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Air-Conditioning and Refrigeration

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9.1 Introduction

Air-Conditioning

Air-conditioning is a process that simultaneously conditions air; distributes it combined with the outdoor air to the conditioned space; and at the same time controls and maintains the required space's temperature, humidity, air movement, air cleanliness, sound level, and pressure differential within predetermined limits for the health and comfort of the occupants, for product processing, or both.

The acronym HVAC&R stands for heating, ventilating, air-conditioning, and refrigerating. The combination of these processes is equivalent to the functions performed by air-conditioning.

Because I-P units are widely used in the HVAC&R industry in the U.S., I-P units are used in this chapter. A table for converting I-P units to SI units is available in Appendix X of this handbook.

Air-Conditioning Systems

An *air-conditioning* or *HVAC&R system* consists of components and equipment arranged in sequential order to heat or cool, humidify or dehumidify, clean and purify, attenuate objectionable equipment noise, transport the conditioned outdoor air and recirculate air to the conditioned space, and control and maintain an indoor or enclosed environment at optimum energy use.

The types of buildings which the air-conditioning system serves can be classified as:

- Institutional buildings, such as hospitals and nursing homes
- Commercial buildings, such as offices, stores, and shopping centers

- Residential buildings, including single-family and multifamily low-rise buildings of three or fewer stories above grade
- Manufacturing buildings, which manufacture and store products

Types of Air-Conditioning Systems

In institutional, commercial, and residential buildings, air-conditioning systems are mainly for the occupants' health and comfort. They are often called *comfort air-conditioning systems*. In manufacturing buildings, air-conditioning systems are provided for product processing, or for the health and comfort of workers as well as processing, and are called *processing air-conditioning systems*.

Based on their size, construction, and operating characteristics, air-conditioning systems can be classified as the following.

Individual Room or Individual Systems. An individual air-conditioning system normally employs either a single, self-contained, packaged room air conditioner (installed in a window or through a wall) or separate indoor and outdoor units to serve an individual room, as shown in [Figure 9.1.1](#). “Self-contained, packaged” means factory assembled in one package and ready for use.

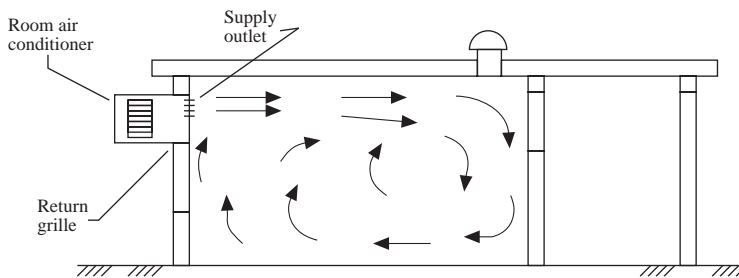


FIGURE 9.1.1 An individual room air-conditioning system.

Space-Conditioning Systems or Space Systems. These systems have their air-conditioning—cooling, heating, and filtration—performed predominantly in or above the conditioned space, as shown in [Figure 9.1.2](#). Outdoor air is supplied by a separate outdoor ventilation system.

Unitary Packaged Systems or Packaged Systems. These systems are installed with either a single self-contained, factory-assembled packaged unit (PU) or two split units: an indoor air handler, normally with ductwork, and an outdoor condensing unit with refrigeration compressor(s) and condenser, as shown in [Figure 9.1.3](#). In a packaged system, air is cooled mainly by direct expansion of refrigerant in coils called DX coils and heated by gas furnace, electric heating, or a heat pump effect, which is the reverse of a refrigeration cycle.

Central Hydronic or Central Systems. A central system uses chilled water or hot water from a central plant to cool and heat the air at the coils in an air handling unit (AHU) as shown in [Figure 9.1.4](#). For energy transport, the heat capacity of water is about 3400 times greater than that of air. Central systems are built-up systems assembled and installed on the site.

Packaged systems are comprised of only air system, refrigeration, heating, and control systems. Both central and space-conditioning systems consist of the following.

Air Systems. An air system is also called an air handling system or the air side of an air-conditioning or HVAC&R system. Its function is to condition the air, distribute it, and control the indoor environment according to requirements. The primary equipment in an air system is an AHU or air handler; both of these include fan, coils, filters, dampers, humidifiers (optional), supply and return ductwork, supply outlets and return inlets, and controls.

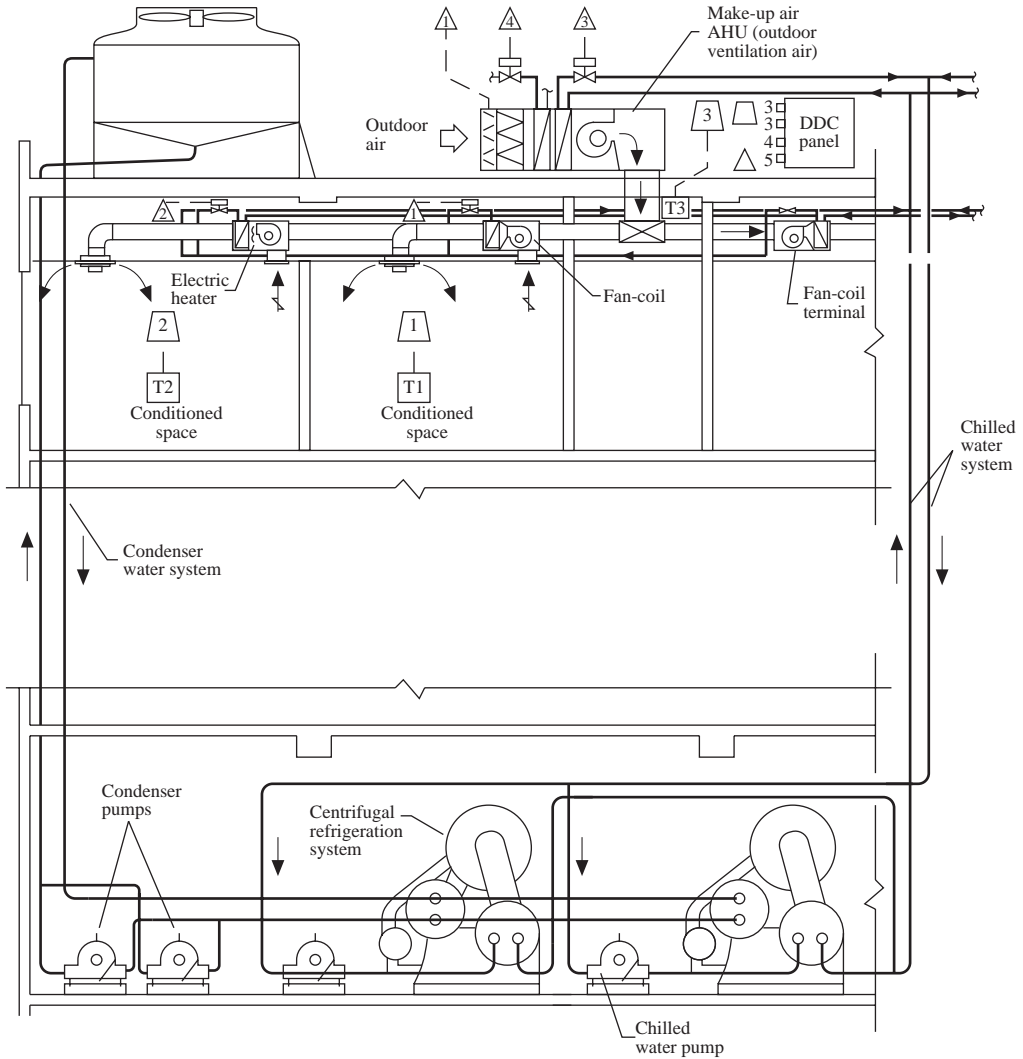


FIGURE 9.1.2 A space-conditioning air-conditioning system (fan-coil system).

Water Systems. These systems include chilled water, hot water, and condenser water systems. A water system consists of pumps, piping work, and accessories. The water system is sometimes called the water side of a central or space-conditioning system.

Central Plant Refrigeration and Heating Systems. The refrigeration system in the central plant of a central system is usually in the form of a chiller package with an outdoor condensing unit. The refrigeration system is also called the refrigeration side of a central system. A boiler and accessories make up the heating system in a central plant for a central system, and a direct-fired gas furnace is often the heating system in the air handler of a rooftop packaged system.

Control Systems. Control systems usually consist of sensors, a microprocessor-based direct digital controller (DDC), a control device, control elements, personal computer (PC), and communication network.

Based on Commercial Buildings Characteristics 1992, Energy Information Administration (EIA) of the Department of Energy of United States in 1992, for commercial buildings having a total floor area

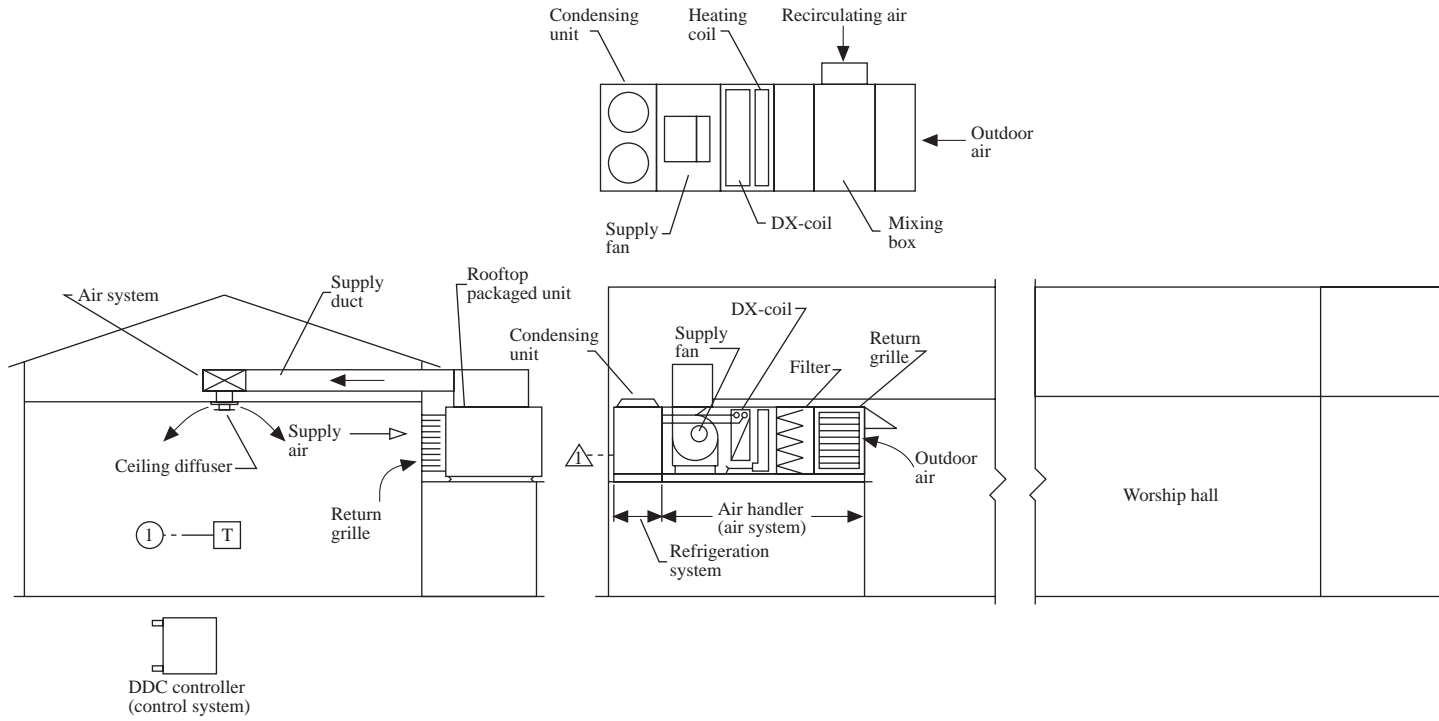


FIGURE 9.1.3 A packaged air-conditioning system.

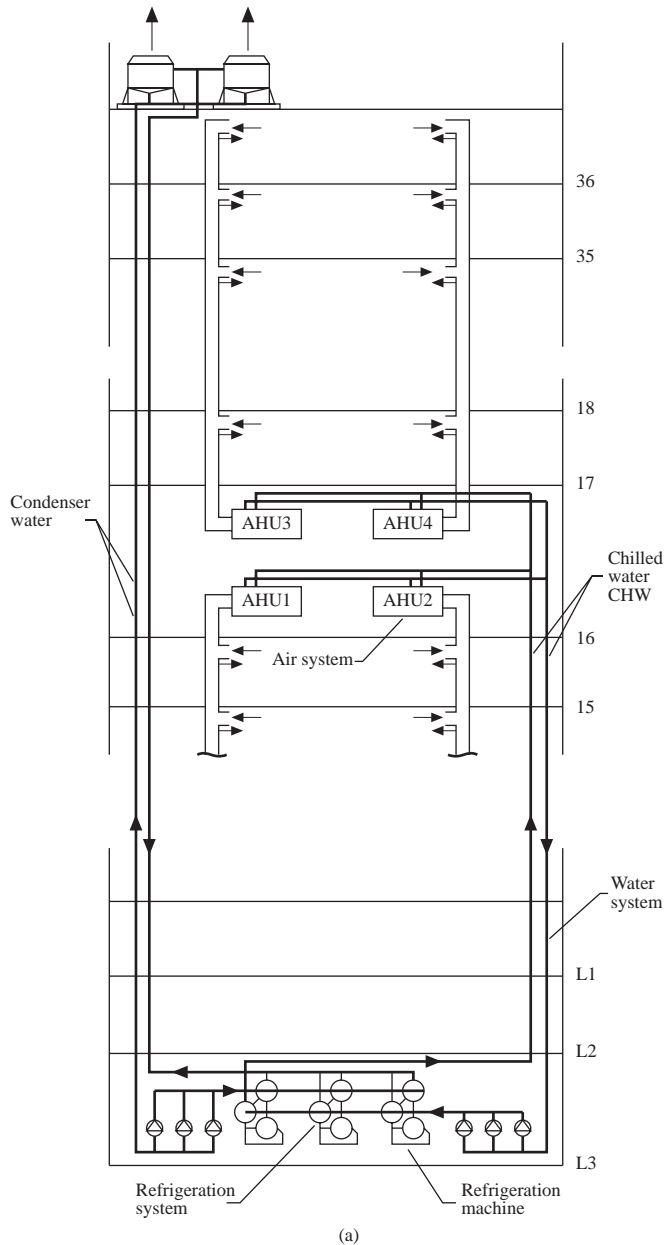


FIGURE 9.1.4a A central air-conditioning system: schematic diagram.

of 67,876 million ft², of which 57,041 million ft² or 84% is cooled and 61,996 million ft² or 91% is heated, the air-conditioning systems for cooling include:

Individual systems	19,239 million ft ²	(25%)
Packaged systems	34,753 million ft ²	(49%)
Central systems	14,048 million ft ²	(26%)

Space-conditioning systems are included in central systems. Part of the cooled floor area has been counted for both individual and packaged systems. The sum of the floor areas for these three systems therefore exceeds the total cooled area of 57,041 million ft².

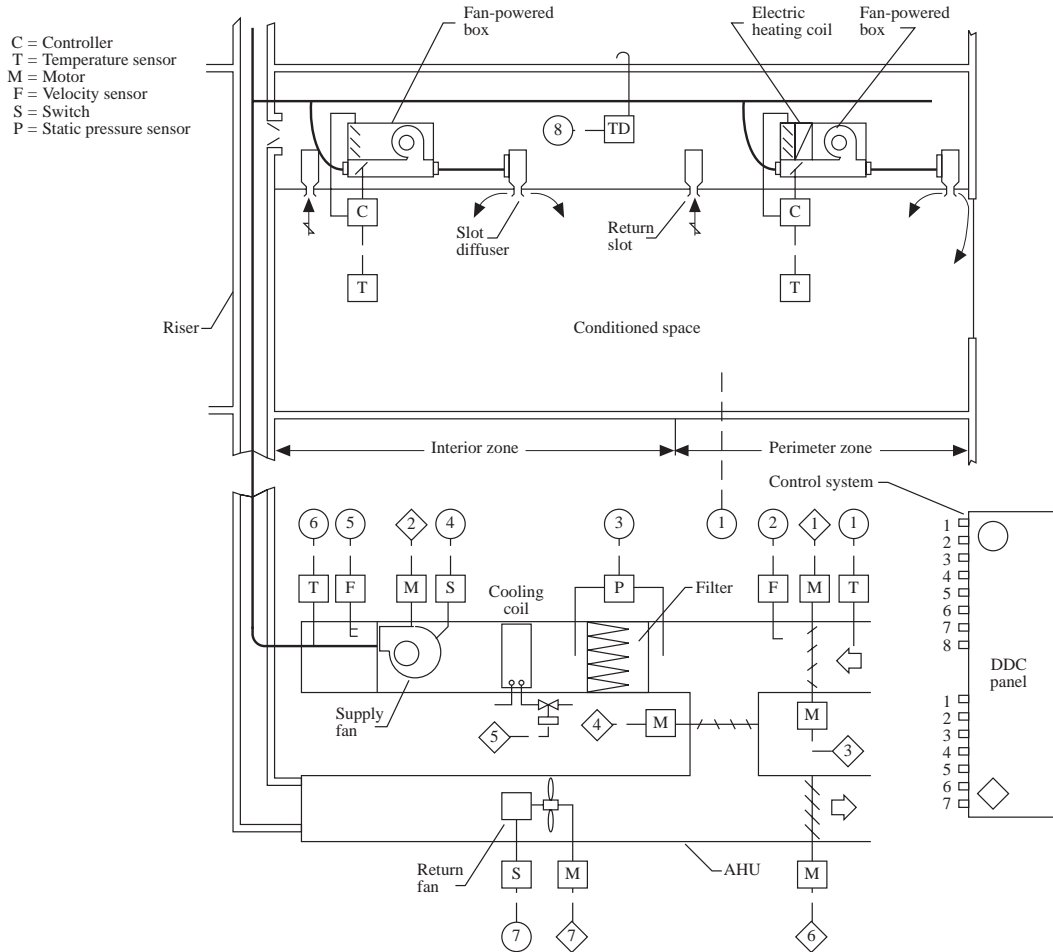


FIGURE 9.1.4b A central air-conditioning system: air and control systems for a typical floor.

Air-Conditioning Project Development and System Design

The goal of an air-conditioning/HVAC&R system is to provide a healthy and comfortable indoor environment with acceptable indoor air quality, while being energy efficient and cost effective.

ASHRAE Standard 62-1989 defines *acceptable indoor air quality* as “air in which there are no known contaminants at harmful concentrations as determined by cognizant authorities and with which a substantial majority (80% or more) of the people exposed do not express dissatisfaction.”

The basic steps in the development and use of an air-conditioning project are design, installation, commissioning, operation, and maintenance. There are two types of air-conditioning projects: *design-bid* and *design-build*. A design-bid project separates the design (engineering consultant) and installation (contractors) responsibilities. In a design-build project, the design is also done by the installation contractor. A design-build project is usually a small project or a project having insufficient time to go through normal bidding procedures.

In the building construction industry, air-conditioning or HVAC&R is one of the *mechanical services*; these also include plumbing, fire protection, and escalators.

Air-conditioning design is a process of selecting the optimum system, subsystem, equipment, and components from various alternatives and preparing the drawings and specifications. Haines (1994) summarized this process in four phases: gather information, develop alternatives, evaluate alternatives,

and sell the best solution. Design determines the basic operating characteristics of a system. After an air-conditioning system is designed and constructed, it is difficult and expensive to change its basic characteristics.

The foundation of a successful project is teamwork and coordination between designer, contractor, and operator and between mechanical engineer, electrical engineer, facility operator, architect, and structural engineer.

Field experience is helpful to the designer. Before beginning the design process it is advisable to visit similar projects that have operated for more than 2 years and talk with the operator to investigate actual performance.

Mechanical Engineer's Responsibilities

The normal procedure in a design-bid construction project and the mechanical engineer's responsibilities are

1. Initiation of a project by owner or developer
2. Organizing a design team
3. Determining the design criteria and indoor environmental parameters
4. Calculation of cooling and heating loads
5. Selection of systems, subsystems, and their components
6. Preparation of schematic layouts; sizing of piping and ductwork
7. Preparation of contract documents: drawings and specifications
8. Competitive biddings by various contractors; evaluation of bids; negotiations and modifications
9. Advice on awarding of contract
10. Monitoring, supervision, and inspection of installation; reviewing shop drawings
11. Supervision of commissioning
12. Modification of drawings to the as-built condition; preparation of the operation and maintenance manual
13. Handing over to the property management for operation

Design Documents

Drawings and *specifications* are legal documents of a construction contract. The designer conveys the owner's or developer's requirements to the contractor through these documents. Drawings and specifications complement each other.

Drawings should clearly and completely show, define, and present the work. Adequate plan and sectional views should be drawn. More often, isometric drawings are used to show the flow diagrams for water or the supply, return, and exhaust air.

Specifications include the legal contract between the owner and the contractor, installer, or vendor and the technical specifications, which describe in detail what kind of material and equipment should be used and how they are to be installed.

Most projects now use a format developed by the Construction Specifications Institute (CSI) called the Masterformat for Specifications. It includes 16 divisions. The 15000 Mechanical division is divided into the following:

Section No.	Title
15050	Basic Mechanical Materials and Methods
15250	Mechanical Insulation
15300	Fire Protection
15400	Plumbing
15500	Heating, Ventilating, and Air-Conditioning
15550	Heat Generation
15650	Refrigeration
15750	Heat Transfer
15850	Air Handling

Section No.	Title
15880	Air Distribution
15950	Controls
15990	Testing, Adjusting, and Balancing

Each section includes general considerations, equipment and material, and field installation. Design criteria and selected indoor environmental parameters that indicate the performance of the HVAC&R system must be clearly specified in the general consideration of Section 15500.

There are two types of specifications: the performance specification, which depends mainly on the required performance criteria, and the or-equal specification, which specifies the wanted vendor. Specifications should be written in simple, direct, and clear language without repetition.

Computer-Aided Design and Drafting

With the wide acceptance of the PC and the availability of numerous types of engineering software, the use of *computer-aided drafting* (CAD) and *computer-aided design and drafting* (CADD) has increased greatly in recent years. According to the 1994 CADD Application and User Survey of design firms reported in *Engineering Systems* (1994[6]), “15% of the design firms now have a computer on every desk” and “Firms with high productivity reported that they perform 95% on CADD.” Word processing software is widely used to prepare specifications.

Drafting software used to reproduce architectural drawings is the foundation of CADD. Automated CAD (AutoCAD) is the leading personal computer-based drafting tool software used in architectural and engineering design firms.

In “Software Review” by Amistadi (1993), duct design was the first HVAC&R application to be integrated with CAD.

- Carrier Corp. DuctLINK and Softdesk HVAC 12.0 are the two most widely used duct design software. Both of them convert the single-line duct layout drawn with CAD to two-dimensional (2D) double-line drawings with fittings, terminals, and diffusers.
- Tags and schedules of HVAC&R equipment, ductwork, and duct fittings can be produced as well.
- DuctLINK and Softdesk can also interface with architectural, electrical, and plumbing drawings through AutoCAD software.

Software for piping system design and analysis can also be integrated with CAD. The software developed at the University of Kentucky, KYCAD/KYPIPE, is intended for the design and diagnosis of large water piping systems, has extensive hydraulic modeling capacities, and is the most widely used. Softdesk AdCADD Piping is relative new software; it is intended for drafting in 2D and 3D, linking to AutoCAD through design information databases.

Currently, software for CADD for air-conditioning and HVAC&R falls into two categories: engineering and product. The engineering category includes CAD (AutoCAD integrated with duct and piping system), load calculations and energy analysis, etc. The most widely used software for load calculations and energy analysis is Department of Energy DOE-2.1D, Trane Company’s TRACE 600, and Carrier Corporation’s softwares for load calculation, E20-II Loads.

Product categories include selection, configuration, performance, price, and maintenance schedule. Product manufacturers provide software including data and CAD drawings for their specific product.

Codes and Standards

Codes are federal, state, or city laws that require the designer to perform the design without violating people’s (including occupants and the public) safety and welfare. Federal and local codes must be followed. The designer should be thoroughly familiar with relevant codes. HVAC&R design codes are definitive concerning structural and electrical safety, fire prevention and protection (particularly for gas- or oil-fired systems), environmental concerns, indoor air quality, and energy conservation.

Conformance with *ASHRAE Standards* is voluntary. However, for design criteria or performance that has not been covered in the codes, whether the ASHRAE Standard is followed or violated is the vital criterion, as was the case in a recent indoor air quality lawsuit against a designer and contractor.

For the purpose of performing an effective, energy-efficient, safe, and cost-effective air-conditioning system design, the following ASHRAE Standards should be referred to from time to time:

- ASHRAE/IES Standard 90.1-1989, Energy Efficient Design of New Buildings Except New Low-Rise Residential Buildings
- ANSI/ASHRAE Standard 62-1989, Ventilation for Acceptable Indoor Air Quality
- ANSI/ASHRAE Standard 55-1992, Thermal Environmental Conditions for Human Occupancy
- ASHRAE Standard 15-1992, Safety Code for Mechanical Refrigeration

9.2 Psychrometrics

Moist Air

Above the surface of the earth is a layer of air called the *atmosphere*, or *atmospheric air*. The lower atmosphere, or homosphere, is composed of moist air, that is, a mixture of dry air and water vapor.

Psychrometrics is the science of studying the thermodynamic properties of moist air. It is widely used to illustrate and analyze the change in properties and the thermal characteristics of the air-conditioning process and cycles.

The composition of dry air varies slightly at different geographic locations and from time to time. The approximate composition of dry air by volume is nitrogen, 79.08%; oxygen, 20.95%; argon, 0.93%; carbon dioxide, 0.03%; other gases (e.g., neon, sulfur dioxide), 0.01%.

The amount of water vapor contained in the moist air within the temperature range 0 to 100°F changes from 0.05 to 3% by mass. The variation of water vapor has a critical influence on the characteristics of moist air.

The equation of state for an ideal gas that describes the relationship between its thermodynamic properties covered in Chapter 2 is

$$pv = RT_R \quad (9.2.1)$$

or

$$pV = mRT_R \quad (9.2.2)$$

where p = pressure of the gas, psf (1 psf = 144 psi)
 v = specific volume of the gas, ft³/lb
 R = gas constant, ftlb_f/lb_m °R
 T_R = absolute temperature of the gas, °R
 V = volume of the gas, ft³
 m = mass of the gas, lb

The most exact calculation of the thermodynamic properties of moist air is based on the formulations recommended by Hyland and Wexler (1983) of the U.S. National Bureau of Standards. The psychrometric charts and tables developed by ASHRAE are calculated and plotted from these formulations. According to Nelson et al. (1986), at a temperature between 0 and 100°F, enthalpy and specific volume calculations using ideal gas Equations (9.2.1) and (9.2.2) show a maximum deviation of 0.5% from the results of Hyland and Wexler's exact formulations. Therefore, ideal gas equations are used in the development and calculation of psychrometric formulations in this handbook.

Although air contaminants may seriously affect human health, they have little effect on the thermodynamic properties of moist air. For thermal analysis, moist air may be treated as a binary mixture of dry air and water vapor.

Applying Dalton's law to moist air:

$$p_{at} = p_a + p_w \quad (9.2.3)$$

where p_{at} = atmospheric pressure of the moist air, psia
 p_a = partial pressure of dry air, psia
 p_w = partial pressure of water vapor, psia

Dalton's law is summarized from the experimental results and is more accurate at low gas pressure. Dalton's law can also be extended, as the Gibbs-Dalton law, to describe the relationship of internal energy, enthalpy, and entropy of the gaseous constituents in a mixture.

Humidity and Enthalpy

The *humidity ratio* of moist air, w , in lb/lb is defined as the ratio of the mass of the water vapor, m_w to the mass of dry air, m_a , or

$$w = m_w/m_a = 0.62198 p_w / (p_{at} - p_w) \quad (9.2.4)$$

The *relative humidity* of moist air, ϕ , or RH, is defined as the ratio of the mole fraction of water vapor, x_w , to the mole fraction of saturated moist air at the same temperature and pressure, x_{ws} . Using the ideal gas equations, this relationship can be expressed as:

$$\phi = x_w/x_{ws}|_{T,p} = p_w/p_{ws}|_{T,p} \quad (9.2.5)$$

and

$$x_w = n_w / (n_a + n_w); \quad x_{ws} = n_{ws} / (n_a + n_{ws})$$

$$x_a + x_w = 1 \quad (9.2.6)$$

where p_{ws} = pressure of saturated water vapor, psia

T = temperature, °F

n_a, n_w, n_{ws} = number of moles of dry air, water vapor, and saturated water vapor, mol

Degree of saturation μ is defined as the ratio of the humidity ratio of moist air, w , to the humidity ratio of saturated moist air, w_s , at the same temperature and pressure:

$$\mu = w/w_s|_{T,p} \quad (9.2.7)$$

The difference between ϕ and μ is small, usually less than 2%.

At constant pressure, the difference in specific enthalpy of an ideal gas, in Btu/lb, is $\Delta h = c_p \Delta T$. Here c_p represents the specific heat at constant pressure, in Btu/lb. For simplicity, the following assumptions are made during the calculation of the *enthalpy* of moist air:

1. At 0°F, the enthalpy of dry air is equal to zero.
2. All water vapor is vaporized at 0°F.
3. The enthalpy of saturated water vapor at 0°F is 1061 Btu/lb.
4. The unit of the enthalpy of the moist air is Btu per pound of dry air and the associated water vapor, or Btu/lb.

Then, within the temperature range 0 to 100°F, the enthalpy of the moist air can be calculated as:

$$h = c_{pd} T + w(h_{g0} + c_{ps} T)$$

$$= 0.240T + w(1061 + 0.444T) \quad (9.2.8)$$

where c_{pd}, c_{ps} = specific heat of dry air and water vapor at constant pressure, Btu/lb°F. Their mean values can be taken as 0.240 and 0.444 Btu/lb°F, respectively.

h_{g0} = specific enthalpy of saturated water vapor at 0°F.

Moist Volume, Density, Specific Heat, and Dew Point

The specific *moist volume* v , in ft^3/lb , is defined as the volume of the mixture of dry air and the associated water vapor when the mass of the dry air is exactly 1 lb:

$$v = V/m_a \quad (9.2.9)$$

where V = total volume of the moist air, ft^3 . Since moist air, dry air, and water vapor occupy the same volume,

$$v = R_a T_R / p_{\text{at}} (1 + 1.6078w) \quad (9.2.10)$$

where R_a = gas constant for dry air.

Moist air density, often called *air density* ρ , in lb/ft^3 , is defined as the ratio of the mass of dry air to the total volume of the mixture, or the reciprocal of the moist volume:

$$\rho = m_a / V = 1/v \quad (9.2.11)$$

The *sensible heat of moist air* is the thermal energy associated with the change of air temperature between two state points. In Equation (9.2.8), $(c_{\text{pd}} + wc_{\text{ps}})T$ indicates the sensible heat of moist air, which depends on its temperature T above the datum 0°F . *Latent heat of moist air*, often represented by $wh_{\text{fg}0}$, is the thermal energy associated with the change of state of water vapor. Both of them are in Btu/lb . Within the temperature range 0 to 100°F , if the average humidity ratio w is taken as 0.0075 lb/lb , the *specific heat of moist air* c_{pa} can be calculated as:

$$c_{\text{pa}} = c_{\text{pd}} + wc_{\text{ps}} = 0.240 + 0.0075 \times 0.444 = 0.243 \text{ Btu}/\text{lb } ^\circ\text{F} \quad (9.2.12)$$

The *dew point temperature* T_{dew} , in $^\circ\text{F}$, is the temperature of saturated moist air of the moist air sample having the same humidity ratio at the same atmospheric pressure. Two moist air samples of similar dew points T_{dew} at the same atmospheric pressure have the same humidity ratio w and the same partial pressure of water vapor p_w .

Thermodynamic Wet Bulb Temperature and Wet Bulb Temperature

The *thermodynamic wet bulb temperature* of moist air, T^* , is equal to the saturated state of a moist air sample at the end of a constant-pressure, ideal adiabatic saturation process:

$$h_1 + (w_s^* - w_1)h_w^* = h_s^* \quad (9.2.13)$$

where h_1, h_s^* = enthalpy of moist air at the initial state and enthalpy of saturated air at the end of the constant-pressure, ideal adiabatic saturation process, Btu/lb

w_1, w_s^* = humidity ratio of moist air at the initial state and humidity ratio of saturated air at the end of the constant-pressure, ideal adiabatic saturation process, lb/lb

h_w^* = enthalpy of water added to the adiabatic saturation process at temperature T^* , Btu/lb

An *ideal adiabatic saturation process* is a hypothetical process in which moist air at initial temperature T_1 , humidity ratio w_1 , enthalpy h_1 , and pressure p flows over a water surface of infinite length through a well-insulated channel. Liquid water is therefore evaporated into water vapor at the expense of the sensible heat of the moist air. The result is an increase of humidity ratio and a drop of temperature until the moist air is saturated at the thermodynamic wet bulb temperature T^* during the end of the ideal adiabatic saturation process.

The thermodynamic wet bulb temperature T^* is a unique fictitious property of moist air that depends only on its initial properties, T_1 , w_1 , or h_1 .

A sling-type *psychrometer*, as shown in Figure 9.2.1, is an instrument that determines the temperature, relative humidity, and thus the state of the moist air by measuring its dry bulb and wet bulb temperatures. It consists of two mercury-in-glass thermometers. The sensing bulb of one of them is dry and is called the dry bulb. Another sensing bulb is wrapped with a piece of cotton wick, one end of which dips into a water tube. This wetted sensing bulb is called the wet bulb and the temperature measured by it is called the *wet bulb temperature* T' .

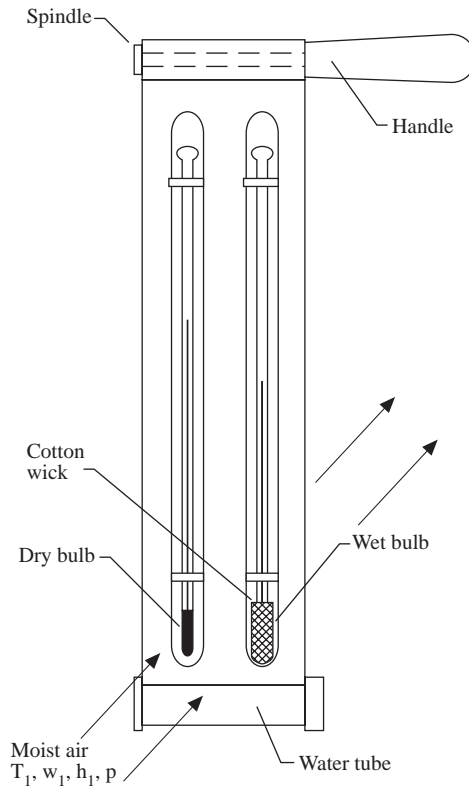


FIGURE 9.2.1 A sling psychrometer.

When unsaturated moist air flows over the surface of the wetted cotton wick, liquid water evaporates from its surface. As it absorbs sensible heat, mainly from the surrounding air, the wet bulb temperature drops. The difference between the dry and wet bulb temperatures is called *wet bulb depression* ($T - T'$). Turning the handle forces the surrounding air to flow over the dry and wet bulbs at an air velocity between 300 to 600 fpm. Distilled water must be used to wet the cotton wick.

At steady state, if heat conduction along the thermometer stems is neglected and the temperature of the wetted cotton wick is equal to the wet bulb temperature of the moist air, as the sensible heat transfer from the surrounding moist air to the cotton wick exactly equals the latent heat required for evaporation, the heat and mass transfer per unit area of the wet bulb surface can be evaluated as:

$$h_c(T - T') + h_r(T_{ra} - T) = h_d(w'_s - w_1) \quad (9.2.14)$$

where h_c , h_r = mean conductive and radiative heat transfer coefficient, Btu/hr ft²°F
 h_d = mean convective mass transfer coefficient, lb/hr ft²

- T = temperature of undisturbed moist air at a distance from the wet bulb, °F
 T_{ra} = mean radiant temperature (covered later), °F
 w_1, w'_s = humidity ratio of the moist air and the saturated film at the interface of cotton wick and surrounding air, lb/lb
 h'_{fg} = latent heat of vaporization at the wet bulb temperature, Btu/lb

The humidity ratio of the moist air is given by:

$$w_1 = w'_s - K'(T - T') \left(1 + \left\{ h_r (T_{ra} - T') / [h_c (T - T')] \right\} \right)$$

$$K' = c_{pa} Le^{0.6667} / h'_{fg} \quad (9.2.15)$$

where K' = wet bulb constant, which for a sling psychrometer = 0.00218 1/°F
 Le = Lewis number

The wet bulb temperature T' depends not only on its initial state but also on the rate of heat and mass transfer at the wet bulb. Therefore, the thermodynamic wet bulb temperature is used in ASHRAE psychrometric charts.

According to Threlkeld (1970), for a sling psychrometer whose wet bulb diameter is 1 in. and for air flowing at a velocity of 400 fpm over the wet bulb, if the dry bulb temperature is 90°F and the measured wet bulb temperature is 70°F, the difference between the measured wet bulb and the thermodynamic wet bulb $(T' - T^*) / (T^* - T')$ is less than 1%.

Psychrometric Charts

A *psychrometric chart* is a graphical presentation of the thermodynamic properties of moist air and various air-conditioning processes and air-conditioning cycles. A psychrometric chart also helps in calculating and analyzing the work and energy transfer of various air-conditioning processes and cycles.

Psychrometric charts currently use two kinds of basic coordinates:

1. h - w charts. In h - w charts, enthalpy h , representing energy, and humidity ratio w , representing mass, are the basic coordinates. Psychrometric charts published by ASHRAE and the Chartered Institution of Building Services Engineering (CIBSE) are h - w charts.
2. T - w charts. In T - w charts, temperature T and humidity ratio w are basic coordinates. Psychrometric charts published by Carrier Corporation, the Trane Company, etc. are T - w charts.

