower temperature so that the occupant experiences the desired temperature when the supply air is mixed with the zone’s existing air. A typical mixed air ventilation system provides supply air at 55°F whereas a typical DV system supplies air at a temperature that is closer to the desired occupant temperature and is often between 65°F and 75°F. The higher the supply air temperature, the lower the chiller or unit HVAC system load which in turn saves energy. However, in humid climates air temperature must still be reduced to 55°F to remove moisture and then mixed for distribution. In addition, the ability of a DV system to use a higher supply air temperature will also allow greater use of free cooling saving additional energy.

Even if conditioning is being done with a DV for the first 6 ft of elevation instead of all the way to a ceiling, load calculations still need to be considered particularly because of the lower temperature difference between the space and the supply air discharge temperature.

Similarly, air is supplied to the zone served by the DV system at a lower velocity than what is typically supplied by a conventional mixed air distribution system. As a result, smaller fans can often be used for a DV system that saves on the initial cost of the fan and ductwork serving the zone. In addition, the lower fan size and reduced airflow that can be used with a DV system should also result in reduced operating costs over the life of the installation. A byproduct of supplying lower velocity air to the zone served by the DV system is reduced noise in the space and better acoustics when compared with a conventional mixed air distribution system.

A.5 CEILING HEIGHT

One restriction on the effective use of a DV system is ceiling height. In order to be effective, the ceiling height of a zone using a DV system must be at least 9 feet above the floor. The nine-foot minimum ceiling height provides space above the six-foot occupied zone to allow the warmed air to gather at the ceiling and be captured and exhausted from the zone by return air vents.

A.6 HEATING WITH DV SYSTEMS

The most common application of a DV system is in zones that require continuous year-round cooling. However, DV can also be used to heat a zone if the heating load is low. In the heating mode, a DV system actually acts as a conventional mixed ventilation distribution system with the conditioned air at or near the floor and the return air system at or near the ceiling. However, care must be taken when locating the return air grilles in the space to avoid short-circuiting the warm air directly from its supply point to its return point. This short-circuiting will occur because the warm air being introduced to the space at floor level will rise due to buoyancy to the ceiling and be exhausted directly from the zone unless the location of the supply diffusers and return vents are located such that the warm supply air will have to mix with the cooler existing air raising the ambient temperature of the space to what the occupants want.

A.7 DV SYSTEMS AND UNDERFLOOR AIR DISTRIBUTION SYSTEMS

Underfloor air distribution (UFAD) systems and DV systems are often considered to be synonymous with one another. However, as noted above, the conditioned air from a DV system can be supplied at or near the floor of the zone served. In other words, the conditioned air supplied by a DV system could be supplied through a vertical wall-mounted diffuser as well as a diffuser installed directly in the floor that is supplied either by in-the-floor ductwork or an under-the-floor supply air plenum. All UFAD systems are DV systems but not all DV systems are UFAD systems.

UFAD systems originated from a raised floor system. Raised floor systems were initially used in computer rooms to provide easy access to alter and run new wires.
or reroute existing wiring as computer systems were moved or updated. The raised floor provided a convenient path not just for wiring alterations but also to supply conditioned air to the computer equipment. Over time, this concept was extended to include office spaces in the belief that it simplified the reconfiguration of offices. Where raised floors are required, UFAD may present an economical alternative. There is considerable difference of opinion that the use of UFAD offers a more economical or effective system when installed simply as a UFAD where the raised floor is not required. This difference of opinion has been exacerbated by technical problems that have been experienced with UFAD systems. The fact that no industry standard has been developed for UFAD has compounded the problems of introducing this technology into the general market and given rise to many operational problems.

A.8 CONCLUSION

For all of these reasons, Displacement Ventilation is a viable Air Distribution system that can be both economical to operate and environmentally beneficial to the building occupants, and warrants consideration when first designing a buildings HVAC system.
B.1 DEDICATED OUTSIDE AIR SYSTEMS OVERVIEW

Historically, most HVAC systems designers handle both the space conditioning and ventilation requirements with a single system. Habit and first cost perceptions have kept most designers on this path. The current wisdom says it is simpler and cheaper to design and install a single system to handle sensible, latent, and ventilation with one duct and air handler system.

But there are many operational realities that are out of alignment with the one duct/air handler approach. Sensible and latent peak loads are rarely coincident. The result is that during many of the annual operational hours the sensible load is driving HVAC system operation and the latent load is being insufficiently handled, especially on cool, rainy days. The resulting least case problem is occupant discomfort and the worse case issue is excess moisture in the building.

A more effective design approach would be to separate the ventilation requirements from the space conditioning using a dedicated outside air system (DOAS). It so happens that a DOAS also provides many additional inherent advantages if it is properly integrated with the primary HVAC system, the local climate, and the needs of the occupants.

A completely separate and independent DOAS has its own duct, controls, and conditioning components that can operate independently from the primary HVAC system. Such a system would typically have the capability to handle all occupant and space condition loads up to 30 percent of the peak load. So, one of the “screens” to evaluate the economics of DOAS are the number of annual operational hours above this threshold.

B.2 DOAS ADVANTAGES

Ventilation air can be supplied directly to the individual spaces in the quantity required. Compliance with code-mandated ventilation requirements is difficult to achieve on an-hour by hour basis. DOAS is arguably the most reliably consistent and verifiable method to achieve an individual building’s ventilation requirements.

During mild weather or periods of low occupancy the DOAS system can handle all the space or occupancy conditioning which permits the much larger HVAC system to be completely turned off. Considering the potential for energy savings, this one benefit—by itself—should justify at least considering and examining the use of DOAS.

Building pressurization can be precisely managed on a floor-by-floor or even a space-by-space basis.

With dehumidification separated from the sensible load, more precise temperature and humidity control can be achieved.

Potentially, the control systems will be less complex since the DOAS and HVAC system have divided the task of code-required ventilation and space conditioning.

DOAS can also be partnered with other types of non-ducted HVAC systems—such as radiant heating and cooling—to provide the ventilation and outdoor air conditioning requirements while retaining the human comfort benefits and energy saving benefits inherent in radiant thermal conditioning. These designs may actually result in a reduced first cost versus a single, central HVAC system.

In high-security buildings DOAS can be used to better thwart the malicious introduction of air-borne contaminants into the building. This can be accomplished both by being selective regarding the location of the DOAS intake—typically near the center of the roof—and by maintaining a constant positive pressure on the building’s envelop. Since all outside air enters the building via a single, secure location specialized filtering and air cleaning systems can be easily in-
installed and operated to provide further protection of the space and occupants. Certain types of US Federal buildings are required to use DOAS for these reasons.

B.3 DOAS DISADVANTAGES

Because in a 100 percent DOAS two separate duct systems, controls, and HVAC conditioning equipment are required, the first cost is generally expected to be higher. That said, since the use of a DOAS system makes the companion HVAC equipment smaller and less complex some claim they can design DOAS systems for less that a conventional HVAC system.

Since a typical 100 percent DOAS only supplies approximately 30 percent of the total airflow volume of a central HVAC system this is also the maximum amount of economizer capacity that can be provided. Whether or not this is a true “system” disadvantage depends on the climate, building operational requirements and the balance of HVAC system type and capabilities.

B.4 FRACTIONAL AND INTEGRATED DOAS SYSTEMS

Of course designers can also apply DOAS principles to other system design variations such as delivering DOAS conditioned air to local terminal units or directly into the primary HVAC’s duct system. While this may better fit the conventional design path along which many find professional comfort, it does so at the expense of many benefits of total 100 percent DOAS designs.

Another possibility is to apply DOAS in building spaces with widely variable ventilation requirements, only. Along this design path, outside air volumes would track the requirements in meeting and convention spaces based on occupancy using sensors. This would allow the primary HVAC system to operate a relatively constant pace and avoid the need of outsized equipment to address widely variable “point” loads.
APPENDIX C

SUSTAINABLE BUILDING
HVAC SYSTEMS
C.1 SUSTAINABLE BUILDINGS

Concern about the environment and the future of our planet has become the focal point of everyday conversation, political debate, and media coverage, not only in the United States, but around the globe. Where this debate has been focused on the industrial, manufacturing, and transportation sectors in the past, energy usage and its associated environmental impacts have become a major issue in the building industry. Buildings consume about 40 percent of the energy used in the United States according to the U.S. Department of Energy’s Energy Information Agency. The amount of energy used in buildings and its percentage of the total United States’ annual energy usage is expected to increase in the future despite increased efficiency and conservation efforts. As a result, more and more building owners, including all levels of government as well as building occupants, are demanding sustainable buildings. To accomplish this, they and are seeking third-party certification to verify and publicly recognize their commitment to the environment.

C.2 WHAT IS A SUSTAINABLE BUILDING?

The term “sustainable” or “green” building is defined in ASTM Standard E2114-06a as a building that provides the specified building performance requirements while minimizing disturbance to and improving the functioning of local, regional, and global ecosystems both during and after its construction and specified service life. The HVAC system is a key element in any sustainable building project because it has a significant impact on the building’s energy usage and operating costs as well as the well being of the building occupants on a daily basis. As a result, HVAC systems are a central part of any sustainable building rating system that is used on a building project.

C.3 SUSTAINABLE BUILDING RATING SYSTEMS

C.3.1 Purpose Of Sustainable Building Rating Systems

The purpose of a sustainable building rating system is to provide an objective standard for certifying that a building is environmentally friendly or “green.” This is best accomplished when the certification process is coordinated by and through a third party familiar with the chosen certification/rating system. Many federal, state, and local governments and governmental agencies are beginning to require that its buildings as well as private buildings under their jurisdiction either be certified or certifiable using a third-party sustainable building rating system. The HVAC contracting firm needs to be aware of the specific requirements included in any sustainable building rating system that is being used on a project because it can impact the HVAC contracting firm’s design if it is a design-build project. It also impacts the procurement and fabrication of HVAC materials and equipment including ductwork, and field operations including HVAC system installation, commissioning, and closeout activities.

C.3.2 Typical U.S. Sustainable Building Rating Systems

There are a number of green building rating systems that are in use or being developed worldwide and in the U.S. The following two green building rating systems are the most commonly encountered green building rating systems in the United States:

- Green Globes™
- Leadership in Energy and Environmental Design™

C.3.2.1 Green Globes™

The Green Globes™ green building rating system is licensed by The Green Building Initiative (GBI) from the Canadian firm ECD Energy and Environmental, Ltd. GBI is a non-profit organization whose mission is to promote the adoption of sustainable building practices. GBI adapted Green Globes™ to the U.S. commercial building market by referencing U.S. codes and standards, measurement units, and analysis tools.

The Green Globes™ rating system addresses seven sustainable building categories that are: project management; site; energy; water; resources; building materials, and solid waste; emissions and other impacts; and indoor environment. GBI provides third-party certification of green buildings for building owners that...
C.2 HVAC Systems Applications

want to pursue certification using the Green Globes™ rating system. Certification under the Green Globes™ rating system results in the building being rated from one to four globes based on the percent of applicable points earned. Additional information about GBI and the Green Globes™ rating system can be found at www.thegbi.org.