Figure 9.2.2 shows an abridged ASHRAE psychrometric chart. In the ASHRAE chart:

- A normal temperature chart has a temperature range of 32 to 120°F, a high-temperature chart 60 to 250°F, and a low-temperature chart -40 to 50°F. Since enthalpy is the basic coordinate, temperature lines are not parallel to each other. Only the 120°F line is truly vertical.
- Thermodynamic properties of moist air are affected by atmospheric pressure. The standard atmospheric pressure is 29.92 in. Hg at sea level. ASHRAE also published charts for high altitudes of 5000 ft, 24.89 in. Hg, and 7500 ft, 22.65 in. Hg. Both of them are in the normal temperature range.
- Enthalpy h -lines incline downward to the right-hand side (negative slope) at an angle of 23.5° to the horizontal line and have a range of 12 to 54 Btu/lb.
- Humidity ratio w -lines are horizontal lines. They range from 0 to 0.28 lb/lb.
- Relative humidity ϕ -lines are curves of relative humidity 10%, 20%, ... 90% and a saturation curve. A saturation curve is a curve of the locus of state points of saturated moist air, that is, $\phi = 100\%$. On a saturation curve, temperature T , thermodynamic wet temperature bulb T^* , and dew point temperature T_{dew} have the same value.

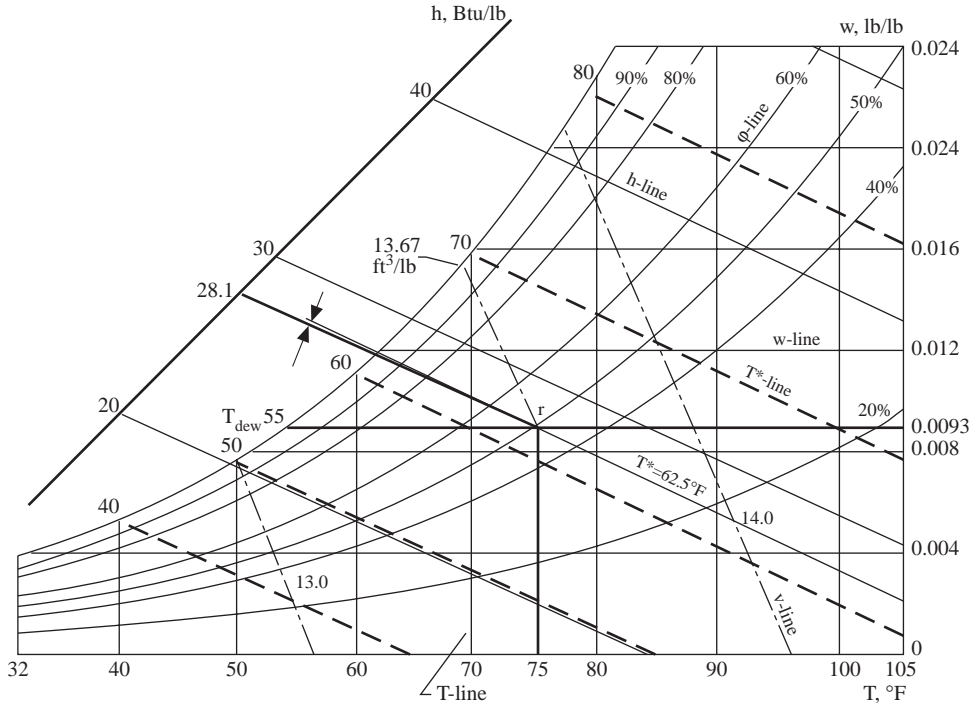


FIGURE 9.2.2 The abridged ASHRAE psychrometric chart and the determination of properties as in Example 9.2.1.

- Thermodynamic wet bulb T^* -lines have a negative slope slightly greater than that of the h -lines. A T^* -line meets the T -line of the same magnitude on the saturation curve.
- Moist volume v -lines have a far greater negative slope than h -lines and T^* -lines. The moist volume ranges from 12.5 to 15 ft^3/lb .

Moist air has seven independent thermodynamic properties or property groups: h , T , ϕ , T^* , p_{at} , $\rho - v$, and $w - p_w - T_{\text{dew}}$. When p_{at} is given, any additional two of the independent properties determine the state of moist air on the psychrometric chart and the remaining properties.

Software using AutoCAD to construct the psychrometric chart and calculate the thermodynamic properties of moist air is available. It can also be linked to the load calculation and energy programs to analyze the characteristics of air-conditioning cycles.

Refer to Wang’s *Handbook of Air Conditioning and Refrigeration* (1993) and *ASHRAE Handbook, Fundamentals* (1993) for details of psychrometric charts and psychrometric tables that list thermodynamic properties of moist air.

Example 9.2.1

An air-conditioned room at sea level has an indoor design temperature of 75°F and a relative humidity of 50%. Determine the humidity ratio, enthalpy, density, dew point, and thermodynamic wet bulb temperature of the indoor air at design condition.

Solution

1. Since the air-conditioned room is at sea level, a psychrometric chart of standard atmospheric pressure of 14.697 psi should be used to find the required properties.
2. Plot the state point of the room air at design condition r on the psychrometric chart. First, find the room temperature 75°F on the horizontal temperature scale. Draw a line parallel to the 75°F

temperature line. This line meets the relative humidity curve of 50% at point r , which denotes the state point of room air as shown in [Figure 9.2.2](#).

3. Draw a horizontal line toward the humidity ratio scale from point r . This line meets the ordinate and thus determines the room air humidity ratio $\phi_r = 0.0093$ lb/lb.
4. Draw a line from point r parallel to the enthalpy line. This line determines the enthalpy of room air on the enthalpy scale, $h_r = 28.1$ Btu/lb.
5. Draw a line through point r parallel to the moist volume line. The perpendicular scale of this line indicates $v_r = 13.67$ ft³/lb.
6. Draw a horizontal line to the left from point r . This line meets the saturation curve and determines the dew point temperature, $T_{\text{dew}} = 55^\circ\text{F}$.
7. Draw a line through point r parallel to the thermodynamic wet bulb line. The perpendicular scale to this line indicates that the thermodynamic wet bulb temperature $T^* = 62.5^\circ\text{F}$.

9.3 Air-Conditioning Processes and Cycles

Air-Conditioning Processes

An *air-conditioning process* describes the change in thermodynamic properties of moist air between the initial and final stages of conditioning as well as the corresponding energy and mass transfers between the moist air and a medium, such as water, refrigerant, absorbent or adsorbent, or moist air itself. The energy balance and conservation of mass are the two principles used for the analysis and the calculation of the thermodynamic properties of the moist air.

Generally, for a single air-conditioning process, heat transfer or mass transfer is positive. However, for calculations that involve several air-conditioning processes, heat supplied to the moist air is taken as positive and heat rejected is negative.

The *sensible heat ratio* (SHR) of an air-conditioning process is defined as the ratio of the change in absolute value of sensible heat to the change in absolute value of total heat, both in Btu/hr:

$$\text{SHR} = |q_{\text{sen}}|/|q_{\text{total}}| = |q_{\text{sen}}|/(|q_{\text{sen}}| + |q_1|) \quad (9.3.1)$$

For any air-conditioning process, the sensible heat change

$$q_{\text{sen}} = 60 \dot{V}_s \rho_s c_{\text{pa}} (T_2 - T_1) = 60 \dot{m}_a c_{\text{pa}} (T_2 - T_1) \quad (9.3.2)$$

where \dot{V}_s = volume flow rate of supply air, cfm

ρ_s = density of supply air, lb/ft³

T_2, T_1 = moist air temperature at final and initial states of an air-conditioning process, °F

and the mass flow rate of supply air

$$\dot{m}_s = \dot{V}_s \rho_s \quad (9.3.3)$$

The latent heat change is

$$q_1 \approx 60 \dot{V}_s \rho_s (w_2 - w_1) h_{\text{fg},58} = 1060 \times 60 \dot{V}_s \rho_s (w_2 - w_1) \quad (9.3.4)$$

where w_2, w_1 = humidity ratio at final and initial states of an air-conditioning process, lb/lb.

In Equation (9.3.4), $h_{\text{fg},58} \approx 1060$ Btu/lb represents the latent heat of vaporization or condensation of water at an estimated temperature of 58°F, where vaporization or condensation occurs in an air-handling unit or packaged unit. Therefore

$$\text{SHR} = \dot{m}_a c_{\text{pa}} (T_2 - T_1) / \left[\dot{m}_a c_{\text{pa}} (T_2 - T_1) + \dot{m}_a (w_2 - w_1) h_{\text{fg},58} \right] \quad (9.3.5)$$

Space Conditioning, Sensible Cooling, and Sensible Heating Processes

In a *space conditioning process*, heat and moisture are absorbed by the supply air at state s and then removed from the conditioned space at the state of space air r during summer, as shown by line sr in Figure 9.3.1, or heat or moisture is supplied to the space to compensate for the transmission and infiltration losses through the building envelope as shown by line $s'r'$. Both processes are aimed at maintaining a desirable space temperature and relative humidity.

The space cooling load q_{tc} , in Btu/hr, can be calculated as:

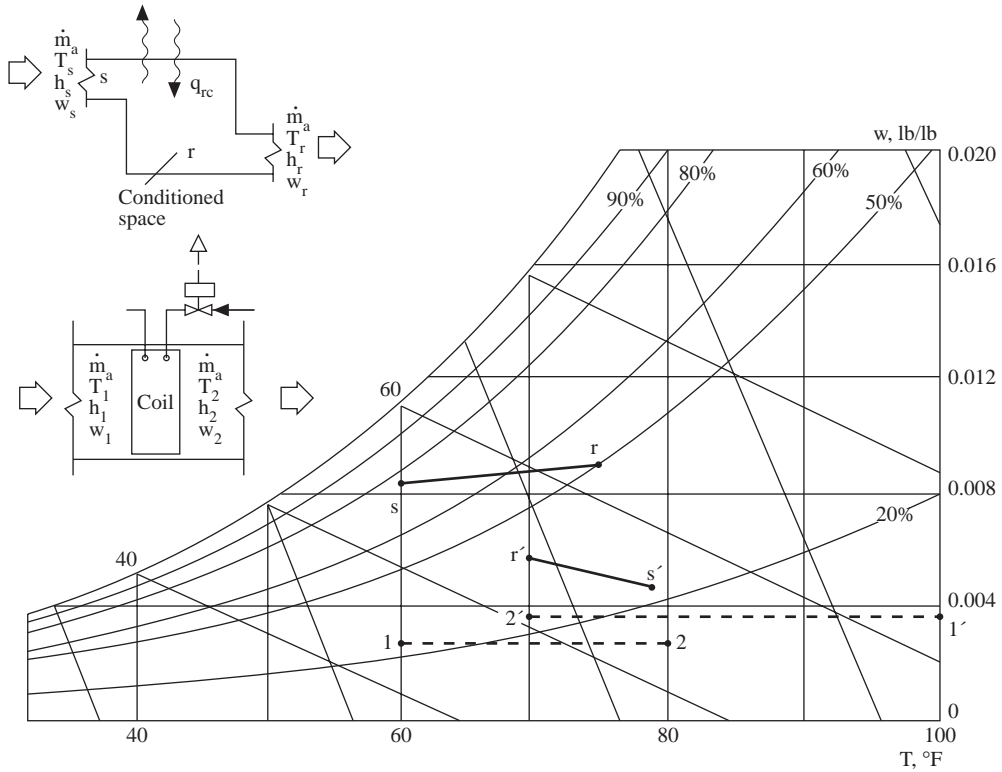


FIGURE 9.3.1 Supply conditioning, sensible heating, and sensible cooling processes.

$$q_{rc} = 60 \dot{m}_a (h_r - h_s) = 60 \dot{V}_s \rho_s (h_r - h_s) \tag{9.3.6}$$

where h_r, h_s = enthalpy of space air and supply air, Btu/lb.

The space sensible cooling load q_{rs} , in Btu/hr, can be calculated from Equation (9.3.2) and the space latent load q_{rl} , in Btu/hr, from Equation (9.3.1). In Equation (9.3.4), T_2 should be replaced by T_r and T_1 by T_s . Also in Equation (9.3.1), w_2 should be replaced by w_r and w_1 by w_s . The space heating load q_{th} is always a sensible load, in Btu/hr, and can be calculated as:

$$q_{th} = 60 \dot{m}_a c_{pa} (T_s - T_r) = 60 \dot{V}_s \rho_s c_{pa} (T_s - T_r) \tag{9.3.7}$$

where T_s, T_r = temperature of supply and space air, °F.

A *sensible heating process* adds heat to the moist air in order to increase its temperature; its humidity ratio remains constant, as shown by line 12 in Figure 9.3.1. A sensible heating process occurs when moist air flows over a heating coil. Heat is transferred from the hot water inside the tubes to the moist air. The rate of heat transfer from the hot water to the colder moist air is often called the heating coil load q_{th} , in Btu/hr, and is calculated from Equation (9.3.2).

A *sensible cooling process* removes heat from the moist air, resulting in a drop of its temperature; its humidity ratio remains constant, as shown by line 1'2' in Figure 9.3.1. The sensible cooling process occurs when moist air flows through a cooling coil containing chilled water at a temperature equal to or greater than the dew point of the entering moist air. The sensible cooling load can also be calculated from Equation (9.3.2). T_2 is replaced by T_1 and T_1 by T_2 .

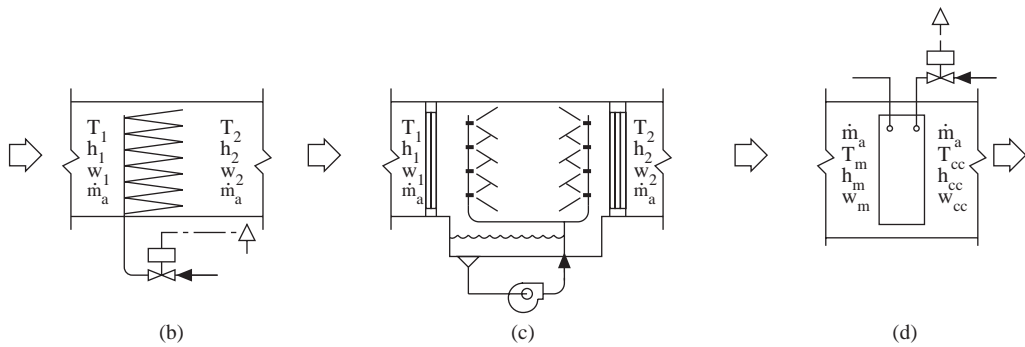
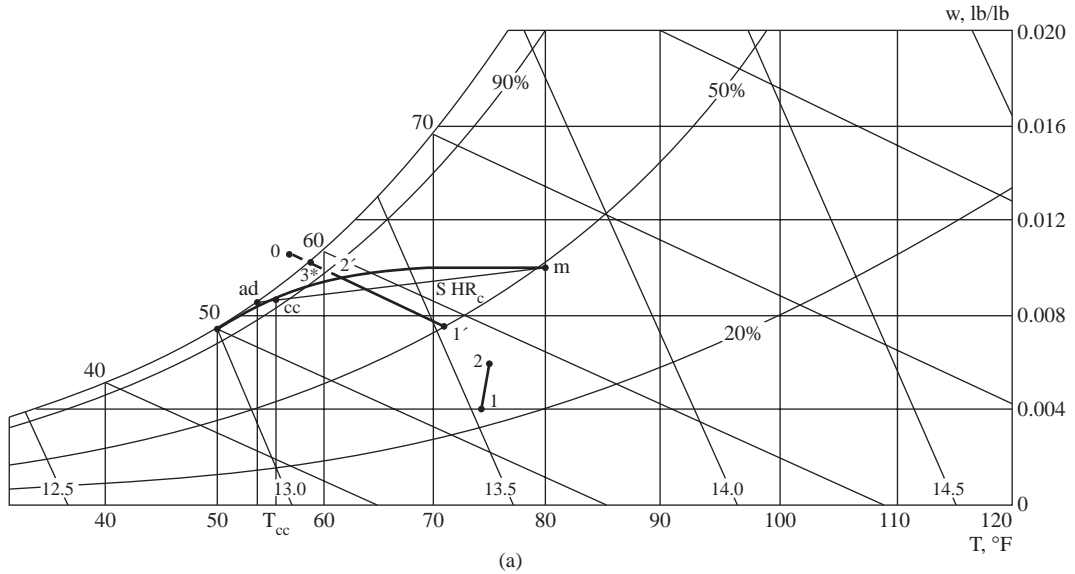


FIGURE 9.3.2 Humidifying and cooling and dehumidifying processes: (a) process on psychrometric chart, (b) steam humidifier, (c) air washer, and (d) water cooling or DX coil.

Humidifying and Cooling and Dehumidifying Processes

In a *humidifying process*, water vapor is added to moist air and increases the humidity ratio of the moist air entering the humidifier if the moist air is not saturated. Large-scale humidification of moist air is usually performed by steam injection, evaporation from a water spray, atomizing water, a wetted medium, or submerged heating elements. Some details of their construction and characteristics are covered in a later section. Dry steam in a steam injection humidifying process is often supplied from the main steam line to a grid-type humidifier and injected into the moist air directly through small holes at a pressure slightly above atmospheric, as shown by line 12 in [Figure 9.3.2\(a\)](#) and (b). The humidifying capacity \dot{m}_{hu} , in lb/min, is given by:

$$\dot{m}_{hu} = \dot{V}_s \rho_s (w_{hl} - w_{he}) \tag{9.3.8}$$

where w_{hl}, w_{he} = humidity ratio of moist air leaving and entering the humidifier, lb/lb. The slight inclination at the top of line 12 is due to the high temperature of the steam. The increase in temperature of the moist air due to steam injection can be calculated as:

$$(T_2 - T_1) = w_{sm} c_{ps} T_s / (c_{pd} + w_{12} c_{ps}) \quad (9.3.9)$$

where T_2, T_1 = temperature of moist air at initial and final states, °F

w_{sm} = ratio of mass flow rate of injected steam to moist air, \dot{m}_s / \dot{m}_a

T_s = temperature of injected steam, °F

w_{12} = average humidity ratio of moist air, lb/lb

An *air washer* is a device that sprays water into moist air in order to humidify, to cool and dehumidify, and to clean the air, as shown in Figure 9.3.2(c). When moist air flows through an air washer, the moist air is humidified and approaches saturation. This actual adiabatic saturation process approximately follows the thermodynamic wet bulb line on the psychrometric chart as shown by line 1'2'. The humidity ratio of the moist air is increased while its temperature is reduced. The cooling effect of this adiabatic saturation process is called *evaporative cooling*.

Oversaturation occurs when the amount of water present in the moist air w_{os} , in lb/lb, exceeds the saturated humidity ratio at thermodynamic wet bulb temperature w_s^* , as shown in Figure 9.3.2(a). When moist air leaves the air washer, atomizing humidifier, or centrifugal humidifier after humidification, it often contains unevaporated water droplets at state point 2', w_w , in lb/lb. Because of the fan power heat gain, duct heat gain, and other heat gains providing the latent heat of vaporization, some evaporation takes place due to the heat transfer to the water drops, and the humidity ratio increases further. Such evaporation of oversaturated drops is often a process with an increase of both humidity ratio and enthalpy of moist air. Oversaturation can be expressed as:

$$w_{os} = w_o - w_s^* = (w_{2'} + w_w) - w_s^* \quad (9.3.10)$$

where $w_{2'}$ = humidity ratio at state point 2', lb/lb

w_o = sum of $w_{2'}$ and w_w , lb/lb

The magnitude of w_w depends mainly on the construction of the humidifier and water eliminator, if any. For an air washer, w_w may vary from 0.0002 to 0.001 lb/lb. For a pulverizing fan without an eliminator, w_w may be up to 0.00135 lb/lb.

Cooling and Dehumidifying Process

In a cooling and dehumidifying process, both the humidity ratio and temperature of moist air decrease. Some water vapor is condensed in the form of liquid water, called a *condensate*. This process is shown by curve m cc on the psychrometric chart in Figure 9.3.2(a). Here m represents the entering mixture of outdoor and recirculating air and cc the conditioned air leaving the cooling coil.

Three types of heat exchangers are used in a cooling and dehumidifying process: (1) water cooling coil as shown in Figure 9.3.2(d); (2) direct expansion DX coil, where refrigerant evaporates directly inside the coil's tubes; and (3) air washer, in which chilled water spraying contacts condition air directly.

The temperature of chilled water entering the cooling coil or air washer T_{we} , in °F, determines whether it is a sensible cooling or a cooling and dehumidifying process. If T_{we} is smaller than the dew point of the entering air T_{ae}'' in the air washer, or T_{we} makes the outer surface of the water cooling coil $T_{s,t} < T_{ae}''$, it is a cooling and dehumidifying process. If $T_{we} \geq T_{ae}''$, or $T_{s,t} \geq T_{ae}''$, sensible cooling occurs. The cooling coil's load or the cooling capacity of the air washer q_{cc} , in Btu/hr, is

$$q_{cc} = 60 \dot{V}_s \rho_s (h_{ae} - h_{cc}) - 60 \dot{m}_w h_w \quad (9.3.11a)$$

where h_{ae}, h_{cc} = enthalpy of moist air entering and leaving the coil or washer, Btu/lb

\dot{m}_w = mass flow rate of the condensate, lb/min

h_w = enthalpy of the condensate, Btu/lb

Since the thermal energy of the condensate is small compared with q_{cc} , in practical calculations the term $60\dot{m}_w h_w$ is often neglected, and

$$q_{cc} = 60\dot{V}_s \rho_s (h_{ae} - h_{cc}) \quad (9.3.11b)$$

The sensible heat ratio of the cooling and dehumidifying process SHR_c can be calculated from

$$SHR_c = q_{cs}/q_{cc} \quad (9.3.12)$$

where q_{cs} = sensible heat removed during the cooling and dehumidifying process, Btu/hr. SHR_c is shown by the slope of the straight line joining points m and cc.

The relative humidity of moist air leaving the water cooling coil or DX coil depends mainly on the outer surface area of the coil including pipe and fins. For coils with ten or more fins per inch, if the entering moist air is around 80°F dry bulb and 68°F wet bulb, the relative humidity of air leaving the coil (off-coil) may be estimated as:

Four-row coil	90 to 95%
Six-row and eight-row coils	96 to 98%

Two-Stream Mixing Process and Bypass Mixing Process

For a *two-stream adiabatic mixing process*, two moist air streams, 1 and 2, are mixed together adiabatically and a mixture m is formed in a mixing chamber as shown by line 1 m 2 in [Figure 9.3.3](#). Since the AHU or PU is well insulated, the heat transfer between the mixing chamber and ambient air is small and is usually neglected. Based on the principle of heat balance and conservation of mass:

$$\begin{aligned} \dot{m}_1 h_1 + \dot{m}_2 h_2 &= \dot{m}_m h_m \\ \dot{m}_1 w_1 + \dot{m}_2 w_2 &= \dot{m}_m w_m \\ \dot{m}_1 T_1 + \dot{m}_2 T_2 &= \dot{m}_m T_m \end{aligned} \quad (9.3.13)$$

In Equation (9.3.13), \dot{m} represents the mass flow rate of air, lb/min; h the enthalpy, in Btu/lb; w the humidity ratio, in lb/lb; and T the temperature, in °F. Subscripts 1 and 2 indicate air streams 1 and 2 and m the mixture; also,

$$\begin{aligned} \dot{m}_1/\dot{m}_m &= (h_2 - h_m)/(h_2 - h_1) = (w_2 - w_m)/(w_2 - w_1) \\ &= (\text{line segment } m1\ 2)/(\text{line segment } 12) \end{aligned} \quad (9.3.14)$$

Similarly,

$$\begin{aligned} \dot{m}_2/\dot{m}_m &= (h_m - h_1)/(h_2 - h_1) = (w_m - w_1)/(w_2 - w_1) \\ &= (\text{line segment } 1\ m1)/(\text{line segment } 12) \end{aligned} \quad (9.3.15)$$

Mixing point m must lie on the line that joins points 1 and 2 as shown in [Figure 9.3.3](#).

If the differences between the density of air streams 1 and 2 and the density of the mixture are neglected,

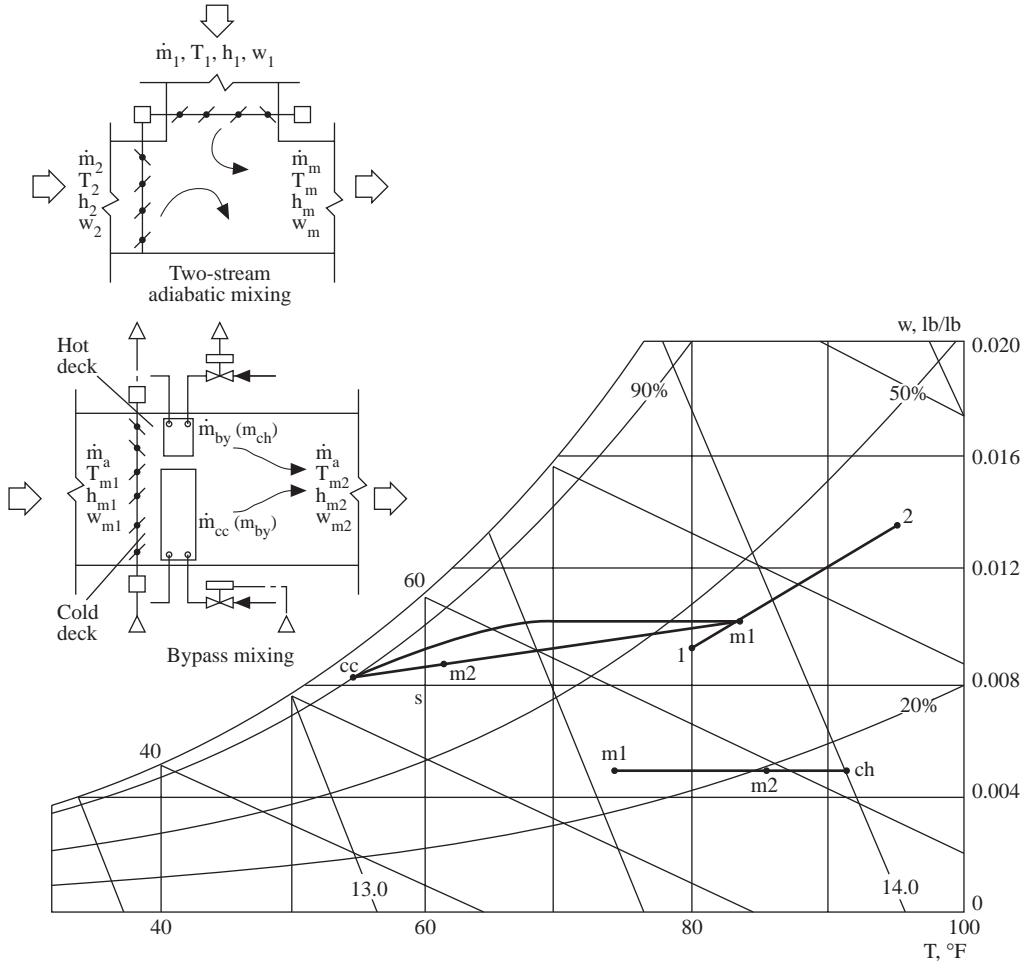


FIGURE 9.3.3 Mixing processes.

$$\begin{aligned} \dot{V}_1 h_1 + \dot{V}_2 h_2 &= \dot{V}_m h_m \\ \dot{V}_1 w_1 + \dot{V}_2 w_2 &= \dot{V}_m w_m \end{aligned} \tag{9.3.16}$$

$$\begin{aligned} \dot{V}_1 T_1 + \dot{V}_2 T_2 &= \dot{V}_m T_m \\ \dot{V}_1 + \dot{V}_2 &= \dot{V}_m \end{aligned} \tag{9.3.17}$$

In a *bypass mixing process*, a conditioned air stream is mixed with a bypass air stream that is not conditioned. The cold conditioned air is denoted by subscript cc, the heated air ch, and the bypass air by.

Equations (9.3.14) and (9.3.17) can still be used but subscript 1 should be replaced by cc or ch and subscript 2 by “by” (bypass).

Let $K_{cc} = \dot{m}_{cc} / \dot{m}_m$ and $K_{ch} = \dot{m}_{ch} / \dot{m}_m$; then the cooling coil’s load q_{cc} and heating coil’s load q_{ch} , both in Btu/hr, for a bypass mixing process are

$$q_{cc} = K_{cc} \dot{V}_s \rho_s (h_m - h_{cc}) \quad (9.3.18)$$

$$q_{ch} = K_{ch} \dot{V}_s \rho_s (h_2 - h_1)$$

In Equation (9.3.18), subscript s denotes the supply air and m the mixture air stream.

Air-Conditioning Cycle and Operating Modes

An *air-conditioning cycle* comprises several air-conditioning processes that are connected in a sequential order. An air-conditioning cycle determines the operating performance of the air system in an air-conditioning system. The *working substance* to condition air may be chilled or hot water, refrigerant, desiccant, etc.

Each type of air system has its own air-conditioning cycle. Psychrometric analysis of an air-conditioning cycle is an important tool in determining its operating characteristics and the state of moist air at various system components, including the volume flow rate of supply air, the coil's load, and the humidifying and dehumidifying capacity.

According to the cycle performance, air-conditioning cycles can be grouped into two categories:

- *Open cycle*, in which the moist air at its end state does not resume its original state. An air-conditioning cycle with all outdoor air is an open cycle.
- *Closed cycle*, in which moist air resumes its original state at its end state. An air-conditioning cycle that conditions the mixture of recirculating and outdoor air, supplies it, recirculates part of the return air, and mixes it again with outdoor air is a closed cycle.

Based on the outdoor weather and indoor operating conditions, the operating modes of air-conditioning cycles can be classified as:

- *Summer mode*: when outdoor and indoor operating parameters are in summer conditions.
- *Winter mode*: when outdoor and indoor operating parameters are in winter conditions.
- *Air economizer mode*: when all outdoor air or an amount of outdoor air that exceeds the minimum amount of outdoor air required for the occupants is taken into the AHU or PU for cooling. The air economizer mode saves energy use for refrigeration.

Continuous modes operate 24 hr a day and 7 days a week. Examples are systems that serve hospital wards and refrigerated warehouses. An *intermittently operated mode* usually shuts down once or several times within a 24-hr operating cycle. Such systems serve offices, class rooms, retail stores, etc. The 24-hr day-and-night cycle of an intermittently operated system can again be divided into:

1. *Cool-down or warm-up period*. When the space is not occupied and the space air temperature is higher or lower than the predetermined value, the space air should be cooled down or warmed up before the space is occupied.
2. *Conditioning period*. The air-conditioning system is operated during the occupied period to maintain the required indoor environment.
3. *Nighttime shut-down period*. The air system or terminal is shut down or only partly operating to maintain a set-back temperature.

Summer, winter, air economizer, and continuously operating modes consist of *full-load* (design load) and part-load operations. *Part load* occurs when the system load is less than the design load. The capacity of the equipment is selected to meet summer and winter system design loads as well as system loads in all operating modes.

Basic Air-Conditioning Cycle — Summer Mode

A *basic air-conditioning system* is a packaged system of supply air at a constant volume flow rate, serving a single zone, equipped with only a single supply/return duct. A *single zone* is a conditioned space for which a single controller is used to maintain a unique indoor operating parameter, probably indoor temperature. A *basic air-conditioning cycle* is the operating cycle of a basic air-conditioning system. Figure 9.1.3 shows a basic air-conditioning system. Figure 9.3.4 shows the basic air-conditioning cycle of this system. In summer mode at design load, recirculating air from the conditioned space, a worship hall, enters the packaged unit through the return grill at point ru. It is mixed with the required minimum amount of outdoor air at point o for acceptable indoor air quality and energy saving. The mixture *m* is then cooled and dehumidified to point cc at the DX coil, and the conditioned air is supplied to the hall through the supply fan, supply duct, and ceiling diffuser. Supply air then absorbs the sensible and latent load from the space, becoming the space air *r*. Recirculating air enters the packaged unit again and forms a closed cycle. *Return air* is the air returned from the space. Part of the return air is exhausted to balance the outdoor air intake and infiltration. The remaining part is the *recirculating air* that enters the PU or AHU.

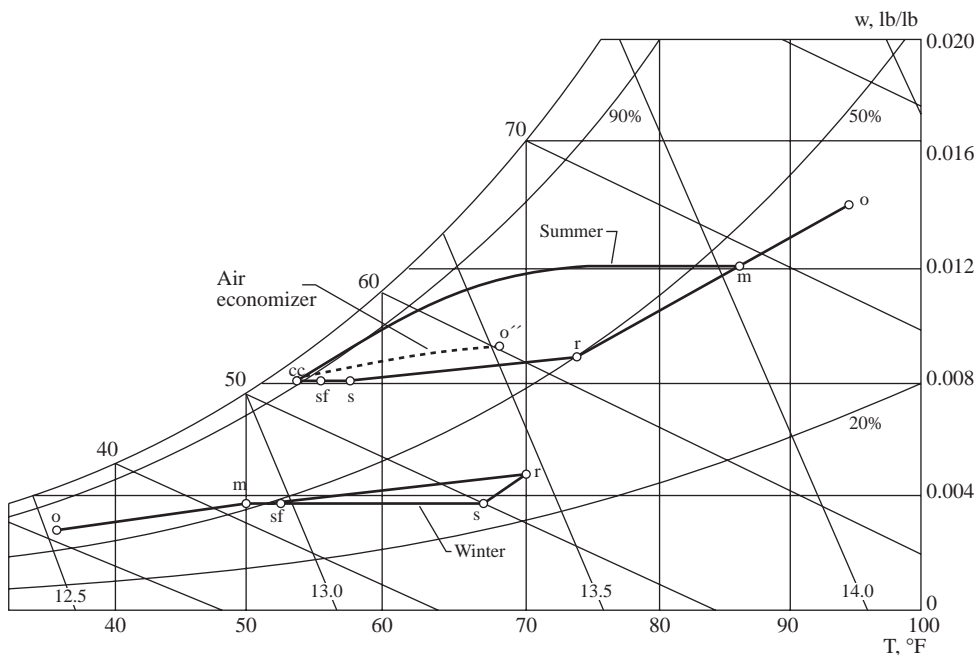


FIGURE 9.3.4 Basic air-conditioning cycle — summer, winter, and air economizer mode.

The summer mode operating cycle consists of the following processes:

1. Sensible heating process, represented by line *r ru*, due to the return system gain $q_{r,s}$, in Btu/hr, when recirculating air flows through the return duct, ceiling plenum, and return fan, if any. In this packaged system, the return system heat gain is small and neglected.
2. Adiabatic mixing process of recirculating air at point *ru* and outdoor air at point *o* in the mixing box, represented by line *ru m o*.
3. Cooling and dehumidifying process *m cc* at the DX coil whose coil load determines the cooling capacity of the system calculated from Equation (9.3.11).
4. Sensible heating process related to the supply system heat gain $q_{s,s}$, in Btu/hr, represented by line *cc sf s*. $q_{s,s}$ consists of the fan power heat gain q_{sf} , line *cc sf*, and duct heat gain q_{sd} , line *sf s*, that is:

$$q_{s,s} = q_{sf} + q_{sd} = \dot{V}_s \rho_s c_{pa} \Delta T_{s,s} \quad (9.3.19)$$

It is more convenient to use the temperature rise of the supply system $\Delta T_{s,s}$ in psychrometric analysis.

5. Supply conditioning process line sr.

Design Supply Volume Flow Rate

Design supply volume flow rate and cooling and heating capacities are primary characteristics of an air-conditioning system. Design supply volume flow rate is used to determine the size of fans, grills, outlets, air-handling units, and packaged units. For most comfort systems and many processing air-conditioning systems, *design supply volume flow rate* $\dot{V}_{s,d}$, in cfm, is calculated on the basis of the capacity to remove the space cooling load at summer design conditions to maintain a required space temperature T_s :

$$\dot{V}_{s,d} = q_{rc,d} / [60\rho_s (h_r - h_s)] = q_{rs,d} / [60\rho_s c_{pa} (T_r - T_s)] \quad (9.3.20)$$

where $q_{rc,d}$, $q_{rs,d}$ = design space cooling load and design sensible cooling load, Btu/hr. In Equation (9.3.20), the greater the $q_{rs,d}$, the higher \dot{V}_s will be. Specific heat c_{pa} is usually considered constant. Air density ρ_s may vary with the various types of air systems used. A greater ρ_s means a smaller $\dot{V}_{s,d}$ for a given supply mass flow rate. For a given $q_{rs,d}$, the supply temperature differential $\Delta T_s = (T_r - T_s)$ is an important parameter that affects $\dot{V}_{s,d}$. Conventionally, a 15 to 20°F ΔT_s is used for comfort air-conditioning systems. Recently, a 28 to 34°F ΔT_s has been adopted for cold air distribution in ice-storage central systems. When ΔT_s has a nearly twofold increase, there is a considerable reduction in \dot{V}_s and fan energy use and saving in investment on ducts, terminals, and outlets.

The summer cooling load is often greater than the winter heating load, and this is why q_{rc} or $q_{rs,d}$ is used to determine $\dot{V}_{s,d}$, except in locations where the outdoor climate is very cold.

Sometimes the supply volume flow rate may be determined from the following requirements:

- To dilute the concentration of air contaminants in the conditioned space C_i , in mg/m³, the design supply volume flow rate is

$$\dot{V}_{s,d} = 2118 \dot{m}_{par} / (C_i - C_s) \quad (9.3.21)$$

where C_s = concentration of air contaminants in supply air, mg/m³
 \dot{m}_{par} = rate of contaminant generation in the space, mg/sec

- To maintain a required space relative humidity ϕ_r and a humidity ratio w_r at a specific temperature, the design supply volume flow rate is

$$\dot{V}_{s,d} = q_{rl,d} / [60\rho_s (w_r - w_s) h_{fg,58}] \quad (9.3.22)$$

where $q_{rl,d}$ = design space latent load, Btu/hr.

- To provide a required air velocity v_r , in fpm, within the working area of a clean room, the supply volume flow rate is given by

$$\dot{V}_{s,d} = A_r v_r \quad (9.3.23a)$$

where A_r = cross-sectional area perpendicular to the air flow in the working area, ft².

- To exceed the outdoor air requirement for acceptable air quality for occupants, the supply volume flow rate must be equal to or greater than

$$\dot{V}_s \geq n \dot{V}_{oc} \quad (9.3.23b)$$

where n = number of occupants

\dot{V}_{oc} = outdoor air requirement per person, cfm/person

- To exceed the sum of the volume flow rate of exhaust air \dot{V}_{ex} and the exfiltrated or relief air \dot{V}_{ef} , both in cfm,

$$\dot{V}_s \geq \dot{V}_{ex} + \dot{V}_{ef} \quad (9.3.24)$$

The design supply volume flow rate should be the largest of any of the foregoing requirements.

Rated Supply Volume Flow Rate

For an air system at atmospheric pressure, since the required mass flow rate of the supply air is a function of air density and remains constant along the air flow,

$$\dot{m}_a = \dot{V}_{cc} \rho_{cc} = \dot{V}_s \rho_s = \dot{V}_{sf} \rho_{sf} \quad (9.3.25)$$

$$\dot{V}_{sf} = \dot{V}_s \rho_s / \rho_{sf}$$

where \dot{V}_{sf} = volume flow rate at supply fan outlet, cfm

ρ_{sf} = air density at supply fan outlet, lb/ft³

A supply fan is rated at *standard air conditions*, that is, dry air at a temperature of 70°F, an atmospheric pressure of 29.92 in. Hg (14.697 psia), and an air density of 0.075 lb/ft³. However, a fan is a constant-volume machine at a given fan size and speed; that is, $\dot{V}_{sf} = \dot{V}_{sf,r}$. Here $\dot{V}_{sf,r}$ represents the rated volume flow rate of a fan at standard air conditions. Therefore,

$$\dot{V}_{sf,r} = \dot{V}_{sf} = q_{rs,d} / \left[60 \rho_{sf} c_{pa} (T_r - T_s) \right] \quad (9.3.26)$$

- For conditioned air leaving the cooling coil at $T_{cc} = 55^\circ\text{F}$ with a relative humidity of 92% and T_{sf} of 57°F , $\rho_{sf,r} = 1/v_{sf} = 1/13.20 = 0.0758$ lb/ft³. From Equation (9.3.26):

$$\dot{V}_{sf,r} = q_{rs,d} / \left[60 \times 0.0758 \times 0.243 (T_r - T_s) \right] = q_{rs,d} / \left[1.1 (T_r - T_s) \right] \quad (9.3.26a)$$

Equation (9.3.26a) is widely used in calculating the supply volume flow rate of comfort air-conditioning systems.

- For cold air distribution, $T_{cc} = 40^\circ\text{F}$ and $\phi_{cc} = 98\%$, if $T_{sf} = 42^\circ\text{F}$, then $v_{sf} = 12.80$ ft³/lb, and the rated supply volume flow rate:

$$\dot{V}_{sf,r} = 12.80 q_{rs,d} / \left[60 \times 0.243 (T_r - T_s) \right] = q_{rs,d} / \left[1.14 (T_r - T_s) \right] \quad (9.3.26b)$$

- For a blow-through fan in which the fan is located upstream of the coil, if $T_{sf} = 82^\circ\text{F}$ and $\phi_{sf} = 43\%$, then $v_{sf} = 13.87$ ft³/lb, and the rated supply volume flow rate:

$$\dot{V}_{sf,r} = 13.87q_{rs,d} / [60 \times 0.243(T_r - T_s)] = q_{rs,d} / [1.05(T_r - T_s)] \tag{9.3.26c}$$

Effect of the Altitude

The higher the altitude, the lower the atmospheric pressure and the air density. In order to provide the required mass flow rate of supply air, a greater $\dot{V}_{sf,r}$ is needed. For an air temperature of 70°F:

$$\dot{V}_{x,ft} = \dot{V}_{sf,r} (p_{sea} / p_{x,ft}) = \dot{V}_{sf,r} (\rho_{sea} / \rho_{x,ft}) \tag{9.3.27}$$

where $\dot{V}_{x,ft}$ = supply volume flow rate at an altitude of x ft, cfm
 $p_{sea}, p_{x,ft}$ = atmospheric pressure at sea level and an altitude of x ft, psia
 $\rho_{sea}, \rho_{x,ft}$ = air density at sea level and an altitude of x ft, psia

Following are the pressure or air density ratios at various altitudes. At 2000 ft above sea level, the rated supply volume flow rate $\dot{V}_{r,2000} = \dot{V}_{sf,r} (p_{sea} / p_{x,ft}) = 1.076 \dot{V}_{sf,r}$ cfm instead of $\dot{V}_{sf,r}$ cfm at sea level.

Altitude, ft	p_{at} , psia	ρ , lb/ft ³	$(p_{sea} / p_{x,ft})$
0	14.697	0.075	1.000
1000	14.19	0.0722	1.039
2000	13.58	0.0697	1.076
3000	13.20	0.0672	1.116
5000	12.23	0.0625	1.200

Off-Coil and Supply Air Temperature

For a given design indoor air temperature T_r , space sensible cooling load q_{rs} , and supply system heat gain $q_{s,s}$, a lower air off-coil temperature T_{cc} as well as supply temperature T_s means a greater supply temperature differential ΔT_s and a lower space relative humidity ϕ_r and vice versa. A greater ΔT_s decreases the supply volume flow rate \dot{V}_s and then the fan and terminal sizes, duct sizes, and fan energy use. The result is a lower investment and energy cost.

A lower T_{cc} and a greater ΔT_s require a lower chilled water temperature entering the coil T_{we} , a lower evaporating temperature T_{ev} in the DX coil or refrigerating plant, and therefore a greater power input to the refrigerating compressors. When an air-conditioning system serves a conditioned space of a single zone, optimum T_{cc} , T_s , and T_{we} can be selected. For a conditioned space of multizones, T_{cc} , T_s , and T_{we} should be selected to satisfy the lowest requirement. In practice, T_s and T_{we} are often determined according to previous experience with similar projects.

In general, the temperature rise due to the supply fan power system heat gain q_{sf} can be taken as 1 to 3°F depending on the fan total pressure. The temperature rise due to the supply duct system heat gain at design flow can be estimated as 1°F/100 ft insulated main duct length based on 1-in. thickness of duct insulation.

Outside Surface Condensation

The outside surface temperature of the ducts, terminals, and supply outlets T_{sur} in the ceiling plenum in contact with the return air should not be lower than the dew point of the space air T_r' , in °F. The temperature rise due to the fan power heat gain is about 2°F. According to Dorgan (1988), the temperature difference between the conditioned air inside the terminal and the outside surface of the terminal with insulation wrap is about 3°F. For a space air temperature of 75°F and a relative humidity of 50%, its dew point temperature is 55°F. If the outside surface temperature $T_s = (T_{cc} + 2 + 3) \leq 55^\circ\text{F}$, condensation may occur on the outside surface of the terminal. Three methods are often used to prevent condensation:

1. Increase the thickness of the insulation layer on the outside surface.
2. Adopt a supply outlet that induces more space air.

- Equip with a terminal that mixes the supply air with the space air or air from the ceiling plenum.

During the cool-down period, due to the high dew point temperature of the plenum air when the air system is started, the supply air temperature must be controlled to prevent condensation.

Example 9.3.1

The worship hall of a church uses a package system with a basic air system. The summer space sensible cooling load is 75,000 Btu/hr with a latent load of 15,000 Btu/hr. Other design data for summer are as follows:

Outdoor summer design temperature: dry bulb 95°F and wet bulb 75°F

Summer indoor temperature: 75°F with a space relative humidity of 50%:

Temperature rise: fan power 2°F

supply duct 2°F

Relative humidity of air leaving cooling coil: 93%

Outdoor air requirement: 1800 cfm

Determine the

- Temperature of supply air at summer design conditions
- Rated volume flow rate of the supply fan
- Cooling coil load
- Possibility of condensation at the outside surface of the insulated branch duct to the supply outlet

Solution

- From Equation 9.3.1 the sensible heat ratio of the space conditioning line is

$$\text{SHR}_s = |q_{rs}| / (|q_{rs}| + |q_{rl}|) = 60,000 / (60,000 + 15,000) = 0.8$$

On the psychrometric chart, from given $T_r = 75^\circ\text{F}$ and $\phi_r = 50\%$, plot space point r. Draw a space conditioning line sr from point r with $\text{SHR}_s = 0.8$.

Since $\Delta T_{s,s} = 2 + 2 = 4^\circ\text{F}$, move line segment cc s (4°F) up and down until point s lies on line sr and point cc lies on the $\phi_{cc} = 93\%$ line. The state points s and cc are then determined as shown in Figure 9.3.4:

$$T_s = 57.5^\circ\text{F}, \phi_s = 82\%, \text{ and } w_s = 0.0082 \text{ lb/lb}$$

$$T_{cc} = 53.5^\circ\text{F}, \phi_{cc} = 93\%, h_{cc} = 21.8 \text{ Btu/lb}, \text{ and } w_{cc} = 0.0082 \text{ lb/lb}$$

- Since $T_{sf} = 53.5 + 2 = 55.5^\circ\text{F}$ and $w_{sf} = 0.0082 \text{ lb/lb}$, $\rho_{sf} = 1/v_{sf} = 1/13.15 = 0.076 \text{ lb/ft}^3$. From Equation 9.4.2, the required rated supply volume flow rate is

$$\begin{aligned} \dot{V}_{sf,r} &= q_{rs,d} / [60\rho_{sf}c_{pa}(T_r - T_s)] \\ &= 60,000 / [60 \times 0.076 \times 0.243(75 - 57.5)] = 3094 \text{ cfm} \end{aligned}$$

- Plot outdoor air state point o on the psychrometric chart from given dry bulb 95°F and wet bulb 75°F. Connect line ro. Neglect the density differences between points r, m, and o; then

$$rm/ro = 1800/3094 = 0.58$$

From the psychrometric chart, the length of line ro is 2.438 in. As shown in Figure 9.3.4, point m is then determined as:

$$T_m = 86.7^\circ\text{F}, \quad h_m = 35 \text{ Btu/lb}$$

From Equation (9.3.11), the cooling coil load is

$$q_{cc} = 60 \dot{V}_s \rho_s (h_m - h_{cc}) = 60 \times 3094 \times 0.076(35 - 21.8) = 186,234 \text{ Btu/lb}$$

4. From the psychrometric chart, since the dew point of the space air $T_r'' = 55^\circ\text{F}$ and is equal to that of the plenum air, the outside surface temperature of the branch duct $T_s = 53.5 + 2 + 3 = 58^\circ\text{F}$ which is higher than $T_r'' = 55^\circ\text{F}$. Condensation will not occur at the outside surface of the branch duct.

Basic Air-Conditioning Cycle — Winter Mode

When the basic air-conditioning systems are operated in winter mode, their air-conditioning cycles can be classified into the following four categories:

Cold Air Supply without Space Humidity Control. In winter, for a fully occupied worship hall, if the heat loss is less than the space sensible cooling load, a cold air supply is required to offset the space sensible cooling load and maintain a desirable indoor environment as shown by the lower cycle in [Figure 9.3.4](#). Usually, a humidifier is not used.

The winter cycle of a cold air supply without humidity control consists of the following air-conditioning processes:

1. Adiabatic mixing process of outdoor air and recirculating air o m r.
2. Sensible heating process due to supply fan power heat gain m sf. Because of the smaller temperature difference between the air in the ceiling plenum and the supply air inside the supply duct, heat transfer through duct wall in winter can be neglected.
3. Supply conditioning line sr.

For a winter-mode basic air-conditioning cycle with a cold air supply without space humidity control, the space relative humidity depends on the space latent load, the humidity ratio of the outdoor air, and the amount of outdoor air intake. In order to determine the space humidity ratio w_r , in lb/lb, and the space relative humidity ϕ_r , in %, Equations (9.3.15) and (9.3.22) should be used to give the following relationships:

$$\begin{aligned} (w_r - w_m) / (w_r - w_o) &= \dot{V}_o / \dot{V}_s \\ (w_r - w_s) &= q_{rl} / \left(60 \dot{V}_s \rho_s h_{fg,58} \right) \\ w_s &= w_m \end{aligned} \quad (9.3.28)$$

For a cold air supply, if there is a high space sensible cooling load, the amount of outdoor air must be sufficient, and the mixture must be cold enough to satisfy the following relationships:

$$\begin{aligned} (T_r - T_s) &= q_{rs} / \left(60 \dot{V}_s \rho_s c_{pa} \right) \\ (T_r - T_s) / (T_r - T_o) &= \dot{V}_o / \dot{V}_s \end{aligned} \quad (9.3.29)$$

The heating coil load for heating of the outdoor air can be calculated using Equation (9.3.7).

Example 9.3.2

For the same packaged air-conditioning system using a basic air system to serve the worship hall in a church as in Example 9.3.1, the space heating load at winter design condition is 10,000 Btu/hr and the latent load is 12,000 Btu/hr. Other winter design data are as follows:

Winter outdoor design temperature	35°F
Winter outdoor design humidity ratio	0.00035 lb/lb
Winter indoor design temperature	70°F
Temperature rise due to supply fan heat gain	2°F
Outdoor air requirement	1800 cfm

Determine (1) the space relative humidity at winter design temperature and (2) the heating coil load.

Solution

1. Assume that the supply air density $\rho_{sf} = 1/v_{sf} = 1/13.0 = 0.0769$ lb/ft³, and the mass flow rate of the supply air is the same as in summer mode. Then from Equation 9.3.28 the humidity ratio difference is

$$(w_r - w_s) = q_{rl} / \left(60 \dot{V}_{sf,r} \rho_{sf} h_{fg,58} \right) = 12,000 / (60 \times 3094 \times 0.0769 \times 1060) = 0.00079 \text{ lb/lb}$$

From Equation 9.3.29, the supply air temperature differential is

$$(T_r - T_s) = q_{rs,d} / \left(60 \dot{V}_{sf,r} \rho_{sf} c_{pa} \right) = 10,000 / (60 \times 3094 \times 0.0769 \times 0.243) = 2.88^\circ\text{F}$$

Since $\dot{V}_o / \dot{V}_s = 1800/3094 = 0.58$ and $w_s = w_m$,

$$(w_r - w_s) / (w_r - w_o) = 0.00079 / (w_r - w_o) = \dot{V}_o / \dot{V}_s = 0.58$$

$$(w_r - w_o) = 0.00079 / 0.58 = 0.00136 \text{ lb/lb}$$

And from given information,

$$w_r = 0.00136 + w_o = 0.00136 + 0.0035 = 0.00486 \text{ lb/lb}$$

From the psychrometric chart, for $T_r = 70^\circ\text{F}$ and $w_r = 0.00486$ lb/lb, point r can be plotted, and ϕ_r is about 32% (see Figure 9.3.4).

2. Since $m_r/o_r = 0.58$, point m can be determined, and from the psychrometric chart $T_m = 50.0^\circ\text{F}$. As $T_s = 70 - 2.88 = 67.12^\circ\text{F}$ and $T_{sf} = T_m + 2 = 50.0 + 2 = 52.0^\circ\text{F}$, from Equation 9.3.7 the heating coil's load is

$$q_{ch} = 60 \dot{V}_s \rho_s c_{pa} (T_s - T_{sf}) = 60 \times 3094 \times 0.0769 \times 0.243 (67.12 - 52.0) = 52,451 \text{ Btu/hr}$$

Warm Air Supply without Space Humidity Control

When the sum of space heat losses is greater than the sum of the internal heat gains in winter, a warm air supply is needed. For many comfort systems such as those in offices and stores, in locations where winter is not very cold, humidification is usually not necessary. The basic air-conditioning cycle for a warm air supply without space humidity control is shown in Figure 9.3.5(a). This cycle is similar to the

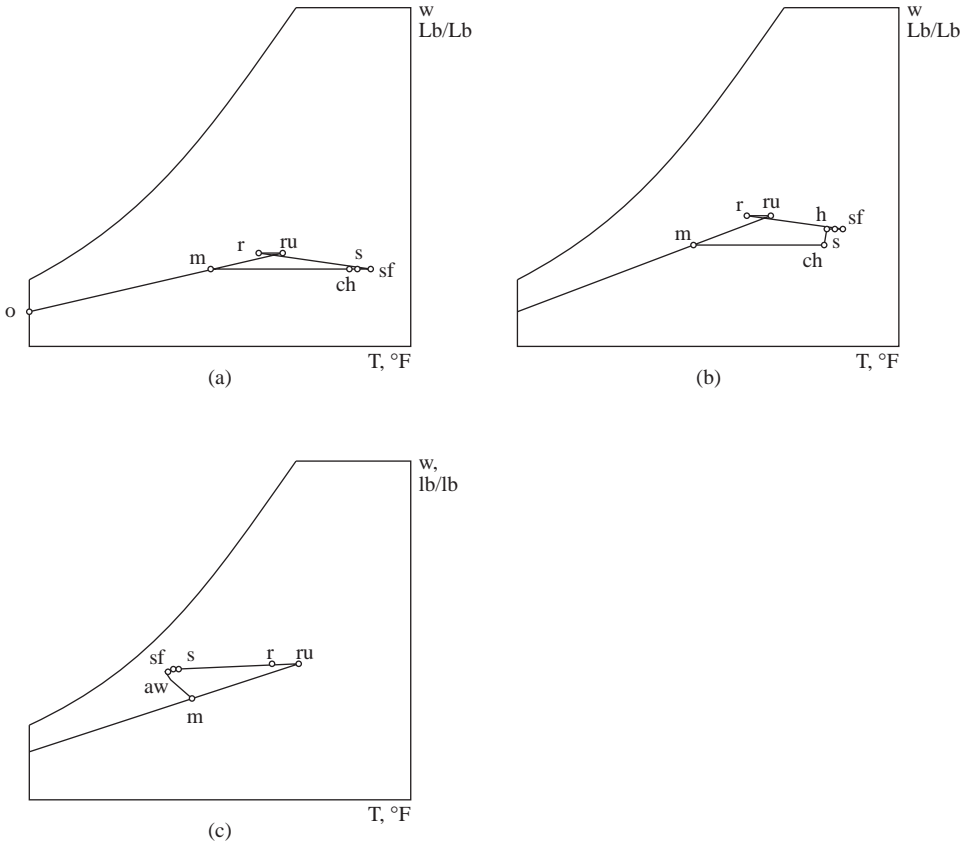


FIGURE 9.3.5 Basic air-conditioning cycle — winter modes: (a) warm air supply without space humidity control, (b) cold air supply without space humidity control, and (c) cold air supply with space humidity control. ch = air leaving heating coil, h = air leaving humidifer, and aw = air leaving air washer.

winter mode cycle of a cold air supply without space humidity control shown in Figure 9.3.4 except that the supply air temperature is higher than space temperature, that is, $T_s > T_r$. To prevent stratification, with the warm supply air staying at a higher level, $(T_s - T_r) > 20^\circ\text{F}$ is not recommended.

Warm Air Supply with Space Humidity Control

This operating cycle (see Figure 9.3.5[b]) is often used for hospitals, nurseries, etc. or in locations where winter is very cold. The state point of supply air must be determined first by drawing a space conditioning line with known SHR_s and then from the calculated supply temperature differential ΔT_s . The difference in humidity ratio ($w_s - w_{ch}$) is the water vapor must be added at the humidifier. Humidifying capacity can be calculated from Equation 9.3.8.

Cold Air Supply with Space Humidity Control

This operating cycle (shown in Figure 9.3.5[c]) is widely used in industrial applications such as textile mills where a cold air supply is needed to remove machine load in winter and maintains the space relative humidity required for the manufacturing process. An outdoor air and recirculating air mixture is often used for the required cold air supply. An air washer is adopted for winter humidification.

Air Economizer Mode

In the *air economizer* mode, as shown by the middle dotted line cycle $o''-cc-sf-s-r$ in [Figure 9.3.4](#), all outdoor air or an outdoor air-recirculating air mixture is used to reduce the refrigeration capacity and improve the indoor air quality during spring, fall, or winter.

When all outdoor air is admitted, it is an open cycle. Outdoor air is cooled and often dehumidified to point cc . After absorbing fan and duct heat gains, it is supplied to the conditioned space. Space air is exhausted entirely through openings, relief dampers, or relief/exhaust fans to the outside. An all-outdoor air-operating mode before the space is occupied is often called an *air purge* operation, used to dilute space air contaminants.

Cool-Down and Warm-Up Modes

In summer, when an air system is shut down during an unoccupied period at night, the space temperature and relative humidity often tend to increase because of infiltration of hot and humid air and heat transfer through the building envelope. The air system is usually started before the space is occupied in cool-down mode to cool the space air until the space temperature falls within predetermined limits.

In winter, the air system is also started before the occupied period to warm up the space air to compensate for the nighttime space temperature setback to 55 to 60°F for energy saving or the drop of space temperature due to heat loss and infiltration.

If dilution of indoor air contaminants is not necessary, only recirculating space air is used during cool-down or warm-up periods in order to save energy.

9.4 Refrigerants and Refrigeration Cycles

Refrigeration and Refrigeration Systems

Refrigeration is the cooling effect of the process of extracting heat from a lower temperature heat source, a substance or cooling medium, and transferring it to a higher temperature heat sink, probably atmospheric air and surface water, to maintain the temperature of the heat source below that of the surroundings.

A *refrigeration system* is a combination of components, equipment, and piping, connected in a sequential order to produce the refrigeration effect. Refrigeration systems that provide cooling for air-conditioning are classified mainly into the following categories:

1. *Vapor compression systems.* In these systems, a compressor(s) compresses the refrigerant to a higher pressure and temperature from an evaporated vapor at low pressure and temperature. The compressed refrigerant is condensed into liquid form by releasing the latent heat of condensation to the condenser water. Liquid refrigerant is then throttled to a low-pressure, low-temperature vapor, producing the refrigeration effect during evaporation. Vapor compression is often called *mechanical refrigeration*, that is, refrigeration by mechanical compression.
2. *Absorption systems.* In an absorption system, the refrigeration effect is produced by means of thermal energy input. After liquid refrigerant produces refrigeration during evaporation at very low pressure, the vapor is absorbed by an aqueous absorbent. The solution is heated by a direct-fired gas furnace or waste heat, and the refrigerant is again vaporized and then condensed into liquid form. The liquid refrigerant is throttled to a very low pressure and is ready to produce the refrigeration effect again.
3. *Gas expansion systems.* In an air or other gas expansion system, air or gas is compressed to a high pressure by compressors. It is then cooled by surface water or atmospheric air and expanded to a low pressure. Because the temperature of air or gas decreases during expansion, a refrigeration effect is produced.

Refrigerants, Cooling Mediums, and Absorbents

A *refrigerant* is a primary working fluid used to produce refrigeration in a refrigeration system. All refrigerants extract heat at low temperature and low pressure during evaporation and reject heat at high temperature and pressure during condensation.

A *cooling medium* is a working fluid cooled by the refrigerant during evaporation to transport refrigeration from a central plant to remote cooling equipment and terminals. In a large, centralized air-conditioning system, it is more economical to pump the cooling medium to the remote locations where cooling is required. Chilled water and brine are cooling media. They are often called secondary refrigerants to distinguish them from the primary refrigerants.

A *liquid absorbent* is a working fluid used to absorb the vaporized refrigerant (water) after evaporation in an absorption refrigeration system. The solution that contains the absorbed vapor is then heated. The refrigerant vaporizes, and the solution is restored to its original concentration to absorb water vapor again.

A numbering system for refrigerants was developed for hydrocarbons and halocarbons. According to ANSI/ASHRAE Standard 34-1992, the first digit is the number of unsaturated carbon-carbon bonds in the compound. This digit is omitted if the number is zero. The second digit is the number of carbon atoms minus one. This is also omitted if the number is zero. The third digit denotes the number of hydrogen atoms plus one. The last digit indicates the number of fluorine atoms. For example, the chemical formula for refrigerant R-123 is CHCl_2CF_3 . In this compound:

No unsaturated carbon-carbon bonds, first digit is 0
 There are two carbon atoms, second digit is $2 - 1 = 1$
 There is one hydrogen atom, third digit is $1 + 1 = 2$

There are three fluorine atoms, last digit is 3

To compare the relative ozone depletion of various refrigerants, an index called the *ozone depletion potential* (ODP) has been introduced. ODP is defined as the ratio of the rate of ozone depletion of 1 lb of any halocarbon to that of 1 lb of refrigerant R-11. For R-11, ODP = 1.

Similar to the ODP, halocarbon global warming potential (HGWP) is an index used to compare the global warming effect of a halocarbon refrigerant with the effect of refrigerant R-11.

Classification of Refrigerants

Nontoxic and nonflammable synthetic chemical compounds called *halogenated hydrocarbons*, or simply *halocarbons*, were used almost exclusively in vapor compression refrigeration systems for comfort air-conditioning until 1986. Because chlorofluorocarbons (CFCs) cause ozone depletion and global warming, they must be replaced. A classification of refrigerants based on ozone depletion follows (see [Table 9.4.1](#)):

Hydrofluorocarbons (HFCs)

HFCs contain only hydrogen, fluorine, and carbon atoms and cause no ozone depletion. HFCs group include R-134a, R-32, R-125, and R-245ca.

HFC's Azeotropic Blends or Simply HFC's Azeotropic

An azeotropic is a mixture of multiple components of volatilities (refrigerants) that evaporate and condense as a single substance and do not change in volumetric composition or saturation temperature when they evaporate or condense at constant pressure. HFC's azeotropics are blends of refrigerant with HFCs. ASHRAE assigned numbers between 500 and 599 for azeotropic. HFC's azeotropic R-507, a blend of R-125/R-143, is the commonly used refrigerant for low-temperature vapor compression refrigeration systems.

HFC's Near Azeotropic

A near azeotropic is a mixture of refrigerants whose characteristics are near those of an azeotropic. Because the change in volumetric composition or saturation temperature is rather small for a near azeotropic, such as, 1 to 2°F, it is thus named. ASHRAE assigned numbers between 400 and 499 for zeotropic. R-404A (R-125/R-134a/R-143a) and R-407B (R-32/R-125/R134a) are HFC's near azeotropic. R-32 is flammable; therefore, its composition is usually less than 30% in the mixture. HFC's near azeotropic are widely used for vapor compression refrigeration systems.

Zeotropic or nonazeotropic, including near azeotropic, shows a change in composition due to the difference between liquid and vapor phases, leaks, and the difference between charge and circulation. A shift in composition causes the change in evaporating and condensing temperature/pressure. The difference in dew point and bubble point during evaporation and condensation is called glide, expressed in °F. Near azeotropic has a smaller glide than zeotropic. The midpoint between the dew point and bubble point is often taken as the evaporating and condensing temperature for refrigerant blends.

Hydrochlorofluorocarbons (HCFCs) and Their Zeotropics

HCFCs contain hydrogen, chlorine, fluorine, and carbon atoms and are not fully halogenated. HCFCs have a much shorter lifetime in the atmosphere (in decades) than CFCs and cause far less ozone depletion (ODP 0.02 to 0.1). R-22, R-123, R-124, etc. are HCFCs. HCFCs are the most widely used refrigerants today.

HCFC's near azeotropic and *HCFC's zeotropic* are blends of HCFCs with HFCs. They are transitional or interim refrigerants and are scheduled for a restriction in production starting in 2004.

Inorganic Compounds

These compounds include refrigerants used before 1931, like ammonia R-717, water R-718, and air R-729. They are still in use because they do not deplete the ozone layer. Because ammonia is toxic and

TABLE 9.4.1 Properties of Commonly Used Refrigerants 40°F Evaporating and 100°F Condensing

		Chemical Formula	Molecular Mass	Ozone Depletion Potential (ODP)	Global Warming Potential (HGWP)	Evaporating Pressure, psia	Condensing Pressure, psia	Compression Ratio	Refrigeration Effect, Btu/lb
Hydrofluorocarbons HFCs									
R-32	Difluoromethane	CH ₂ F ₂	52.02	0.0	0.14	135.6	340.2	2.51	
R-125	Pentafluoroethane	CHF ₂ CF ₃	120.03	0.0	0.84	111.9	276.2	2.47	37.1
R-134a	Tetrafluoroethane	CF ₃ CH ₂ F	102.03	0.0	0.26	49.7	138.8	2.79	65.2
R-143a	Trifluoroethane	CH ₃ CF ₃	84.0	0.0					
R-152a	Difluoroethane	CH ₃ CHF ₂	66.05	0.0		44.8	124.3	2.77	
R-245ca	Pentafluoropropane	CF ₃ CF ₂ CH ₃	134.1	0.0					
HFC's azeotropics									
R-507	R-125/R-143 (45/55)			0.0	0.98				
HFC's near azeotropic									
R-404A	R-125/R-143a (44/52/4)			0.0	0.94				
R-407A	R-32/R-125/R-134a (20/40/40)			0.0	0.49				
R-407C	R-32/R-125/R-134a (23/25/52)			0.0	0.70				
Hydrochlorofluorocarbons HCFCs and their azeotropics									
R-22	Chlorodifluoromethane	CHClF ₂	86.48	0.05	0.40	82.09	201.5	2.46	69.0
R-123	Dichlorotrifluoroethane	CHCl ₂ CF ₃	152.93	0.02	0.02	5.8	20.8	3.59	62.9
R-124	Chlorotetrafluoroethane	CHFClCF ₃	136.47	0.02		27.9	80.92	2.90	5.21
HCFC's near azeotropics									
R-402A	R-22/R-125/R-290 (38/60/2)			0.02	0.63				
HCFC's azeotropics									
R-401A	R-22/R-124/R-152a (53/34/13)			0.37	0.22				
R-401B	R-22/R-124/R-152a (61/28/11)			0.04	0.24				

TABLE 9.4.1 Properties of Commonly Used Refrigerants 40°F Evaporating and 100°F Condensin (continued)

		Chemical Formula	Molecular Mass	Ozone Depletion Potential (ODP)	Global Warming Potential (HGWP)	Evaporating Pressure, psia	Condensing Pressure, psia	Compression Ratio	Refrigeration Effect, Btu/lb
Inorganic compounds									
R-717	Ammonia	NH ₃	17.03	0	0	71.95	206.81	2.87	467.4
R-718	Water	H ₂ O	18.02	0					
R-729	Air		28.97	0					
Chlorofluorocarbons CFCs, halons BFCs and their azeotropic									
R-11	Trichlorofluoromethane	CCl ₃ F	137.38	1.00	1.00	6.92	23.06	3.33	68.5
R-12	Dichlorodifluoromethane	CCl ₂ F ₂	120.93	1.00	3.20	50.98	129.19	2.53	50.5
R-13B1	Bromotrifluoromethane	CBrF ₃	148.93	10					
R-113	Trichlorotrifluoroethane	CCl ₃ FCF ₂	187.39	0.80	1.4	2.64	10.21	3.87	54.1
R-114	Dichlorotetrafluoroethane	CCl ₂ FCF ₃	170.94	1.00	3.9	14.88	45.11	3.03	42.5
R-500	R-12/R-152a (73.8/26.2)		99.31			59.87	152.77	2.55	60.5
R-502	R-22/R-115 (48.8/51.2)		111.63	0.283	4.10				

TABLE 9.4.1 Properties of Commonly Used Refrigerants 40°F Evaporating and 100°F Condensing (continued)

Replacement of	Trade Name	Specific Volume of Vapor ft ³ /lb	Compressor Displacement cfm/ton	Power Consumption hp/ton	Critical Temperature °F	Discharge Temperature °F	Flammability	Safety
Hydrofluorocarbons HFCs								
		0.63			173.1			
		0.33			150.9	103	Nonflammable	A1
	R-12	0.95			213.9		Nonflammable	A1
	R143a							
	R-152a	1.64			235.9		Lower flammable	A2
	R-245ca							
HFC's azeotropics								
	R-502							
	Genetron AZ-50							
HFC's near azeotropic								
	R-22							A1/A1 ^a
	SUVA HP-62							A1/A1 ^a
	R-22							A1/A1 ^a
	KLEA 60							A1/A1 ^a
	R-22							A1/A1 ^a
	KLEA 66							A1/A1 ^a
Hydrochlorofluorocarbons HCFC's and their azeotropics								
		0.66	1.91	0.696	204.8	127	Nonflammable	A1
	R-11	5.88	18.87	0.663	362.6		Nonflammable	B1
		1.30	5.06	0.698	252.5			
HCFC's near azeotropics								
	R-502							A1/A1 ^a
	SUVA HP-80							A1/A1 ^a
HCFC's azeotropics								
	R-12							A1/A1 ^a
	MP 39							A1/A1 ^a
	R-12							A1/A1 ^a
	MP 66							A1/A1 ^a
Inorganic compounds								
		3.98	1.70	0.653	271.4	207	Lower flammability	B2
							Nonflammable	
							Nonflammable	

TABLE 9.4.1 Properties of Commonly Used Refrigerants 40°F Evaporating and 100°F Condensing (continued)

Replacement of	Trade Name	Specific Volume of Vapor ft ³ /lb	Compressor Displacement cfm/ton	Power Consumption hp/ton	Critical Temperature °F	Discharge Temperature °F	Flammability	Safety
Chlorofluorocarbons CFCs, halons BFCs, and their azeotropics								
	R-11	5.43	15.86	0.636	388.4	104	Nonflammable	A1
	R-12	5.79	3.08	0.689	233.6	100	Nonflammable	A1
	R-13B1	0.21			152.6	103	Nonflammable	A1
	R-113	10.71	39.55	0.71	417.4	86	Nonflammable	A1
	R-114	2.03	9.57	0.738	294.3	86	Nonflammable	A1
	R-500 R-12/R-152a (73.8/26.2)	0.79	3.62	0.692	221.9	105	Nonflammable	A1
	R-502 R-22/R-115 (48.8/51.2)					98	Nonflammable	A1

Source: Adapted with permission from *ASHRAE Handbooks 1993 Fundamentals*. Also from refrigerant manufacturers.

^a First classification is that safety classification of the formulated composition. The second is the worst case of fractionation.

flammable, it is used in industrial applications. Inorganic compounds are assigned numbers between 700 and 799 by ASHRAE.

Chlorofluorocarbons, Halons, and Their Azeotropic

CFCs contain only chlorine, fluorine, and carbon atoms. CFCs have an atmospheric lifetime of centuries and cause ozone depletion (ODP from 0.6 to 1). R-11, R-12, R-113, R-114, R-115... are all CFCs.

Halons or BFCs contain bromide, fluorine, and carbon atoms. R-13B1 and R-12B1 are BFCs. They cause very high ozone depletion (ODP for R-13B1 = 10). Until 1995, R-13B1 was used for very low temperature vapor compression refrigeration systems.

Phaseout of CFCs, BFCs, HCFCs, and Their Blends

On September 16, 1987, the European Economic Community and 24 nations, including the United States, signed a document called the Montreal Protocol. It is an agreement to restrict the production and consumption of CFCs and BFCs in the 1990s because of ozone depletion.

The Clean Air Act amendments, signed into law in the United States on November 15, 1990, concern two important issues: the phaseout of CFCs and the prohibition of deliberate venting of CFCs and HCFCs.

In February 1992, President Bush called for an accelerated ban of CFCs in the United States. In late November 1992, representatives of 93 nations meeting in Copenhagen agreed to phase out CFCs beginning January 1, 1996. Restriction on the use of HCFCs will start in 2004, with a complete phaseout by 2030.

In the earlier 1990s, R-11 was widely used for centrifugal chillers, R-12 for small and medium-size vapor compression systems, R-22 for all vapor compression systems, and CFC/HCFC blend R-502 for low-temperature vapor compression systems. Because of the phaseout of CFCs and BFCs before 1996 and HCFCs in the early years of the next century, alternative refrigerants have been developed to replace them:

- R-123 (an HCFC of ODP = 0.02) to replace R-11 is a short-term replacement that causes a slight reduction in capacity and efficiency. R-245ca (ODP = 0) may be the long-term alternative to R-11.
- R-134a (an HFC with ODP = 0) to replace R-12 in broad applications. R-134a is not miscible with mineral oil; therefore, a synthetic lubricant of polyolester is used.
- R-404A (R-125/R-134a/143a) and R-407C (R-32/R-125/R-134a) are both HFCs near azeotropic of ODP = 0. They are long-term alternatives to R-22. For R-407C, the composition of R-32 in the mixture is usually less than 30% so that the blend will not be flammable. R-407C has a drop of only 1 to 2% in capacity compared with R-22.
- R-507 (R-125/R-143a), an HFC's azeotropic with ODP = 0, is a long-term alternative to R-502. Synthetic polyolester lubricant oil will be used for R-507. There is no major performance difference between R-507 and R-502. R-402A (R-22/R-125/R-290), an HCFC's near azeotropic, is a short-term immediate replacement, and drop-in of R-502 requires minimum change of existing equipment except for reset of a higher condensing pressure.

Required Properties of Refrigerants

A refrigerant should not cause ozone depletion. A low global warming potential is required. Additional considerations for refrigerant selection are

1. *Safety*, including toxicity and flammability. ANSI/ASHRAE Standard 34-1992 classifies the *toxicity* of refrigerants as Class A and Class B. Class A refrigerants are of low toxicity. No toxicity was identified when their time-weighted average concentration was less than or equal to 400 ppm, to which workers can be exposed for an 8-hr workday and 40-hr work week without adverse effect. Class B refrigerants are of higher toxicity and produce evidence of toxicity.

ANSI/ASHRAE Standard 34-1982 classifies the *flammability* of refrigerants as Class 1, no flame propagation; Class 2, lower flammability; and Class 3, higher flammability.

The safety classification of refrigerants is based on the combination of toxicity and flammability: A1, A2, A3, B1, B2, and B3. R-134a and R-22 are in the A1 group, lower toxicity and nonflammable; R-123 in the B1 group, higher toxicity and nonflammable; and R-717 (ammonia) in the B2 group, higher toxicity and lower flammability.

2. *Effectiveness of refrigeration cycle.* High effectiveness is a desired property. The power consumed per ton of refrigeration produced, hp/ton or kW/ton, is an index for this assessment. Table 9.4.1 gives values for an ideal single-stage vapor compression cycle.
3. *Oil miscibility.* Refrigerant should be miscible with mineral lubricant oil because a mixture of refrigerant and oil helps to lubricate pistons and discharge valves, bearings, and other moving parts of a compressor. Oil should also be returned from the condenser and evaporator for continuous lubrication. R-22 is partially miscible. R-134a is hardly miscible with mineral oil; therefore, synthetic lubricant of polyolester will be used.
4. *Compressor displacement.* Compressor displacement per ton of refrigeration produced, in cfm/ton, directly affects the size of the positive displacement compressor and therefore its compactness. Ammonia R-717 requires the lowest compressor displacement (1.70 cfm/ton) and R-22 the second lowest (1.91 cfm/ton).
5. Desired properties:
 - Evaporating pressure p_{ev} should be higher than atmospheric. Then noncondensable gas will not leak into the system.
 - Lower condensing pressure for lighter construction of compressor, condenser, piping, etc.
 - A high thermal conductivity and therefore a high heat transfer coefficient in the evaporator and condenser.
 - Dielectric constant should be compatible with air when the refrigerant is in direct contact with motor windings in hermetic compressors.
 - An inert refrigerant that does not react chemically with material will avoid corrosion, erosion, or damage to system components. Halocarbons are compatible with all containment materials except magnesium alloys. Ammonia, in the presence of moisture, is corrosive to copper and brass.
 - Refrigerant leakage can be easily detected. Halide torch, electronic detector, and bubble detection are often used.