C.3.2.2 Leadership in Energy and Environmental Design™

The U.S. Green Building Council’s (USGBC) developed Leadership in Energy and Environmental Design (LEED™) green building rating systems. The USGBC also maintains the Leadership in Energy and Environmental Design (LEED™) green building rating systems on an ongoing basis. The USGBC is an industry organization whose membership is made up of all parts of the construction industry including owners, designers, contractors, and others. The USGBC promotes the construction of environmentally friendly buildings through its sponsorship of the LEED™ green building rating systems. Additional information about USGBC and the LEED™ green building rating systems can be found at www.usgbc.org.

The LEED™ green building rating systems are the most common rating systems encountered in commercial and institutional construction in the United States today. Therefore, the following sections will focus on the LEED™ green building rating systems and certification process. There are also other rating systems including the Green Globes™ rating system that are similar in scope and application.

C.4 LEED™ Certification

LEED™ certification of buildings is based on the ability of the owner to demonstrate that the building project meets the requirements of the LEED™ green building rating system being used. The USGBC has developed or is in the process of developing a number of different rating systems for different project and building types. The most common LEED™ sustainable building rating system encountered is LEED™ 2009 for New Construction and Major Renovations which is the rating system used for new building construction and major building renovations. Other USGBC sustainable building rating systems address core and shell development for speculative buildings and commercial interiors for tenant build-out within an existing building core and shell. In addition, the USGBC also has specific green building rating systems for different types of buildings such as school, retail, and health care either available or under development.

C.5 LEED™ Certification

LEED™ certification of a building project starts with the owner’s decision that the project will be a “green” project. The second step is to select the appropriate LEED™ building rating system. This will typically be LEED™ 2009 for New Construction and Major Renovations that addresses the following seven aspects of a sustainable building:

- Sustainable Site (SS)
- Water Efficiency (WE)
- Energy and Atmosphere (EA)
- Materials and Resources (MR)
- Indoor Environmental Quality (IEQ)
- Innovation in Design (ID)
- Regional Priority (RP)

Each of these seven aspects of a sustainable building represents a category in the rating system. These categories include both mandatory requirements that are referred to as prerequisites as well as points that are referred to as credits. Credits are awarded to a project seeking LEED™ certification for meeting criteria in excess of the prerequisite requirements. Based on the number of credits achieved, the building can be certified as “LEED Certified” which requires the minimum number of credits, or achieve a higher certification of “Certified Silver,” “Certified Gold,” or “Certified Platinum,” the latter being the highest certification level available through the USGBC.
C.6 EXAMPLE LEED™ HVAC REQUIREMENTS

The Energy and Atmosphere (EA) Category in the U.S. Green Building Council’s (USGBC) Leadership in Energy and Environmental Design (LEED™) 2009 for New Construction and Major Renovations requires that the building comply with ASHRAE/IESNA Standard 90.1-2007. This requirement impacts the selection and layout of HVAC equipment, system, and control requirements. Further, EA Category Credit 1 requires that the building energy performance be improved beyond the minimum energy specified in ASHRAE/IESNA Standard 90.1 in order to earn points toward LEED™ accreditation. EA Category Credit 4 addresses refrigerant management and EA Category Credit 5 requires the continuous metering of HVAC and other building equipment and system operation.

Indoor air quality is also addressed in LEED™ 2009 for New Construction and Major Renovations both during construction and following occupancy under the Indoor Environmental Quality (IEQ) Category. The prerequisite for this category requires that buildings comply with ASHRAE standard 62.1-2007. Further, credits are available in the IEQ category for post-occupancy outdoor air delivery monitoring (Credit 1) and increased ventilation (Credit 2). Potentially, LEED™ requirements could impact the building’s HVAC control system because IEQ Category Credit 6.2 covers the controllability of HVAC systems for thermal comfort.

C.7 SUSTAINABLE BUILDING INFORMATION FOR THE HVAC CONTRACTOR

The LEED™ requirements cited in this section are only a few of the many requirements that may impact the HVAC system selection, procurement, installation, and operation. There are a number of other requirements that can impact the HVAC system design and procurement, as well as affect the HVAC contracting firm’s shop fabrication and field installation work on site. Copies of the LEED™ green building rating systems and checklists are available as downloads at the USGBC website. To further assist the project team in meeting the requirements of the LEED™ green building rating systems, the USGBC also publishes reference guides such as the Green Building Design and Construction Reference Guide (2009 edition) that provide detailed information about the intent of each prerequisite and credit, requirements and submittals required, design strategies, case studies, and other information.

The HVAC Contractor’s Guide To Bidding Green Building Projects as published by SMACNA to help SMACNA-member HVAC contractors understand sustainable building requirements and how these requirements can impact the HVAC contractor’s office, fabrication shop, and field operations. The HVAC contracting firm should be thoroughly familiar with the LEED™ or other sustainable building rating system required on a project prior to submitting a bid or proposal for that project.

Finally, any contractor considering bidding on a project that is seeking compliance or certification within any “green” program or rating system should verify what project-specific requirements they need to meet or what other contractors they need to work in conjunction with to meet the requirements of the green rating program. In spite of well-developed green rating methodologies, the short history of the green rating programs has shown that there are often expectations that are not included in the specifications. And, some “green professionals” who may be responsible for evaluating compliance with green requirements may interpret in a far reaching manner that may not always be conducive to the construction environment.
Thermal energy storage (TES) is a technology that stores energy in the form of heat (or “cold”) for use at a later time. Historically, both heat and cool energy has been stored for later use. Typically, energy is stored at night to take advantage of lower utility system demands and then released during the day when demands on the utility system is greater. Either heat or cool energy can be stored in materials with thermal mass and the total energy stored can be increased by taking advantage of the greater energy available in phase change—such as the freeze-thaw cycles of water. The most prevalent form of TES currently in use in the United States is often referred to as Cool Storage or Ice Storage. While the balance of this description will focus only on the most prevalent TES—ice storage—many of the engineering principles are the same or similar for other forms or variations of TES.

Providing air conditioning for commercial buildings during summer daytime hours is the largest single contributor to electric utility system peak demand. In the afternoon, as more air conditioning is needed to maintain comfortable temperatures for building occupants, the increased demand for electricity adds to the load created by lighting, operating equipment, computers and other uses. This requires the electric power suppliers to bring peak load generating equipment on line to handle these periods of highest demand.

Commercial users, whose large air conditioning loads contribute significantly to the need for these peaking generating stations, are charged more for energy demand because they contribute heavily to these peak demand periods. These charges are either in the form of higher energy charges (kWh) or via demand charges (kW) which is based on the customer’s highest peak demand for electricity. The demand charge is typically based on the amount of electricity required over a utility-specified time period, usually 15 or 30 minutes, and is assessed based on the highest demand on a monthly or even a yearly basis.

Thermal energy storage (TES) shifts space conditioning loads from the utility peak periods into off-peak hours. In most utility service areas this will significantly lower energy or demand charges during the air conditioning season. In some TES designs and building types there is also the potential to lower total energy usage using TES—this is particularly true when air-source cooling equipment is providing the cooling since outside air temperatures are lower at night.

The most prevalent TES methods use a standard chiller to produce ice at night during off-peak periods when the building’s electrical loads are minimal. The ice is made and stored in ice tanks. The stored ice provides cooling to meet the building’s air conditioning load requirement the following day allowing chillers to be downsized (partial storage) or completely turned off (full storage). TES is a proven method of reducing building operating costs with several thousand installations worldwide.

TES can not only substantially reduce operating costs but it also has the potential to substantially reduce capital cost outlays for HVAC equipment. Engineers can specify half-sized chillers operating up to 24 hours a day rather than full-size chillers that may only operate 10 or 12 hours per day.

In retrofit applications, TES systems can often provide cooling for an addition or increased loads in a building without adding chiller capacity. In conventional air conditioning system design, cooling loads are measured in terms of “Tons of Refrigeration” (or kW’s required) or more simply “Tons.” TES systems, however, are measured by the term “Ton-Hours” (or kW·h) or how many tons of cooling are required over a 24-hour design day.

### D.1 FULL OR PARTIAL STORAGE

There are a number of operational and control strategies that can be used to take advantage of the benefits of cool storage but there are only two basic design approaches—full or partial storage.

Full storage uses a chiller sized in the normal manner with sufficient cool storage capacity to allow the chiller to be turned off during peak utility periods—the daytime. During the day, the only loads are lights, computers, and the fans and pumps that circulate the stored cooling energy to cool the building and its occupants.

Partial storage uses half-sized chillers with enough storage to “carry” half the building’s cooling load through the peak daytime hours as the chiller “carries” the other half of the cooling load. This effectively reduces the air conditioning load 50 percent and the initial chiller capacity a similar factor. Of course, storage capacity adds “back” to the initial cost but there are
other potentially offsetting savings such as smaller electrical system requirements.

In new construction projects the electric rates are one of the primary determinants of which design strategy is the most cost effective for a project. High demand charges may completely offset the cost of full-sized chillers and full-sized cool storage capacity.

Full storage systems are often used in retrofit applications using the existing chiller capacity since the equipment investment is already embedded.

Cool storage has historically been designed in two formats, constructed and modular storage.

The essential component of a modular storage system is an insulated tank—typically constructed from a durable plastic such as polyethylene. Inside the insulated tank is a heat exchanger that is typically plastic tubing so that it can endure the freeze/thaw cycles of the water. Tanks can be provided in a variety of sizes ranging from 45 to over 500 ton-hours. At night, a fluid comprised of water and an antifreeze solution—typically ethylene glycol—is cooled by a chiller and circulated through the heat exchanger. Heat is extracted until most of the water in the tank is frozen. In a well-designed modular system the ice would be built uniformly throughout the tank but a small percentage of the water does not become surrounded by ice during the freezing process, preventing damage to the tank.

The following day, the ice cools the heat exchanger solution circulating through the piping in the tank and, using a temperature modulating valves and a bypass loop to bypass the tank, mix to achieve the desired operating temperature—typically around 45°F (7.2°C). The 45°F (7.2°C) fluid enters the coil, where it cools the building using the balance of the HVAC system such as air coils, chilled beams or some other “cooling” delivery method.

It should be noted that, while making ice at night, the chiller must cool the water-glycol solution to 25°F (-3.9°C), rather than produce the more typical 45°F (7.2°C) water temperatures required for conventional air conditioning systems. This has the effect of “e-rating” the nominal chiller capacity by 30 to 35 percent. Compressor efficiency, however, will vary only slightly (either better or worse) because lower nighttime air temperatures result in cooler condenser temperatures and help keep the unit operating efficiently.

D.1.1 Where Does Storage Makes Sense?