Ideal Single-Stage Vapor Compression Cycle

Refrigeration Process

A refrigeration process shows the change of the thermodynamic properties of the refrigerant and the energy and work transfer between the refrigerant and surroundings.

Energy and work transfer is expressed in British thermal units per hour, or Btu/hr. Another unit in wide use is ton of refrigeration, or ton. A ton = 12,000 Btu/hr of heat removed; i.e., 1 ton of ice melting in 24 hr = 12,000 Btu/hr.

Refrigeration Cycles

When a refrigerant undergoes a series of processes like evaporation, compression, condensation, throttling, and expansion, absorbing heat from a low-temperature source and rejecting it to a higher temperature sink, it is said to have undergone a refrigeration cycle. If its final state is equal to its initial state, it is a *closed cycle*; if the final state does not equal the initial state, it is an *open cycle*. Vapor compression refrigeration cycles can be classified as single stage, multistage, compound, and cascade cycles.

A *pressure-enthalpy diagram* or *p-h diagram* is often used to calculate the energy transfer and to analyze the performance of a refrigeration cycle, as shown in Figure 9.4.1. In a *p-h* diagram, pressure p , in psia or psig logarithmic scale, is the ordinate, and enthalpy h , in Btu/lb, is the abscissa. The saturated liquid and saturated vapor line encloses a two-phase region in which vapor and liquid coexist. The two-phase region separates the subcooling liquid and superheated vapor regions. The constant-temperature

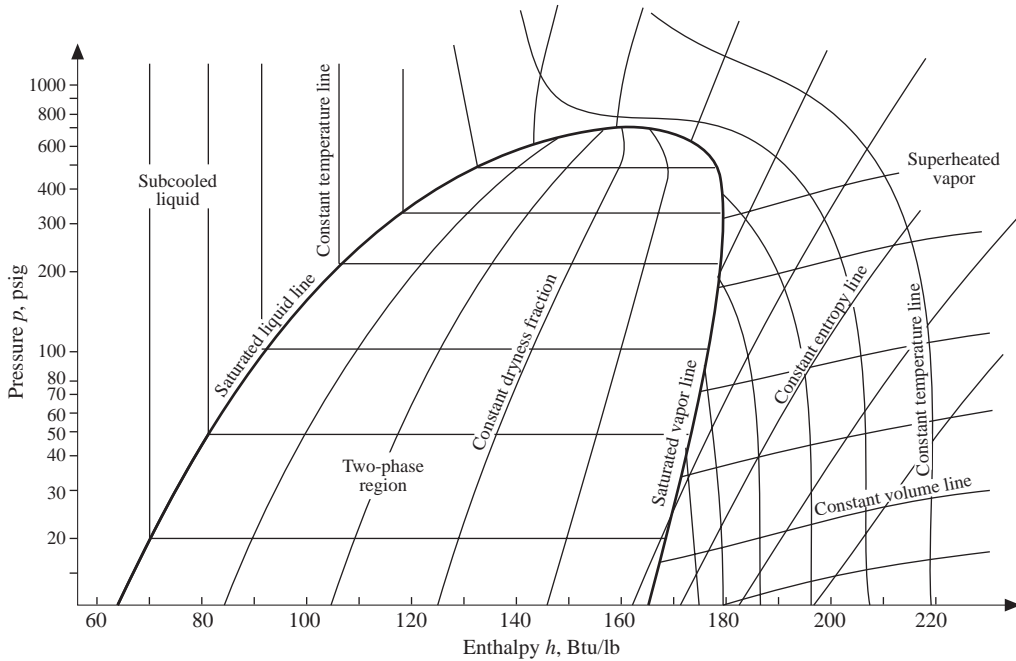


FIGURE 9.4.1 Skeleton of pressure-enthalpy diagram for R-22.

line is nearly vertical in the subcooling region, horizontal in the two-phase region, and curved down sharply in the superheated region.

In the two-phase region, a given saturated pressure determines the saturated temperature and vice versa. The constant-entropy line is curved upward to the right-hand side in the superheated region. Each kind of refrigerant has its own *p-h* diagram.

Refrigeration Processes in an Ideal Single-Stage Cycle

An *ideal cycle* has isentropic compression, and pressure losses in the pipeline, valves, and other components are neglected. All refrigeration cycles covered in this section are ideal. Single stage means a single stage of compression.

There are four refrigeration processes in an ideal single-stage vapor compression cycle, as shown in Figure 9.4.2(a) and (b):

1. Isothermal evaporation process 4–1 — The refrigerant evaporates completely in the evaporator and produces refrigeration effect q_{rf} , in Btu/lb:

$$q_{rf} = (h_1 - h_4) \tag{9.4.1}$$

where h_1, h_4 = enthalpy of refrigerant at state points 1 and 4, respectively, Btu/lb.

2. Isentropic compression process 1–2 — Vapor refrigerant is extracted by the compressor and compressed isentropically from point 1 to 2. The work input to the compressor W_{in} , in Btu/lb, is

$$W_{in} = (h_2 - h_1) \tag{9.4.2}$$

where h_2 = enthalpy of refrigerant at state point 2, Btu/lb.

The greater the difference in temperature/pressure between the condensing pressure p_{con} and evaporating pressure p_{ev} , the higher will be the work input to the compressor.

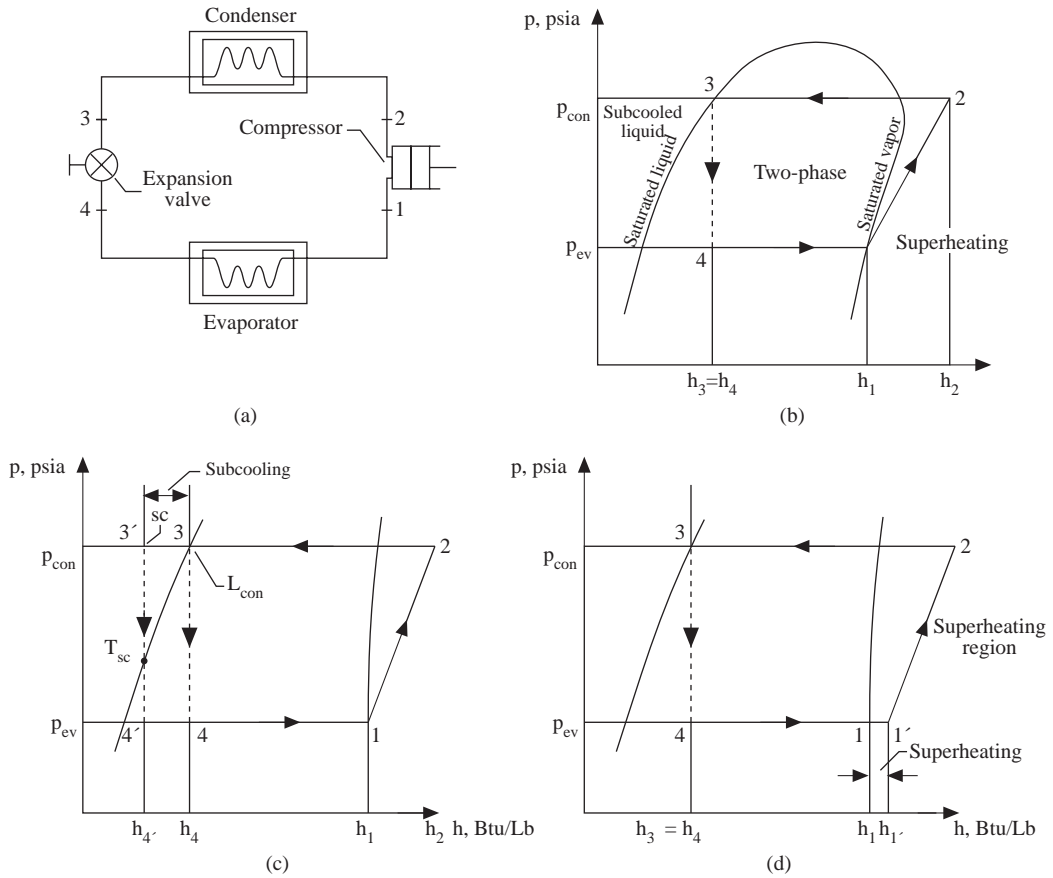


FIGURE 9.4.2 A single-stage ideal vapor compression refrigeration cycle: (a) schematic diagram, (b) p - h diagram, (c) subcooling, and (d) superheating.

3. Isothermal condensation process 2-3 — Hot gaseous refrigerant discharged from the compressor is condensed in the condenser into liquid, and the latent heat of condensation is rejected to the condenser water or ambient air. The heat rejection during condensation, q_{2-3} , in Btu/lb, is

$$-q_{2-3} = (h_2 - h_3) \tag{9.4.3}$$

where h_3 = enthalpy of refrigerant at state point 3, Btu/lb.

4. Throttling process 3-4 — Liquid refrigerant flows through a throttling device (e.g., an expansion valve, a capillary tube, or orifices) and its pressure is reduced to the evaporating pressure. A portion of the liquid flashes into vapor and enters the evaporator. This is the only irreversible process in the ideal cycle, usually represented by a dotted line. For a throttling process, assuming that the heat gain from the surroundings is negligible:

$$h_3 = h_4 \tag{9.4.4}$$

The mass flow rate of refrigerant \dot{m}_r , in lb/min, is

$$\dot{m}_r = q_{rc} / 60q_{rf} \tag{9.4.5}$$

where q_{rc} = refrigeration capacity of the system, Btu/hr.

The ideal single-stage vapor compression refrigeration cycle on a p - h diagram is divided into two pressure regions: high pressure (p_{con}) and low pressure (p_{ev}).

Coefficient of Performance of Refrigeration Cycle

The *coefficient of performance* (COP) is a dimensionless index used to indicate the performance of a thermodynamic cycle or thermal system. The magnitude of COP can be greater than 1.

- If a *refrigerator* is used to produce a refrigeration effect, COP_{ref} is

$$COP_{ref} = q_{rf} / W_{in} \quad (9.4.6)$$

- If a *heat pump* is used to produce a useful heating effect, its performance denoted by COP_{hp} is

$$COP_{hp} = q_{2-3} / W_{in} \quad (9.4.7)$$

- For a heat recovery system when both refrigeration and heating effects are produced, the COP_{hr} is denoted by the ratio of the sum of the absolute values of q_{rf} and q_{2-3} to the work input, or

$$COP_{hr} = (|q_{rf}| + |q_{2-3}|) / W_{in} \quad (9.4.8)$$

Subcooling and Superheating

Condensed liquid is often subcooled to a temperature lower than the saturated temperature corresponding to the condensing pressure p_{con} , in psia or psig, as shown in [Figure 9.4.2\(c\)](#). *Subcooling* increases the refrigeration effect to $q_{rf,sc}$ as shown in [Figure 9.4.2\(c\)](#):

$$q_{rf,sc} = (h_{4'} - h_1) > (h_4 - h_1) \quad (9.4.9)$$

The enthalpy of subcooled liquid refrigerant h_{sc} approximately equals the enthalpy of the saturated liquid refrigerant at subcooled temperature $h_{s,sc}$, both in Btu/lb:

$$h_{sc} = h_{3'} = h_{4'} = h_{l,con} - c_{pr}(T_{con} - T_{sc}) \approx h_{s,sc} \quad (9.4.10)$$

where $h_{3'}$, $h_{4'}$ = enthalpy of liquid refrigerant at state points 3' and 4' respectively, Btu/lb

$h_{l,con}$ = enthalpy of saturated liquid at condensing temperature, Btu/lb

c_{pr} = specific heat of liquid refrigerant at constant pressure, Btu/lb °F

T_{con} = condensing temperature or saturated temperature of liquid refrigerant at condensing pressure, °F

T_{sc} = temperature of subcooled liquid refrigerant, °F

The purpose of *superheating* is to prevent liquid refrigerant flooding back into the compressor and causing slugging damage as shown in [Figure 9.4.2\(d\)](#). The degree of superheating depends mainly on the types of refrigerant feed, construction of the suction line, and type of compressor. The state point of vapor refrigerant after superheating of an ideal system must be at the evaporating pressure with a specific degree of superheat and can be plotted on a p - h diagram for various refrigerants.

Refrigeration Cycle of Two-Stage Compound Systems with a Flash Cooler

A *multistage system* employs more than one compression stage. Multistage vapor compression systems are classified as compound systems and cascade systems. A *compound system* consists of two or more

compression stages connected in series. It may have one high-stage compressor (higher pressure) and one low-stage compressor (lower pressure), several compressors connected in series, or two or more impellers connected internally in series and driven by the same motor.

The *compression ratio* R_{com} is defined as the ratio of discharge pressure p_{dis} to the suction pressure at the compressor inlet p_{suc} :

$$R_{com} = p_{dis}/p_{suc} \tag{9.4.11}$$

Compared to a single-stage system, a multistage has a smaller compression ratio and higher compression efficiency for each stage of compression, greater refrigeration effect, lower discharge temperature at the high-stage compressor, and greater flexibility. At the same time, a multistage system has a higher initial cost and more complicated construction.

The pressure between the discharge pressure of the high-stage compressor and the suction pressure of the low-stage compressor of a multistage system is called *interstage pressure* p_i , in psia. Interstage pressure for a two-stage system is usually determined so that the compression ratios are nearly equal between two stages for a higher COP. Then the interstage pressure is

$$p_i = \sqrt{(p_{con} p_{ev})} \tag{9.4.12}$$

where p_{con} , p_{ev} = condensing and evaporating pressures, psia.

For a multistage system of n stages, then, the compression ratio of each stage is

$$R_{com} = (p_{con}/p_{suc})^{1/n} \tag{9.4.13}$$

Figure 9.4.3(a) shows a schematic diagram and Figure 9.4.3(b) the refrigeration cycle of a two-stage compound system with a flash cooler. A *flash cooler*, sometimes called an economizer, is used to subcool the liquid refrigerant to the saturated temperature corresponding to the interstage pressure by vaporizing a portion of the liquid refrigerant in the flash cooler.

Based on the principle of heat balance, the fraction of evaporated refrigerant, x , or quality of the mixture in the flash cooler is

$$x = (h_{5'} - h_8)/(h_7 - h_8) \tag{9.4.14}$$

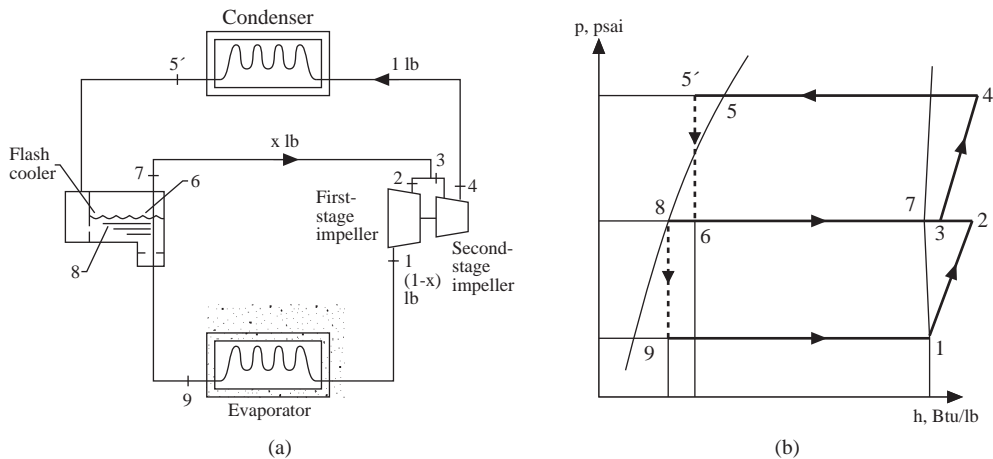


FIGURE 9.4.3 Two-stage compound system with a flash cooler: (a) schematic diagram and (b) refrigeration cycle.

where h_5, h_7, h_8 = enthalpy of the refrigerant at state points 5', 7, and 8, respectively, Btu/lb. The coefficient of performance of the refrigeration cycle of a two-stage compound system with a flash cooler, COP_{ref} , is given as

$$COP_{ref} = q_{rf} / W_{in} = (1 - x)(h_1 - h_9) / [(1 - x)(h_2 - h_1) + (h_4 - h_3)] \tag{9.4.15}$$

where h_1, h_2, h_3, h_4, h_9 = enthalpy of refrigerant at state points 1, 2, 3, 4, and 9, respectively, Btu/lb. The mass flow rate of refrigerant flowing through the condenser, \dot{m}_r , in lb/min, can be calculated as

$$\dot{m}_r = q_{rc} / 60q_{rf} \tag{9.4.16}$$

Because a portion of liquid refrigerant is flashed into vapor in the flash cooler and goes directly to the second-stage impeller inlet, less refrigerant is compressed in the first-stage impeller. In addition, the liquid refrigerant in the flash cooler is cooled to the saturated temperature corresponding to the interstage temperature before entering the evaporator, which significantly increases the refrigeration effect of this compound system. Two-stage compound systems with flash coolers are widely used in large central air-conditioning systems.

Cascade System Characteristics

A *cascade system* consists of two independently operated single-stage refrigeration systems: a lower system that maintains a lower evaporating temperature and produces a refrigeration effect and a higher system that operates at a higher evaporating temperature as shown in Figure 9.4.4(a) and (b). These two separate systems are connected by a *cascade condenser* in which the heat released by the condenser in the lower system is extracted by the evaporator in the higher system.

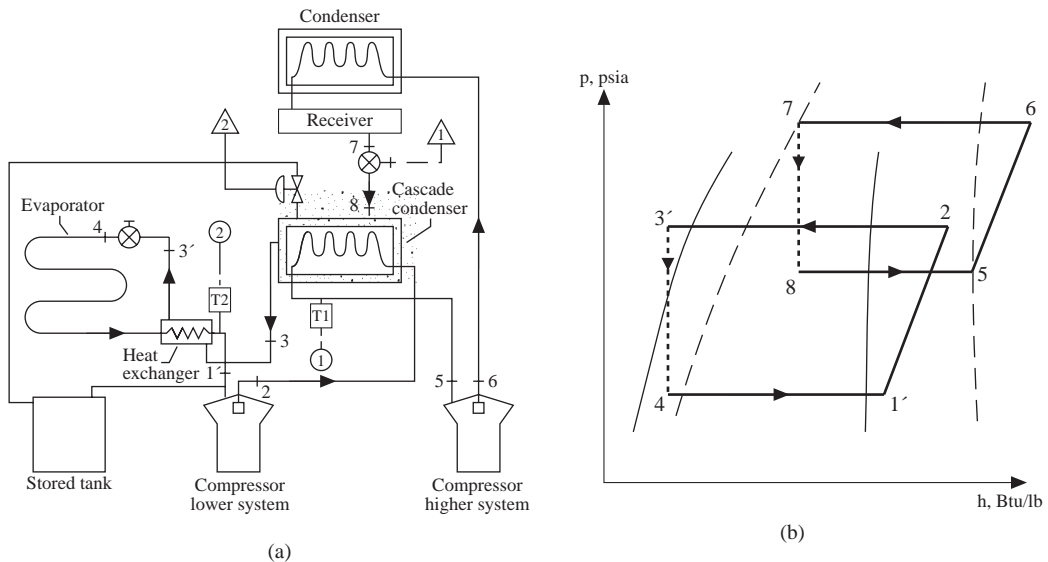


FIGURE 9.4.4 Cascade system: (a) schematic diagram and (b) refrigeration cycle.

A heat exchanger is often used between the liquid refrigerant from the condenser and the vapor refrigerant leaving the evaporator of the lower system. When the system is shut down in summer, a relief valve connected to a stored tank should be used to relieve the higher pressure of refrigerant at the higher storage temperature.

The main advantages of a cascade system compared with a compound system are that different refrigerants, oils, and equipment can be used for the lower and higher systems. Disadvantages are the overlap of the condensing temperature of the lower system and the evaporating temperature of the higher system because of the heat transfer in the cascade condenser and a more complicated system.

The refrigeration effect q_{rf} of the cascade system is

$$q_{\text{rf}} = (h_1 - h_4) \quad (9.4.17)$$

where h_1, h_4 = enthalpy of the refrigerant leaving and entering the evaporator of the lower system, Btu/lb. The total work input to the compressors in both higher and lower systems W_{in} , in Btu/lb, can be calculated as

$$W_{\text{in}} = (h_2 - h_{1'}) + \dot{m}_h (h_6 - h_5) / \dot{m}_l \quad (9.4.18)$$

where h_2 = enthalpy of refrigerant discharged from the compressor of the lower system
 $h_{1'}$ = enthalpy of the vapor refrigerant leaving the heat exchanger
 h_6, h_5 = enthalpy of the refrigerant leaving and entering the high-stage compressor
 \dot{m}_h, \dot{m}_l = mass flow rate of the refrigerant of the higher and lower systems, respectively

The coefficient of performance of a cascade system COP_{ref} is

$$\text{COP}_{\text{ref}} = q_{\text{rf}} / W_{\text{in}} = \dot{m}_l (h_1 - h_4) / \left[\dot{m}_l (h_2 - h_{1'}) + \dot{m}_h (h_6 - h_5) \right] \quad (9.4.19)$$

9.5 Outdoor Design Conditions and Indoor Design Criteria

Outdoor Design Conditions

In principle, the capacity of air-conditioning equipment should be selected to offset or compensate for the space load so that indoor design criteria can be maintained if the outdoor weather does not exceed the design values. Outdoor and indoor design conditions are used to calculate the design space loads. In energy use calculations, hour-by-hour outdoor climate data of a design day should be adopted instead of summer and winter design values.

ASHRAE Handbook 1993 Fundamentals (Chapter 24 and 27) and *Wang's Handbook of Air Conditioning and Refrigeration* (Chapter 7) both list tables of climate conditions for the U.S. and Canada based on the data from the National Climate Data Center (NCDC), U.S. Air Force, U.S. Navy, and Canadian Atmospheric Environment Service. In these tables:

- *Summer design dry bulb temperature* in a specific location $T_{o,s}$, in °F, is the rounded higher integral number of the statistically determined summer outdoor design dry bulb temperature $T_{o,ss}$ so that the average number of hours of occurrence of outdoor dry bulb temperature T_o higher than $T_{o,ss}$ during June, July, August, and September is less than 1, 2.5, or 5% of the total number of hours in these summer months (2928 hr). The data are an average of 15 years. An occurrence of less than 2.5% of 2928 hr of summer months, that is, $0.025 \times 2928 = 73$ hr, is most widely used.
- *Summer outdoor mean coincident wet bulb temperature* $T'_{o,s}$, in °F, is the mean of all the wet bulb temperatures at the specific summer outdoor design dry bulb temperature $T_{o,s}$ during the summer months.
- *Summer outdoor 2.5% design wet bulb temperature* is the design wet bulb temperature that has an average annual occurrence of $T'_o > T'_{o,s}$ less than 73 hr. This design value is often used for evaporative cooling design.
- *Mean daily range*, in °F, is the difference between the average daily maximum and the average daily minimum temperature during the warmest month.
- In *ASHRAE Handbook 1993 Fundamentals*, *solar heat gain factors* (SHGFs), in Btu/h.ft², are the average solar heat gain per hour during cloudless days through double-strength sheet (DSA) glass. The *maximum SHGFs* are the maximum values of SHGFs on the 21st of each month for a specific latitude.
- *Winter outdoor design dry bulb temperature* $T_{o,w}$, in °F, is the rounded lower integral value of the statically determined winter outdoor design temperature $T_{o,ws}$, so that the annual average number of hours of occurrence of outdoor temperature $T_o > T_{o,ws}$ is equal to or exceeds 99%, or 97.5% of the total number of hours in December, January, and February (2160 hr).

A *degree day* is the difference between a base temperature and the mean daily outdoor air temperature $T_{o,m}$ for any one day, in °F. The total numbers of heating degree days HDD65 and cooling degree days CDD65 referring to a base temperature of 65°F per annum are

$$\begin{aligned} \text{HDD65} &= \sum_{n=1} (65 - T_{o,m}) \\ \text{CDD65} &= \sum_{m=1} (T_{o,m} - 65) \end{aligned} \tag{9.5.1}$$

where n = number of days for which $T_{o,m} < 65^\circ\text{F}$
 m = number of days for which $T_{o,m} > 65^\circ\text{F}$

Indoor Design Criteria and Thermal Comfort

Indoor design criteria, such as space temperature, humidity, and air cleanliness, specify the requirements for the indoor environment as well as the quality of an air-conditioning or HVAC&R project.

The human body requires energy for physical and mental activity. This energy comes from the oxidation of food. The rate of heat release from the oxidation process is called the *metabolic rate*, expressed in met (1 met = 18.46 Btu/h.ft²). The metabolic rate depends mainly on the intensity of the physical activity of the human body. Heat is released from the human body by two means: *sensible heat exchange* and *evaporative heat loss*. Experience and experiments all show that there is thermal comfort only under these conditions:

- Heat transfer from the human body to the surrounding environment causes a steady state of thermal equilibrium; that is, there is no heat storage in the body core and skin surface.
- Evaporative loss or regulatory sweating is maintained at a low level.

The physiological and environmental factors that affect the thermal comfort of the occupants in an air-conditioned space are mainly:

1. Metabolic rate M determines the amount of heat that must be released from the human body.
2. *Indoor air temperature* T_r and *mean radiant temperature* T_{rad} , both in °F. The *operating temperature* T_o is the weighted sum of T_r and T_{rad} . T_{rad} is defined as the temperature of a uniform black enclosure in which the surrounded occupant would have the same radiative heat exchange as in an actual indoor environment. T_r affects both the sensible heat exchange and evaporative losses, and T_{rad} affects only sensible heat exchange. In many indoor environments, $T_{rad} \approx T_r$.
3. Relative humidity of the indoor air ϕ_r , in %, which is the primary factor that influences evaporative heat loss.
4. Air velocity of the indoor air v_r , in fpm, which affects the heat transfer coefficients and therefore the sensible heat exchange and evaporative loss.
5. *Clothing insulation* R_{cl} , in clo (1 clo = 0.88 h.ft².°F/Btu), affects the sensible heat loss. Clothing insulation for occupants is typically 0.6 clo in summer and 0.8 to 1.2 clo in winter.

Indoor Temperature, Relative Humidity, and Air Velocity

For comfort air-conditioning systems, according to ANSI/ASHRAE Standard 55-1981 and ASHRAE/IES Standard 90.1-1989, the following indoor design temperatures and air velocities apply for conditioned spaces where the occupant's activity level is 1.2 met, indoor space relative humidity is 50% (in summer only), and $T_r = T_{rad}$:

	Clothing insulation (clo)	Indoor temperature (°F)	Air velocity (fpm)
Winter	0.8–0.9	69–74	<30
Summer	0.5–0.6	75–78	<50

If a suit jacket is the clothing during summer for occupants, the summer indoor design temperature should be dropped to 74 to 75°F.

Regarding the indoor humidity:

1. Many comfort air-conditioning systems are not equipped with humidifiers. Winter indoor relative humidity should not be specified in such circumstances.
2. When comfort air-conditioning systems are installed with humidifiers, ASHRAE/IES Standard 90.1-1989 requires that the humidity control prevent “the use of fossil fuel or electricity to produce humidity in excess of 30% ... or to reduce relative humidity below 60%.”

3. Indoor relative humidity should not exceed 75% to avoid increasing bacterial and viral populations.
4. For air-conditioning systems that use flow rate control in the water cooling coil, space indoor relative humidity may be substantially higher in part load than at full load.

Therefore, for comfort air-conditioning systems, the recommended indoor relative humidities, in %, are

	Tolerable range	Preferred value
Summer	30–65	40–50
Winter		
With humidifier		25–30
Without humidifier		Not specified

In surgical rooms or similar health care facilities, the indoor relative humidity is often maintained at 40 to 60% year round.

Indoor Air Quality and Outdoor Ventilation Air Requirements

According to the National Institute for Occupational Safety and Health (NIOSH), 1989, the causes of indoor air quality complaints in buildings are inadequate outdoor ventilation air, 53%; indoor contaminants, 15%; outdoor contaminants, 10%; microbial contaminants, 5%; construction and furnishings, 4%; unknown and others, 13%. For space served by air-conditioning systems using low- and medium-efficiency air filters, according to the U.S. Environmental Protection Agency (EPA) and Consumer Product Safety Commission (CPSC) publication “A Guide to Indoor Air Quality” (1988) and the field investigations reported by Bayer and Black (1988), *indoor air contaminants* may include some of the following:

1. *Total particulate concentration.* This concentration comprises particles from building materials, combustion products, mineral fibers, and synthetic fibers. In February 1989, the EPA specified the allowable indoor concentration of particles of 10 μm and less in diameter (which penetrate deeply into lungs) as:
 - 50 $\mu\text{g}/\text{m}^3$ (0.000022 grain/ft³), 1 year
 - 150 $\mu\text{g}/\text{m}^3$ (0.000066 grain/ft³), 24 hr
 In these specifications, “1 year” means maximum allowable exposure per day over the course of a year.
2. *Formaldehyde and organic gases.* Formaldehyde is a colorless, pungent-smelling gas. It comes from pressed wood products, building materials, and combustion. Formaldehyde causes eye, nose, and throat irritation as well as coughing, fatigue, and allergic reactions. Formaldehyde may also cause cancer. Other organic gases come from building materials, carpeting, furnishings, cleaning materials, etc.
3. *Radon.* Radon, a colorless and odorless gas, is released by the decay of uranium from the soil and rock beneath buildings, well water, and building materials. Radon and its decay products travel through pores in soil and rock and infiltrate into buildings along the cracks and other openings in the basement slab and walls. Radon at high levels causes lung cancer. The EPA believes that levels in most homes can be reduced to 4 pCi/l (picocuries per liter) of air. The estimated national average is 1.5 pCi/l, and levels as high as 200 pCi/l have been found in houses.
4. *Biologicals.* These include bacteria, fungi, mold and mildew, viruses, and pollen. They may come from wet and moist walls, carpet furnishings, and poorly maintained dirty air-conditioning systems and may be transmitted by people. Some biological contaminants cause allergic reactions, and some transmit infectious diseases.

5. *Combustion products.* These include environmental tobacco smoke, nitrogen dioxide, and carbon monoxide. *Environmental tobacco* smoke from cigarettes is a discomfort factor to other persons who do not smoke, especially children. Nicotine and other tobacco smoke components cause lung cancer, heart disease, and many other diseases. *Nitrogen dioxide* and *carbon monoxide* are both combustion products from unvented kerosene and gas space heaters, woodstoves, and fireplaces. Nitrogen dioxide (NO₂) causes eye, nose, and throat irritation; may impair lung function; and increases respiratory infections. Carbon monoxide (CO) causes fatigue at low concentrations; impaired vision, headache, and confusion at higher concentrations; and is fatal at very high concentrations. Houses without gas heaters and gas stoves may have CO levels varying from 0.5 to 5 parts per million (ppm).
6. *Human bioeffluents.* These include the emissions from breath including carbon dioxide exhaled from the lungs, body odors from sweating, and gases emitted as flatus.

There are three basic means of reducing the concentration of indoor air contaminants and improving indoor air quality: (1) eliminate or reduce the source of air pollution, (2) enhance the efficiency of air filtration, and (3) increase the ventilation (outdoor) air intake. Dilution of the concentrations of indoor contaminants by outdoor ventilation air is often the simple and cheapest way to improve indoor air quality. The amount of outdoor air required for metabolic oxidation is rather small.

Abridged outdoor air requirements listed in ANSI/ASHRAE Standard 62-1989 are as follows:

Applications	cfm/person
Hotels, conference rooms, offices	20
Retail stores	0.2–0.3 cfm/ft ²
Classrooms, theaters, auditoriums	15
Hospital patient rooms	25

These requirements are based on the analysis of dilution of CO₂ as the representative human bioeffluent to an allowable indoor concentration of 1000 ppm. Field measurements of daily maximum CO₂ levels in office buildings reported by Persily (1993) show that most of them were within the range 400 to 820 ppm. The quality of outdoor air must meet the EPA's National Primary and Secondary Ambient Air Quality Standards, some of which is listed below:

Pollutants	Long-term concentration			Short-term concentration		
	µg/m ³	ppm	Exposure	µg/m ³	ppm	Exposure
Particulate	75		1 year	260		24 hr
SO ₂	80	0.03	1 year	365	0.14	24 hr
CO				40,000	35	1 hr
				10,000	9	8 hr
NO ₂	100	0.055	1 year			
Lead	1.5		3 months			

Here exposure means average period of exposure.

If unusual contaminants or unusually strong sources of contaminants are introduced into the space, or recirculated air is to replace part of the outdoor air supply for occupants, then acceptable indoor air quality is achieved by controlling known and specific contaminants. This is called an indoor air quality procedure. Refer to ANSI/ASHRAE Standard 62-1989 for details.

Clean Rooms

Electronic, pharmaceutical, and aerospace industries and operating rooms in hospitals all need strict control of air cleanliness during manufacturing and operations. According to ASHRAE Handbook 1991 HVAC Applications, clean rooms can be classified as follows based on the particle count per ft³:

Class	Particle size	
	0.5 μm and larger	5 μm and larger
	Particle count per ft^3 not to exceed	
1	1	0
10	10	0
100	100	
1000	1000	
10,000	10,000	65
100,000	100,000	700

For clean rooms, space temperature is often maintained at $72 \pm 2^\circ\text{F}$ and space humidity at $45 \pm 5\%$. Here, $\pm 2^\circ\text{F}$ and $\pm 5\%$ are allowable tolerances. Federal Standard 209B specifies that the ventilation (outdoor air) rate should be 5 to 20% of the supply air.

Space Pressure Differential

Most air-conditioning systems are designed to maintain a slightly higher pressure than the surroundings, a positive pressure, to prevent or reduce infiltration and untreated air entering the space directly. For laboratories, restrooms, or workshops where toxic, hazardous, or objectional gases or contaminants are produced, a slightly lower pressure than the surroundings, a negative pressure, should be maintained to prevent or reduce the diffusion of these contaminants' exfiltrate to the surrounding area.

For comfort air-conditioning systems, the recommended pressure differential between the indoor and outdoor air is 0.02 to 0.05 in. WG. WG indicates the pressure at the bottom of a top-opened water column of specific inches of height; 1 in. WG = 0.03612 psig.

For clean rooms, Federal Standard No. 209B, Clean Rooms and Work Stations Requirements (1973), specifies that the minimum positive pressure between the clean room and any adjacent area with lower cleanliness requirements should be 0.05 in. WG with all entryways closed. When the entryways are open, an outward flow of air is to be maintained to prevent migration of contaminants into the clean room. In comfort systems, the space pressure differential is usually not specified in the design documents.

Sound Levels

Noise is any unwanted sound. In air-conditioning systems, noise should be attenuated or masked with another less objectionable sound.

Sound power is the capability to radiate power from a sound source excited by an energy input. The intensity of sound power is the output from a sound source expressed in watts (W). Due to the wide variation of sound output at a range of 10^{20} to 1, it is more convenient to use a logarithmic expression to define a *sound power level* L_w , in dB:

$$L_w = 10 \log(w/10^{-12} \text{ W}) \text{ re } 1 \text{ pW} \quad (9.5.2)$$

where w = sound source power output, W.

The human ear and microphones are sound pressure sensitive. Similarly to the sound power level, the *sound pressure level* L_p , in dB, is defined as:

$$L_p = 20 \log(p/2 \times 10^{-5} \text{ Pa}) \text{ re } 20 \mu\text{Pa} \quad (9.5.3)$$

where p = sound pressure, Pa.

The sound power level of any sound source is a fixed output. It cannot be measured directly; it can only be calculated from the measured sound pressure level. The sound pressure level at any one point is affected by the distance from the source and the characteristics of the surroundings.

Human ears can hear frequencies from 20 Hz to 20 kHz. For convenience in analysis, sound frequencies are often subdivided into eight octave bands. An *octave* is a frequency band in which the frequency of the upper limit of the octave is double the frequency of the lower limit. An octave band is represented by its center frequency, such as 63, 125, 250, 500, 1000, 2000, 4000, and 8000 Hz. On 1000 Hz the octave band has a higher limit of 1400 Hz and a lower limit of 710 Hz. Human ears do not respond in the same way to low frequencies as to high frequencies.

The object of noise control in an air conditioned space is to provide background sound low enough that it does not interfere with the acoustical requirements of the occupants. The distribution of background sound should be balanced over a broad range of frequencies, that is, without whistle, hum, rumble, and beats.

The most widely used criteria for sound control are the noise criteria NC curves. The shape of NC curves is similar to the equal-loudness contour representing the response of the human ear. NC curves also intend to indicate the permissible sound pressure level of broad-band noise at various octave bands rated by a single NC curve. NC curves are practical and widely used.

Other criteria used are room criteria RC curves and A-weighted sound level, dBA. RC curves are similar to NC curves except that the shape of the RC curves is a close approximation to a balanced, bland-sounding spectrum. The A-weighted sound level is a single value and simulates the response of the human ear to sound at low sound pressure levels.

The following are abridged indoor design criteria, NC or RC range, listed in *ASHRAE Handbook 1987 Systems and Applications*:

Type of area	Recommended NC or RC range (dB)
Hotel guest rooms	30–35
Office	
Private	30–35
Conference	25–30
Open	30–35
Computer equipment	40–45
Hospital, private	25–30
Churches	25–30
Movie theaters	30–35

For industrial factories, if the machine noise in a period of 8 hr exceeds 90 dBA, Occupational Safety and Health Administration Standard Part 1910.95 requires the occupants to use personal protection equipment. If the period is shorter, the dBA level can be slightly higher. Refer to this standard for details.

9.6 Load Calculations

Space Loads

Space, Room, and Zone

Space indicates a volume or a site without partitions, or a partitioned room or a group of rooms. A *room* is an enclosed or partitioned space that is considered as a single load. An air-conditioned room does not always have an individual zone control system. A *zone* is a space of a single room or group of rooms having similar loads and operating characteristics. An air-conditioned zone is always installed with an individual control system. A typical floor in a building may be treated as a single zone space, or a *multizone space* of perimeter, interior, east, west, south, north, ... zones.

Space and equipment loads can be classified as:

1. *Space heat gain* q_{es} , in Btu/hr, is the rate of heat transfer entering a conditioned space from an external heat source or heat releases to the conditioned space from an internal source. The rate of sensible heat entering the space is called *sensible heat gain* q_{es} , whereas the rate of latent heat entering the space is called *latent heat gain* q_{el} . In most load calculations, the time interval is often 1 hr, and therefore q_{es} , q_{es} , and q_{el} are all expressed in Btu/hr.
2. *Space cooling load* or simply *cooling load* q_{rc} , also in Btu/hr, is the rate at which heat must be removed from a conditioned space to maintain a constant space temperature and an acceptable relative humidity. The sensible heat removed is called *sensible cooling load* q_{rs} , and the latent heat removed is called *latent cooling load* q_{rl} , both in Btu/hr.
3. *Space heat extraction rate* q_{ex} , in Btu/hr, is the rate at which heat is removed from the conditioned space. When the space air temperature is constant, $q_{ex} = q_{rc}$.
4. *Space heating load* q_{th} , in Btu/hr, is the rate at which heat must be added to the conditioned space to maintain a constant temperature.
5. *Coil load* q_c , in Btu/hr, is the rate of heat transfer at the coil. The cooling *coil load* q_{cc} is the rate of heat removal from the conditioned air by the chilled water or refrigerant inside the coil. The *heating coil load* q_{ch} is the rate of heat energy addition to the conditioned air by the hot water, steam, or electric elements inside the coil.
6. *Refrigeration load* q_{rl} , in Btu/hr, is the rate at which heat is extracted by the evaporated refrigerant at the evaporator. For packaged systems using a DX coil, $q_{rl} = q_{cc}$. For central systems:

$$q_{rl} = q_{cc} + q_{pi} + q_{pu} + q_{s,t} \quad (9.6.1)$$

where q_{pi} = chilled water piping heat gain, Btu/hr
 q_{pu} = pump power heat gain, Btu/hr
 $q_{s,t}$ = storage tank heat gain, if any, Btu/hr

Heat gains q_{pi} and q_{pu} are usually about 5 to 10% of the cooling coil load q_{cc} .

Convective Heat and Radiative Heat

Heat enters a space and transfer to the space air from either an external source or an internal source is mainly in the form of *convective heat* and *radiative heat* transfer.

Consider radiative heat transfer, such as solar radiation striking the outer surface of a concrete slab as shown in [Figure 9.6.1\(a\) and \(b\)](#). Most of the radiative heat is absorbed by the slab. Only a small fraction is reflected. After the heat is absorbed, the outer surface temperature of the slab rises. If the slab and space air are in thermal equilibrium before the absorption of radiative heat, heat is convected from the outer surface of the slab to the space air as well as radiated to other surfaces. At the same time, heat is conducted from the outer surface to the inner part of the slab and stored there when the temperature of the inner part of the slab is lower than that of its outer surface. Heat convected from the outer surface of the concrete slab to the space air within a time interval forms the sensible cooling load.

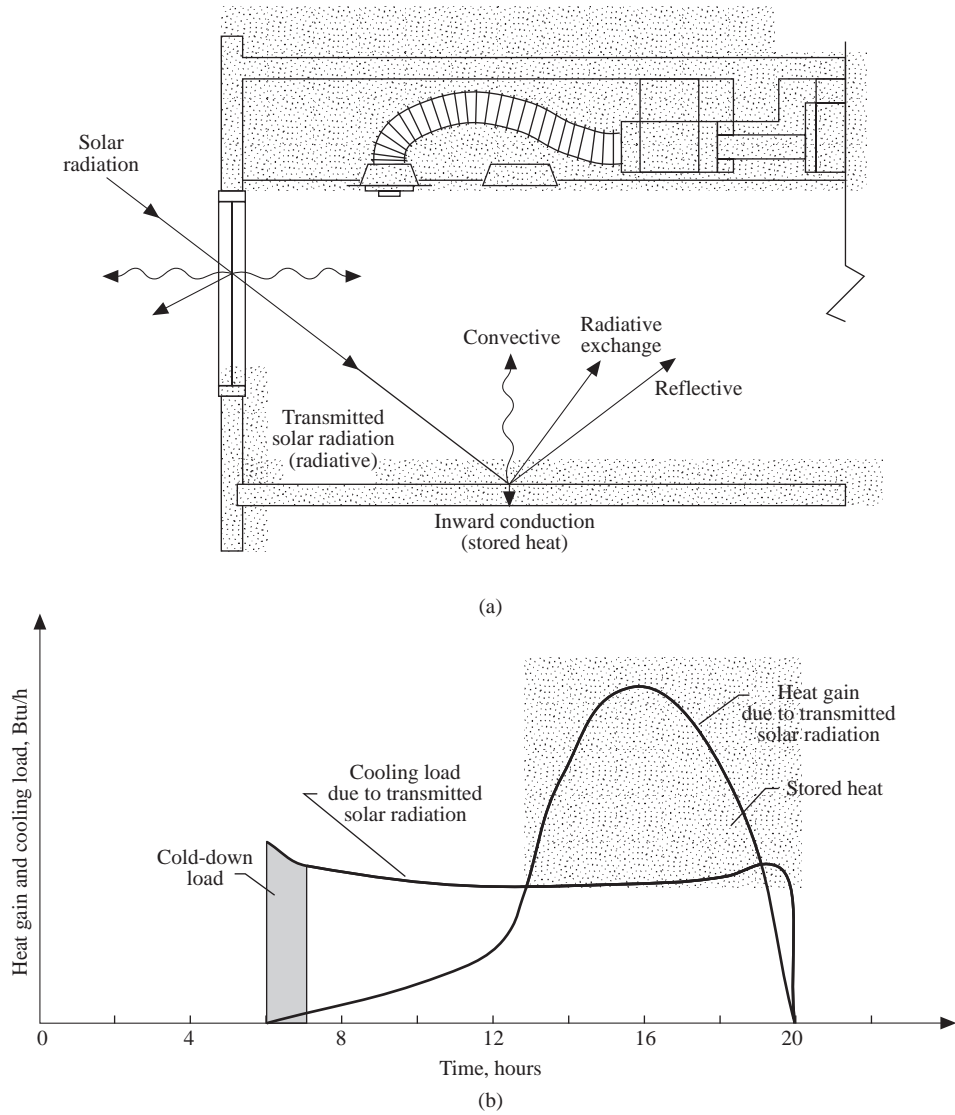


FIGURE 9.6.1 Solar heat gain from west window and its corresponding space cooling load for a night shutdown air system: (a) convective and radiative heat transfer and (b) heat gain and cooling load curves.

The sensible heat gain entering the conditioned space does not equal the sensible cooling load during the same time interval because of the stored heat in the building envelope. Only the convective heat becomes cooling load instantaneously. The sum of the convective heats from the outer surfaces, including the outer surfaces of the internal heat gains in a conditioned space, becomes cooling load. This phenomenon results in a smaller cooling load than heat gain, as shown in Figure 9.6.1(a) and (b). According to *ASHRAE Handbook 1993 Fundamentals*, the percentages of convective and radiative components of the sensible heat gains are as follows:

Sensible heat gains	Convective (%)	Radiative (%)
Solar radiation with internal shading	42	58
Fluorescent lights	50	50
Occupants	67	33
External wall, inner surface	40	60

Load Profile, Peak Load, and Block Load

A *load profile* shows the variation of space, zone, floor, or building load in a certain time period, such as a 24-hr day-and-night cycle. In a load profile, load is always plotted against time. The load profile depends on the outdoor climate as well as the space operating characteristics.

Peak load is the maximum cooling load in a load profile. *Block load* is the sum of the zone loads and floor loads at a specific time. The sum of the zone peak loads in a typical floor does not equal the block load of that floor because the zone peak loads may all not appear at the same time.

Moisture Transfer in Building Envelope

Moisture transfer takes place along two paths:

1. Moisture migrates in the building envelope in both liquid and vapor form. It is mainly liquid if the relative humidity of the ambient air exceeds 50%. Liquid flow is induced by capillary flow and moisture content gradient. Vapor diffusion is induced by vapor pressure gradients. *Moisture content* is defined as the ratio of the mass of moisture contained in a solid to the mass of bone-dry solid. During the migration, the moisture content and the vapor pressure are in equilibrium at a specific temperature and location.
2. Air leakage and its associated water vapor infiltrate or exfiltrate through the cracks, holes, and gaps between joints because of poor construction of the building envelope. The driving potential of this air leakage and associated water vapor is the pressure differential across the building envelope. If the insulating material is of open-cell structure, air leakage and associated water vapor may penetrate the perforated insulating board through cracks and gaps. Condensation, even freezing, will occur inside the perforated insulation board if the temperature of the board is lower than the dew point of the leaked air or the freezing point of the water.

In most comfort air-conditioning systems, usually only the space temperature is controlled within limits. A slight variation of the space relative humidity during the operation of the air system is often acceptable. Therefore, the store effect of moisture is ignored except in conditioned spaces where both temperature and relative humidity need to be controlled or in a hot and humid area where the air system is operated at night shutdown mode. In most cases, latent heat gain is considered equal to latent cooling load instantaneously. For details refer to Wang's *Handbook of Air Conditioning and Refrigeration* (1993), Chapters 6 and 7.

Cooling Load Calculation Methodology

Basic considerations include the following:

- It is assumed that equations of heat transfer for cooling load calculation within a time interval are linear. It is also assumed that the superposition principle holds. When a number of changes occur simultaneously in the conditioned space, they will proceed as if independent of each other. The total change is the sum of the responses caused by the individual changes.
- Space load calculations are often performed by computer-aided design (CAD), with market-available software like DOE-2.1D, TRACE-600, and Carrier E20-II Loads.
- Peak load calculations evaluate the maximum load to size and select the equipment. The energy analysis program compares the total energy use in a certain period with various alternatives in order to determine the optimum one.
- The methodology of various cooling load calculations is mainly due to their differences in the conversion of space radiative heat gains into space cooling loads. Convective heat, latent heat, and sensible heat gains from infiltration are all equal to cooling loads instantaneously.
- Space cooling load is used to calculate the supply volume flow rate and to determine the size of the air system, ducts, terminals, and diffusers. The coil load is used to determine the size of the

cooling coil and the refrigeration system. Space cooling load is a component of the cooling coil load.

The Rigorous Approach

The rigorous approach to the calculation of the space cooling load consists of (1) finding the inside surface temperatures of the building structures that enclose the conditioned space due to heat balance at time t and (2) calculating the sum of the convective heats transferred from these surfaces as well as from the occupants, lights, appliances, and equipment in the conditioned space at time t .

The inside surface temperature of each surface $T_{i,t}$, in °F, can be found from the following simultaneous heat balance equations:

$$q_{i,t} = \left[h_{ci}(T_{r,t} - T_{i,t}) + \sum_{j=1}^m g_{ij}(T_{j,t} - T_{i,t}) \right] A_i + S_{i,t} + L_{i,t} + P_{i,t} + E_{i,t} \quad (9.6.2)$$

where h_{ci} = convective heat transfer coefficient, Btu/hr.ft².°F
 g_{ij} = radiative heat transfer factor between inside surface i and inside surface j , Btu/hr.ft².°F
 $T_{r,t}$ = space air temperature at time t , °F
 $T_{i,t}$, $T_{j,t}$ = average temperature of inside surfaces i and j at time t , °F
 A_i = area of inside surface i , ft²
 $S_{i,t}$, $L_{i,t}$, $P_{i,t}$, $E_{i,t}$ = solar radiation transmitted through windows and radiative heat from lights, occupants, and equipment absorbed by inside surface i at time t , Btu/hr

In Equation (9.6.2), $q_{i,t}$ in Btu/hr, is the conductive heat that comes to surface i at time t because of the temperature excitation on the outer opposite surface of i . This conductive heat can be found by solving the partial differential equations or by numerical solutions. The number of inside surfaces i is usually equal to 6, and surface i is different from j so that radiative exchange can proceed. $q_{i,t}$ could also be expressed in Btu/min or even Btu/sec.

The space sensible cooling load q_{rs} , in Btu/hr, is the sum of the convective heat from the inside surfaces, including the convective heat from the inner window glass due to the absorbed solar radiation and the infiltration:

$$q_{rs} = \left[\sum_{i=1}^6 h_{ci}(T_{i,t} - T_{r,t}) \right] A_i + 60 \dot{V}_{if} \rho_o c_{pa} (T_{o,t} - T_{r,t}) + S_{c,t} + L_{c,t} + P_{c,t} + E_{c,t} \quad (9.6.3)$$

where \dot{V}_{if} = volume flow rate of infiltrated air, cfm
 ρ_o = air density of outdoor air, lb/ft³
 c_{pa} = specific heat of moist air, Btu/lb.°F
 $T_{o,t}$ = temperature of outdoor air at time t , °F
 $S_{c,t}$, $L_{c,t}$, $P_{c,t}$, $E_{c,t}$ = heat convected from the windows, lights, occupants, and equipment, Btu/hr

Equations (9.6.2) and (9.6.3), and partial differential equations to determine conductive heat $q_{i,t}$ must be solved simultaneously. Using a rigorous approach to find the space cooling load requires numerous computer calculations. It is laborious and time consuming. The rigorous approach is impractical and is suitable for research work only.

Transfer Function Method (TFM)

The *transfer function* of a system relates its output in Laplace transform Y to its input in Laplace transform G by a ratio K , that is,

$$K = Y/G = (v_0 + v_1 z^{-1} + v_2 z^{-2} + \dots) / (1 + w_1 z^{-1} + w_2 z^{-2} + \dots) \quad (9.6.4)$$

$$z = e^{s\Delta}$$

where Δ = time interval.

In Equation (9.6.4), K , Y , and G are all expressed in z -transforms of the time series function. Coefficients v_n and w_n are called *transfer function coefficients*, or weighting factors. *Weighting factors* are used to weight the importance of the effect of current and previous heat gains as well as the previous space sensible cooling load on the current space sensible cooling load $q_{rs,t}$. Then, the output $q_{rs,t}$ can be related to the input, the space sensible heat gain $q_{es,t}$ through $q_{rs,t} = Kq_{es,t}$.

Mitalas and Stevenson (1967) and others developed a method for determining the transfer function coefficients of a zone of given geometry and details of the calculated space heat gains and the previously known space sensible cooling load through rigorous computation or through tests and experiments. In DOE 2.1A (1981) software for custom weighting factors (tailor made according to a specific parametric zone) is also provided. Sowell (1988) and Spitler et al. (1993) expanded the application of TFM to zones with various parameters: zone geometry, different types of walls, roof, floor, ceilings, building material, and mass locations. Mass of construction is divided into light construction, 30 lb/ft² of floor area; medium construction, 70 lb/ft²; and heavy construction, 130 lb/ft². Data are summarized into groups and listed in tabular form for user's convenience.

Cooling Load Temperature Difference/Solar Cooling Load Factor/Cooling Load Factor (CLTD/SCL/CLF) Method

The *CLTD/SCL/CLF method* is a one-step simplification of the transfer function method. The space cooling load is calculated directly by multiplying the heat gain q_e with CLTD, SCL, or CLF instead of first finding the space heat gains and then converting into space cooling loads through the room transfer function. In the CLTD/SCL/CLF method, the calculation of heat gains is the same as in the transfer function method.

The CLTD/SCL/CLF method was introduced by Rudoy and Duran (1975). McQuiston and Spitler (1992) recommended a new SCL factor. In 1993 they also developed the CLTD and CLF data for different zone geometries and constructions.

Finite Difference Method

Since the development of powerful personal computers, the finite difference or numerical solution method can be used to solve transient simultaneous heat and moisture transfer in space cooling load calculations. Wong and Wang (1990) emphasized the influence of moisture stored in the building structure on the cool-down load during the night shut-down operating mode in locations where the summer outdoor climate is hot and humid. The finite difference method is simple and clear in concept as well as more direct in computation than the transfer function method. Refer to Wang's *Handbook of Air Conditioning and Refrigeration* for details.

Total Equivalent Temperature Differential/Time Averaging (TETD/TA) Method

In this method, the heat gains transmitted through external walls and roofs are calculated from the Fourier series solution of one-dimensional transient heat conduction. The conversion of space heat gains to space cooling loads takes place by (1) averaging the radiative heat gains to the current and successive hours according to the mass of the building structure and experience and (2) adding the instantaneous convective fraction and the allocated radiative fraction in that time period. The TETD/TA method is simpler and more subjective than the TFM.

Conduction Heat Gains

Following are the principles and procedures for the calculation of space heat gains and their conversion to space cooling loads by the TFM. TFM is the method adopted by software DOE-2.1D and is also one

of the computing programs adopted in TRACE 600. Refer to *ASHRAE Handbook 1993 Fundamentals*, Chapters 26 and 27, for detail data and tables.

Surface Heat Transfer Coefficients

ASHRAE Handbook 1993 Fundamentals adopts a constant *outdoor heat transfer coefficient* $h_o = 3.0$ Btu/hr.ft².°F and a constant *inside heat transfer coefficient* $h_i = 1.46$ Btu/hr.ft².°F during cooling load calculations. However, many software programs use the following empirical formula (*TRACE 600 Engineering Manual*):

$$h_{o,t} = 2.00 + 0.27v_{\text{wind},t} \quad (9.6.5)$$

where $h_{o,t} = h_o$ at time t , Btu/hr.ft².°F
 $v_{\text{wind},t}$ = wind velocity at time t , ft/sec

Coefficient h_i may be around 2.1 Btu/hr.ft².°F when the space air system is operating, and h_i drops to only about 1.4 Btu/hr.ft².°F when the air system is shut down. The *R value*, expressed in hr.ft².°F/Btu, is defined as the reciprocal of the overall heat transfer coefficient *U* value, in Btu/hr.ft².°F. The *R* value is different from the thermal resistance R^* , since $R^* = 1/UA$, which is expressed in hr.°F/Btu.

Sol-air temperature T_{sol} is a fictitious outdoor air temperature that gives the rate of heat entering the outer surface of walls and roofs due to the combined effect of incident solar radiation, radiative heat exchange with the sky vault and surroundings, and convective heat exchange with the outdoor air. T_{sol} , in °F, is calculated as

$$T_{\text{sol}} = T_o + \alpha I_t / h_o - \epsilon \Delta R / h_o \quad (9.6.6)$$

where T_o = outdoor air temperature, °F
 α = absorptance of incident solar radiation on outer surface
 ϵ = hemispherical emittance of outer surface, assumed equal to 1
 I_t = total intensity of solar radiation including diffuse radiation, Btu/hr.ft²

Tabulated sol-air temperatures that have been calculated and listed in *ASHRAE Handbook 1993 Fundamentals* are based on the following: if $\epsilon = 1$ and $h_o = 3$ Btu/hr.ft².°F, $\Delta R/h_o$ is -7°F for horizontal surfaces and 0°F for vertical surfaces, assuming that the long-wave radiation from the surroundings compensates for the loss to the sky vault.

External Wall and Roof

The sensible heat gain through a wall or roof at time t , $q_{e,t}$, in Btu/hr, can be calculated by using the sol-air temperature at time $t-n\delta$, $T_{\text{sol},t-n\delta}$, in °F, as the outdoor air temperature and a constant indoor space air temperature T_r , in °F, as

$$q_{e,t} = A \left[\sum_{n=0} b_n T_{\text{sol},t-n\delta} - \sum_{n=1} d_n (q_{e,t-n\delta} / A) \right] - T_r \sum_{n=0} c_n \quad (9.6.7)$$

where A = surface area of wall or roof, ft²
 $q_{e,t-n\delta}$ = heat gain through wall or roof at time $t-n\delta$, Btu/hr
 n = number of terms in summation

In Equation (9.6.7), b_n , c_n , and d_n are *conduction transfer function* coefficients.

Ceiling, Floor, and Partition Wall

If the variation of the temperature of the adjacent space T_{aj} , in °F, is small compared with the differential $(T_{aj} - T_r)$, or even when T_{aj} is constant, the heat gain to the conditioned space through ceiling, floor, and partition wall $q_{aj,t}$, in Btu/hr, is

$$q_{aj,t} = UA(T_{aj} - T_r) \quad (9.6.8)$$

where U = overall heat transfer coefficient of the ceiling, floor, and partition (Btu/hr.ft².°F).

Heat Gain through Window Glass

Shading Devices. There are two types of shading devices: indoor shading devices and outdoor shading devices. *Indoor shading devices* increase the reflectance of incident radiation. *Venetian blinds* and *draperies* are the most widely used indoor shading devices. Most horizontal venetian blinds are made of plastic, aluminum, or rigid woven cloth slats spaced 1 to 2 in. apart. Vertical venetian blinds with wider slats are also used. *Draperies* are made of fabrics of cotton, rayon, or synthetic fibers. They are usually loosely hung, wider than the window, often pleated, and can be drawn open and closed as needed. Draperies also increase thermal resistance in winter.

External shading devices include overhangs, side fins, louvers, and pattern grilles. They reduce the sunlit area of the window glass effectively and therefore decrease the solar heat gain. External shading devices are less flexible and are difficult to maintain.

Shading Coefficient (SC). The shading coefficient is an index indicating the glazing characteristics and the associated indoor shading device to admit solar heat gain. SC of a specific window glass and shading device assembly at a summer design solar intensity and outdoor and indoor temperatures can be calculated as

$$SC = \frac{\text{solar heat gain of specific type of window glass assembly}}{\text{solar heat gain of reference (DSA) glass}} \quad (9.6.9)$$

Double-strength sheet glass (DSA) has been adopted as the reference glass with a transmittance of $\tau = 0.86$, reflectance $\rho = 0.08$, and absorptance $\alpha = 0.06$.

Solar Heat Gain Factors (SHGFs). SHGFs, in Btu/hr.ft², are the average solar heat gains during cloudless days through DSA glass. $SHGF_{max}$ is the maximum value of SHGF on the 21st day of each month for a specific latitude as listed in *ASHRAE Handbook 1993 Fundamentals*. At high elevations and on very clear days, the actual SHGF may be 15% higher.

Heat Gain through Window Glass. There are two kinds of heat gain through window glass: heat gain due to the solar radiation transmitted and absorbed by the window glass, $q_{es,t}$, and conduction heat gain due to the outdoor and indoor temperature difference, $q_{ec,t}$, both in Btu/hr:

$$q_{es,t} = (A_s \times SHGF \times SC) + (A_{sh} \times SHGF_{sh} \times SC) \quad (9.6.10)$$

where A_s , A_{sh} = sunlit and shaded areas of the glass (ft²).

In Equation (9.6.10), $SHGF_{sh}$ represents the SHGF of the shaded area of the glass having only diffuse radiation. Generally, the SHGF of solar radiation incident on glass facing north without direct radiation can be considered as $SHGF_{sh}$. Because the mass of window glass is small, its heat storage effect is often neglected; then the conduction heat gain at time t is

$$q_{ec,t} = U_g (A_s + A_{sh}) (T_{o,t} - T_r) \quad (9.6.11)$$

where U_g = overall heat transfer coefficient of window glass, Btu/hr.ft².°F
 $T_{o,t}$ = outdoor air temperature at time t , °F

The inward heat transfer due to the solar radiation absorbed by the glass and the conduction heat transfer due to the outdoor and indoor temperature difference is actually combined. It is simple and more convenient if they are calculated separately.

Internal Heat Gains

Internal heat gains are heat released from the internal sources.

People

The sensible heat gain and latent heat gain per person, $q_{es,t}$ and $q_{el,t}$, both in Btu/hr, are given as

$$\begin{aligned} q_{es,t} &= N_t q_{os} \\ q_{el,t} &= N_t q_{ol} \end{aligned} \quad (9.6.12)$$

where N_t = number of occupants in the conditioned space at time t .

Heat gains q_{os} and q_{ol} depend on the occupant's metabolic level, whether the occupant is an adult or a child, male or female, and the space temperature. For a male adult seated and doing very light work, such as system design by computer in an office of space temperature 75°F, q_{os} and q_{ol} are both about 240 Btu/hr.

Lights

Heat gain in the conditioned space because of the electric lights, $q_{e,li}$, in Btu/hr, is calculated as

$$q_{e,li} = \sum_{n=1} \left[3.413 W_{li} N_{li} F_{ul} F_{al} (1 - F_{lp}) \right] = 3.413 W_{lA} A_n \quad (9.6.13)$$

where W_{li} = watt input to each lamp, W

W_{lA} = lighting power density, W/ft²

A_n = floor area of conditioned space, ft²

N_{li} , n = number of lamps (each type) and number of types of electric lamp

F_{al} , F_{ul} = use factor of electric lights and allowance factor for ballast loss, usually taken as 1.2

F_{lp} = heat gain carried away by the return air to plenum or by exhaust device; it varies from 0.2 to 0.5

Machines and Appliances

The sensible and latent heat gains in the conditioned space from machines and appliances, $q_{e,ap}$ and $q_{l,ap}$, both in Btu/hr, can be calculated as

$$\begin{aligned} q_{e,ap} &= 3.413 W_{ap} F_{ua} F_{load} F_{ra} = q_{is} F_{ua} F_{ra} \\ q_{l,ap} &= q_{il} F_{ua} \end{aligned} \quad (9.6.14)$$

where W_{ap} = rated power input to the motor of the machines and appliances, W

F_{ua} = use factor of machine and appliance

F_{load} = ratio of actual load to the rated power

F_{ra} = radiation reduction factor because of the front shield of the appliance

q_{is} , q_{il} = sensible and latent heat input to the appliance, Btu/hr

Infiltration

Infiltration is the uncontrolled inward flow of unconditioned outdoor air through cracks and openings on the building envelope because of the pressure difference across the envelope. The pressure difference is probably caused by wind pressure, stack effect due to outdoor–indoor temperature difference, and the operation of an air system(s).

Today new commercial buildings have their external windows well sealed. If a positive pressure is maintained in the conditioned space when the air system is operating, infiltration is normally considered as zero.

When the air system is shut down, or for hotels, motels, and high-rise residential buildings, ASHRAE/IES Standard 90.1-1989 specifies an infiltration of 0.038cfm/ft² of gross area of the external wall, 0.15 air change per hour (ach) for the perimeter zone.

When exterior windows are not well sealed, the outdoor wind velocity is high at winter design conditions, or there is a door exposed to the outdoors directly, an infiltration rate of 0.15 to 0.4 ach for perimeter zone should be considered.

When the volume flow rate of infiltration is determined, the sensible heat gain due to infiltration $q_{s,if}$ and latent heat gain due to infiltration $q_{l,if}$, in Btu/hr, are

$$q_{s,if} = 60 \dot{V}_{if} \rho_o c_{pa} (T_o - T_r) \quad (9.6.15)$$

$$q_{l,if} = 60 \times 1060 \dot{V}_{if} \rho_o (w_o - w_r)$$

where \dot{V}_{if} = volume flow rate of infiltration, cfm
 ρ_o = air density of outdoor air, lb/ft³
 c_{pa} = specific heat of moist air, 0.243 Btu/lb.°F
 w_o, w_r = humidity ratio of outdoor and space air, lb/lb
 $h_{fg,58}$ = latent heat of vaporization, 1060 Btu/lb

Infiltration enters the space directly and mixes with space air. It becomes space cooling load instantaneously. Ventilation air is often taken at the AHU or PU and becomes sensible and latent coil load components.

Conversion of Heat Gains into Cooling Load by TFM

The space sensible cooling load $q_{rs,t}$, in Btu/hr, is calculated as

$$q_{rs,t} = q_{s-e,t} + q_{s-c,t} \quad (9.6.16)$$

where $q_{s-e,t}$ = space sensible cooling load converted from heat gains having radiative and convective components, Btu/hr
 $q_{s-c,t}$ = space sensible cooling load from convective heat gains, Btu/hr

Based on Equation (9.6.4), space sensible cooling load $q_{s-e,t}$ can be calculated as

$$q_{s-e,t} = \sum_{i=1} (v_o q_{e,t} + v_1 q_{e,t-\delta} + v_2 q_{e,t-2\delta} + \dots) - (w_1 q_{r,t-\delta} + w_2 q_{r,t-2\delta} + \dots) \quad (9.6.17)$$

where $q_{e,t}, q_{e,t-\delta}, q_{e,t-2\delta}$ = space sensible heat gains having both radiative and convective heats at time $t, t-\delta,$ and $t-2\delta$, Btu/hr

$q_{r,t-\delta}, q_{r,t-2\delta}$ = space sensible cooling load at time $t-\delta,$ and $t-2\delta$, Btu/hr

In Equation (9.6.17), v_n and w_n are called *room transfer function coefficients* (RTFs). RTF is affected by parameters like zone geometry; wall, roof, and window construction; internal shades; zone location; types of building envelope; and air supply density. Refer to RTF tables in *ASHRAE Handbook 1993 Fundamentals*, Chapter 26, for details.

Space sensible cooling load from convective heat gains can be calculated as

$$q_{s-c,t} = \sum_{k=1} q_{ec,t} \quad (9.6.18)$$

where $q_{ec,t}$ = each of k space sensible heat gains that have convective heat gains only (Btu/hr). Space latent cooling load at time t , $q_{rl,t}$ in Btu/hr, can be calculated as

$$q_{rl,t} = \sum_{m=1} q_{el,t} \quad (9.6.19)$$

where $q_{el,t}$ = each of m space latent heat gains (Btu/hr).

Space Air Temperature and Heat Extraction Rate

At equilibrium, the space sensible heat extraction rate at time t , $q_{xs,t}$, is approximately equal to the space sensible cooling load, $q_{rs,t}$, when zero offset proportional plus integral or proportional-integral-derivative control mode is used. During the cool-down period, the sensible heat extraction rate of the cooling coil or DX coil at time t , $q_{xs,t}$, or the sensible cooling coil load, in Btu/hr, is greater than the space sensible cooling load at time t and the space temperature T_r , in °F then drops gradually. According to *ASHRAE Handbook 1993 Fundamentals*, the relationship between the space temperature T_r and the sensible heat extraction rate $q_{xs,t}$ can be expressed as

$$\sum_{i=0}^1 p_i (q_{xs,t} - q_{rs,t-i\delta}) = \sum_{i=0}^2 g_i (T_r - T_{r,t-i\delta}) \quad (9.6.20)$$

where $q_{rs,t-i\delta}$ = space sensible cooling load calculated on the basis of constant space air temperature T_r , Btu/hr

$T_{r,t-i\delta}$ = space temperature at time $t-i\delta$

In Equation (9.6.20), p_i , g_i are called *space air transfer function coefficients*. Space air temperature T_r can be considered an average reference temperature within a time interval.

Cooling Coil Load

Cooling coil load q_{cc} , in Btu/hr, can be calculated from Equation (9.3.11). The sensible and latent cooling coil loads can then be calculated as

$$q_{cs} = 60 \dot{V}_s \rho_s c_{pa} (T_m - T_{cc}) \quad (9.6.21)$$

$$q_{cl} = 60 \dot{V}_s \rho_s (w_m - w_{cc})$$

where q_{cs} , q_{cl} = sensible and latent cooling coil load, Btu/hr

T_m , T_{cc} = air temperature of the mixture and leaving the cooling coil, °F

w_m , w_{cc} = humidity ratio of the air mixture and air leaving the cooling coil, lb/lb

Heating Load

The *space heating load* or simply *heating load* is always the possible maximum heat energy that must be added to the conditioned space at winter design conditions to maintain the indoor design temperature. It is used to size and select the heating equipment. In heating load calculations, solar heat gain, internal heat gains, and the heat storage effect of the building envelope are usually neglected for reliability and simplicity.

Normally, space heating load q_{th} , in Btu/hr, can be calculated as

$$q_{th} = q_{trans} + q_{if,s} + q_{ma} + q_{hu}$$

$$= \left[\left(\sum_{n=1} AU \right) + 60 \dot{V}_{if} \rho_o c_{pa} \dot{m}_m c_m \right] (T_r - T_o) + \dot{m}_w h_{fg,58} \quad (9.6.22)$$

where A = area of the external walls, roofs, glasses, and floors, ft²

U = overall heat transfer coefficient of the walls, roofs, glasses, and floors, Btu/hr.ft².°F

c_m, \dot{m}_m = specific heat and mass flow rate of the cold product entering the space per hour, Btu/lb.°F and lb/hr

\dot{m}_w = mass flow rate of water evaporated for humidification, lb/hr

$h_{fg,58}$ = latent heat of vaporization at 58°F, 1060 Btu/lb

In Equation 9.88, q_{trans} indicates the *transmission loss* through walls, roofs, glasses, and floors, $q_{if,s}$ the *sensible infiltration heat loss*, q_{ma} the heat required to heat the cold product that enters the conditioned space, and q_{hu} the heat required to raise the space air temperature when water droplets from a space humidifier are evaporated in the conditioned space. For details, refer to *ASHRAE Handbook 1993 Fundamentals*.

9.7 Air Handling Units and Packaged Units

Terminals and Air Handling Units

A *terminal unit*, or *terminal*, is a device or equipment installed directly in or above the conditioned space to cool, heat, filter, and mix outdoor air with recirculating air. Fan-coil units, VAV boxes, fan-powered VAV boxes, etc. are all terminals.

An *air handling unit* (AHU) handles and conditions the air, controls it to a required state, and provides motive force to transport it. An AHU is the primary equipment of the air system in a central air-conditioning system. The basic components of an AHU include a supply fan with a fan motor, a water cooling coil, filters, a mixing box except in a makeup AHU unit, dampers, controls, and an outer casing. A return or relief fan, heating coil(s), and humidifier are optional depending on requirements. The supply volume flow rate of AHUs varies from 2000 to about 60,000 cfm.

AHUs are classified into the followings groups according to their structure and location.

Horizontal or Vertical Units

Horizontal AHUs have their fan, coils, and filters installed at the same level as shown in Figure 9.7.1(a). They need more space and are usually for large units. In *vertical units*, as shown in Figure 9.7.1(b), the supply fan is installed at a level higher than coils and filters. They are often comparatively smaller than horizontal units.

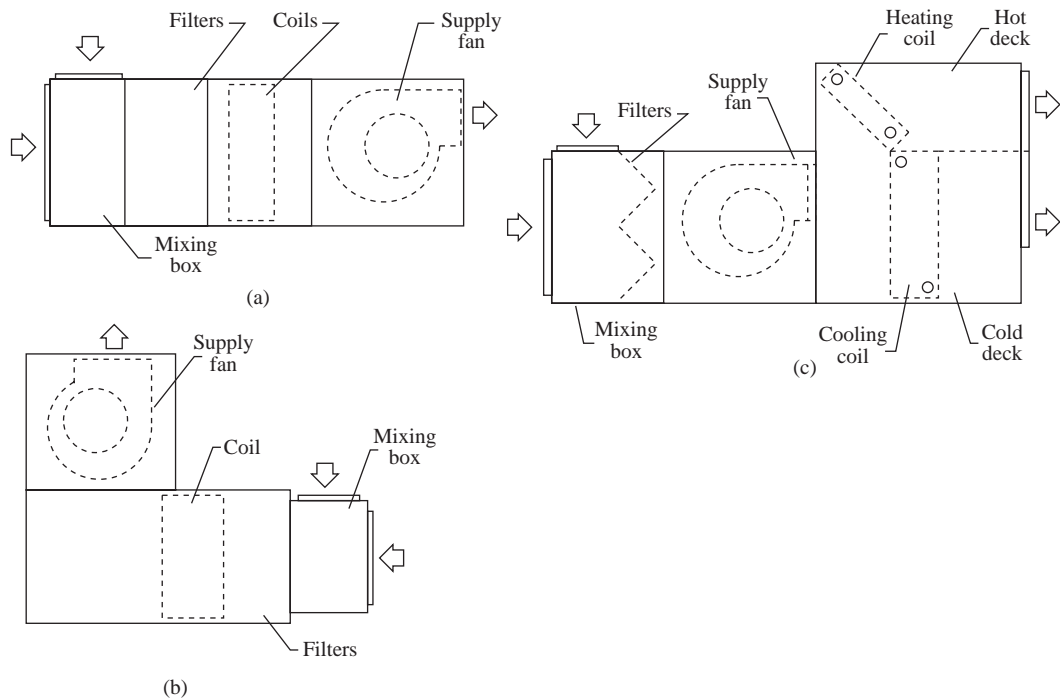


FIGURE 9.7.1 Type of air handling units: (a) horizontal draw-through unit, (b) vertical draw-through unit, and (c) multizone blow-through unit.

Draw-Through or Blow-Through Units

In a *draw-through unit*, as shown in Figure 9.7.1(a), the supply fan is located downstream of the coils. Air is evenly distributed over the coil section, and the fan discharge can easily be connected to a supply duct of nearly the same air velocity. In a *blow-through unit*, as shown in Figure 9.7.1(c), the supply fan

is located upstream of the coils. It usually has hot and cold decks with discharge dampers connected to warm and cold ducts, respectively.

Factory-Fabricated and Field Built-Up Units

Factory-fabricated units are standard in construction and layout, low in cost, of higher quality, and fast in installation. *Field built-up units* or *custom-built units* are more flexible in construction, layout, and dimensions than factory-built standardized units.

Rooftop and Indoor Units

A *rooftop AHU*, sometimes called a penthouse unit, is installed on the roof and will be completely weatherproof. An *indoor AHU* is usually located in a fan room or ceiling and hung like small AHU units.

Make-Up Air and Recirculating Units

A *make-up AHU*, also called a primary-air unit, is used to condition outdoor air entirely. It is a once-through unit. There is no return air and mixing box. *Recirculating units* can have 100% outdoor air intake or mixing of outdoor air and recirculating air.

Packaged Units

A *packaged unit* (PU) is a self-contained air conditioner. It conditions the air and provides it with motive force and is equipped with its own heating and cooling sources. The packaged unit is the primary equipment in a packaged air-conditioning system and is always equipped with a DX coil for cooling, unlike an AHU. R-22, R-134a, and others are used as refrigerants in packaged units. The portion that handles air in a packaged unit is called an *air handler* to distinguish it from an AHU. Like an AHU, an indoor air handler has an indoor fan, a DX coil (indoor coil), filters, dampers, and controls. Packaged units can be classified according to their place of installation: rooftop, indoor, and split packaged units.

Rooftop Packaged Units

A *rooftop packaged unit* is mounted on the roof of the conditioned space as shown in [Figure 9.7.2](#). From the types of heating/cooling sources provided, rooftop units can be subdivided into:

- Gas/electric rooftop packaged unit, in which heating is provided by gas furnace and cooling by electric power-driven compressors.
- Electric/electric rooftop packaged unit, in which electric heating and electric power-driven compressors provide heating and cooling.
- Rooftop packaged heat pump, in which both heating and cooling are provided by the same refrigeration system using a four-way reversing valve (heat pump) in which the refrigeration flow changes when cooling mode is changed to heating mode and vice versa. Auxiliary electric heating is provided if necessary.