Energy storage systems meet the cooling requirements of almost every type of facility: offices, schools, arenas, hotels, churches, large retail stores, hospitals, restaurants, supermarkets, data centers with emergency cooling, and batch type industrial processes.

In the case of facilities that have 24 hour operation thermal energy storage may have a good life cycle value. Hotels have offices, ballrooms, and meeting rooms with intermittent use that can add to the electrical demand peaks. Thermal energy storage can minimize the demand penalty associated with those events. Hospitals have surgical suites and out-patient centers that have daytime only use. Thermal energy storage can minimize the demand penalty within those facilities as well. Schools are a perfect application for thermal energy storage because the cooling loads are so variable and cool storage minimizes the demand charges.

D.1.2 How would one evaluate thermal energy storage?

Thermal energy storage should reduce on-peak electrical demand and may reduce chiller plant capacity depending of the operational strategy. When properly sized and applied, an ice thermal energy storage for new construction design, the first cost of the mechanical system may be comparable to conventional chiller systems. Saving energy and energy costs are an additional benefit. Design considerations include available space for storage tanks, chiller location: series-upstream, series-downstream, parallel, system charging supply and return temps and cooling loop supply and return temps.

Whatever the reason for a chiller replacement (i.e., age, reliability, inefficiency, capacity issues, or refrigerant phase out) a thermal energy storage system should be considered as part of the replacement solution. The future demand for energy and the cost of energy will continue to rise. New building design standards are favoring higher system energy efficiencies,
lower peak demand and incorporation of “green” technologies. Thermal energy storage is a natural choice.

D.1.3 How it would be integrated with other types of HVAC systems?

Thermal energy storage uses standard air conditioning chillers with factory mounted ice-making controls and increases total cooling capacity without increasing electrical power supply components such as electric service, transformers, and switch gear. The cooling loop fluid may need to be changed from just water to a water/anti-freeze mixture.
**Absolute Pressure** – Pressure measured relative to the absolute zero pressure - the pressure that would occur at absolute vacuum.

**Absorption** – A process whereby a material extracts one or more substances present in an atmosphere or mixture of gases or liquids accompanied by the material's physical and chemical changes.

**Actuator** – A controlled motor, relay or solenoid in which the electric energy or pneumatic is converted to a rotary, linear, or switching action. An actuator can effect a change in the controlled variable by operating the final control elements a number of times. Valves and dampers are examples of mechanisms which can be controlled by actuators.

**Adjustable Differential** – A means of changing the difference between the control cut-in and cut-out points.

**Aerodynamic Noise** – Also called generated noise, self-generated noise; is noise of aerodynamic origin in a moving fluid arising from flow instabilities. In duct systems, aerodynamic noise is caused by airflow through elbows, dampers, branch wyes, pressure reduction devices, silencers and other duct components.

**Air, Ambient** – Generally speaking, the air surrounding an object.

**Air, Dry** – Air without contained water vapor; air only.

**Air, outdoor** – Air taken from outdoors and, therefore, not previously circulated through the system.

**Air, Recirculated** – Return air passed through the conditioner before being again supplied to the conditioned space.

**Air, Reheating of** – In an air conditioning system, the final step in treatment, in the event the temperature or humidity is too low.

**Air, Return** – Air returned from conditioned or refrigerated space.

**Air, Saturated** – Moist air in which the partial pressure of the water vapor is equal to the vapor pressure of water at the existing temperature. This occurs when dry air and saturated water vapor coexist at the same dry-bulb temperature.

**Air, Standard Dry** – Dry air at or near sea level.

**Air Changes** – A method of expressing the amount of air leakage into or out of a building or room in terms of the number of building volumes or room volumes exchanged.

**Air Conditioner, Unitary** – An evaporator, compressor, and condenser combination; designed in one or more assemblies, the separate parts designed to be shipped or assembled together.

**Air Conditioning, Comfort** – The process of treating air so as to control its temperature, humidity, cleanliness and distribution to meet the comfort requirements of the occupants of the conditioned space.

**Air Conditioning Unit** – An assembly of equipment for the treatment of air so as to control its temperature, humidity, cleanliness and distribution to meet the requirements of a conditioned space.

**Air Diffuser** – A circular, square, or rectangular air distribution outlet, generally located in the ceiling and comprised of deflecting members discharging supply air in various directions and planes, and arranged to promote mixing of primary supply air with secondary room air.

**Air Washer** – A water spray system or device for cleaning, or humidifying the air.

**Airborne Sound** – Sound which reaches the point of interest by radiation through the air.

**Alternating Current (AC)** – A source of power for an electrical circuit which periodically reverses the polarity of its charge.

**Anemometer** – An instrument for measuring the velocity of a fluid.

**Anticipating Control** – One which, by artificial means, is activated sooner than it would be without such means, to produce a smaller differential of the controlled property. Heat and cool anticipators are commonly used in zone thermostats.

**Anticipators** – A small heater element in two-position temperature controllers which deliberately cause false indications of temperature in the controller in an attempt to minimize the differential and smooth out the temperature variation in the controlled space.

**Approach** – In an evaporative cooling device, the difference between the average temperatures of the circu-
lating water leaving the device and the average wet-bulb temperature of the entering air. In a conduction heat exchanger device, the temperature difference between the leaving treated fluid and the entering working fluid.

Aspect Ratio – In air distribution outlets, the ratio of the length of the core opening of a grille, face, or register to the width. In rectangular ducts, the ratio of the width to the depth.

Attenuation – The sound reduction process in which sound energy is absorbed or diminished in intensity as the result of energy conversion from sound to motion or heat.

B –

Background Noise – Sound other than the wanted signal. In room acoustics, the irreducible noise level measured in the absence of any building occupants.

Bimetallic Element – One formed of two metals having different coefficients of thermal expansion such as are used in temperature indicating and controlling devices.

Boiling Point – The temperature at which the vapor pressure of a liquid equals the absolute external pressure at the liquid-vapor interface.

Breakout Noise – The term used to describe the transmission or radiation of noise from some part of the duct system to an occupied space in the building.

British Thermal Unit (Btu) – The Btu is defined as the heat required to raise the temperature of a pound of water from 59°F to 60°F.

Bulb – The name given to the temperature sensing device located in the fluid for which control or indication is provided. The bulb may be liquid-filled, gas-filled, or gas-and-liquid filled. Changes in temperature produce pressure changes within the bulb which are transmitted to the controller or indicator.

Bypass – A pipe or duct, usually controlled by a valve or damper, for conveying a fluid around a component of a system.

C –

Calibration – Process of dividing and numbering the scale of an instrument; also of correcting or determining the error of an existing scale, or of evaluating one quantity in terms of readings of another.

Capillary Tube – The capillary tube is a metering device made from a small diameter tube approximately which feeds liquid directly to the evaporator. Usually limited to small systems to perform all of the functions of the thermal expansion valve.

Ceiling Outlet – A round, square, rectangular, or linear air diffuser located in the ceiling which provides a distribution pattern of primary and secondary air over the occupied zone and induces low velocity secondary air motion through the occupied zone.

Celsius – A thermometric scale in which the freezing point of water is called 0°C and its boiling point 100°C at normal atmospheric pressure.

Change of State – Change from one phase, such as solid, liquid or gas, to another.

Changeover – The process of switching an air conditioning system from heating to cooling, or vice versa.

Channel – Term used to describe output of a load management system. Usually corresponds to a specific relay.

Circuit – An electrical arrangement requiring a source of voltage, a closed loop of wiring, an electric load and some means for opening and closing it.

Circuit Breaker – A switch-type mechanism that opens automatically when it senses an overload in the form of excess current.

Coefficient of Discharge – For an air diffuser, the ratio of net area or effective area at vena contracta of an orificed airstream to the free area of the opening.

Coefficient of Performance (COP), Heat Pump – The ratio of the compressor heating effect (heat pump) to the rate of energy input to the shaft of the compressor, in consistent units, in a complete heat pump, under designated operating conditions.

Coil – A cooling or heating element made of pipe or tubing.

Cold Deck – The cooling section of a mixed air zoning system.

Comfort Cooling – Refrigeration for comfort as opposed to refrigeration for storage or manufacture.
Comfort Zone – (Average) the range of effective temperatures over which the majority (50 percent or more) of adults feels comfortable; (extreme) the range of effective temperatures over which one or more adults feel comfortable.

Compressibility – The ease which a fluid may be reduced in volume by the application of pressure, depends upon the state of the fluid as well as the type of fluid itself. In TAB work, consider that water may not be compressed. Air is a compressible gas, but that factor is usually not considered during normal testing and balancing procedures.

Compressor – The pump which provides the pressure differential to cause fluid to flow and in the pumping process increases pressure of the refrigerant to the high side condition. The compressor is the separation between low side and high side.

Condensate – The liquid formed by condensation of a vapor. In steam heating, water condensed from steam; in air conditioning, water extracted from air, as by condensation on the cooling coil of a refrigeration machine.

Condensation – Process of changing a vapor into liquid by extracting heat. Condensation of steam or water vapor is effected in either steam condensers or dehumidifying coils and the resulting water is called condensate.

Condenser – The heat exchanger in which the heat absorbed by the evaporator and some of the heat of compression introduced by the compressor are removed from the system. The gaseous refrigerant changes to a liquid, again taking advantage of the relatively large heat transfer by the change of state in the condensing process.

Conditions, Standard – A set of physical, chemical, or other parameters of a substance or system which defines an accepted reference state or forms a basis for comparison.

Control Diagram (ladder diagram) – A diagram that shows the control scheme only. Power wiring is not shown. The control items are shown between two vertical lines; hence, the name-ladder diagram.

Control Point – The value of the controlled variable which the controller operates to maintain.

Controlled Device – A device which receives the converted signal from the transmission system and translates it into the appropriate action in the environmental system. For example, a valve opens or closes to regulate fluid flow in the system.

Controller – An instrument which receives the signal from the sensing device and translates that signal into the appropriate corrective measure. The correction is then sent to the system controlled devices through the transmission system.

Convection – Transfer of heat by movement of fluid. Convection, Forced – Convection resulting from forced circulation of a fluid, as by a fan or pump.

Convection, Natural – Circulation of gas or liquid (usually air or water) due to differences in density resulting from temperature changes.

Cooling, Evaporative – Involves the adiabatic exchange of heat between air and water spray or a wetted surface. The water assumes the wet-bulb temperature of the air, which remains constant during its traverse of the exchanger.

Cooling, Regenerative – Process of utilizing heat which must be rejected or absorbed in one part of the cycle to function usefully in another part of the cycle by heat transfer.

Cooling Coil – An arrangement of pipes or tubing which transfers heat from the air to a contained fluid such as refrigerant, brine, or water.

Cooling Effect, Latent –

Cooling Effect, Total – Difference between the total cooling effect and the dehumidifying effect, expressed in BTUH or kWs.

Cooling Range – In a water cooling device, the difference between the average temperatures of the water entering and leaving the device.