Rooftop packaged units are single packaged units. Their cooling capacity may vary from 3 to 220 tons with a corresponding volume flow rate of 1200 to 80,000 cfm. Rooftop packaged units are the most widely used packaged units.

Indoor Packaged Units

An *indoor packaged unit* is also a single packaged and factory-fabricated unit. It is usually installed in a fan room or a machinery room. A small or medium-sized indoor packaged unit could be floor mounted directly inside the conditioned space with or without ductwork. The cooling capacity of an indoor packaged unit may vary from 3 to 100 tons and volume flow rate from 1200 to 40,000 cfm.

Indoor packaged units are also subdivided into:

- Indoor packaged cooling units

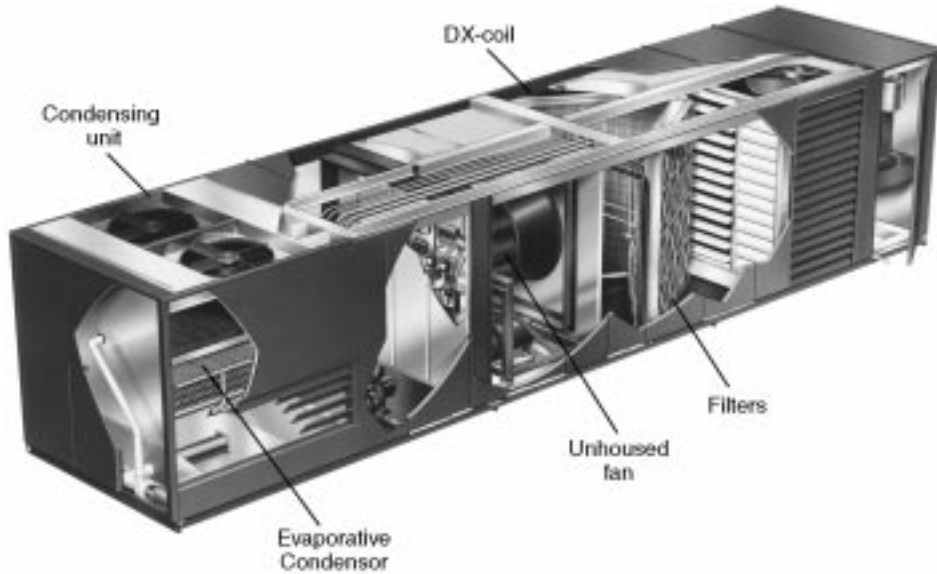


FIGURE 9.7.2 A cut view of a rooftop package unit. (Source: Mammoth, Inc. Reprinted by permission.)

- Indoor packaged cooling/heating units, in which heating may be provided from a hot water heating coil, a steam heating coil, and electric heating
- Indoor packaged heat pumps

Indoor packaged units have either an air-cooled condenser on the rooftop or a shell-and-tube or double-tube water-cooled condenser inside the unit.

Split Packaged Units

A *split packaged unit* consists of two separate pieces of equipment: an indoor air handler and an outdoor condensing unit. The indoor air handler is often installed in the fan room. Small air handlers can be ceiling hung. The condensing unit is usually located outdoors, on a rooftop or podium or on the ground.

A split packaged unit has its compressors and condenser in its outdoor condensing unit, whereas an indoor packaged unit usually has its compressors indoors. The cooling capacity of split packaged units varies from 3 to 75 tons and the volume flow rate from 1200 to 30,000 cfm.

Rating Conditions and Minimum Performance

Air Conditioning and Refrigeration Institute (ARI) Standards and ASHRAE/IES Standard 90.1-1989 specified the following rating indices:

- Energy efficiency ratio (EER) is the ratio of equipment cooling capacity, in Btu/hr, to the electric input, in W, under rating conditions.
- SEER is the seasonal EER, or EER during the normal annual usage period.
- IPLV is the integrated part-load value. It is the summarized single index of part-load efficiency of PUs based on weighted operations at several load conditions.
- HSPF is the heating seasonal performance factor. It is the total heating output of a heat pump during its annual usage period for heating, in Btu, divided by the total electric energy input to the heat pump during the same period, in watt-hours.

According to ARI standards, the minimum performance for air-cooled, electrically operated single packaged units is

	q_{rc} (Btu/hr)	T_o (°F)	EER	T_o (°F)	IPLV
Air-cooled	<65,000	95	9.5		
	$65,000 \leq q_{rc} < 135,000$	95	8.9	80	8.3
	$135,000 \leq q_{rc} < 760,000$		8.5		7.5

For water- and evaporatively cooled packaged units including heat pumps, refer to ASHRAE/IES Standard 90.1-1989 and also ARI Standards.

Coils

Coils, Fins, and Water Circuits

Coils are indirect contact heat exchangers. Heat transfer or heat and mass transfer takes place between conditioned air flowing over the coil and water, refrigerant, steam, or brine inside the coil for cooling, heating, dehumidifying, or cooling/dehumidifying. Chilled water, brine, and refrigerants that are used to cool and dehumidify the air are called *coolants*. Coils consist of tubes and external fins arranged in rows along the air flow to increase the contact surface area. Tubes are usually made of copper; in steam coils they are sometimes made of steel or even stainless steel. Copper tubes are staggered in 2, 3, 4, 6, 8, or up to 10 rows.

Fins are extended surfaces often called *secondary surfaces* to distinguish them from the *primary surfaces*, which are the outer surfaces of the tubes. Fins are often made from aluminum, with a thickness $F_t = 0.005$ to 0.008 in., typically 0.006 in. Copper, steel, or sometimes stainless steel fins are also used. Fins are often in the form of continuous plate fins, corrugated plate fins to increase heat transfer, crimped spiral or smooth spiral fins that may be extruded from the aluminum tubes, and spine pipes, which are shaved from the parent aluminum tubes. Corrugated plate fins are most widely used.

Fin spacing S_f is the distance between two fins. *Fin density* is often expressed in fins per inch and usually varies from 8 to 18 fins/in.

In a water cooling coil, *water circuits* or *tube feeds* determine the number of water flow passages. The greater the finned width, the higher the number of water circuits and water flow passages.

Direct Expansion (DX) Coil

In a *direct expansion coil*, the refrigerant, R-22, R-134a, or others, is evaporated and expanded directly inside the tubes to cool and dehumidify the air as shown in Figure 9.7.3(a). Refrigerant is fed to a distributor and is then evenly distributed to various copper tube circuits typically 0.375 in. in diameter. Fin density is usually 12 to 18 fins/in. and a four-row DX coil is often used. On the inner surface of the copper tubes, microfins, typically at 60 fins/in. and a height of 0.008 in., are widely used to enhance the boiling heat transfer.

Air and refrigerant flow is often arranged in a combination of counterflow and cross flow and the discharge header is often located on the air-entering side. Refrigerant distribution and loading in various circuits are critical to the coil's performance. Vaporized vapor refrigerant is superheated 10 to 20°F in order to prevent any liquid refrigerant from flooding back to the reciprocating compressors and damaging them. Finally, the vapor refrigerant is discharged to the suction line through the header.

For comfort air-conditioning systems, the evaporating temperature of refrigerant T_{ev} inside the tubes of a DX coil is usually between 37 and 50°F. At such a temperature, the surface temperature of the coil is often lower than the dew point of the entering air. Condensation occurs at the coil's outside surface, and the coil becomes a wet coil. A condensate *drain pan* is necessary for each vertically banked DX coil, and a trap should be installed to overcome the negative pressure difference between the air in the coil section and the ambient air.

Face velocity of the DX coil v_a , in fpm, is closely related to the blow-off of the water droplets of the condensate, the heat transfer coefficients, the air-side pressure drop, and the size of the air system. For corrugated fins, the upper limit is 600 fpm, with an air-side pressure drop of 0.20 to 0.30 in. WG/row.

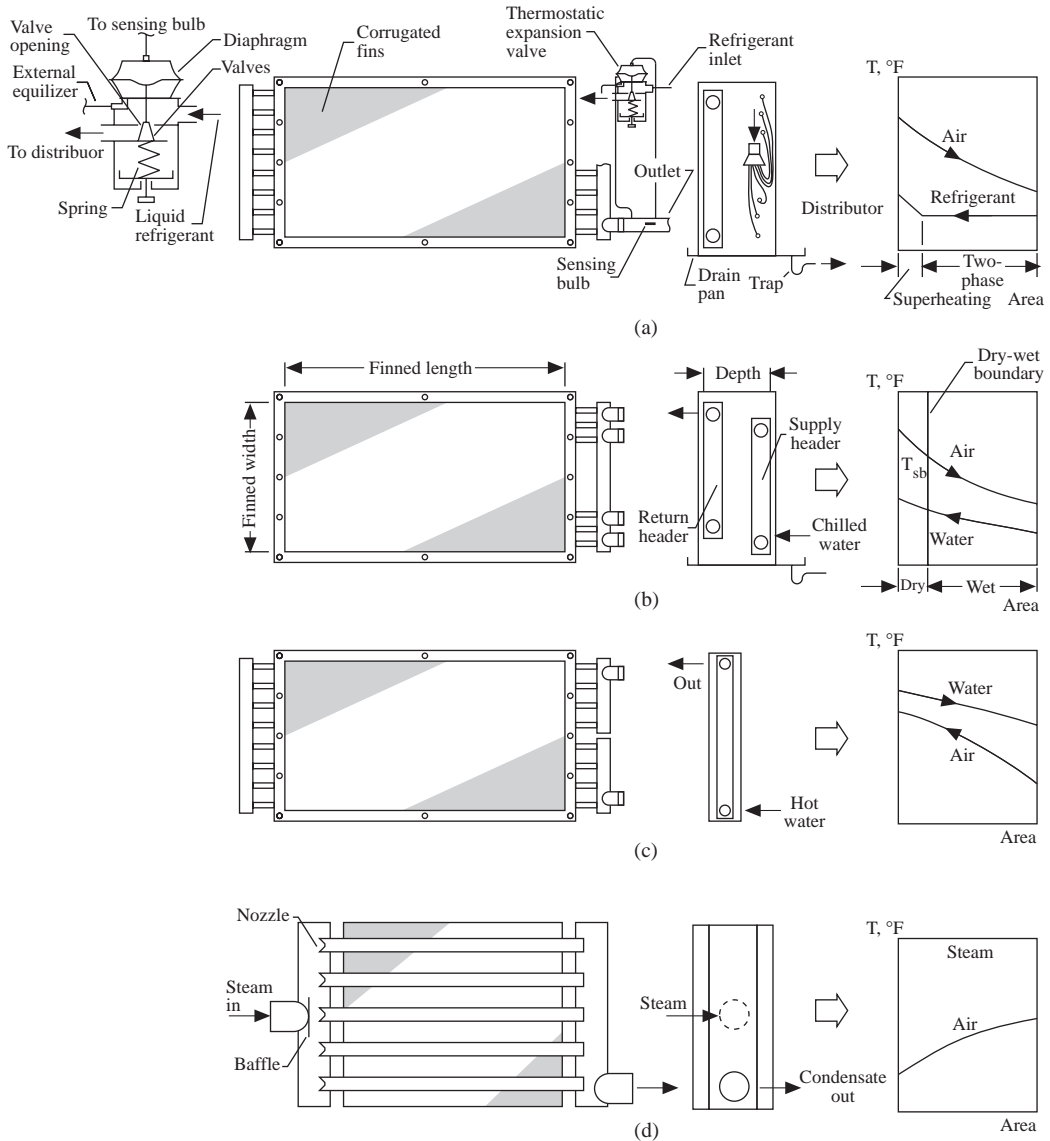


FIGURE 9.7.3 Types of coils: (a) direct expansion coil, (b) water cooling coil, (c) water heating coil, and (d) steam heating coil.

A large DX coil is often divided into two refrigerant sections, each with its own expansion valve, distributor, and discharge header.

For a packaged unit of a specific model, size, face velocity and condition of entering air and outdoor air, the DX coil’s cooling capacities in nominal tons, number of rows, and fin density are all fixed values.

Water Cooling Coils — Dry-Wet Coils

In a water cooling coil, chilled water at a temperature of 40 to 50°F, brine, or glycol-water at a temperature of 34 to 40°F during cold air distribution enters the coil. The temperature of chilled water, brine, or glycol-water is usually raised 12 to 24°F before it leaves the water cooling coil.

The water tubes are usually copper tubes of 1/2 to 5/8 in. diameter with a tube wall thickness of 0.01 to 0.02 in. They are spaced at a center-to-center distance of 0.75 to 1.25 in. longitudinally and 1 to 1.5

in. transversely. These tubes may be staggered in 2, 3, 4, 6, 8, or 10 rows. Chilled water coils are often operated at a pressure of 175 to 300 psig.

As in a DX coil, the air flow and water flow are in a combination of counterflow and cross flow. The outer surface of a chilled water cooling coil at the air entering side T_{se} is often greater than the dew point of the entering air T_{ae}'' , or $T_{se} > T_{ae}''$. The outer surface temperature of coil at the air leaving side T_{sl} may be smaller than T_{ae}'' , or $T_{sl} < T_{ae}''$. Then the water cooling coil becomes a dry-wet coil with part of the dry surface on the air entering side and part of the wet surface on the air leaving side. A *dry-wet boundary* divides the dry and wet surfaces. At the boundary, the tube outer surface temperature $T_{sb} = T_{ae}''$ as shown in [Figure 9.7.3\(b\)](#). A condensate drain pan is necessary for a dry-wet coil.

A water cooling coil is selected from the manufacturer's selection program or from its catalog at (1) a dry and wet bulb of entering air, such as 80°F dry bulb and 67°F wet bulb; (2) an entering water temperature, such as 44 or 45°F; (3) a water temperature rise between 10 and 24°F; and (4) a coil face velocity between 400 and 600 fpm. The number of rows and fins per inch is varied to meet the required sensible and cooling coil load, in Btu/hr.

Water Cooling Coil-Dry Coil

When the temperature of chilled water entering the water cooling coil $T_{we} \geq T_{ae}''$, condensation will not occur on the outer surface of the coil. This coil becomes a sensible cooling-dry coil, and the humidity ratio of the conditioned air w_a remains constant during the sensible cooling process.

The construction of a sensible cooling-dry coil, such as material, tube diameter, number of rows, fin density, and fin thickness, is similar to that of a dry-wet coil except that a dry coil always has a poorer surface heat transfer coefficient than a wet coil, and therefore a greater coil surface area is needed; the maximum face velocity of a dry coil can be raised to $v_a \leq 800$ fpm; and the coil's outer surface is less polluted. The effectiveness of a dry coil ϵ_{dry} is usually 0.55 to 0.7.

Water Heating Coil

The construction of a water heating coil is similar to that of a water cooling coil except that in water heating coils hot water is supplied instead of chilled water and there are usually fewer rows, only 2, 3, and 4 rows, than in water cooling coils. Hot water pressure in water heating coils is often rated at 175 to 300 psig at a temperature up to 250°F. [Figure 9.7.3\(c\)](#) shows a water heating coil.

Steam Heating Coil

In a steam heating coil, latent heat of condensation is released when steam is condensed into liquid to heat the air flowing over the coil, as shown in [Figure 9.7.3\(d\)](#). Steam enters at one end of the coil, and the condensate comes out from the opposite end. For more even distribution, a baffle plate is often installed after the steam inlet. Steam heating coils are usually made of copper, steel, or sometimes stainless steel.

For a steam coil, the coil core inside the casing should expand or contract freely. The coil core is also pitched toward the outlet to facilitate condensate drainage. Steam heating coils are generally rated at 100 to 200 psig at 400°F.

Coil Accessories and Servicing

Coil accessories include air vents, drain valves, isolation valves, pressure relief valves, flow metering valves, balancing valves, thermometers, pressure gauge taps, condensate drain taps, and even distribution baffles. They are employed depending on the size of the system and operating and servicing requirements.

Coil cleanliness is important for proper operation. If a medium-efficiency air filter is installed upstream of the coil, dirt accumulation is often not a problem. If a low-efficiency filter is employed, dirt accumulation may block the air passage and significantly increase the pressure drop across the coil. Coils should normally be inspected and cleaned every 3 months in urban areas when low-efficiency filters are used. Drain pans should be cleaned every month to prevent buildup of bacteria and microorganisms.

Coil Freeze-Up Protection

Improper mixing of outdoor air and recirculating air in the mixing box of an AHU or PU may cause coil freeze-up when the outdoor air temperature is below 32°F. Outdoor air should be guided by a baffle plate and flow in an opposite direction to the recirculating air stream so that they can be thoroughly mixed without stratification.

Run the chilled water pump for the idle coil with a water velocity of 2.5 ft/sec, so that the cooling coil will not freeze when the air temperature drops to 32°F. A better method is to drain the water completely. For a hot water coil, it is better to reset the hot water temperature at part-load operation instead of running the system intermittently. A steam heating coil with inner distributor tubes and outer finned heating tubes provides better protection against freeze-up.

Air Filters

Air Cleaning and Filtration

Air cleaning is the process of removing airborne particles from the air. Air cleaning can be classified into air filtration and industrial air cleaning. Industrial air cleaning involves the removal of dust and gaseous contaminants from manufacturing processes as well as from the space air, exhaust air, and flue gas for air pollution control. In this section, only air filtration is covered.

Air filtration involves the removal of airborne particles presented in the conditioned air. Most of the airborne particles removed by air filtration are smaller than 1 μm , and the concentration of these particles in the airstream seldom exceeds 2 mg/m^3 . The purpose of air filtration is to benefit the health and comfort of the occupants as well as meet the cleanliness requirements of the working area in industrial buildings.

An *air filter* is a kind of air cleaner that is installed in AHUs, PUs, and other equipment to filter the conditioned air by inertial impaction or interception and to diffuse and settle fine dust particles on the fibrous medium. The filter medium is the fabricated material that performs air filtration.

Operating performance of air filters is indicated by their:

- *Efficiency* or effectiveness of dust removal
- *Dust holding capacity* m_{dust} , which is the amount of dust held in the air filter, in grains/ft²
- *Initial pressure drop* when the filter is clean Δp_{fi} and *final pressure drop* Δp_{ff} when the filter's m_{dust} is maximum, both in in. WG
- *Service life*, which is the operating period between Δp_{fi} and Δp_{ff}

Air filters in AHUs and PUs can be classified into low-, medium-, and high-efficiency filters and carbon activated filters.

Test Methods

The performance of air filters is usually tested in a test unit that consists of a fan, a test duct, the tested filter, two samplers, a vacuum pump, and other instruments. Three test methods with their own test dusts and procedures are used for the testing of low-, medium-, and high-efficiency air filters.

The *weight arrestance test* is used for low-efficiency air filters to assess their ability to remove coarse dusts. Standard synthetic dusts that are considerably coarser than atmospheric dust are fed to the test unit. By measuring the weight of dust fed and the weight gain due to the dust collected on the membrane of the sampler after the tested filter, the arrestance can be calculated.

The *atmospheric dust spot efficiency test* is used for medium-efficiency air filters to assess their ability to remove atmospheric dusts. *Atmospheric dusts* are dusts contained in the outdoor air, the outdoor atmosphere. Approximately 99% of atmospheric dusts are dust particles $<0.3 \mu\text{m}$ that make up 10% of the total weight; 0.1% of atmospheric dusts is particles $>1 \mu\text{m}$ that make up 70% of the total weight.

Untreated atmospheric dusts are fed to the test unit. Air samples taken before and after the tested filter are drawn through from identical fiber filter-paper targets. By measuring the light transmission of these discolored white filter papers, the efficiency of the filter can be calculated. Similar atmospheric

dust spot test procedures have been specified by American Filter Institute (AFI), ASHRAE Standard 52.1, and former National Bureau of Standards (NBS).

The *DOP penetration and efficiency test* or simply *DOP test* is used to assess high-efficiency filters removing dusts particles of $0.18\ \mu\text{m}$. According to U.S. Military Standard MIL-STD-282 (1956), a smoke cloud of uniform dioctyl phthalate (DOP) droplets $0.18\ \mu\text{m}$ in diameter, generated from the condensation of the DOP vapor, is fed to the test unit. By measuring the concentration of these particles in the air stream upstream and downstream of the tested filter using an electronic particle counter or laser spectrometer, the penetration and efficiency of the air filter can be calculated.

Low-Efficiency Air Filters

ASHRAE weight arrestance for low-efficiency filters is between 60 and 95%, and ASHRAE dust spot efficiency for low-efficiency filters is less than 20%. These filters are usually in panels as shown in Figure 9.7.4(a). Their framework is typically 20×20 in. or 24×24 in. Their thickness varies from 1 to 4 in.

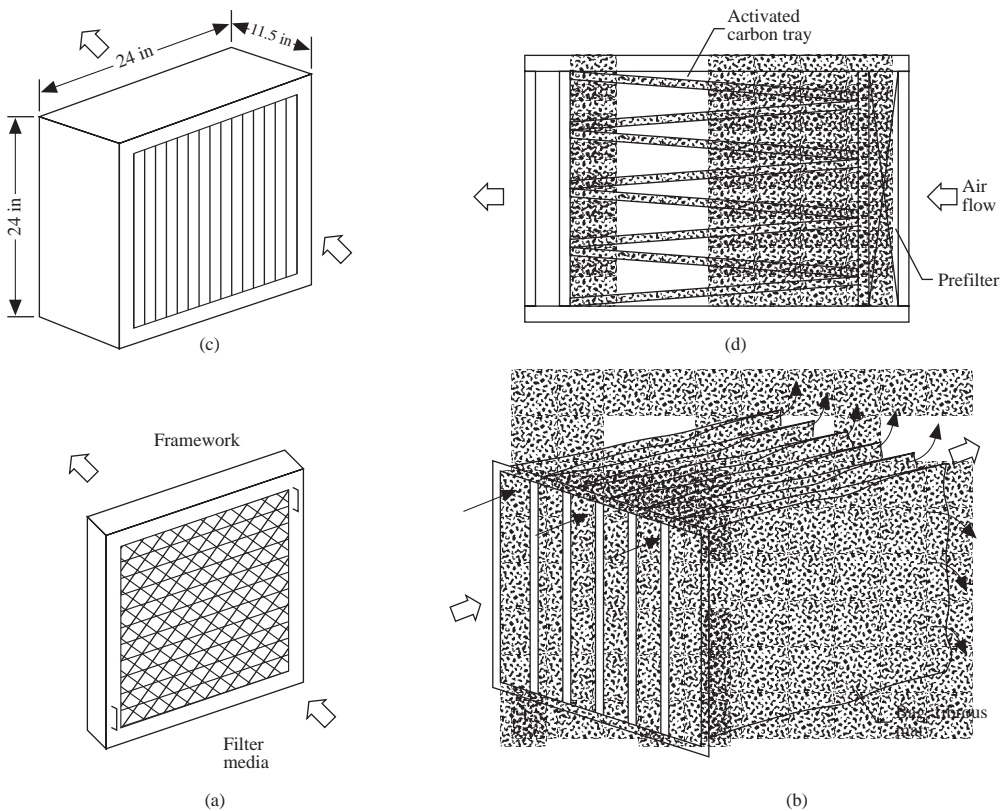


FIGURE 9.7.4 Various types of air filters: (a) low efficiency, (b) medium efficiency, (c) HEPA and ULPA filters, and (d) activated carbon filter.

For low-efficiency filters, the filter media are often made of materials such as

- Corrugated wire mesh and screen strips coated with oil, which act as adhesives to enhance dust removal. Detergents may be used to wash off dusts so that the filter media can be cleaned and reused — they are therefore called *viscous and reusable*.
- Synthetic fibers (nylon, terylene) and polyurethane foam can be washed, cleaned, and reused if required — *dry and reusable*.

- Glass fiber mats with fiber diameter greater than 10 μm . The filter medium is discarded when its final pressure drop is reached — *dry and disposable*. The face velocity of the panel filter is usually between 300 and 600 fpm. The initial pressure drop varies from 0.05 to 0.25 in. WG and the final pressure drop from 0.2 to 0.5 in. WG.

Medium-Efficiency Air Filters

These air filters have an ASHRAE dust spot efficiency usually between 20 and 95%. Filter media of medium-efficiency filters are usually made of glass fiber mat with a fiber diameter of 10 to 1 μm using nylon fibers to join them together. They are usually dry and disposable. In addition:

- As the dust spot efficiency increases, the diameter of glass fibers is reduced, and they are placed closer together.
- Extended surfaces, such as pleated mats or bags, are used to increase the surface area of the medium as shown in [Figure 9.7.4\(b\)](#). Air velocity through the medium is 6 to 90 fpm. Face velocity of the air filter is about 500 fpm to match the face velocity of the coil in AHUs and PUs.
- Initial pressure drop varies from 0.20 to 0.60 in. WG and final pressure drop from 0.50 to 1.20 in. WG.

High-Efficiency Particulate Air (HEPA) Filters and Ultra-Low-Penetration Air (ULPA) Filters

HEPA filters have a DOP test efficiency of 99.97% for dust particles $\geq 0.3 \mu\text{m}$ in diameter. *ULPA filters* have a DOP test efficiency of 99.999% for dust particles $\geq 0.12 \mu\text{m}$ in diameter.

A typical HEPA filter, shown in [Figure 9.7.4\(d\)](#), has dimensions of $24 \times 24 \times 11.5$ in. Its filter media are made of glass fibers of submicrometer diameter in the form of pleated paper mats. The medium is dry and disposable. The surface area of the HEPA filter may be 50 times its face area, and its rated face velocity varies from 190 to 390 fpm, normally at a pressure drop of 0.50 to 1.35 in. WG for clean filters. The final pressure drop is 0.8 to 2 in. WG. Sealing of the filter pack within its frame and sealing between the frame and the gaskets are critical factors that affect the penetration and efficiency of the HEPA filter.

An ULPA filter is similar to a HEPA filter in construction and filter media. Both its sealing and filter media are more efficient than those of a HEPA filter.

To extend the service life of HEPA filters and ULPA filters, both should be protected by a medium-efficiency filter, or a low-efficiency and a medium-efficiency filter in the sequence low–medium just before the HEPA or ULPA filters. HEPA and ULPA filters are widely used in clean rooms and clean spaces.

Activated Carbon Filters

These filters are widely used to remove objectional odors and irritating gaseous airborne particulates, typically 0.003 to 0.006 μm in size, from the air stream by adsorption. *Adsorption* is physical condensation of gas or vapor on the surface of an activated substance like activated carbon. Activated substances are extremely porous. One pound of activated carbon contains 5,000,000 ft^2 of internal surface.

Activated carbon in the form of granules or pellets is made of coal, coconut shells, or petroleum residues and is placed in trays to form activated carbon beds as shown in [Figure 9.7.4\(d\)](#). A typical carbon tray is $23 \times 23 \times 5/8$ in. thick. Low-efficiency prefilters are used for protection. When air flows through the carbon beds at a face velocity of 375 to 500 fpm, the corresponding pressure drop is 0.2 to 0.3 in. WG.

Humidifiers

A *humidifier* adds moisture to the air. Air is humidified by: (1) heating the liquid to evaporate it; (2) atomizing the liquid water into minute droplets by mechanical means, compressed air, or ultrasonic vibration to create a larger area for evaporation; (3) forcing air to flow through a wetted element in

which water evaporates; and (4) injecting steam into air directly before it is supplied to the conditioned space.

For comfort air-conditioning systems, a steam humidifier with a separator as shown in [Figure 9.7.5\(a\)](#) is widely used. Steam is supplied to a jacketed distribution manifold. It enters a separating chamber with its condensate. Steam then flows through a control valve, throttles to a pressure slightly above atmospheric, and enters a dry chamber. Due to the high temperature in the surrounding separating chamber, the steam is superheated. Dry steam is then discharged into the ambient air stream through the orifices on the inner steam discharge tubes.

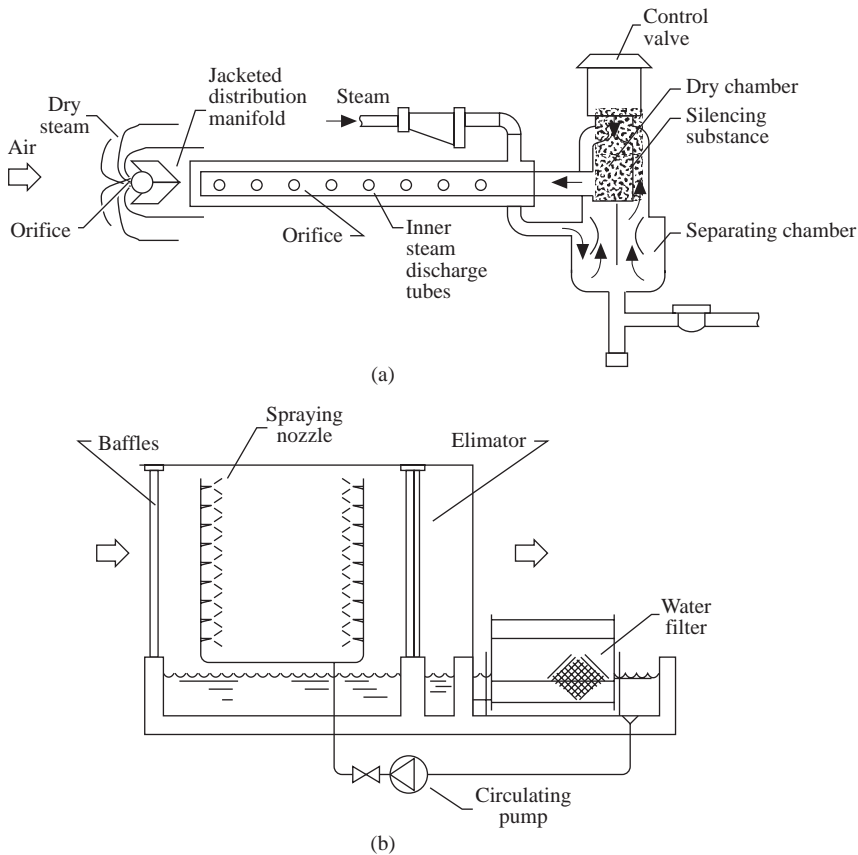


FIGURE 9.7.5 Steam grid humidifier (a) and air washer (b).

For an air system of cold air supply with humidity control during winter mode operation, an air washer is economical for large-capacity humidification in many industrial applications.

An air washer is a humidifier, a cooler, a dehumidifier, and an air cleaner. An air washer usually has an outer casing, two banks of spraying nozzles, one bank of guide baffles at the entrance, one bank of eliminators at the exit, a water tank, a circulating pump, a water filter, and other accessories as shown in [Figure 9.7.5\(b\)](#). Outer casing, baffles, and eliminators are often made of plastics or sometimes stainless steel. Spraying nozzles are usually made of brass or nylon, with an orifice diameter of 1/16 to 3/16 in., a smaller orifice for humidification, and a larger orifice for cooling and dehumidification. An eccentric inlet connected to the discharge chamber of the spraying nozzle gives centrifugal force to the water stream and atomizes the spraying water. Water is supplied to the spraying nozzle at a pressure of 15 to 30 psig. The distance between two spraying banks is 3 to 4.5 ft, and the total length of the air water from 4 to 7 ft. The air velocity inside an air washer is usually 500 to 800 fpm.

Selection of AHUs and PUs

- The size of an AHU is usually selected so that the face velocity of its coil is 600 fpm or less in order to prevent entrained condensate droplets. The cooling and heating capacities of an AHU can be varied by using coils of different numbers of rows and fin densities. The size of a PU is determined by its cooling capacity. Normally, the volume flow rate per ton of cooling capacity in PUs is 350 to 400 cfm. In most packaged units whose supply fans have belt drives, the fan speed can be selected so that the volume flow rate is varied and external pressure is met.
- ASHRAE/IES Standard 90.1-1989 specifies that the selected equipment capacity may exceed the design load only when it is the smallest size needed to meet the load. Selected equipment in a size larger always means a waste of energy and investment.
- To improve the indoor air quality, save energy, and prevent smudging and discoloring building interiors, a medium-efficiency filter of dust spot efficiency $\geq 50\%$ and an air economizer are preferable for large AHUs and PUs.

9.8 Refrigeration Components and Evaporative Coolers

Refrigeration Compressors

A *refrigeration compressor* is the heart of a vapor compression system. It raises the pressure of refrigerant so that it can be condensed into liquid, throttled, and evaporated into vapor to produce the refrigeration effect. It also provides the motive force to circulate the refrigerant through condenser, expansion valve, and evaporator.

According to the compression process, refrigeration compressors can be divided into *positive displacement* and *nonpositive displacement* compressors. A positive displacement compressor increases the pressure of the refrigerant by reducing the internal volume of the compression chamber. Reciprocating, scroll, rotary, and screw compressors are all positive displacement compressors. The centrifugal compressor is the only type of nonpositive displacement refrigeration compressor widely used in refrigeration systems today.

Based on the sealing of the refrigerant, refrigeration compressors can be classified as

- *Hermetic compressors*, in which the motor and the compressor are sealed or welded in the same housing to minimize leakage of refrigerant and to cool the motor windings by using suction vapor
- *Semihhermetic compressors*, in which motor and compressor are enclosed in the same housing but are accessible from the cylinder head for repair and maintenance
- *Open compressors*, in which compressor and motor are enclosed in two separate housings

Refrigeration compressors are often driven by motor directly or by gear train.

Performance Indices

Volumetric efficiency η_v of a refrigeration compressor is defined as

$$\eta_v = \dot{V}_{a,v} / \dot{V}_p \quad (9.8.1)$$

where $\dot{V}_{a,v}$ = actual induced volume of the suction vapor at suction pressure, cfm
 \dot{V}_p = calculated displacement of the compressor, cfm

Isentropic efficiency η_{isen} , *compression efficiency* η_{cp} , compressor efficiency η_{com} , and *mechanical efficiency* η_{mec} are defined as

$$\begin{aligned} \eta_{isen} &= (h_2 - h_1) / (h'_2 - h_1) = \eta_{cp} \eta_{mec} = \eta_{com} \\ \eta_{cp} &= W_{sen} / W_v \\ \eta_{mec} &= W_v / W_{com} \end{aligned} \quad (9.8.2)$$

where h_1, h_2, h'_2 = enthalpy of the suction vapor, ideal discharged hot gas, and actual discharged hot gas, respectively, Btu/lb

W_{isen}, W_v, W_{com} = isentropic work = $(h_2 - h_1)$, work delivered to the vapor refrigerant, and work delivered to the compressor shaft, Btu/lb

The actual power input to the compressor P_{com} , in hp, can be calculated as

$$\begin{aligned} P_{com} &= \dot{m}_r (h_2 - h_1) / (42.41 \eta_{isen} \eta_{mo}) \\ \dot{m}_r &= \dot{V}_p \eta_v \rho_{suc} \\ \eta_{mo} &= P_{com} / P_{mo} \end{aligned} \quad (9.8.3)$$

where \dot{m}_r = mass flow rate of refrigerant, lb/min
 ρ_{suc} = density of suction vapor, lb/ft³
 P_{mo} = power input to the compressor motor, hp

Power consumption, kW/ton refrigeration, is an energy index used in the HVAC&R industry in addition to EER and COP.

Currently used refrigeration compressors are reciprocating, scroll, screw, rotary, and centrifugal compressors.

Reciprocating Compressors

In a reciprocating compressor, as shown in [Figure 9.8.1\(a\)](#), a crankshaft connected to the motor shaft drives 2, 3, 4, or 6 single-acting pistons moving reciprocally in the cylinders via a connecting rod.

The refrigeration capacity of a reciprocating compressor is a fraction of a ton to about 200 tons. Refrigerants R-22 and R-134a are widely used in comfort and processing systems and sometimes R-717 in industrial applications. The maximum compression ratio R_{com} for a single-stage reciprocating compressor is about 7. Volumetric efficiency η_v drops from 0.92 to 0.65 when R_{com} is raised from 1 to 6. Capacity control of reciprocating compressor including: on-off and cylinder unloader in which discharge gas is in short cut and return to the suction chamber.

Although reciprocating compressors are still widely used today in small and medium-sized refrigeration systems, they have little room for significant improvement and will be gradually replaced by scroll and screw compressors.

Scroll Compressors

A scroll compressor consists of two identical spiral scrolls assembled opposite to each other, as shown in [Figure 9.8.1\(b\)](#). One of the scrolls is fixed, and the other moves in an orbit around the motor shaft whose amplitude equals the radius of the orbit. The two scrolls are in contact at several points and therefore form a series of pockets.

Vapor refrigerant enters the space between two scrolls through lateral openings. The lateral openings are then sealed and the formation of the two trapped vapor pockets indicates the end of the suction process. The vapor is compressed and the discharge process begins when the trapped gaseous pockets open to the discharge port. Compressed hot gas is then discharged through this opening to the discharge line. In a scroll compressor, the scrolls touch each other with sufficient force to form a seal but not enough to cause wear.

The upper limit of the refrigeration capacity of currently manufactured scroll compressors is 60 tons. A scroll compressor has $\eta_v > 95\%$ at $R_{\text{com}} = 4$ and $\eta_{\text{isen}} = 80\%$. A scroll compressor also has only about half as many parts as a reciprocating compressor at the same refrigeration capacity. Few components result in higher reliability and efficiency. Power input to the scroll compressor is about 5 to 10% less than to the reciprocating compressor. A scroll compressor also operates more smoothly and is quieter.

Rotary Compressors

Small rotary compressors for room air conditioners and refrigerators have a capacity up to 4 tons. There are two types of rotary compressors: rolling piston and rotating vane. A typical rolling piston rotary compressor is shown in [Figure 9.8.1\(c\)](#). A rolling piston mounted on an eccentric shaft is kept in contact with a fixed vane that slides in a slot. Vapor refrigerant enters the compression chamber and is compressed by the eccentric motion of the roller. When the rolling piston contacts the top housing, hot gas is squeezed out from the discharge valve.

Screw Compressors

These are also called *helical rotary compressors*. Screw compressors can be classified into single-screw compressors, in which there is a single helical rotor and two star wheels, and twin-screw compressors. Twin-screw compressors are widely used.

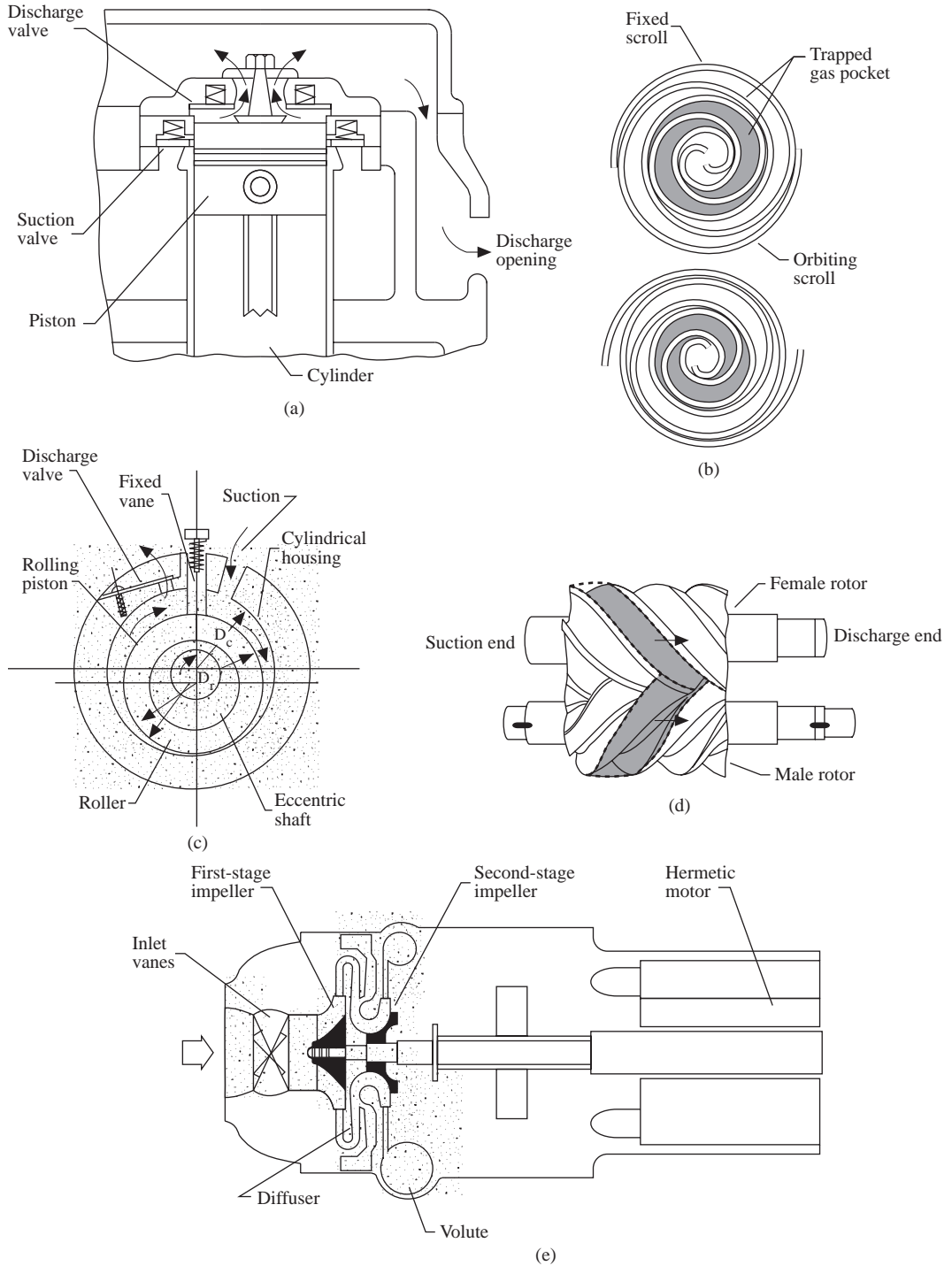


FIGURE 9.8.1 Various types of refrigeration compressors: (a) reciprocating, (b) scroll, (c) rotary, (d) twin-screw, and (e) centrifugal.

A typical twin-screw compressor, as shown in [Figure 9.8.1\(d\)](#) consists of a four-lobe male rotor and a six-lobe female rotor, a housing with suction and discharge ports, and a sliding valve to adjust the

capacity during part load. Normally, the male rotor is the driver. Twin-screw compressors are often direct driven and of hermetic type.

Vapor refrigerant is extracted into the interlobe space when the lobes are separated at the suction port. During the successive rotations of the rotor, the volume of the trapped vapor is compressed. When the interlobe space is in contact with the discharge port, the compressed hot gas discharges through the outlet. Oil injection effectively cools the rotors and results in a lower discharge temperature. Oil also provides a sealing effect and lubrication. A small clearance of 0.0005 in. as well as the oil sealing minimizes leakage of the refrigerant.

The refrigeration capacity of twin-screw compressors is 50 to 1500 tons. The compression ratio of a twin-screw compressor can be up to 20:1. R-22 and R-134a are the most widely used refrigerants in comfort systems. In a typical twin-screw compressor, η_v decreases from 0.92 to 0.87 and η_{isen} drops from 0.82 to 0.67 when R_{com} increases from 2 to 10. Continuous and stepless capacity control is provided by moving a sliding valve toward the discharge port, which opens a shortcut recirculating passage to the suction port.

Twin-screw compressors are more efficient than reciprocating compressors. The low noise and vibration of the twin-screw compressor together with its positive displacement compression results in more applications today.

Centrifugal Compressors

A *centrifugal compressor* is a turbomachine and is similar to a centrifugal fan. A hermetic centrifugal compressor has an outer casing with one, two, or even three impellers internally connected in series and is driven by a motor directly or by a gear train. At the entrance to the first-stage impeller are inlet guide vanes positioned at a specific opening to adjust refrigerant flow and therefore the capacity of the centrifugal compressor.

Figure 9.8.1(e) shows a two-stage hermetic centrifugal compressor. The total pressure rise in a centrifugal compressor, often called head lift, in psi, is due to the conversion of the velocity pressure into static pressure. Although the compression ratio R_{com} of a single-stage centrifugal compressor using R-123 and R-22 seldom exceeds 4, two or three impellers connected in series satisfy most of the requirements in comfort systems.

Because of the high head lift to raise the evaporating pressure to condensing pressure, the discharge velocity at the exit of the second-stage impeller approaches the acoustic velocity of saturated vapor v_{ac} of R-123, 420 ft/sec at atmospheric pressure and a temperature of 80°F. Centrifugal compressors need high peripheral velocity and rotating speeds (up to 50,000 rpm) to produce such a discharge velocity. It is not economical to manufacture small centrifugal compressors. The available refrigeration capacity for centrifugal compressors ranges from 100 to 10,000 tons. Centrifugal compressors have higher volume flow per unit refrigeration capacity output than positive displacement compressors. Centrifugal compressors are efficient and reliable. Their volumetric efficiency almost equals 1. At design conditions, their η_{isen} may reach 0.83, and it drops to 0.6 during part-load operation. They are the most widely used refrigeration compressors in large air-conditioning systems.

Refrigeration Condensers

A *refrigeration condenser* or simply a *condenser* is a heat exchanger in which hot gaseous refrigerant is condensed into liquid and the latent heat of condensation is rejected to the atmospheric air, surface water, or well water. In a condenser, hot gas is first desuperheated, then condensed into liquid, and finally subcooled.

The capacity of a condenser is rated by its *total heat rejection* Q_{rej} , in Btu/hr, which is defined as the total heat removed from the condenser during desuperheating, condensation, and subcooling. For a refrigeration system using a hermetic compressor, Q_{rej} can be calculated as

$$\dot{Q}_{\text{rej}} = U_{\text{con}} A_{\text{con}} \Delta T_m = 60 \dot{m}_r (h_2 - h'_3) = q_{\text{rl}} + (2545 P_{\text{com}}) / \eta_{\text{mo}} \quad (9.8.4)$$

where U_{con} = overall heat transfer coefficient across the tube wall in the condenser, Btu/hr.ft².°F
 A_{con} = condensing area in the condenser, ft²
 ΔT_m = logarithmic temperature difference, °F
 \dot{m}_r = mass flow rate of refrigerant, lb/min
 h_2, h'_3 = enthalpy of suction vapor refrigerant and hot gas, Btu/lb
 q_{rl} = refrigeration load at the evaporator, Btu/hr

A factor that relates \dot{Q}_{rej} and q_{rl} is the *heat rejection factor* F_{rej} , which is defined as the ratio of total heat rejection to the refrigeration load, or

$$F_{\text{rej}} = \dot{Q}_{\text{rej}} / q_{\text{rl}} = 1 + (2545 P_{\text{com}}) / (q_{\text{rl}} \eta_{\text{mo}}) \quad (9.8.5)$$

Fouling factor R_f , in hr.ft².°F/Btu, is defined as the additional resistance caused by a dirty film of scale, rust, or other deposits on the surface of the tube. ARI Standard 550-88 specifies the following for evaporators and condensers:

Field fouling allowance	0.00025 hr.ft ² .°F/Btu
New evaporators and condensers	0

According to the cooling process used during condensation, refrigeration condensers can be classified as air-cooled, water-cooled, and evaporative-cooled condensers.

Air-Cooled Condensers

In an *air-cooled condenser*, air is used to absorb the latent heat of condensation released during desuperheating, condensation, and subcooling.

An air-cooled condenser consists of a condenser coil, a subcooling coil, condenser fans, dampers, and controls as shown in Figure 9.8.2(a). There are refrigeration circuits in the condensing coil. Condensing coils are usually made of copper tubes and aluminum fins. The diameter of the tubes is 1/4 to 3/4 in., typically 3/8 in., and the fin density is 8 to 20 fins/in. On the inner surface of the copper tubes, microfins, typically 60 fins/in. with a height of 0.008 in., are used. A condensing coil usually has only two to three rows due to the low pressure drop of the propeller-type condenser fans. A subcooling coil is located at a lower level and is connected to the condensing coil.

Hot gas from the compressor enters the condensing coil from the top. When the condensate increases, part of the condensing area can be used as a subcooling area. A receiver is necessary only when the liquid refrigerant cannot all be stored in the condensing and subcooling coils during the shut-down period in winter.

Cooling air is drawn through the coils by a condenser fan(s) for even distribution. Condenser fans are often propeller fans for their low pressure and large volume flow rate. A damper(s) may be installed to adjust the volume flow of cooling air.

In air-cooled condensers, the volume flow of cooling air per unit of total heat rejection $\dot{V}_{\text{ca}} / \dot{Q}_{\text{u.rej}}$ is 600 to 1200 cfm/ton of refrigeration capacity at the evaporator, and the optimum value is about 900 cfm/ton. The corresponding cooling air temperature difference — cooling air leaving temperature minus outdoor temperature ($T_{\text{ca,l}} - T_o$) — is around 13°F.

The condenser temperature difference (CTD) for an air-cooled condenser is defined as the difference between the saturated condensing temperature corresponding to the pressure at the inlet and the air intake temperature, or ($T_{\text{con,i}} - T_o$). Air-cooled condensers are rated at a specific CTD, depending on the evaporating temperature of the refrigeration system T_{ev} in which the air-cooled condenser is installed. For a refrigeration system having a lower T_{ev} , it is more economical to equip a larger condenser with a smaller CTD. For a comfort air-conditioning system having a T_{ev} of 45°F, CTD = 20 to 30°F.

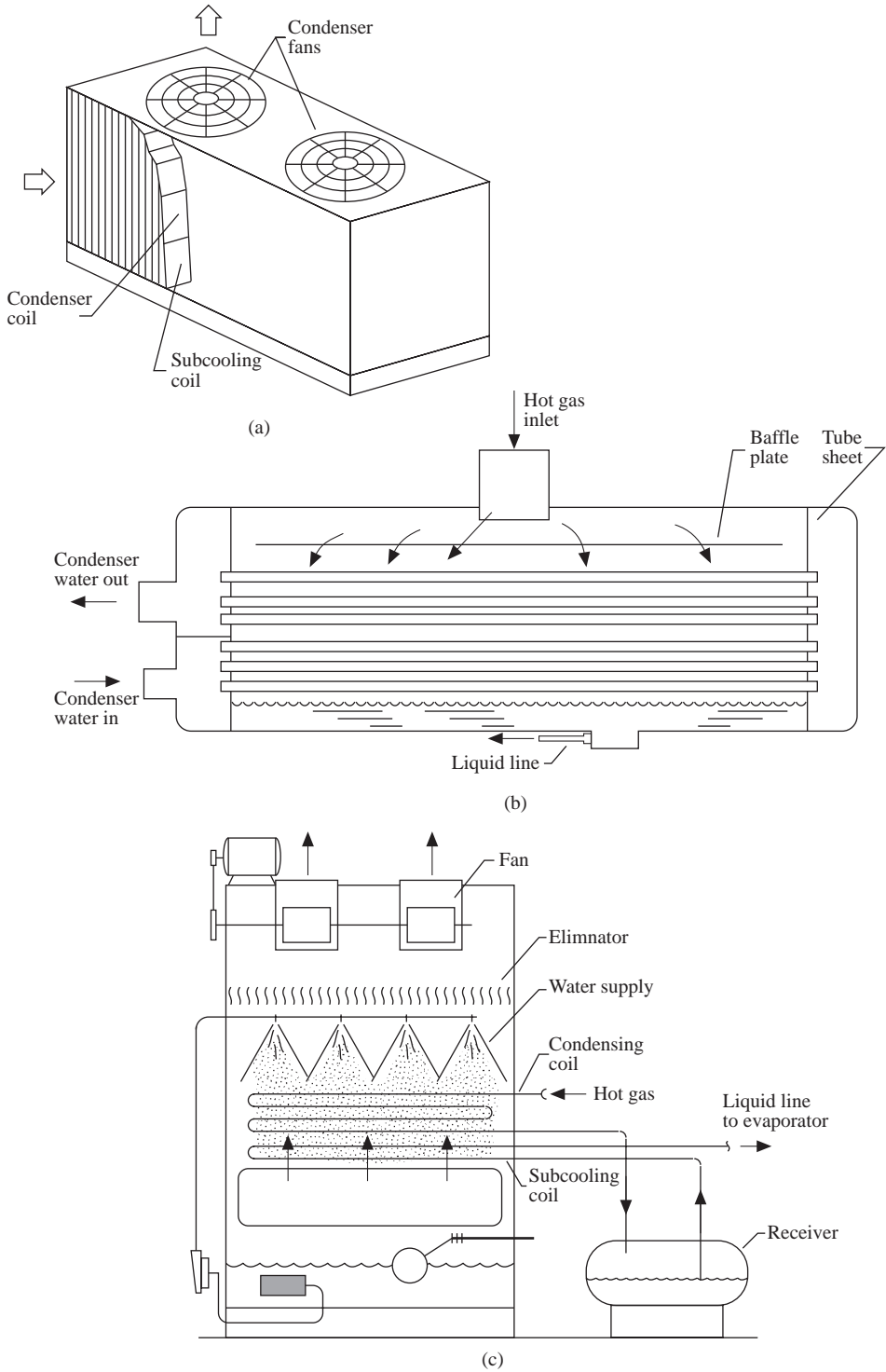


FIGURE 9.8.2 Various types of refrigeration condensers: (a) air-cooled, (b) shell-and-tube water-cooled, and (c) evaporative cooled.

A higher condensing temperature T_{con} , a higher condensing pressure p_{con} , and a higher compressor power input may be due to an undersized air-cooled condenser, lack of cooling air or low $\dot{V}_{\text{ca}}/Q_{\text{u,rej}}$ value, a high entering cooling air temperature at the roof, a dirty condensing coil, warm air circulation because of insufficient clearance between the condenser and the wall, or a combination of these. The clearance should not be less than the width of the condensing coil.

If p_{con} drops below a certain value because of a lower outdoor temperature, the expansion valve in a reciprocating vapor compression system may not operate properly. At a low ambient temperature T_o , the following controls are often used:

- Duty cycling, turning the condenser fans on and off until all of them are shut down, to reduce cooling air volume flow
- Modulating the air dampers to reduce the volume flow
- Reducing the fan speed

Some manufacturers' catalogs start low ambient control at $T_o = 65^\circ\text{F}$ and some specify a minimum operating temperature at $T_o = 0^\circ\text{F}$.

Water-Cooled Condensers

In a *water-cooled condenser*, latent heat of condensation released from the refrigerant during condensation is extracted by water. This cooling water, often called condenser water, is taken directly from river, lake, sea, underground well water or a cooling tower.

Two types of water-cooled condensers are widely used for air-conditioning and refrigeration: double-tube condensers and horizontal shell-and-tube condensers.

A *double-tube condenser* consists of two tubes, one inside the other. Condenser water is pumped through the inner tube and refrigerant flows within the space between the inner and outer tubes in a counterflow arrangement. Because of its limited condensing area, the double-tube condenser is used only in small refrigeration systems.

A horizontal *shell-and-tube water-cooled condenser* using halocarbon refrigerant usually has an outer shell in which copper tubes typically 5/8 to 3/4 in. in diameter are fixed in position by tube sheets as shown in [Figure 9.8.2\(b\)](#). Integral external fins of 19 to 35 fins/in. and a height of 0.006 in. and spiral internal grooves are used for copper tubes to increase both the external and the inner surface area and their heat transfer coefficients.

Hot gas from the compressor enters the top inlet and is distributed along the baffle to fill the shell. Hot gas is then desuperheated, condensed, subcooled into liquid, and discharged into the liquid line at the bottom outlet. Usually one sixth of the volume is filled with subcooled liquid refrigerant. Subcooling depends on the entering temperature of condenser water T_{ce} , in $^\circ\text{F}$, and usually varies between 2 and 8°F .

Condenser water enters the condenser from the bottom for effective subcooling. After extracting heat from the gaseous refrigerant, condenser water is discharged at a higher level. Two-pass or three-pass water flow arrangements are usually used in shell-and-tube water-cooled condensers. The two-pass arrangement means that water flows from one end to the opposite end and returns to the original end. Two-pass is the standard setup. In a shell-and-tube water-cooled condenser, the condensing temperature T_{con} depends mainly on the entering temperature of condenser water T_{ce} , the condenser area, the fouling factor, and the configuration of the copper tube.

Evaporative Condenser

An *evaporative condenser* uses the evaporation of water spray on the outer surface of the condensing tubes to remove the latent heat of condensation of refrigerant during condensation.

An evaporative condenser consists of a condensing coil, a subcooling coil, a water spray, an induced draft or sometimes forced draft fan, a circulating water pump, a water eliminator, a water basin, an outer casing, and controls as shown in [Figure 9.8.2\(c\)](#). The condensing coil is usually made of bare copper, steel, or sometimes stainless steel tubing.

Water is sprayed over the outside surface of the tubing. The evaporation of a fraction of condenser water from the saturated air film removes the sensible and latent heat rejected by the refrigerant. The wetted outer surface heat transfer coefficient h_{wet} is about four or five times greater than the dry surface heat transfer coefficient h_o , in Btu/hr.ft².°F. The rest of the spray falls and is collected by the basin. Air enters from the inlet just above the basin. It flows through the condensing coil at a face velocity of 400 to 700 fpm, the water spray bank, and the eliminator. After air absorbs the evaporated water vapor, it is extracted by the fan and discharged at the top outlet. The water circulation rate is about 1.6 to 2 gpm/ton, which is far less than that of the cooling tower.

An evaporative condenser is actually a combination of a water-cooled condenser and a cooling tower. It is usually located on the rooftop and should be as near the compressor as possible. Clean tube surface and good maintenance are critical factors for evaporative condensers. An evaporative condenser also needs low ambient control similar as in an air-cooled condenser.

Comparison of Air-Cooled, Water-Cooled, and Evaporative Condensers

An air-cooled condenser has the highest condensing temperature T_{con} and therefore the highest compressor power input. For an outdoor dry bulb temperature of 90°F and a wet bulb temperature of 78°F, a typical air-cooled condenser has $T_{\text{con}} = 110^\circ\text{F}$. An evaporative condenser has the lowest T_{con} and is most energy efficient. At the same outdoor dry and wet bulb temperatures, its T_{con} may be equal to 95°F, even lower than that of a water-cooled condenser incorporating with a cooling tower, whose T_{con} may be equal to 100°F. An evaporative condenser also consumes less water and pump power. The drawback of evaporative condensers is that the rejected heat from the interior zone is difficult to recover and use as winter heating for perimeter zones and more maintenance is required.

Evaporators and Refrigerant Flow Control Devices

An *evaporator* is a heat exchanger in which the liquid refrigerant is vaporized and extracts heat from the surrounding air, chilled water, brine, or other substance to produce a refrigeration effect.

Evaporators used in air-conditioning can be classified according to the combination of the medium to be cooled and the type of refrigerant feed, as the following.

Direct expansion DX coils are air coolers, and the refrigerant is fed according to its degree of superheat after vaporization. DX coils were covered earlier.

Direct expansion ice makers or *liquid overfeed ice makers* are such that liquid refrigerant is forced through the copper tubes or the hollow inner part of a plate heat exchanger and vaporized. The refrigeration effect freezes the water in the glycol-water that flows over the outside surface of the tubes or the plate heat exchanger. In direct expansion ice makers, liquid refrigerant completely vaporizes inside the copper tubes, and the superheated vapor is extracted by the compressor. In liquid overfeed ice makers, liquid refrigerant floods and wets the inner surface of the copper tubes or the hollow plate heat exchanger. Only part of the liquid refrigerant is vaporized. The rest is returned to a receiver and pumped to the copper tubes or plate heat exchanger again at a circulation rate two to several times greater than the evaporation rate.

Flooded shell-and-tube liquid coolers, or simply *flooded liquid coolers*, are such that refrigerant floods and wets all the boiling surfaces and results in high heat transfer coefficients. A flooded shell-and-tube liquid cooler is similar in construction to a shell-and-tube water-cooled condenser, except that its liquid refrigeration inlet is at the bottom and the vapor outlet is at the top. Water velocity inside the copper tubes is usually between 4 and 12 ft/sec and the water-side pressure normally drops below 10 psi. Flooded liquid coolers can provide larger evaporating surface area and need minimal space. They are widely used in large central air-conditioning systems.

Currently used refrigerant flow control devices include thermostatic expansion valves, float valves, multiple orifices, and capillary tubes.

A *thermostatic expansion valve* throttles the refrigerant pressure from condensing to evaporating pressure and at the same time regulates the rate of refrigerant feed according to the degree of superheat

of the vapor at the evaporator's exit. A thermostatic expansion valve is usually installed just prior to the refrigerant distributor in DX coils and direct-expansion ice makers.

A thermostatic expansion valve consists of a valve body, a valve pin, a spring, a diaphragm, and a sensing bulb near the outlet of the DX coil, as shown in Figure 9.7.3(a). The sensing bulb is connected to the upper part of the diaphragm by a connecting tube.

When the liquid refrigerant passes through the opening of the thermostatic expansion valve, its pressure is reduced to the evaporating pressure. Liquid and a small fraction of vaporized refrigerant then flow through the distributor and enter various refrigerant circuits. If the refrigeration load of the DX coil increases, more liquid refrigerant evaporizes. This increases the degree of superheat of the leaving vapor at the outlet and the temperature of the sensing bulb. A higher bulb temperature exerts a higher saturated pressure on the top of the diaphragm. The valve pin then moves downward and widens the opening. More liquid refrigerant is allowed to enter the DX coil to match the increase of refrigeration load. If the refrigeration load drops, the degree of superheat at the outlet and the temperature of the sensing bulb both drop, and the valve opening is narrower. The refrigeration feed decreases accordingly. The degree of superheat is usually 10 to 20°F. Its value can also be adjusted manually by varying the spring tension.

A *float valve* is a valve in which a float is used to regulate the valve opening to maintain a specific liquid refrigerant level. A lower liquid level causes a lower valve pin and therefore a wider opening and vice versa.

In a centrifugal refrigeration system, two or more orifice plates, *multiple orifices*, are sometimes installed in the liquid line between the condenser and the flash cooler and between the flash cooler and the flooded liquid cooler to throttle their pressure as well as to regulate the refrigerant feed.

A *capillary tube*, sometimes called a *restrictor tube*, is a fixed length of small-diameter tubing installed between the condenser and the evaporator to throttle the refrigerant pressure from p_{con} to p_{ev} and to meter the refrigerant flow to the evaporator. Capillary tubes are usually made of copper. The inside diameter D_{cap} is 0.05 to 0.06 in. and the length L_{cap} from an inch to several feet. There is a trend to use short capillary tubes of $L_{\text{cap}}/D_{\text{cap}}$ between 3 and 20. Capillary tubes are especially suitable for a heat pump system in which the refrigerant flow may be reversed.

Evaporative Coolers

An evaporative cooling system is an air-conditioning system in which air is cooled evaporatively. It consists of evaporative coolers, fan(s), filters, dampers, controls, and others. A mixing box is optional. An evaporative cooler could be a stand-alone cooler or installed in an air system as a component. There are three types of evaporative coolers: (1) direct evaporative coolers, (2) indirect evaporative coolers, and (3) indirect–direct evaporative coolers.

Direct Evaporative Cooler

In a *direct evaporative cooler*, the air stream to be cooled directly contacts the water spray or wetted medium as shown in Figure 9.8.3(a). Evaporative pads made of wooden fibers with necessary treatment at a thickness of 2 in., rigid and corrugated plastics, impregnated cellulose, or fiber glass all dripping with water are wetted mediums.

The direct evaporation process 12 takes place along the thermodynamic wet bulb line on the psychrometric chart. Saturation effectiveness ϵ_{sat} is an index that assesses the performance of a direct evaporative cooler:

$$\epsilon_{\text{sat}} = (T_{\text{ae}} - T_{\text{al}}) / (T_{\text{ae}} - T_{\text{ae}}^*) \quad (9.8.6)$$

where T , T^* = temperature and thermodynamic wet bulb temperature of air stream, °F. Subscript ae indicates the entering air and al the leaving air. ϵ_{sat} usually varies between 0.75 and 0.95 at a water–air ratio of 0.1 to 0.4.

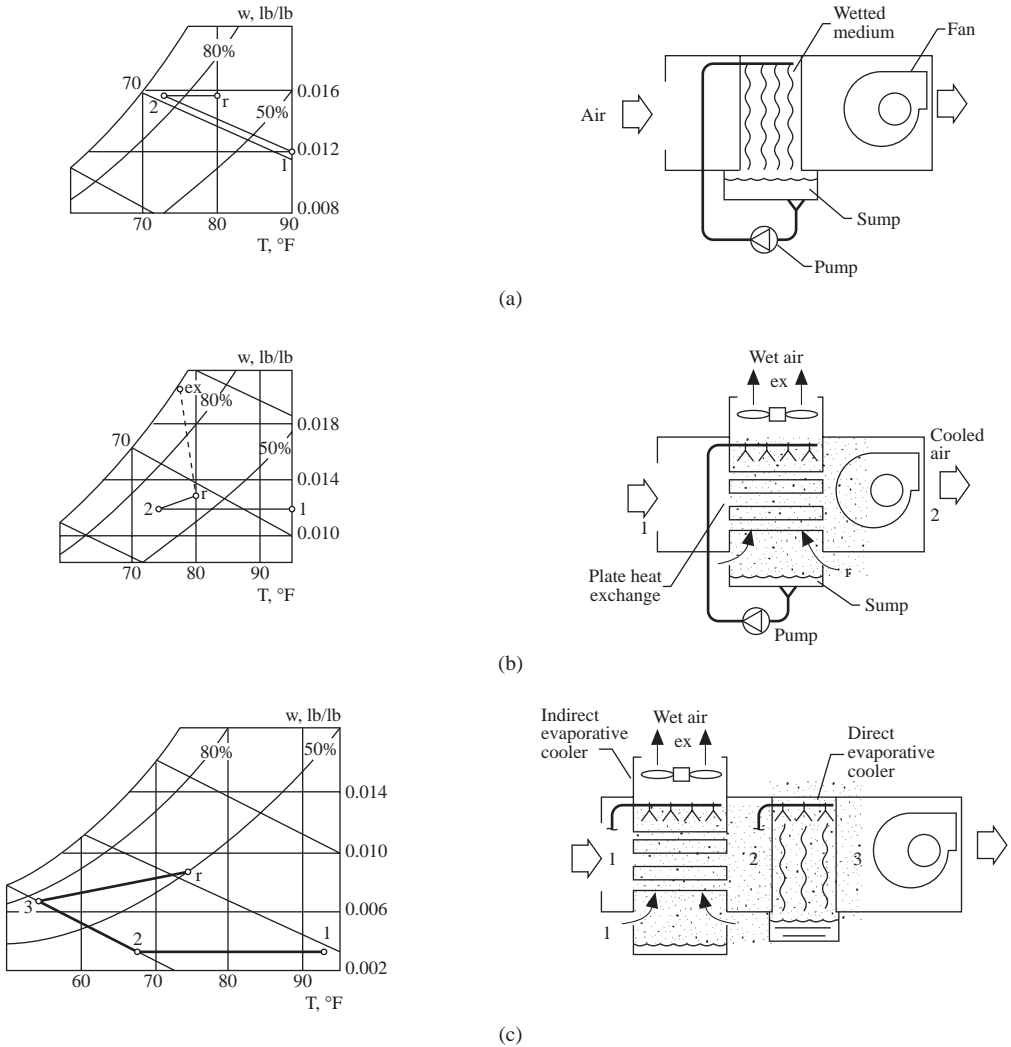


FIGURE 9.8.3 Types of evaporative coolers: (a) direct, (b) indirect, and (c) indirect–direct.

Indirect Evaporative Coolers

In an *indirect evaporative cooler*, the cooled-air stream to be cooled is separated from a wetted surface by a flat plate or tube wall as shown in Figure 9.8.3(b). A wet-air stream flows over the wetted surface so that liquid water is evaporated and extracts heat from the cooled-air stream through the flat plate or tube wall. The cooled-air stream is in contact with the wetted surface indirectly.

The core part of an indirect evaporative cooler is a plate heat exchanger. It is made of thin polyvinyl chloride plates 0.01 in. thick and spaced from 0.08 to 0.12 in. apart to form horizontal passages for cooled air and vertical passages for wet air and water. As in a direct evaporative cooler, there are also fan(s), water sprays, circulating pump, air intake, dampers, controls, etc.

An indirect evaporative cooling process is represented by a horizontal line on a psychrometric chart, which shows that humidity ratio remains constant. If the space air is extracted and used as the wet air intake, the wet air will be exhausted at point x at nearly saturated state.

The performance of an indirect evaporative cooler can be assessed by its performance factor e_{in} , which is calculated as:

$$e_{\text{in}} = (T_{\text{ca,e}} - T_{\text{ca,l}}) / (T_{\text{ca,e}} - T_{\text{s,a}}) \quad (9.8.7)$$

where $T_{\text{ca,e}}$, $T_{\text{ca,l}}$ = temperature of cooled air entering and leaving the indirect evaporative cooler, °F, and $T_{\text{s,a}}$ = temperature of the saturated air film on the wet air side and is about 3°F higher than the wet bulb temperature of the entering air, °F.

An indirect evaporative cooler could be so energy efficient as to provide evaporative cooling with an EER up to 50 instead of 9 to 12 for a reciprocating compression refrigeration system.

Direct–Indirect Evaporative Cooler. A direct–indirect evaporative cooler is a two-stage evaporating cooler, as shown in [Figure 9.8.3\(c\)](#), in which the first-stage indirect evaporative cooler is connected in series with a second-stage direct evaporative cooler for the purpose of increasing the evaporating effect.

Operating Characteristics. The saturation effectiveness ϵ_{sat} and performance factor e_{in} are both closely related to the air velocity flowing through the air passages. For a direct evaporative cooler, face velocity is usually less than 600 fpm to reduce drift carryover. For an indirect evaporative cooler, face velocity v_s is usually between 400 to 1000 fpm. A higher v_s results at a greater air-side pressure drop.

Scofield et al. (1984) reported the performance of an indirect–direct evaporative cooler in Denver, Colorado. Outdoor air enters the indirect cooler at a dry bulb of 93°F and a wet bulb of 67.5° and was evaporatively cooled to 67.5°F dry bulb and 49.8°F wet bulb with an $e_{\text{in}} = 0.76$ as shown in [Figure 9.8.3\(c\)](#). In the direct cooler, conditioned air was further cooled to a dry bulb of 53.5°F and the wet bulb remained at 49.8°F at a saturation effectiveness $\epsilon_{\text{sat}} = 0.8$.

In locations where outdoor wet bulb $T'_o \leq 60^\circ\text{F}$, a direct evaporative can often provide an indoor environment of 78°F and a relative humidity of 60%. In locations $T'_o \leq 68^\circ\text{F}$, an indirect–direct evaporative cooler can maintain a comfortable indoor environment. In locations $T'_o \geq 72^\circ\text{F}$, an evaporative cooler with a supplementary vapor compression refrigeration may be cost effective. Because the installation cost of an indirect–direct cooler is higher than that of refrigeration, cost analysis is required to select the right choice. Evaporative coolers are not suitable for dehumidification except in locations where $T'_o \leq 60^\circ\text{F}$.

9.9 Water Systems

Types of Water Systems

In central and space conditioning air-conditioning systems, water that links the central plant and the air handling units or terminals, that extracts condensing heat, or that provides evaporative cooling may be classified as

- *Chilled water system*, in which chilled water is first cooled in the centrifugal, screw, and reciprocating chillers in a central plant. Chilled water is then used as a cooling medium to cool the air in the cooling coils in AHUs and terminals.
- *Evaporative-cooled water system*, used to cool air directly or indirectly in evaporative coolers.
- *Hot water system*, in which hot water is heated in the boiler and then used to heat the air through heating coils in AHUs, terminals, or space finned-tube heaters.
- *Dual-temperature water system*, in which chilled water and hot water are supplied to and returned from the coils in AHUs and terminals through separate or common main and branch pipes. Using common main and branch pipes requires a lengthy changeover from chilled water to hot water or vice versa for a period of several hours.
- *Condenser water system*, which is a kind of cooling water system used to extract the latent heat of condensation from the condensing refrigerant in a water-cooled condenser and heat of absorption from the absorber.

Water systems can also be classified according to their operating characteristics.

Closed System

In a closed system, water forms a closed loop for water conservation and energy saving when it flows through the coils, chillers, boilers, heaters, or other heat exchangers and water is not exposed to the atmosphere.

Open System

In an open system, water is exposed to the atmosphere.

Once-Through System

In a once-through system, water flows through a heat exchanger(s) only once without recirculation.

Basics

Volume Flow and Temperature Difference

The rate of heat transfer between water and air or water and refrigerant when water flows through a heat exchanger q_w , in Btu/hr, can be calculated as

$$q_w = 500 \dot{V}_{\text{gal}} (T_{\text{wl}} - T_{\text{we}}) = 500 \dot{V}_{\text{gal}} \Delta T_w \quad (9.9.1)$$

where \dot{V}_{gal} = volume flow rate of water, gpm
 $T_{\text{wl}}, T_{\text{we}}$ = temperature of water leaving and entering the heat exchanger, °F
 ΔT_w = temperature rise or drop of water when it flows through a heat exchanger, °F

The temperature of chilled water leaving the water chiller T_{el} should not be lower than 38°F in order to prevent freezing in the evaporator. Otherwise, brine or glycol-water should be used. The T_{el} of chilled water entering the coil T_{we} and the temperature difference of chilled water leaving and entering the coil ΔT_w directly affect the temperature of air leaving the cooling coil T_{ce} . The lower T_{we} , the higher will be the compressor power input. The smaller ΔT_w , the greater will be the water volume flow rate, the pipe

size, and the pump power. For chilled water in conventional comfort systems, T_{we} is usually 40 to 45°F and ΔT_w 12 to 24°F. Only in cold air distribution, T_{we} may drop to 34°F. For a cooling capacity of 1 ton refrigeration, a ΔT_w of 12°F requires a $\dot{V}_{gal} = 2$ gpm.

For hot water heating systems in buildings, hot water often leaves the boiler and enters the heating coil or heaters at a temperature T_{we} of 190 to 200°F. It returns at 150 to 180°F. For dual-temperature systems, hot water is usually supplied at 100 to 150°F and returns at a ΔT_w of 20 to 40°F.

Pressure Drop

Usually the pressure drop of water in pipes due to friction for HVAC&R systems, H_f , is in the range 0.75 ft/100 ft length of pipe to 4 ft/100 ft. A pressure loss of 2.5 ft/100 ft is most often used. ASHRAE/IES Standard 90.1-1989 specifies that water piping systems should be designed at a pressure loss of no more than 4.0 ft/100 ft. Figure 9.9.1(a), (b), and (c) shows the friction charts for steel, copper, and plastic pipes for closed water systems.

Water Piping

The piping materials of various water systems for HVAC&R are as follows:

Chilled water	Black and galvanized steel
Hot water	Black steel, hard copper
Condenser water	Black steel, galvanized ductile iron, polyvinyl chloride (PVC)

The pipe thickness varies from Schedule 10, a light wall pipe, to Schedule 160, a very heavy wall pipe. Schedule 40 is the standard thickness for a pipe of up to 10 in. diameter. For copper tubing, type K is the heaviest, and type L is generally used as the standard for pressure copper tubes.

Steel pipes of small diameter are often joined by threaded cast-iron fittings. Steel pipes of diameter 2 in. and over, welded joints, and bolted flanges are often used.

In a water system, the maximum allowable working pressure for steel and copper pipes at 250°F varies from 125 to 400 psig, depending on the pipe wall thickness. Not only pipes, but also their joints and fittings should be considered.

During temperature changes, pipes expand and contract. Both operating and shut-down periods should be taken into consideration. Bends like U-, Z-, and L-bends, loops, and sometimes packed expansion joints, bellows, or flexible metal hose mechanical joints are used.

ASHRAE/IES Standard 90.1-1989 specifies that for chilled water between 40 to 55°F, the minimum thickness of the external pipe insulation varies from 0.5 to 1 in. depending on the pipe diameter. For chilled water temperature below 40°F, the minimum thickness varies from 1 to 1.5 in. For hot water temperatures not exceeding 200°F, the minimum thickness varies from 0.5 to 1.5 in. depending on the pipe diameter and the hot water temperature.

Corrosion, Impurities, and Water Treatments

Corrosion is a destructive process caused by a chemical or electrochemical reaction on metal or alloy. In water systems, dissolved *impurities* cause corrosion and scale and the growth of microorganisms like algae, bacteria, and fungi. *Scale* is the deposit formed on a metal surface by precipitation of the insoluble constituents. In addition to the dissolved solids, unpurified water may contain suspended solids.

Currently used chemicals include crystal modifiers to change the crystal formation of scale and sequestering chemicals. Growth of bacteria, algae, and fungi is usually treated by biocides to prevent the formation of an insulating layer resulting in lower heat transfer as well as restricted water flow. Chlorine and its compounds are effective and widely used. Blow-down is an effective process in water treatment and should be considered as important as chemical treatments.