Core Area – The total plane area of that portion of a grille, included within lines tangent to the outer edges of the openings through which air can pass.

Corresponding Values – Simultaneous values of various properties of a fluid, such as pressure, volume, temperature, etc., for a given condition of the fluid.
Corrosive – Having a chemically destructive effect on materials but, more often, the term is applied to metals.

Counterflow – In heat exchange between two fluids where each fluid flows in the opposite direction of the other.

Current (I) – The electric flow in an electric circuit, which is expressed in amperes (amps).

Cycle – A complete course of operation of working fluid back to a starting point, measured in thermodynamic terms (functions). Also, in general, any repeated process of any system.

– D –

Damper – A device used to vary the volume of air passing through an air outlet, air inlet or duct.

Deadband – In HVAC, a temperature range in which neither heating nor cooling is turned on; in load management, a kilowatt range in which loads are neither shed nor restored.

Decibel (dB) – A decibel is a division of a logarithmic scale for expressing the ratio of two quantities proportional to power or energy. The number of decibels denoting such a ratio is ten times the logarithm of the ratio.

Degree Day – A unit, based upon temperature difference and time, used in estimating fuel consumption and specifying nominal heating load of a building in winter. For any one day, when the mean temperature is less than 65°F, there exist as many degree days as there are Fahrenheit degrees difference in temperature between the mean temperature for the day and 65°F.

Dehumidification – The condensation of water vapor from air by cooling below the dewpoint or removal of water vapor from air by chemical or physical methods.

Dehumidifier – (1) An air cooler or washer used for lowering the moisture content of the air passing through it; (2) An absorption or adsorption device for removing moisture from air.

Demand Charge – That part of an electric bill based on kW demand and the demand interval, expressed in dollars per kilowatt. Demand charges offset construction and maintenance of a utility's need for a large generating capacity.

Demand Control – A device which controls the kW demand level by shedding loads when the kW demand exceeds a predetermined set point.

Demand Interval – The period of time during which kW demand is monitored by a utility service, usually 15 or 30 minutes long.

Demand Load – The actual amount of load on a circuit at any time. The sum of all the loads which are ON. Equal to the connected load minus the loads that are OFF.

Demand Reading – Highest or maximum demand for electricity an individual customer registers in a given interval, example, 15 minute interval. The metered demand reading sets the demand charge for the month.

Density – The ratio of the mass of a specimen of a substance to the volume of the specimen. The mass of a unit volume of a substance. When weight can be used without confusion, as synonymous with mass, density is the weight per unit volume.

Desiccant – Any absorbent or adsorbent, liquid or solid, that will remove water or water vapor from a material. In a refrigeration circuit, the desiccant should be insoluble in the refrigerant.

Design Working Pressure – The maximum allowable working pressure for which a specific part of a system is designed.

Dewpoint, Apparatus – That temperature which would result if the psychrometric process occurring in a dehumidifier, humidifier or surface-cooler were carried to the saturation condition of the leaving air while maintaining the same ratio of sensible to total heat load in the process.

Dew Point Depression – The difference between dry bulb and dew point temperatures (FDB - OF DP).

Dew Point Temperature – (tDP) The temperature at which moist air becomes saturated (100% relative humidity) with water vapor when cooled at constant pressure.

Differential – The difference between the points where a controller turns “on” and “off.” If a thermostat turns a furnace on at 68°F and the differential is 3°F, the burner will be turned off at 71°F.

Diffuse Sound Field – A diffuse sound field is a space in which at every point the flow of sound energy in all directions is equally probable. (It is often assumed that
in a diffuse field, the sound pressure level, averaged through time, is everywhere the same.)

**Diffuser** – A circular, square, or rectangular air distribution outlet, generally located in the ceiling and comprised of deflecting members discharging supply air in various directions and planes, and arranged to promote mixing of primary air with secondary room air.

**Direct Acting** – Instruments that increase control pressure as the controlled variable (such as temperature or pressure) increases; while *reverse acting* instruments increase control pressure as the controlled variable decreases.

**Direct Current (DC)** – A source of power for an electrical circuit which does not reverse the polarity of its charge.

**Discharge Stop Valve** – The manual service valve at the leaving connection of the compressor.

**Domestic Hot Water** – Potable hot water as distinguished from hot water used for heating.

**Draft** – A current of air, when referring to the pressure difference which causes a current of air or gases to flow through a flue, chimney, heater, or space; or to a localized effect caused by one or more factors of high air velocity, low ambient temperature, or direction of air flow, whereby more heat is withdrawn from a person’s skin than is normally dissipated.

**Drift** – As related to controls, term used to describe the difference between the set point and the actual operating or control points related to cooling towers, moisture droplets leaving cooling tower discharge and falling downwind.

**Drip** – A pipe, or a steam trap and a pipe considered as a unit, which conducts condensation from the steam side of a piping system to the water or return side of the system.

**Droop** – Terms used to describe the difference between the set point and the actual operating or control point.

**Drop** – The vertical distance that the lower edge of a horizontally projected airstream drops between the outlet and the end of its throw.

**Dry bulb, Room** – The dry bulb (dewpoint, etc.) temperature of the conditioned room or space.

**Dry Bulb Temperature** – The temperature registered by an ordinary thermometer. The dry bulb temperature represents the measure of sensible heat, or the intensity of heat.

**Dry Bulb Temperature, Adjusted (t"db)** – The average of the air temperature \( t_a \) and the mean radiant temperature \( t_r \) at a given location. The adjusted dry bulb temperature \( t_{adb} \) is approximately equivalent to operative temperature \( t_o \) at air motions less than 80 fpm when \( t_r \) is less than 120°F.

**Duct** – A passageway made of sheet metal or other suitable material, not necessarily leak tight, used for conveying air or other gas at low pressures.

**Dust** – An air suspension (aerosol) or particles of any solid material, usually with particle size less than 100 microns.

**Dynamic Discharge Head** – Static discharge head plus friction head plus velocity head.

**Dynamic Suction Head** – Positive static suction head minus friction head and minus velocity head.

**Dynamic Suction Lift** – The sum of suction lift and velocity head at the pump suction when the source is below pump centerline.

**E**

**Economizer** – A system of dampers, temperature and humidity sensors, and actuators which maximizes the use of outdoor air for cooling.

**Effect, Humidifying** – Latent heat of water vaporization at the average evaporating temperature times the number of pounds of water evaporated per hour in Btuh.

**Effect, Sun** – Solar energy transmitted into space through windows and building materials.

**Effect, Total Cooling** – The difference between the total enthalpy of the dry air and water vapor mixture entering a unit per hour and the total enthalpy of the dry air and water vapor (and water) mixture leaving the unit per hour, expressed in Btu per hour.

**Effective Area** – The net area of an outlet or inlet device through which air can pass, equal to the free area times the coefficient of discharge.

**Effectiveness (Efficiency)** – The ratio of the actual amount of heat transferred by a heat recovery device
to the maximum heat transfer possible between the air-streams (sensible heat/sensible heat, sensible heat/total heat, or total heat/total heat).

**Electrical Circuit** – A power supply, a load, and a path for current flow are the minimum requirements for an electrical circuit.

**Electromechanical** – Converting electrical input into mechanical action. A relay is an electromechanical switch.

**Electro-Pneumatic (EP) Switches** – Switches that open or close an air line valve from an electrical impulse.

**Energy** – Expressed in kilowatt-hours (kWh) or watt-hours (Wh), and is equal to the product of power and time.

\[
\text{energy} = \text{power} \times \text{time} \quad \text{kilowatt-hours} = \text{kilowatts} \times \text{hours} \quad \text{watt-hours} = \text{watts} \times \text{hours}
\]

**Energy (Consumption) Charge** – That part of an electric bill based on kWh consumption (expressed in cents per kWh). Energy charge covers cost of utility fuel, general operating costs, and part of the amortization of the utility’s equipment.

**Engine** – Prime mover; device for transforming fuel or heat energy into mechanical energy.

**Enthalpy** – The total quantity of heat energy contained in a substance, also called total heat; the thermodynamic property of a substance defined as the sum of its internal energy plus the quantity \( P v / J \), where \( P \) = pressure of the substance, \( v \) = its volume, and \( J \) = the mechanical equivalent of heat.

**Enthalpy, Specific** – A term sometimes applied to enthalpy per unit weight.

**Entrainment** – The capture of part of the surrounding air by the airstream discharged from an outlet (sometimes called secondary air motion).

**Entropy** – The ratio of the heat added to a substance to the absolute temperature at which it is added.

**Entropy, Specific** – A term sometimes applied to entropy per unit weight.

**Equal Friction Method** – A method of duct sizing wherein the selected duct friction loss value is used constantly throughout the design of a low pressure duct system.

**Equivalent Duct Diameter** – The equivalent of round duct diameter to rectangular duct.

**Evaporation** – Change of state from liquid to vapor. Evaporative Cooling – The adiabatic exchange of heat between air and a water spray or wetted surface. The water approaches the wet-bulb temperature of the air, which remains constant during its traverse of the exchanger.

**Evaporator** – The heat exchanger in which the medium being cooled, usually air or water, gives up heat to the refrigerant through the exchanger transfer surface. The liquid refrigerant boils into a gas in the process of the heat absorption.

**Extended Surface** – Heat transfer surface, one or both sides of which are increased in area by the addition of fins, discs, or other means.

**Face Area** – The total plane area of the portion of a grille, coil, or other items bounded by a line tangent to the outer edges of the openings through which air can pass.

**Face Velocity** – The velocity obtained by dividing the air quantity by the component face area.

**Fahrenheit** – A thermometric scale in which \( 32^\circ F \) denotes freezing and \( 212^\circ F \) the boiling point of water under normal pressure at sea level (14.696 psi).

**Fail Safe** – In load management, returning all loads to conventional control during a power failure. In electrical controls, accomplished by a relay whose contacts are normally closed. In pneumatics, the use of springs or similar permanent motive devices to move the controlled device to the desired position in the event of pneumatic control system failure or pressure loss.

**Fan, Centrifugal** – A fan rotor or wheel within a scroll type housing and including driving mechanism supports for either belt drive or direct connection.

**Fan Performance Curve** – Fan performance curve refers to the constant speed performance curve. This is a graphical presentation of static or total pressure and power input over a range of air volume flow rate at a stated inlet density and fan speed. It may include static and mechanical efficiency curves. The range of air
volume flow rate which is covered generally extends from shutoff (zero air volume flow rate) to free delivery (zero fan static pressure). The pressure curves are generally referred to as the pressure-volume curves.

**Fan Propeller** – A propeller or disc type wheel within a mounting ring or plate and including driving mechanism supports for either belt drive or direct connection.

**Fan, Tubaxial** – A propeller or disc type wheel within a cylinder and including driving mechanism supports for either belt drive or direct connection.

**Fan, Vanaxial** – A disc type wheel within a cylinder, a set of air guide vanes located either before or after the wheel and including driving mechanism supports for either belt drive or direct connection.

**Feed Line** – A pipe that supplies water to items such as a boiler or a domestic hot water tank.

**Filter** – A device to remove solid material from a fluid. **Filter-Drier** – A combination device used as a strainer and moisture remover.

**Fin** – An extended surface to increase the heat transfer area, as metal sheets attached to tubes.

**Flanking Transmission (Sound)** – The reduction in apparent transmission loss of a wall caused by sound being carried around the wall by other paths. (Structure-borne, leaks, etc.)

**Floating Action Controllers** – Essentially two position type controllers which vary the position of the controlled devices but which are arranged to stop before reaching a maximum or minimum position.

**Fluid** – Gas, vapor, or liquid.

**Fluid Head** – The static pressure of fluid expressed in terms of the height of a column of the fluid, or of some manometric fluid, which it would support.

**Fluid Statics** – Fluid Statics as applied to TAB work, refers to a condition of a quantity of fluid at rest. It is the direct result of gravity and weight. Static pressure is used in both air and water testing to determine the potential for the movement of fluid within a system. Pressures in air systems are normally measured in units of inches of water (in. wg). A pressure unit of one inch of water is equivalent to the static pressure found at the base of a column of water one inch high. Pressures in water systems are normally measured in pounds per square inch (psi), but are converted to feet of water (ft. wg) for the purpose of evaluating pump and equipment performance.

**Free Area** – The total minimum area of the openings in the air outlet or inlet through which air can pass.

**Freezing Point** – Temperature at which a given liquid substance will solidify or freeze on removal of heat. Freezing point of water is 32°F.

**Friction** – Friction is the resistance found at the duct and piping walls. Resistance creates a static pressure loss in systems. The primary purpose of a fan or pump is to produce a design volume of fluid at a pressure equal to the frictional resistance of the system and the other dynamic pressure losses of the components.

**Friction Head** – The pressure in psi or feet of the liquid pumped which represents system resistance that must be overcome.

**Full Load Current** – See Running Current.

**Fumes** – Solid particles commonly formed by the condensation of vapors from normally solid materials such as molten metals. Fumes may also be formed by sublimation, distillation, calcination, or chemical reaction wherever such processes create airborne particles predominantly below one micron in size. Such solid particles sometimes serve as condensation nuclei for water vapor to form smog.

**– G –**

**Gas** – Usually a highly superheated vapor which, within acceptable limits of accuracy, satisfies the perfect gas laws.

**Gradual Switches** – Manual adjustment devices which proportion the control condition in accordance with the position of the switch.

**Grains of Moisture** – The unit of measurement of actual moisture contained in a sample of air. (7000 grains = one pound of water).

**Gravity, Specific** – Density compared to density of standard material; reference usually to water or to air.

**Grille** – A louvered or perforated covering for an air passage opening which can be located on a wall, ceiling or floor.
Head, Static – The static pressure of fluid expressed in terms of the height of a column of the fluid, or of some manometric fluid, which it would support.

Head, Velocity – In a flowing fluid, the height of the fluid or of some manometric fluid equivalent to its velocity pressure.

Heat – The form of energy that is transferred by virtue of a temperature difference.

Heat, Latent – Change of enthalpy during a change of state, usually expressed in Btu per lb. With pure substances, latent heat is absorbed or rejected at constant pressure.

Heat, Sensible – Heat which is associated with a change in temperature; specific heat exchange of temperature; in contrast to a heat interchange in which a change of state (latent heat) occurs.

Heat, Specific – The ratio of the quantity of heat required to raise the temperature of a given mass of any substance one degree to the quantity required to raise the temperature of an equal mass of a standard substance (usually water at 59°F) one degree.

Heat, Total (Enthalpy) – The sum of sensible heat and latent heat between an arbitrary datum point and the temperature and state under consideration.

Heat Exchanger – A device specifically designed to transfer heat between two physically separated fluids.

Heat of Fusion – Latent heat involved in changing between the solid and the liquid states.

Heat of Vaporization – Latent heat involved in the change between liquid and vapor states.

Heat Transmission Coefficient – Anyone of a number of coefficients used in the calculation of heat transmission by conduction, convection, and radiation, through various materials and structures.

Heating, Regenerative (or Cooling) – Process of utilizing heat, which must be rejected or absorbed in one part of the cycle, to perform a useful function in another part of the cycle by heat transfer.

High Limit Control – A device which normally monitors the condition of the controlled medium and interrupts system operation if the monitored condition becomes excessive.

High Pressure Cutout – A pressure actuated switch to protect the compressor from pressure often caused by high condenser temperatures and pressure due to fouling and lack of water or air.

High Side – Parts of the refrigerating system subjected to condenser pressure or higher; the system from the compression side of the compressor through the condenser to the expansion point of the evaporator.

Horsepower – Unit of power in foot-pound-second system; work done at the rate of 550 ft-lb per sec, or 33,000 ft-lb per min.

Hot Deck – The heating section of a multizone system.

Hot Gas Bypass – The piping and manual, but more often automatic, valve used to introduce compressor discharge gas directly into the evaporator. This type of arrangement will maintain compressor operation at light loads down to zero by falsely loading the evaporator and compressor.

Hot Gas Piping – The compressor discharge piping which carries the hot refrigerant gas from the compressor to the condenser. Velocities must be high enough to carry entrained oil.

Humidifier – A device to add moisture to air. Humidifying Effect – The latent heat of vaporization of water at the average evaporating temperature times the weight of water evaporated per unit of time.

Humidistat – A regulatory device, actuated by changes in humidity, used for the automatic control of relative humidity.

Humidity – Water vapor within a given space. Humidity, Absolute – The weight of water vapor per unit volume.

Humidity, Percentage – The ratio of the specific humidity of humid air to that of saturated air at the same temperature and pressure, usually expressed as a percentage (degree of saturation; saturation ratio).

Humidity Ratio – The ratio of the mass of the water vapor to the mass of dry air contained in the sample.

Humidity, Relative – The ratio of the mol fraction of water vapor present in the air, to the mol fraction of
water vapor present in saturated air at the same temperature and barometric pressure; approximately, it equals the ratio of the partial pressure or density of the water vapor in the air, to the saturation pressure or density, respectively, of water vapor at the same temperature.

**Humidity, Specific** – Weight of water vapor (steam) associated with 1 lb. weight of dry air, also called humidity.

**Hunting** – A condition which occurs when the desired condition cannot be maintained. The controller, controlled device and system, individually or collectively, continuously override or “overshoot” the control point with a resulting fluctuation and loss of control of the condition to be maintained.

**Hydrostatic Pressure** – The pressure at any point in a liquid at rest; equal to the depth of the liquid multiplied by its density.

**Hygroscopic** – Absorptive of moisture, readily absorbing and retaining moisture.

**Inch of Water (in. wg)** – A unit of pressure equal to the pressure exerted by a column of liquid water 1 inch high at a temperature of 39.2°F.

**Indicator** – A term used to describe any device such as a thermometer or pressure gauge which is used to indicate the condition at a point in the system but which does not provide any controlling action or effect on the system operation.

**Inductance** – The process when a second conductor is placed next to a conductor carrying AC current (but not touching it), the ever changing magnetic field will induce a current in the second conductor.

**Induction** – The capture of part of the ambient air by the jet action of the primary airstream discharging from a controlled device.

**Inductive Loads** – Loads whose voltage and current are out-of-phase. True power consumption for inductive loads is calculated by multiplying its voltage, current, and the power factor of the load.

**Infiltration** – Air flowing inward as through a wall, crack, etc.

**Input Override Relay** – A relay that allows the duty cycle to be inhibited on specific channels because of inputs from outdoor temperature, space temperature, case temperature, time-of-day, etc. Sometimes called "duty cycle control relay."

**Inrush Current** – The current that flows the instant after the switch controlling current flow to a load is closed. Also called "locked rotor current."

**Insertion Loss** – The insertion loss of an element of an acoustic transmission system is the positive or negative change in acoustic power transmission that results when the element is introduced.

**Instantaneous Rate** – Method for determining when load shedding should occur. Actual energy usage is measured and compared to a present kilowatt level. If the actual kilowatt level exceeds a designated set point, loads will be shed until the actual rate drops below the set point.

**Insulation, Thermal** – A material having a relatively high resistance to heat flow and used principally to retard heat flow.

**Interstage Differential** – In a multistage HVAC system, the change in temperature at the thermostat needed to turn additional heating or cooling equipment on.

**Isentropic** – An adjective describing a reversible adiabatic process; a change taking place at constant entropy.

**Isobaric** – An adjective used to indicate a change taking place at constant pressure.

**Isothermal** – An adjective used to indicate a change taking place at constant temperature.

**Junction Box** – Metal box in which tap to circuit conductors is made. Junction box is not an outlet, since no load is fed from it directly.

**Kilovolt Ampere** – Product of the voltage times the current. Different from kilowatts because of inductive loads in an electrical system. Abbreviated – kVA kilowatts is equal to KVA times power factor.

**Kilowatt** – 1000 watts. Abbreviated – kW.

**Kilowatt-Hour** – A measure of electrical energy con-
kW Demand – The maximum rate of electric power usage required to operate a facility during a period of time, usually a month or billing period. Often called "demand".

kWh Consumption – The amount of electric energy used over a period of time; the number of kWh used per month. Often called "consumption".

Lag – A delay in the effect of a changed condition at one point in the system, on some other condition to which it is related. Also, the delay in action of the sensing element of a control, due to the time required for the sensing element to reach equilibrium with the property being controlled; i.e., temperature lag, flow lag, etc.

Latent Heat – The amount of heat necessary to change a quantity of water to water vapor without changing either temperature or pressure. When water is vaporized and passes into the air, the latent heat of vaporization passes into the air along with the vapor. Likewise, latent heat is removed when water vapor is condensed.

Law of Partial Pressure, Dalton’s – Each constituent of a mixture of gases behaves thermodynamically as if it alone occupied the space. The sum of the individual pressures of the constituents equals the total pressure of the mixture.

Light Emitting Diode – A low current and voltage light used as an indicator on load management equipment. Abbreviated – LED.

Limit – Control applied in the line or low voltage control circuit to break the circuit of conditions move outside a preset range. In a motor, a switch which cuts off power to the motor windings when the motor reaches its full open position.

Limit Control – A temperature, pressure, humidity, dew point or other control that overrides the demand control and/or duty cycler to prevent any affect on the business operation from load management, malfunction, or abnormal conditions. Also called “load override.”

Line Side – The side of a device electrically closest to the source of current.

Line Voltage – In the control industry, the normal electric supply voltages, which are usually 120 or 240 volts.

Liquefaction – A change of state to liquid; generally used instead of condensation in cases of substances ordinarily gaseous.

Liquid Sight Glass – The glass ported fitting in the liquid line used to indicate adequate refrigerant charge. When bubbles appear in the glass, there is insufficient refrigerant in the system.

Liquid Solenoid Valve – The electrically operated automatic shutoff valve in the liquid piping which closes on system shutdown to close off receiver discharge when used in pump down cycle and which prevents refrigerant migration in any system.

Load – In refrigeration systems, the amount of heat per unit time imposed on the refrigerant system or the required rate of heat removal. In electrical systems, the kVA imposed by the driven device or devices.

Load Factor – This is a ratio expressing a customer's average actual use of the utility's capacity provided versus the maximum amount used.

Load Management – The control of electrical loads to reduce kW demand and kWh consumption.

Load Programmer – Any device which turns loads on and off on a real time, time interval, or kW demand basis.

Load Side – The side of a device electrically farthest from the current source.

Locked Rotor Current – See “Inrush Current.”

Loudness – The subjective human definition of the intensity of a sound. Human reaction to sound is highly dependent on the sound pressure and frequency.

Loudness Level – A subjective method of rating loudness in which a 1000 Hz tone is varied in intensity until it is judged by listeners to be equally as loud as a given sound sample. The loudness level in “phons” is taken as the sound pressure level, in decibels, of the 1000 Hz tone.

Louver – An assembly of sloping vanes intended to permit air to pass through and to inhibit transfer of water droplets.

Low Limit Control – A device which normally monitors the condition of the controlled medium and inter-
rupts system operation if the monitored condition drops below the desired minimum value.

**Low Side** – The refrigerating system from the expansion point to the point where the refrigerant vapor is compressed; where the system is at or below evaporated pressure.

**Low Temperature Cutout** – A pressure or temperature actuated device with sensing element in the evaporator, which will shut the system down at its control setting to prevent freezing chilled water or to prevent coil frosting. Direct expansion equipment may not use this device.

**Low Voltage** – In the control industry, a power supply of 25 volts or less.

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**M**

**Manometer** – An instrument for measuring pressures – especially a U-tube partially filled with a liquid, usually water, mercury, or a light oil, so constructed that the amount of displacement of the liquid indicates the pressure being exerted on the instrument.

**Mass** – The quantity of matter in a body as measured by the ratio of the force required to produce given acceleration, to the acceleration.

**Master (Central) Control** – Control of all outlets from one point.

**Media** – The heat transfer material used in rotary heat exchangers, also referred to as *matrix*. In filters, the material that captures contaminates from a fluid stream.

**Melting Point** – For a given pressure, the temperature at which the solid and liquid phases of the substance are in equilibrium.

**Microbar** – A unit of pressure equal to 1 dyne/cm² (one millionth of the pressure of the atmosphere).

**Micron** – A unit of length, the thousandth part of 1 mm of the millionth of a meter.

**Microprocessor** – A small computer used in load management to analyze energy demand and consumption such that loads are turned on and off according to a predetermined program.

**Modulation** – Of a control, tending to adjust by increments and decrements.

**Modulating Control** – A mode of automatic control in which the action of the final control element is proportional to the deviation, from set point, of the controlled medium.

**Modulating Controllers** – Constantly reposition themselves in proportion to the requirements of the system, theoretically being able to maintain an accurately constant condition.

**Motor Control Center** – A single metal enclosed assembly containing a number of motor controllers and possibly other devices such as switches and control devices.

**Multipole** – Connects to more than 1 pole such as a 2-pole circuit breaker.

**Multistage Thermostat** – A thermostat which controls auxiliary equipment for heating or cooling in response to a greater demand for heating or cooling.

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**N**

**N.C.** – *Normally closed* contacts of a relay. Contacts are close-circuited when the relay is de-energized.

**N.O.** – *Normally open* contacts of a relay. Contacts are open-circuited when relay is deenergized.

**Neutral** – The circuit conductor that is normally grounded or at zero voltage difference to the ground.

**Noise** – Any undesired sounds, usually of different frequencies, resulting in an objectionable or irritating sensation.

**Normally open (or Normally closed)** – The position of a valve, damper, relay contacts, or switch when external power or pressure is not being applied to the device. Valves and dampers usually are returned to a "normal" position by a spring.

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**O**

**Offset** – Term used to describe the difference between the set point and the actual operating or control point. Also a value used in control software for calibration purposes, i.e. analog offset.

**Ohm (R) or (V)** – A measure of pure resistance in an electrical circuit.

**Ohm's Law** – The relationship between current and voltage in a circuit. It states that current is proportional to voltage and inversely proportional to resistance. Ex-
pressed algebraically, in DC circuits $I = E/R$; in AC circuits $I = E/Z$.

“On-off” Control – A two-position action that allows operation at either maximum or minimum condition, or on or off, depending on the position of the controller.

Open Circuit – The condition when either deliberately or accidentally, an electrical conductor or connection is broken or open with a switch.

Operating Point – The value of the controlled condition at which the controller actually operates. Also called control point.

Optimum Operative Temperature – Temperature that satisfies the greatest possible number of people at a given clothing and activity level.

Outlet, Ceiling – A round, square, rectangular, or linear air diffuser located in the ceiling, which provides a horizontal distribution pattern of primary and secondary air over the occupied zone and induces low velocity secondary air motion through the occupied zone.

Outlet, Slotted – A long, narrow air distribution outlet comprised of deflecting members, located in the ceiling sidewall, or sill, with an aspect ratio greater than 10, designed to distribute supply air in varying directions and planes, and arranged to promote mixing of primary air and secondary room air.

Outlet, Vaned – A register or grille equipped with vertical and/or horizontal adjustable vanes.

Outlet Velocity – The average velocity of air emerging from an opening, fan or outlet, measured in the plane of the opening.

Output – Capacity, duty, performance, net refrigeration produced by system.

Outside Air Opening – Any opening used as an entry for air from outdoors.

Overall Coefficient of Heat Transfer (thermal transmittance) – The time rate of heat flow through a body per unit area, under steady conditions, for a unit temperature difference between the fluids on the two sides of the body.

Overcurrent Device – A device such as a fuse or a circuit breaker designed to protect a circuit against excessive current by opening the circuit.

Overload – A condition of excess current; more current flowing than the circuit was designed to carry.

Override – A manual or automatic action taken to bypass the normal operation of a device or system.

Parallel Circuit – One where all the elements are connected across the voltage source. Therefore, the voltage on each element is the same but the current through each may be different.

Peak Demand – The greatest amount of kilowatts needed during a demand interval.

Peak Load Pricing – A pricing principle that charges more for purchases that contribute to the peak demand and, thereby, cause the expansion of productive capacity when the peak demand exceeds the peak capacity (less minimum excess capacity). In the electric power industry, this means charging more for electricity bought on or near the seasonal peak of the utility or on or near the daily peak of the utility. The latter requires special meters; the former does not.

Performance Factor – Ratio of the useful output capacity of a system to the input required to obtain it. Units of capacity and input need not be consistent.

Phase – Part of an AC voltage cycle. Residential electrical service is 2-phase; commercial facilities are usually 3-phase AC voltage.

Pilot Duty Relay – A relay used for switching loads such as another relay or solenoid valve coils. The pilot duty relay contacts are located in a second control circuit. Pilot duty relays are rated in volt-amperes (VA).

Plenum Chamber – An air compartment connected to one or more distributing ducts.

Pneumatic – Operated by air pressure. Pneumatic-Electric (PE) Switches – Device that operates an electric switch from a change of air pressure.

Point of Operation – Used to designate the single set fan performance values which correspond to the point of intersection of the system curve and the fan pressure-volume curve.

Point of Rating – A statement of fan performance values which correspond to one specific point on the fan pressure-volume curve.
**Polarity** – The direction of current flow in a DC circuit. By convention, current flows from plus to minus. Electron flow is actually in the opposite direction.

**Pole** – An electrical connection point. In a panel, the point of connection. On a device, the terminal that connects to the power.

**Potable** – Water that is safe to drink.

**Potentiometer** – An electromechanical device having a terminal connected to each point and to the resistive element, and a third terminal connected to the wiper contact. The electrical input is divided as the contact moves over the element, thus making it possible to mechanically change the resistance.

**Power (P)** – Expressed in watts (W) or kilowatts (kW), and is equal to – in DC circuits, $P = EI$ and $P = I^2R$, in AC circuits, $P = E*I*power factor$.

**Power Factor (pf)** – A quantity that relates the voltamperes of an AC circuit to the wattage (power = volt-amperes x power factor). Power factor also is the ratio of the circuit resistance (R) to the impedance (Z), expressed as a decimal between zero and one (p.f. = $R/Z$). When the power factor equals one, all consumed power produces useful work.

**Power Factor Charge** – A utility charge for “poor” power factor. It is more expensive to provide power to a facility with a poor power factor (usually less than 0.8).

**Power Factor Correction** – Installing capacitors on the utility service's supply line to improve the power factor of the building.

**Power Supply** – The voltage and current source for an electrical circuit. A battery, a utility service, and a transformer are power supplies.

**Predicting Method** – Method for determining when load shedding should occur. A formula is used to arrive at a preset kilowatt limit. Then the actual amount of energy accumulated during the utility's demand intervals is measured. A projection is made of the actual rate of energy usage during the rest of the interval. If the predicted value exceeds the preset limit, loads will be shed.

**Preheating** – In air conditioning, to heat the air ahead of other processes.

**Pressure** – The normal force exerted by a homogeneous liquid or gas, per unit of area, on the wall of its container.

**Pressure, Absolute** – Pressure referred to that of a perfect vacuum. It is the sum of gauge pressure and atmospheric pressure.

**Pressure, Atmospheric** – It is the pressure indicated by a barometer. Standard atmosphere is the pressure equivalent to 14.696 psi or 29.921 in. of mercury at 32°F.

**Pressure, Critical** – Vapor pressure corresponding to the substance's critical state at which the liquid and vapor have identical properties.

**Pressure, Gage** – Pressure above atmospheric. Pressure, Hydrostatic – The normal force per unit area that would be exerted by a moving fluid on an infinitesimally small body immersed in it if the body were carried along with the fluid.

**Pressure, Static (SP)** – The normal force per unit area that would be exerted by a moving fluid on a small body immersed in it if the body were carried along with the fluid. Practically, it is the normal force per unit area at a small hole in a wall of the duct through which the fluid flows (piezometer) or on the surface of a stationary tube at a point where the disturbances, created by inserting the tube, cancel. It is supposed that the thermodynamic properties of a moving fluid depend on static pressure in exactly the same manner as those of the same fluid at rest depend upon its uniform hydrostatic pressure.

**Pressure, Total (TP)** – In the theory of the flow of fluids, the sum of the static pressure and the velocity pressure at the point of measurement. Also called dynamic pressure.

**Pressure, Vapor** – The partial pressure exerted by the water vapor contained in air.

**Pressure, Velocity (Vp)** – In moving fluid, the pressure capable of causing an equivalent velocity, if applied to move the same fluid through an orifice such that all pressure energy expended is converted into kinetic energy.

**Pressure Drop** – Pressure loss in fluid pressure, as from one end of a duct to the other, due to friction, dynamic losses, and changes in velocity pressure.

**Pressure Regulator** – Automatic valve between the evaporator outlet and compressor inlet that is respons-
ive to pressure or temperature; it functions to throttle the vapor flow when necessary to prevent the evaporator pressure from falling below a preset level. Also used with pneumatic systems to control system pressure.

**Primary Air** – The initial airstream discharged by an air outlet (the air being supplied by a fan or supply duct) prior to any entrainment of the ambient air.

**Primary Control** – A device which directly or indirectly controls the control agent in response to needs indicated by the controller. Typically a motor, valve, relay, etc.

**Primary Element** – The portion of the controller which first uses energy derived from the controlled medium to produce a condition representing the value of the controlled variable; for example, a thermostat bimetal.

**“Process” Hot Water** – Hot water needed for manufacturing processes over and above the “domestic hot water” that is for the personal use of industrial workers.

**Properties, Thermodynamic** – Basic qualities used in defining the condition of a substance, such as temperature, pressure, volume, enthalpy, entropy.

**Proportional Band** – The range of values of a proportional positioning controller through which the controlled variable must pass to move the final control element through its full operating range. Commonly used equivalents are “throttling range” and “modulating range.”

**Proportional Control** – See Modulating Control.

**Psychrometer** – An instrument for ascertaining the humidity or hygrometric state of the atmosphere.

**Psychrometric Chart** – A graphical representation of the thermodynamic properties of moist air.

**Pulsing Demand Meter** – A meter which generates a pulse in correspondence with each revolution of a kWh meter. Pulses are recorded on paper or magnetic tape. Pulse can also be the signal to demand control equipment.

**Pyrometer** – An instrument for measuring high temperatures.

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**Radiation, Thermal** – The transmission of heat through space by wave motion; the passage of heat from one object to another without warming the space between.

**Radius of Diffusion** – The horizontal axial distance an airstream travels after leaving an air outlet before the maximum stream velocity is reduced to a specified terminal level.

**Random Incidence** – If an object is in a diffuse sound field, the sound is said to strike the object at random incidence.

**Refrigerant** – The fluid used for heat transfer in a refrigerating system, which absorbs heat at a low temperature and a low pressure of the fluid and rejects heat at a higher temperature and a higher pressure of the fluid, usually involving changes of state of the fluid.

**Register** – A grille equipped with an integral damper or control valve.

**Relative Humidity (RH)** – The ratio of water vapor in the air as compared to the maximum amount of water vapor that may be contained.

**Relay** – An electromechanical switch that opens or closes contacts in response to some controlled action. Relay contacts can be normally open (N.O.) and/or normally closed (N.C.). Relays may be electric, pneumatic, or a combination of both. PE and EP switches are relays.

**Relay, Thermal** – A switching relay in which a small heater warms a bimetal element which bends to provide the switching force.

**Remote Temperature Set Point** – Ability to set a temperature control point for a location outside the space, often used in public areas.

**Reset** – A process of automatically adjusting the control point of a given controller to compensate for changes in outdoor temperature. The hot deck control point is normally reset upward as the outdoor temperature drops. The cold deck control point is normally reset downward as the outdoor temperature increases.

**Reset Controllers** – Two controllers operating together; one sensing a condition other than that of the controlled space and changing the set point of the second controller, which is directly responsible for the result in the controlled space. The resetting controller
is commonly called the master, and the controller being reset is commonly called the submaster (slave).

**Reset Ratio** – The ratio of change in outdoor temperature to the change in control point temperature. For example, a 2:1 reset ratio means that the control point will increase 1 degree for every 2 degrees change in outdoor temperature.

**Resistance (Ω)** – The opposition which limits the amount of current that can be produced by an applied voltage in an electrical circuit, measured in ohms.

**Resistive Loads** – Electrical loads whose power factor is one. Usually contain heating elements.

**Return Air** – Air returned from conditioned or refrigerated space.

**Riser Diagram** – Electrical block-type diagram showing connection of major items of equipment. It is also applied to signal equipment connections. Also, generally applied to multistory buildings for vertical hydronic piping and ductwork.

**Riser Shaft** – A vertical shaft designed to house electric cables, piping and ductwork.

**Room Dry Bulb** – The actual temperature of the conditioned room or space as measured with an accurate thermometer.

**Running Current** – The current that flows through a load after inrush current has stabilized. Usually referred to as “full load current.”

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**Secondary Air** – The air surrounding an outlet that is captured or entrained by the initial outlet discharge airstream (furnished by a supply duct or fan).

**Sensible Heat** – Sensible heat is any heat transfer that causes a change in temperature. Heating and cooling of air and water that may be measured with a thermometer is sensible heat. Heating or cooling coils that simply increase or decrease the air temperature without a change in moisture content are examples of sensible heat.

**Sensible Heat Factor** – The ratio of sensible heat to total heat.

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**Sensible Heat Ratio, Air Cooler** – The ratio of sensible cooling effect to total cooling effect of an air cooler.

**Sensing Device** – A device that keeps track of the measured condition and its fluctuations so that when sufficient variation occurs it will originate the signal to revise the operation of the system and offset the change. Example – a thermostat “bulb.” A sensing device may be an integral part of a controller.

**Sensing Element** – The first system element or group of elements. The sensing element performs the initial measurement operation.

**Sensitivity** – The ability of a control instrument to measure and act upon variations of the measured condition.

**Sensor** – A sensing element.

**Sequencer** – A mechanical or electrical device that may be set to initiate a series of events and to make the events follow in sequence.

**Sequencing Control** – A control which energizes successive stages of heating or cooling equipment as its sensor detects the need for increased heating or cooling capacity. May be electronic or electromechanical.

**Series Circuit** – One with all the elements connected end to end. The current is the same throughout but the voltage can be different across each element.

**Set Point** – The value of the controlled condition at which the instrument is set to operate.

**Shed** – To de-energize a load in order to maintain a kW demand set point.

**Shed Mode** – A method of demand control that reduces kW demand through shedding and restoring loads.

**Shielded Cable** – Special cable used with equipment that generates a low voltage output. Used to minimize the effects of frequency “noise” on the output signal.

**Short Cycling** – Unit runs and then stops at short intervals; generally this excessive cycling rate is hard on the system equipment.

**Single-phasing** – The condition when one phase of a multi-phase (polyphase) motor circuit is broken or opened. Motors running when this occurs may continue to run but with lower power output and overheating.
Solenoid Air Valves – EP switches with an electromagnetic coil in the valve topworks that opens or closes the valve from normal position. A spring returns the valve to the normal position when the coil is de-energized.

Specific Heat – Specific heat \( (C_p) \) is the amount of heat energy in Btu's required to raise the temperature of one pound of substance one degree Fahrenheit. The following are specific heat values at standard conditions –

\[
\text{water-}C_p = 1.00 \text{ Btu/lb/°F} \\
\text{air-}C_p = 0.24 \text{ Btu/lb/°F}
\]

Using these values in simple equations, gallons per minute or cubic feet per minute may be determined in a system if the Btu per hour and the temperature difference are known.

Specific Volume – The reciprocal of density and is used to determine the cubic feet of volume, if the pounds of weight are known. Both density and specific volume are affected by temperature and pressure. The specific volume of air under standard conditions is 13.33 cubic feet per pound and the specific volume of water at standard conditions is 0.016 cubic feet per pound.

Spread – The divergence of the airstream in a horizontal or vertical plane after it leaves the outlet.

Stage Differential – Change in temperature at the thermostat needed to turn heating or cooling equipment off once it is turned on.

Staging Interval – The minimum time period for shedding or restoring two sequential loads.

Standard Air Density \((d)\) – Standard air density has been set at 0.075 lb/cu. ft. This corresponds approximately to dry air at 70°F and 29.92 in. Hg. In metric units, the standard air density is 1.2041 kg/m³ at 20°C and at 101.325 kPa.

Standard Conditions – The standard conditions referred to in environmental system work for air are – dry air at 70°F, and at an atmospheric pressure of 29.92 inches mercury (in. Hg.). For water, standard conditions are 68°F at the same barometric pressure. At these standard conditions, the density of air is 0.075 pounds per cubic foot and the density of water is 62.4 pounds per cubic foot.

Standard Rating – A rating based on tests performed at Standard Rating Conditions

Starter – Basic contactor with motor overloads, etc., added-a motor starter is an adaptation of the basic contactor which includes overload relays. Starters for large motors may include reactors, step resistors, disconnects, or other features required in a more sophisticated starter package.

State – Refers to the form of a fluid, either liquid, gas or solid. Liquids used in environmental systems are water, thermal fluids such as ethylene glycol solutions, and refrigerants in the liquid state. Gases are steam, evaporated refrigerants and the air-water vapor mixture found in the atmosphere. Some substances, including commonly used refrigerants, may exist in any of three states. A simple example is water, which may be solid (ice), liquid (water), or gas (steam or water vapor).

Static Head – The pressure due to the weight of a fluid above the point of measurement.

Static Regain Method – A method of duct sizing wherein the duct velocities are systematically reduced, allowing a portion of the velocity pressure to convert to static pressure offsetting the duct friction losses.

Static Suction Head – The positive vertical height in feet from the pump centerline to the top of the level of the liquid source.

Static Suction Lift – The distance in feet between the pump centerline and the source of liquid below the pump centerline.

Step Controller – See Sequencer.

Stratified Air – Unmixed air in a duct that is in thermal layers that have temperature variations of more than five degrees.

Structure-Borne Noise – A condition when the sound waves are being carried by a solid material. Sound waves in this state are inaudible to the human ear, since they cannot carry energy to it. Airborne noise can be created from the radiation of structureborne noise into the air. Structure-borne noise may be propagated by shear waves, tension-compression waves, bending waves, or complicated combination of waves.

Subcooling – The difference between the temperature of a pure condensable fluid below saturation and the temperature at the liquid saturated state, at the same pressure.
Sublimation – A change of state directly from solid to gas without appearance of liquid.

Suction Head – The positive pressure on the pump inlet when the source of liquid supply is above the pump centerline.

Suction Lift – The combination of static suction lift and friction head in the suction piping when the source of liquid is below the pump centerline.

Suction Piping – The piping which returns gaseous refrigerant to the compressor. Sizes must be large enough to maintain minimum friction to prevent reduced compressor and system capacity but must be small enough to produce adequate velocity to return oil to the compressor.

Sun Effect – Solar energy transmitted into space through windows and building materials.

Superheat – The difference between the temperature of a pure condensable fluid above saturation and the temperature at the dry saturated state, at the same pressure.

Switching Relays – Relays are devices which operate by a variation in the conditions of one electrical circuit to affect the operation of devices in the same or another circuit. General purpose switching relays are used to increase switching capability and isolate electrical circuits, such as in systems where the heating and cooling equipment have separate power supplies, and provide electrical interlocks within the system.

System – A series of ducts, conduits, elbows, branch piping, etc. designed to guide the flow of air, gas or vapor to and from one or more locations. A fan provides the necessary energy to overcome the resistance to flow of the system and causes air or gas to flow through the system. Some components of a typical system are louvers, grilles, diffusers, filters, heating and cooling coils, energy recovery devices, burner assemblies, volume dampers, mixing boxes, sound attenuators, the ductwork and related fittings.

System, Central Fan – A mechanical, indirect system of heating, ventilating, or air conditioning, in which the air is treated or handled by equipment located outside the rooms served, usually at a central location, and conveyed to and from the rooms by means of a fan and a system of distributing ducts.

System, Closed – A heating or refrigerating piping system in which circulating water or brine is completely enclosed, under pressure above atmospheric, and shut off from the atmosphere except for an expansion tank.

System, Duct – A series of ducts, conduits, elbows, branch piping, etc. designed to guide the flow of air, gas or vapor to and from one or more locations. A fan provides the necessary energy to overcome the resistance to flow of the system and causes air or gas to flow through the system. Some components of a typical system are louvers, grilles, diffusers, filters, heating and cooling coils, energy recovery devices, burner assemblies, volume dampers, mixing boxes, sound attenuators, the ductwork and related fittings.

System, Gravity Circulation – A heating or refrigerating system in which the heating or cooling fluid circulation is effected by the motive head due to difference in densities of cooler and warmer fluids in the two sides of the system.

System, Run-Around – A regenerative-type, closed, secondary system in which continuously circulated fluid abstracts heat from the primary system fluid at one place, returning this heat to the primary system fluid at another place.

System, Unitary – A complete, factory-assembled and factory-tested refrigerating system comprising one or more assemblies which may be shipped as one unit or separately but which are designed to be used together.

System Effect Factor – A pressure loss factor which recognizes the effect of fan inlet restrictions, fan outlet restrictions, or other conditions influencing fan performance when installed in the system.

– T –

Temperature, Absolute Zero – The zero point on the absolute temperature scale, 459.69 degrees below the zero of the Fahrenheit scale, 273.16 degrees below the zero of the Celsius scale.

Temperature, Critical – The saturation temperature corresponding to the critical state of the substance at which the properties of the liquid and vapor are identical.

Temperature, Dewpoint – The temperature at which the condensation of water vapor in a space begins for a given state of humidity and pressure as the temperature of the vapor is reduced. The temperature corresponding to saturation (100 percent relative humidity) for a given absolute humidity at constant pressure.
**Temperature, Drybulb** – The temperature of a gas or mixture of gases indicated by an accurate thermometer after correction for radiation.

**Temperature, Saturation** – The temperature at which no further moisture can be added to the airwater vapor mixture. Equals dew point temperature.

**Temperature, Wet-Bulb** – Thermodynamic wet bulb temperature is the temperature at which liquid or solid water, by evaporating into air, can bring the air to saturation adiabatically at the same temperature. Wet bulb temperature (without qualification) is the temperature indicated by a wet bulb psychrometer constructed and used according to specifications.

**Temperature, Wet Bulb Depression** – Difference between dry bulb and wet bulb temperatures.

**Temperature Difference, Mean** – Mean of difference between temperatures of a fluid receiving and a fluid yielding heat.

**Terminal Velocity** – The maximum airstream velocity at the end of the throw.

**Therm** – Measurement used by gas utilities for billing purposes. 1 Therm = 100 cubic feet of gas = 100,000 Btu.

**Thermal Expansion Valve** – The metering device or flow control which regulates the amount of liquid refrigerant which is allowed to enter the evaporator.

**Thermistor** – Semiconductor material that responds to temperature changes by changing its resistance.

**Thermocouple** – Device for measuring temperature utilizing the fact that an electromotive force is generated whenever two junctions of two dissimilar metals in an electric circuit are at different temperature levels.

**Thermodynamics, Laws of** – Two laws upon which rest the classical theory of thermodynamics. These laws have been stated in many different, but equivalent ways.

*The First Law* – (1) When work is expanded in generating heat, the quantity of heat produced is proportional to the work expended; and, conversely, when heat is employed in the performance of work, the quantity of heat which disappears is proportional to the work done (Joule); (2) If a system is caused to change from an initial state to a final state by adiabatic means only, the work done is the same for all adiabatic paths connecting the two states (Zemansky); (3) In any power cycle or refrigeration cycle, the net heat absorbed by the working substance is exactly equal to the net work done.

*The Second Law* – (1) It is impossible for a self acting machine, unaided by any external agency, to convey heat from a body of lower temperature to one of higher temperature (Clausius); (2) It is impossible to derive mechanical work from heat taken from a body unless there is available a body of lower temperature into which the residue not so used may be discharged (Kelvin); (3) It is impossible to construct an engine that, operating in a cycle, will produce no effect other than the extraction of heat from a reservoir and the performance of an equivalent amount of work (Zemansky).

**Throttling Range** – The amount of change in the variable being controlled to make the controlled device more through the full length of its stroke.

**Throw** – The horizontal or vertical axial distance an airstream travels after leaving an air outlet before the maximum stream velocity is reduced to a specified terminal level; e.g., 200, 150, 100 or 50 fpm.

**Total Dynamic Head** – Dynamic discharge head (static discharge head, plus friction head, plus velocity head) plus dynamic suction lift, or dynamic discharge head minus dynamic suction head.

**Total Heat (Enthalpy)** – Total heat is the sum of the sensible heat and latent heat in an exchange process. In many cases, the addition or subtraction of latent and sensible heat at terminal coils appears simultaneously. Total heat also is called *enthalpy*, both of which can be defined as the quantity of heat energy contained in that substance.

**Transducer** – The means by which the controller converts the signal from the sensing device into the means necessary to have the appropriate effect on the controlled device. For example, a change in air pressure in the pneumatic transmission piping.

**Transformer** – The system power supply-a transformer is an inductive stationary device which transfers electrical energy from one circuit to another. The transformer has two windings, primary and secondary. A changing voltage applied to one of these, usually the primary, induces a current to flow in the other winding. A coupling transformer transfers energy at the same voltage; a step-down transformer transfers energy at a lower voltage, and a step-up transformer transfers energy at a higher voltage.
Transmission – The means by which a signal is moved from one point of origin to the point of action.

Transmittance, Thermal (U factor) – The time rate of heat flow per unit area under steady conditions from the fluid on the warm side of a barrier to the fluid on the cold side, per unit temperature difference between the two fluids.

– U –

Utility Service – The utility company. Also, the amount and configuration of voltage supplied by a utility company. There are four main types of commercial utility services – 208V AC wye, 480V AC wye, 240V AC delta, and 480V AC delta.

Utility Transformer – Primary and secondary coils of wire which reduce (step down) the utility supply voltage for use within a facility.

Unitary System – A room unit which performs part or all of the air conditioning functions. It may or may not be used with a central fan system.

Unloader – A device on or in a compressor for equalizing the high and low side pressures for a brief period during starting, in order to decrease the starting load on the motor; also a device for controlling compressor capacity by rendering one or more cylinders ineffective.

– V –

Vacuum Breaker – A device to prevent a suction in a water pipe.

Valve, Modulating – A valve which can be positioned anywhere between fully on and fully off to proportion the rate of flow in response to a modulating controller (see modulating control).

Valve, Two-Position – A valve which is either fully on or fully off with no positions between. Also called an “on-off valve.”

Vapor – A gas, particularly one near to equilibrium with the liquid phase of the substance and which does not follow the gas laws. Usually used instead of gas for a refrigerant, and, in general, for any gas below the critical temperature.

Vapor, Saturated – Vapor in equilibrium with its liquid; i.e., when the numbers per unit time of molecules passing in two directions through the surface dividing the two phases are equal.

Vapor, Superheated – Vapor at a temperature which is higher than the saturation temperature (i.e., boiling point) at the existing pressure.

Vapor, Water – Used commonly in air conditioning parlance to refer to steam in the atmosphere.

Vapor Barrier – A moisture-impervious layer applied to the surfaces enclosing a humid space to prevent moisture travel to a point where it may condense due to lower temperature.

Vapor Pressure – Vapor pressure denotes the lowest absolute pressure that a given liquid at a given temperature will remain liquid before evaporating into its gaseous form or state.

Velocity – A vector quantity which denotes, at once, the time rate and the direction of a linear motion.

Velocity Head – The pressure needed to accelerate the fluid being pumped.

Velocity Reduction Method – A method of duct sizing where arbitrary reductions are made in velocity after each branch or outlet.

Velocity, Outlet – The average discharge velocity of primary air being discharged from the outlet, normally measured in the plane of the opening.

Velocity, Room – The average, sustained, residual air velocity level in the occupied zone of the conditioned space; e.g., 65, 50, 35 fpm.

Velocity, Terminal – The highest sustained airstream velocity existing in the mixed air path at the end of the throw.

Ventilation – The process of supplying or removing air, by natural or mechanical means, to or from any space. Such air may or may not have been conditioned.

Viscosity – That property of semifluids, fluids, and gases by virtue of which they resist an instantaneous change of shape or arrangement of parts. It is the cause of fluid friction whenever adjacent layers of fluid move with relation to each other.

Volatility – Volatility, surface tension and capillary action of a fluid are incidental to environmental systems.

Volatility is the rapidity with which liquids evaporates extremely rapidly and therefore is highly volatile.

Voltage (E) – The electromotive force in an electrical circuit. The difference in potential between two unlike
charges in an electrical circuit is its voltage measured in "volts" (V).

**Voltage Drop** – The voltage drop around a circuit including wiring and loads must equal the supply voltage.

**Volume** – Cubic feet per pound of dry air in the airwater vapor mixture as used in psychrometrics.

**Volume, Specific** – The volume of a substance per unit mass; the reciprocal of density.

– W –

**Water Hammer** – Banging of pipes caused by the shock of closing valves.

**Watt (W)** – A measure of electric power equal to a current flow of one ampere under one volt of pressure; or one joule per second in SI units.

**Watt Transducer** – A device which converts a current signal into a proportional millivolt signal. Used to interface between current transformers and a load management panel.

**Wavelength** – The distance between two similar and successive points on an alternating wave. The wavelength is equal to the velocity of the propagation divided by the frequency of the alternations.

**Weight** – The amount of force a substance exerts under pull by the earth's gravitational field and that force is measured in pounds in the United States.

**Wet Bulb Temperature (WB)** – The temperature registered by a thermometer whose bulb is covered by a saturated wick and exposed to a current of rapidly moving air. The wet bulb temperature also represents the dew point temperature of the air, where the moisture of the air condenses on a cold surface.

**Wet bulb Depression** – Difference between dry bulb and wet bulb temperatures.

– Z –

**Zoning** – The practice of dividing a building into small sections for heating and cooling control. Each section is selected so that one thermostat can be used to determine its requirements.