Piping Arrangements

Main and Branch Pipes. In a piping circuit as shown in Figure 9.9.2(a), chilled water from a chiller or hot water from a boiler is often supplied to a *main pipe* and then distributed to *branch pipes* that

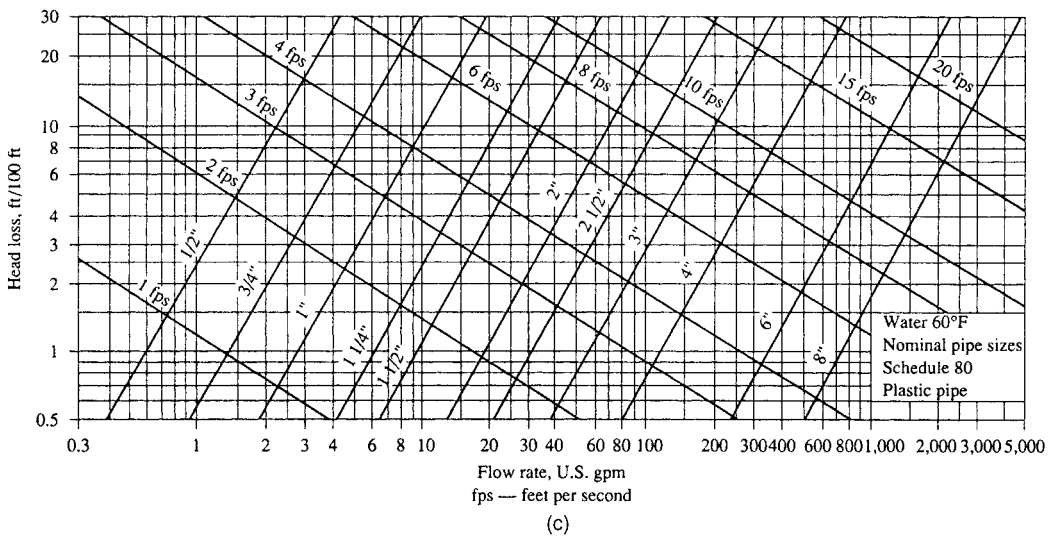
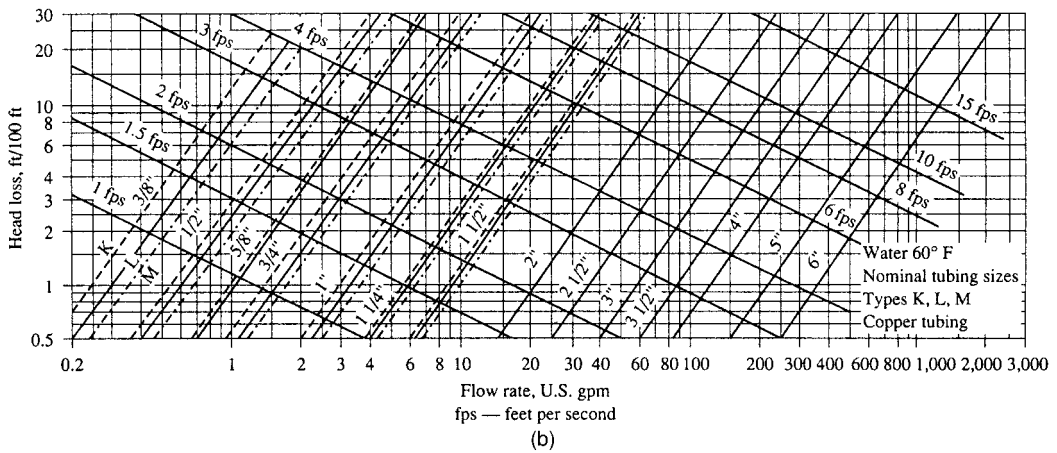
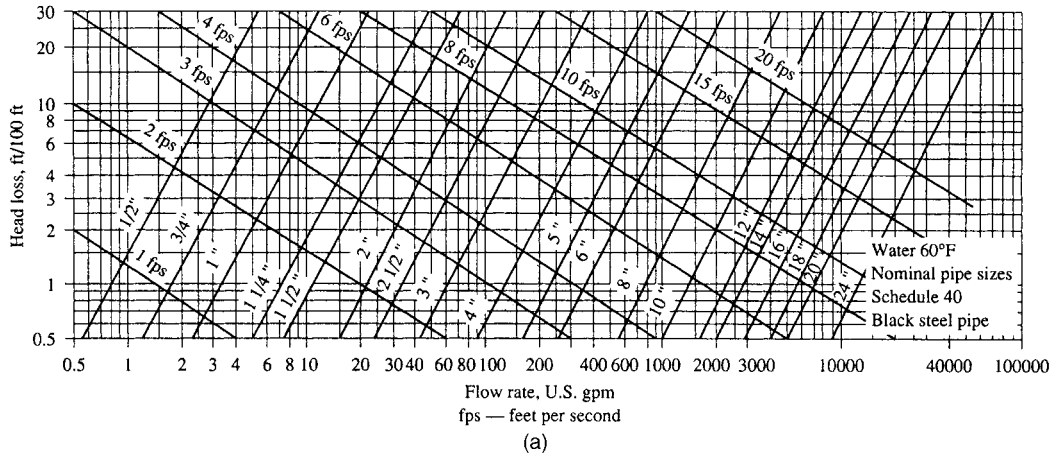


FIGURE 9.9.1 Friction chart for water in pipes: (a) steel pipe (schedule 40), (b) copper tubing, and (c) plastic pipe (schedule 80). (Source: ASHRAE Handbook 1993 Fundamentals. Reprinted with permission.)

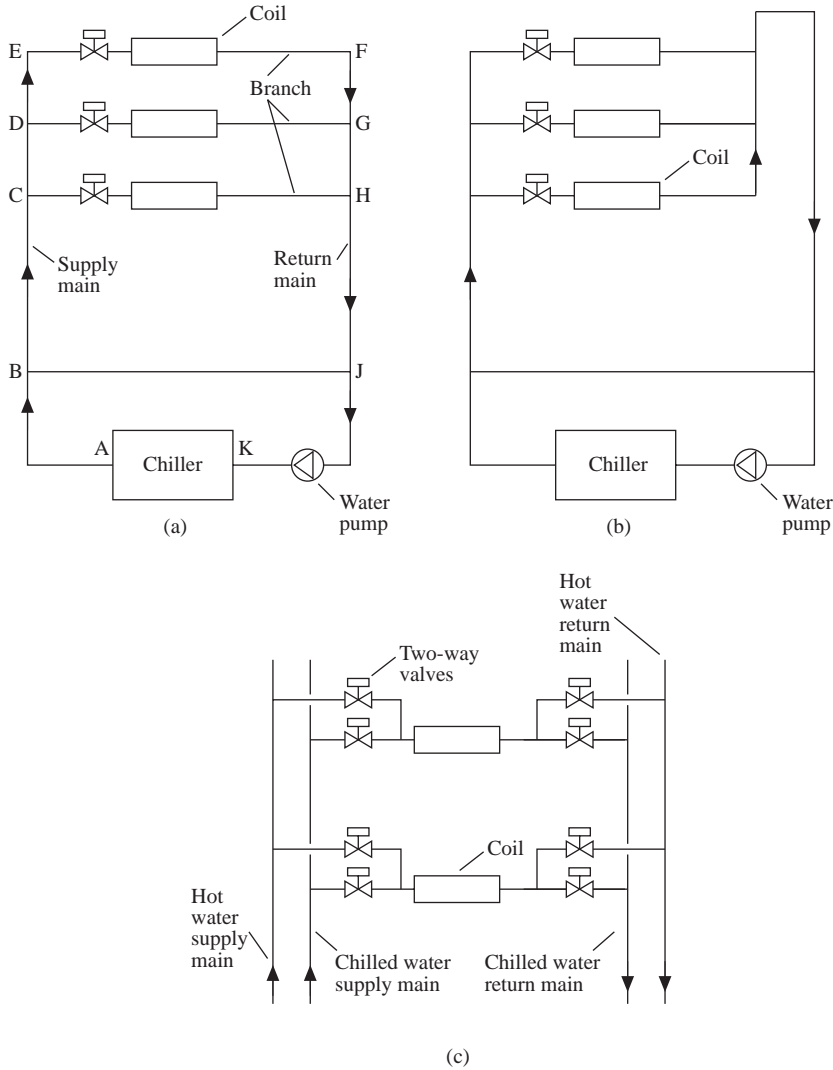


FIGURE 9.9.2 Piping arrangements: (a) two-pipe direct return system, (b) two-pipe reverse system, and (c) four-pipe system.

connect to coils and heat exchangers. Chilled or hot water from the coils and heat exchangers is accumulated by the return main pipe through return branch pipes and then returned to the chiller or boiler.

Constant Flow and Variable Flow. In a constant-flow water system, the volume flow rate at any cross-sectional plane of the supply and return mains remains constant during the entire operating period. In a variable-flow water system, the volume flow rate varies when the system load changes during the operating period.

Direct Return and Reverse Return. In a *direct return* system, the water supplies to and returns from various coils through various piping circuits. ABCHJKA, ... ABCDEFGHJKA are not equal in length, as shown in Figure 9.9.2(a). Water flow must be adjusted and balanced by using balance valves to provide required design flow rates at design conditions. In a *reverse-return* system, as shown in Figure 9.9.2(b), the piping lengths for various piping circuits including the branch and coil are almost equal. Water flow rates to various coils are easier to balance.

Two-Pipe or Four-Pipe. In a dual-temperature water system, the piping from the chiller or boiler to the coils can be either a *two-pipe* system with a supply main and return main as shown in Figure 9.9.2(a) or (b) or a *four-pipe* system with a chilled water supply main, a hot water supply main, a chilled water return main, and a hot water return main as shown in Figure 9.9.2(c). The two-pipe system needs a changeover from chilled to hot water and vice versa. A four-pipe system is more expensive to install.

Plant-Building Loop

System Description

The chillers/boilers in a central plant are often located in the basement, machinery room, or on the rooftop of the building. Generally, a fairly constant-volume water flow is required in the evaporator to protect it from freezing at part load. ASHRAE/IES Standard 90.1-1989 specifies that water systems using control valves to modulate the coil load must be designed for variable flow for energy savings.

Current chilled water systems or dual-temperature water systems often adopt a *plant-building loop* that consists of two piping loops as shown in Figure 9.9.3(a) for reliable and energy-efficient operation:

Plant Loop. A plant loop ABJKA is comprised of chiller(s)/boiler(s); plant chilled/hot water pump(s) usually of split-case, double suction, or end suction type; a diaphragm expansion tank to allow water expansion and contraction during water temperature changes; an air separator to purge dissolved air from water; corresponding piping and fittings; and control systems. The plant loop is operated at constant flow.

Building Loop. A building loop BCDEFGHJB contains coils; building circulating water pumps, often variable-speed pumps, one being a standby pump; corresponding pipes and fittings; and control systems. A differential pressure transmitter is often installed at the farthest end from the building pump between the supply and return mains. The building loop is operated at variable flow.

A short common pipe connects these two loops and combines them into a plant-building loop combination. It is also called a *primary-secondary loop*, with the plant loop the primary loop and the building loop the secondary loop. The location of the pump in a water system should be arranged so that the pressure at any point in the system is greater than atmospheric pressure to prevent air leaking into the system.

Operating Characteristics

In a dual-temperature water system, when the coil load q_{cc} , in Btu/hr, drops, two-way valves close to a smaller opening. The pressure drop across the coils then increases. As soon as the increase of the pressure differential is sensed by the transmitter, a DDC controller reduces the flow rate of the variable-speed pump to match the reduction of the coil load during part load. The pressure differential transmitter functions similarly to a duct static pressure sensor in a VAV system.

Since the plant loop is at constant flow, excess chilled water bypasses the building loop and flows back to the chiller(s) through the common pipe. When the reduction of coil loads in the building loop is equal to or greater than the refrigeration capacity of a single chiller, the DDC controller will shut down a chiller and its associated chilled water pump. The operating characteristics of the hot water in a dual-temperature water system are similar to those in a chilled water system.

Due to the coil's heat transfer characteristics, the reduction of a certain fraction of the design sensible cooling coil load $q_{cs,d}$ does not equal the reduction of the same fraction of design water volume flow $\dot{V}_{w,d}$ (Figure 9.9.3[c]). Roughly, a $0.7 q_{cs,d}$ needs only a $0.45 \dot{V}_{w,d}$, and the temperature difference $\Delta T_w = (T_{wl} - T_{we})$ will be about 150% of that at the design condition. When an equal-percentage contour two-way control valve is used, the valve stem displacement of the cooling coil's two-way valve is approximately linearly proportional to the sensible cooling coil's load change.

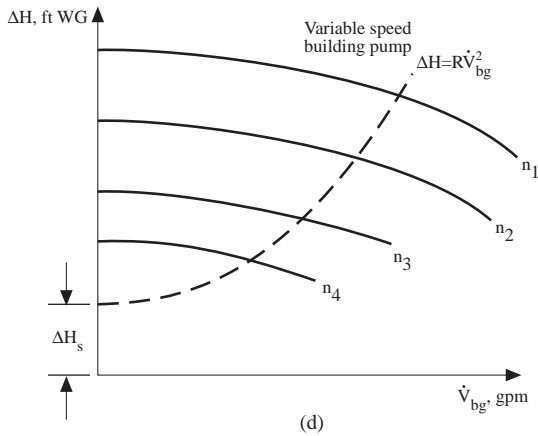
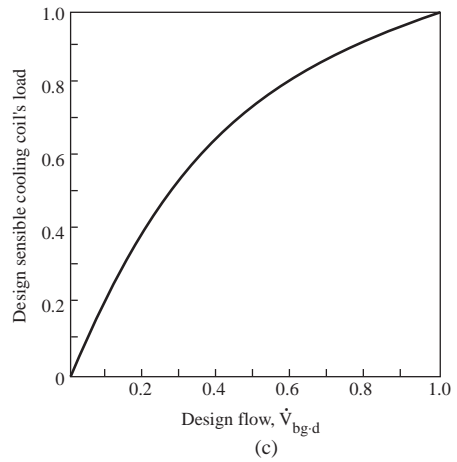
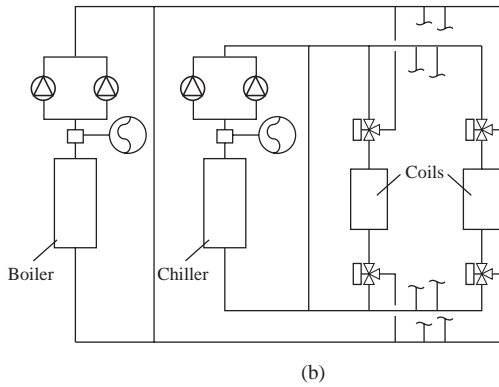
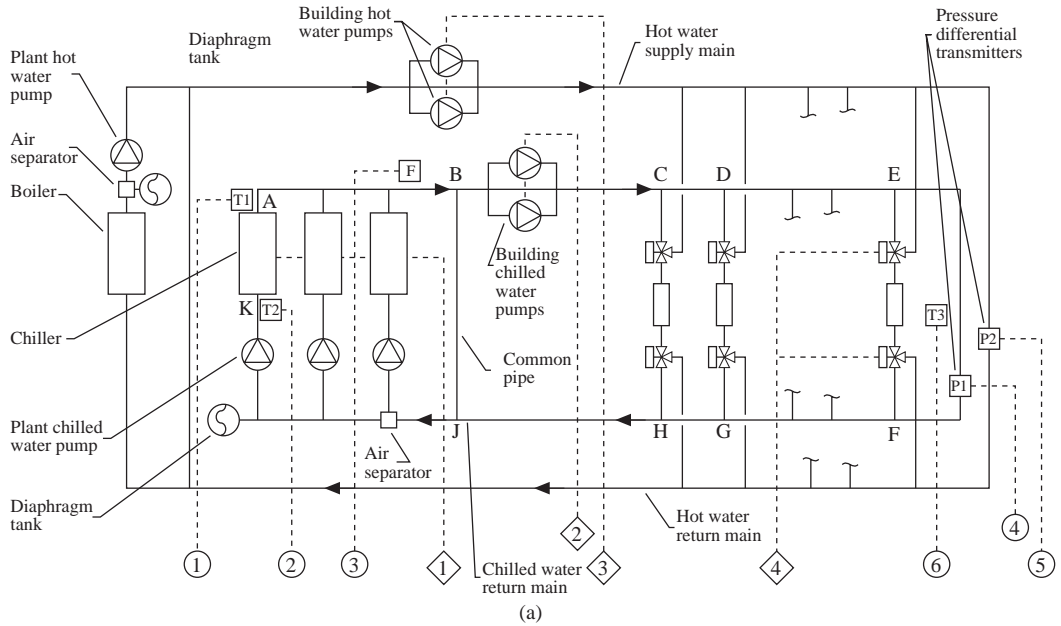


FIGURE 9.9.3 A dual-temperature water system: (a) plant-building loop, (b) single-loop, (c) design sensible load and design volume flow relationship, and (d) variable-speed building pump curves.

Comparison of a Single-Loop and a Plant-Building Loop

Compared with a single-loop water system as shown in [Figure 9.9.3\(b\)](#) that has only a single-stage water pump(s), the plant-building loop:

- Provides variable flow at the building loop and thus saves pump power at reduced flow during part load.
- Separates plant and building loops and makes design, operation, maintenance, and control simpler and more stable as stated in Carlson (1968). The pressure drop across the two-way control valve is considerably reduced as the control valve is only a component of the building loop.

Plant-Distribution-Building Loop

In universities, medical centers, and airports, buildings are often scattered from the central plant. A campus-type chilled water system using a plant-distribution-building loop is often reliable in operation, requires less maintenance, has minimal environmental impact, and sometimes provides energy cost savings.

In a plant-distribution-building loop, plant loop water is combined with many building loops through a distribution loop as shown in [Figure 9.9.4\(a\)](#). The water flow in the plant loop is constant, whereas in both distribution and building loops it is variable. Chilled water often leaves the chiller at a temperature of 40 to 42°F. It is extracted by the distribution pumps and forced into the distribution supply main. At the entrance of each building, chilled water is again extracted by the building pumps and supplied to the cooling coils in AHUs and fan-coil units. From the coils, chilled water is returned through the building return main and the distribution return main and enters the chillers at a temperature around 60°F at design volume flow. System performances of the plant and building loops in a plant-distribution-building loop are similar to those in a plant-building loop.

A campus-type central plant may transport several thousand gallons of chilled water to many buildings, and the farthest building may be several thousand feet away from the central plant. A smaller pressure gradient and end pressure differential between distribution supply and return mains, $\Delta H_{s,d}$, save energy. Meanwhile, a larger diameter of pipes is needed. For a distribution loop, a pressure drop of 0.5 to 1 ft head loss per 100 ft of piping length may be used. A low pressure differential may be offset at the control valves without affecting the normal operation of the coils. Usually, direct return is used for distribution loops.

Using a two-way distribution from the central plant with two separate supply and return loops often reduces the main pipe length and diameter.

A differential pressure transmitter may be located at the farthest end of the distribution mains. As the pressure loss in the building loop is taken care of by the variable-speed building pump(s), a set point of 5 ft WG pressure differential across the supply and return mains seems appropriate.

Chilled water and hot water distribution pipes are often mounted in underground accessible tunnels or trenches. They are well insulated except when chilled water return temperature $T_{ret} > 55^\circ\text{F}$. Under such circumstances, the return main may not be insulated, depending on local conditions and cost analysis.

Many retrofits of campus-type chilled water systems required an existing refrigeration plant(s) and a new developed plant(s) connected to the same plant-distribution-building loop. A stand-alone DDC panel is often used to optimize the turning on and off of the chillers in the existing and developed plants according to the system loads and available sources (chillers).

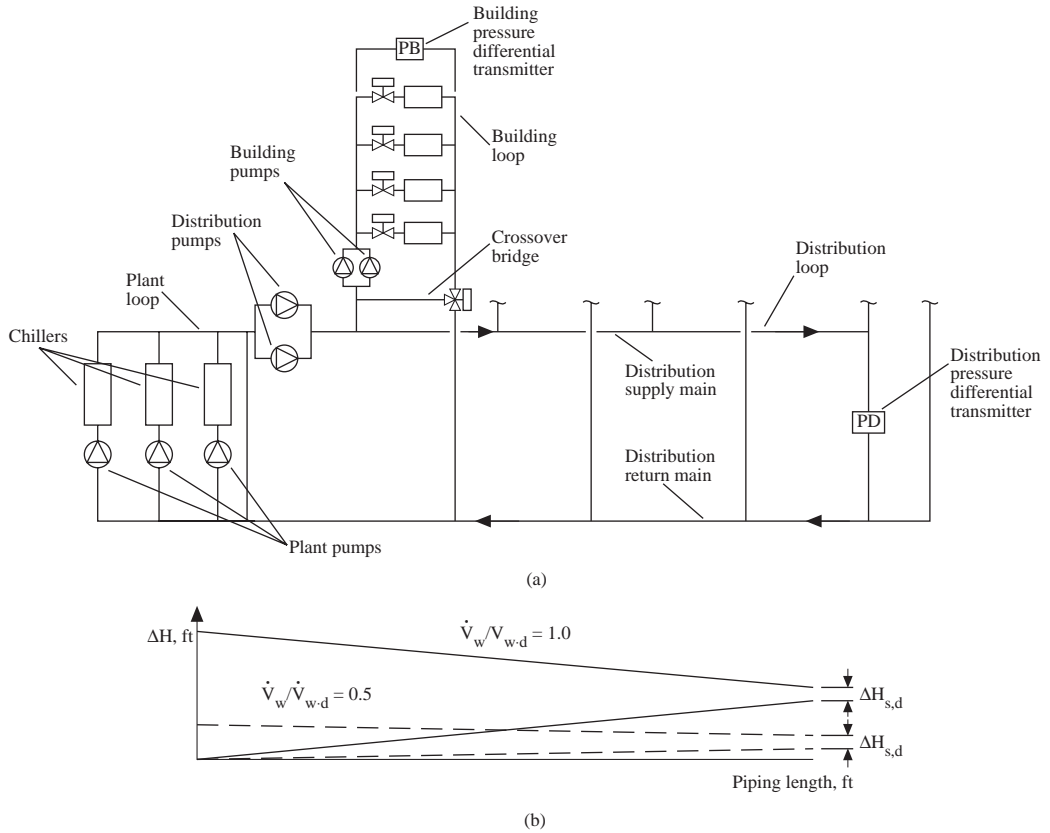


FIGURE 9.9.4 A chilled water system using a plant-distribution-building loop: (a) schematic diagram and (b) pressure gradient for distribution loop.

9.10 Heating Systems

Types of Heating Systems

According to EIA Commercial Buildings Characteristics 1992, for the 57.8 billion ft² of heated commercial buildings in the United States in 1992, the following types of heating systems were used:

Warm air heating systems using warm air furnace	27%
Hot water heating systems using boilers	33%
Heat pumps	13%
District heating	8%
Individual space heaters and others heaters	19%

Modera (1989) reported that nearly 50% of U.S. residential houses are using warm air heating systems with direct-fired warm air furnaces.

Warm Air Furnaces

A *warm air furnace* is a device in which gaseous or liquid fuel is directly fired or electric resistance heaters are used to heat the warm supply air. Natural gas, liquefied petroleum gas (LPG), oil, electric energy, or occasionally wood may be used as the fuel or energy input. Among these, natural gas is most widely used. In a warm air furnace, the warm air flow could be *upflow*, in which the warm air is discharged at the top, as shown in Figure 9.10.1(a) and (b); *downflow*, with the warm air discharged at the bottom; or *horizontal flow*, with the warm air discharged horizontally.

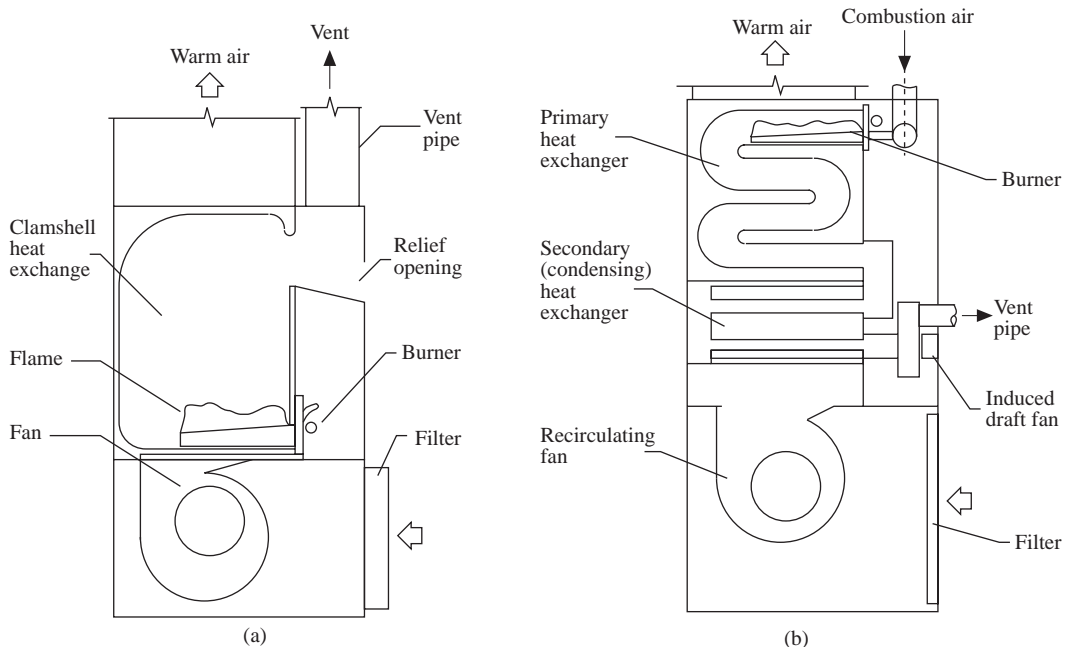


FIGURE 9.10.1 Upflow warm air gas furnace: (a) a natural-vent gas furnace and (b) a power-vent high-efficiency gas furnace.

Natural Vent Combustion Systems. There are two types of combustion systems in a natural gas-fired warm air furnace: natural vent or power vent combustion systems. In a *natural vent* or *atmospheric vent* combustion system, the buoyancy of the combustion products carries the flue gas flowing through the heat exchanger and draft hood, discharging from the chimney or vent pipe. The gas burner is an

atmospheric burner. In an atmospheric burner, air is extracted for combustion by the suction effect of the high-velocity discharged gas and the buoyance effect of the combustion air. An atmospheric burner can be either an in-shot or an up-shot burner or multiple ports. Atmospheric burners are simple, require only a minimal draft of air, and need sufficient gas pressure for normal functioning.

Two types of ignition have been used in burners: standing pilot and spark ignition. In *standing pilot ignition*, the small pilot flame is monitored by a sensor and the gas supply is shut off if the flame is extinguished. *Spark ignition* fires intermittently only when ignition is required. It saves gas fuel if the furnace is not operating.

In a natural vent combustion system, the heat exchanger is often made from cold-rolled steel or aluminized steel in the shape of a clamshell or S. A fan or blower is always used to force the recirculating air flowing over the heat exchanger and distribute the heated air to the conditioned space. A low-efficiency disposable air filter is often located upstream of the fan to remove dust from the recirculating air. A draft hood is also installed to connect the flue gas exit at the top of the heat exchanger to a vent pipe or chimney. A relief air opening is employed to guarantee that the pressure at the flue gas exit is atmospheric and operates safely even if the chimney is blocked. The outer casing of the furnace is generally made of heavy-gauge steel with access panels.

Power Vent Combustion Systems. In a *power vent* combustion system, either a forced draft fan is used to supply the combustion air or an induced draft fan is used to induce the flue gas to the vent pipe or chimney. A power vent is often used for a large gas furnace or a high-efficiency gas furnace with condensing heat exchangers.

Gas burners in a power vent system are called *power burners*. The gas supply to the power burner is controlled by a pressure regulator and a gas valve to control the firing rate. Intermittent spark ignition and *hot surface ignition* that ignites the main burners directly are often used.

Usually, there are two heat exchangers in a power vent combustion system: a primary heat exchanger and a secondary or condensing heat exchanger. The primary heat exchanger constitutes the heating surface of the combustion chamber. When the water vapor in the flue gas is condensed by indirect contact with the recirculating air, part of the latent heat of condensation released is absorbed by the air. Thus the furnace efficiency is increased in the *secondary or condensing heat exchanger*. Both primary and secondary heat exchangers are made from corrosion-resistant steel. A fan is also used to force the recirculating air to flow over the heat exchangers and to distribute the heated air to the conditioned space.

Most natural gas furnaces can use LPG. LPG needs a pressure of 10 in. WG at the manifold, compared with 3 to 4 in. for natural gas. It also needs more primary air for gas burners. Oil furnaces are usually of forced draft and installed with pressure-atomizing burners. The oil pressure and the orifice size of the injection nozzle control the firing rate.

Furnace Performance Indices. The performance of a gas-fired furnace is usually assessed by the following indices:

- Thermal efficiency E_t , in percent, is the ratio of the energy output of heated air or water to the fuel energy input during specific test periods using the same units:

$$E_t = 100(\text{fuel energy output})/(\text{fuel energy input}) \quad (9.10.1)$$

- Annual fuel utilization efficiency (AFUE), in percent, is the ratio of the annual output energy from heated air or water to the annual input energy using the same units:

$$\text{AFUE} = (100 \text{ annual output energy})/(\text{annual input energy}) \quad (9.10.2)$$

- Steady-state efficiency (SSE) is the efficiency of a given furnace according to an ANSI test procedure, in percent:

$$\text{SSE} = 100(\text{fuel input} - \text{fuel loss})/(\text{fuel input}) \quad (9.10.3)$$

Jakob et al. (1986) and Locklin et al. (1987), in a report on ASHRAE Special Project SP43, gave the following performance indices based on a nighttime setback period of 8 hr with a setback temperature of 10°F:

Description	AFUE (%)	SSE (%)
Natural vent		
Pilot ignition	64.5	77
Intermittent ignition	69	77
Intermittent ignition plus vent damper	78	77
Power vent		
Noncondensing	81.5	82.5
Condensing	92.5	93

ASHRAE/IES Standard 90.1-1989 specifies a minimum AFUE of 78% for both gas-fired and oil-fired furnaces of heating capacity <225,000 Btu/hr. For a gas-fired warm air furnace $\geq 225,000$ Btu/hr at maximum rating capacity (steady state) the minimum E_t is 80%, and at minimum rating capacity, E_t is 78%.

Operation and Control. Gas is usually brought from the main to the pressure regulator, where its pressure is reduced to 3.5 in. WG. Gas then flows through a valve and mixes with the necessary amount of outside primary air. The mixture mixes again with the ambient secondary air and is burned. The combustion products flow through the heat exchanger(s) due either to natural draft or power vent by a fan. The flue gas is then vented to the outside atmosphere through a vent pipe or chimney.

Recirculating air is pulled from the conditioned space through the return duct and is mixed with the outdoor air. The mixture is forced through the heat exchanger(s) by a fan and is then heated and distributed to the conditioned space. The fan is often started 1 min after the burner is fired in order to prevent a cold air supply.

At part-load operation, the reduction of the heating capacity of the warm air furnace is usually controlled by the gas valve and the ignition device. For small furnaces, the gas valve is often operated at on-off control. For large furnaces, a two-stage gas valve operates the furnace at 100, 50, and 0% heating capacity as required.

Low NO_x Emissions. NO_x means nitrogen oxides NO and NO₂. They are combustion products in the flue gas and become air pollutants with other emissions like SO₂ and CO when they are discharged to the atmosphere. NO_x cause ozone depletion as well as smog.

Southern California regulations required that NO_x emissions should be 30 ppm or less for gas-fired warm air furnaces and boilers. Many gas burner manufactures use induced flue gas recirculation to cool the burner's flame, a cyclonic-type burner to create a swirling high-velocity flame, and other technologies to reduce NO_x and other emissions while retaining high furnace and boiler efficiencies.

Hot Water Boilers

Types of Hot Water Boilers. A hot water boiler is an enclosed pressure vessel used as a heat source for space heating in which water is heated to a required temperature and pressure without evaporation. Hot water boilers are fabricated according to American Society of Mechanical Engineers (ASME) codes for boilers and pressure vessels. Boilers are generally rated on the basis of their gross output delivered at the boiler's outlet. Hot water boilers are available in standard sizes from 50 to 50,000 MBtu/hr (1 MBtu/hr = 1000 Btu/hr).

According to EIA Characteristics of Commercial Buildings (1991), the percentages of floor area served by different kinds of fuel used in hot water and steam boilers in 1989 in the United States are gas-fired, 69%; oil-fired, 19%; electric, 7%; others, 5%.

Hot water boilers can be classified as *low-pressure boilers*, whose working pressure does not exceed 160 psig and working temperature is 250°F or less, and *medium-* and *high-pressure boilers*, whose working pressure is above 160 psig and working temperature above 250°F. Most of the hot water boilers are low-pressure boilers except those in campus-type or district water heating systems.

Based on their construction and material, hot water boilers can be classified as fire tube boilers, water tube boilers, cast iron sectional boilers, and electric boilers. Water tube boilers are used mainly to generate steam. Cast iron sectional boilers consist of many vertical inverted U-shaped cast iron hollow sections. They are lower in efficiency and used mainly for residential and small commercial buildings. Electric boilers are limited in applications because of their higher energy cost in many locations in the United States.

Scotch Marine Boiler. In a *fire tube* hot water boiler, the combustion chamber and flue gas passages are in tubes. These tubes are enclosed by an outer shell filled with water. A recently developed fire tube model is the modified Scotch Marine packaged boiler, which is a compact, efficient, and popular hot water boiler used today.

A *Scotch Marine* boiler, as shown in [Figure 9.10.2](#), consists of a gas, oil, or gas/oil burner, a mechanical forced-draft fan, a combustion chamber, fire tubes, flue gas vent, outer shell, and corresponding control system. A packaged boiler is a one-piece factory-assembled boiler.

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FIGURE 9.10.2 A Scotch Marine packaged boiler.

Gas or oil and air are measured, mixed, and injected into the combustion chamber in which they are initially ignited by an ignition device. The mixture burns and the combustion process is sustained when there is a high enough temperature in the combustion chamber. Because the number of fire tubes decreases continuously in the second, third, and fourth passes due to the volume contraction at lower flue gas temperatures, the gas velocity and heat transfer coefficients are maintained at reasonably high values.

Return water enters the side of the boiler. It sinks to the bottom, rises again after heating, and then discharges at the top outlet.

The dew point of the flue gas is often 130°F. If the temperature of the return water in a *condensing boiler* is below 125°F, it can be used as a condensing cooling medium to condense the water vapor contained in the flue gas. Latent heat of condensation of water vapor can then be recovered. Corrosion in the condensing heat exchanger and the flue gas passage should be avoided.

The *chimney*, or *stack*, is the vertical pipe or structure used to discharge flue gas, which usually has a temperature rise of 300 to 400°F above the ambient temperature. The chimney or stack should be extended to a certain height above adjacent buildings according to local codes.

Operation and Safety Controls. During part-load operation, reduction of heating capacity is achieved by sensing the temperature of return water and controlling the firing rate of the gas burners in on-off, high-low-off, or modulating modes.

For gas burners, two pressure sensors are often provided to maintain the gas pressure within a narrow range. For modulating controls, the ratio of maximum to minimum input is called the *turn-down* ratio. The minimum input is usually 5 to 25% of the maximum input, that is, a turn-down ratio of 20:1 to 4:1. The boiler should be shut off if the input is less than the minimum.

Pressure and temperature relief valves should be installed in each boiler. An additional high limit control is often equipped to shut down the boiler in case the sensed water pressure or temperature exceeds the predetermined limits. A flame detector is often used to monitor the flame and an airflow sensor to verify the combustion airflow. As soon as the flame is extinguished or the combustion airflow is not sensed, the fuel valve closes and the boiler is shut down. ASHRAE/IES Standard 90.1-1989 specifies that gas-fired boilers should have a minimum AFUE of 80%.

Low-Pressure Warm Air Heating Systems

A *low-pressure warm air heating system* is often equipped with an upflow gas-fired furnace having a furnace heat capacity Q_f to air flow \dot{V}_a ratio, Q_f/\dot{V}_a , of 50 to 70 Btu/hr.cfm and a temperature rise immediately after the furnace of 50 to 70°F. The external pressure loss of the supply and return duct system is usually not higher than 0.5 in. WG. The supply temperature differential ($T_s - T_r$) is often 20 to 35°F. The heating system is often integrated with a cooling system, forming a heating/cooling system.

Low-pressure warm air heating systems usually have a heating capacity not exceeding 100,000 Btu/hr. They are often used in residences and sometimes in small commercial buildings.

System Characteristics. A low-pressure warm air heating system is equipped with either supply and return ducts or a supply duct and a return plenum. Recirculating air is then returned from living, dining, bed, and study rooms to the return plenum through door undercuts in case the doors are closed.

The location of the furnace has a significant effect on the efficiency of the heating system. According to Locklin et al. (1987), if the gas furnace of a low-pressure warm air heating system is installed in a closet and its supply duct is mounted inside the conditioned space, its system efficiency may be 20% higher than that of installations whose furnace and supply duct are in the attic or basement.

When a low-pressure warm air heating system is operating, the supply duct leakage in the attic or basement raises its pressure to a positive value and promotes exfiltration. Return duct leakage extracts the ambient air, lowers the attic or basement pressure to a negative value, and induces infiltration. Gammage et al. (1986) reported that both types of leakage increase the whole house infiltration to 0.78 ach when the low-pressure warm air heating system is operating. The infiltration rate is only 0.44 ach when the low-pressure warm air heating system is shut off.

If the supply temperature differential $\Delta T_s = (T_s - T_r)$ exceeds 30°F, or if there is a high ceiling, thermal stratification may form in the conditioned space. Greater supply volume flow rates and suitable locations of the supply and return outlets may reduce thermal stratification and vertical temperature difference.

Part-Load Operation and Control. For a low-pressure warm air heating system, a space thermostat is often used to control the gas valve of the furnace operated in on-off or high-low-off mode. The proportion of on and off times in an operating cycle can be varied to meet the change of space heating load. The time period of an on-off operating cycle is usually 5 to 15 min.

A warm-air heating system that has an external pressure higher than 0.5 in. WG is often integrated with a cooling system and becomes a part of an air-conditioning system.

Low-Temperature Hot Water Heating System Using Fin-Tube Heaters

In a *low-temperature hot water heating system*, the operating temperature is 250°F or less with a maximum working pressure not exceeding 150 psig, usually less than 30 psig. Low-temperature hot water heating systems are widely used for space heating in residences and commercial buildings.

Fin-Tube Heaters. A *fin-tube heater* is a device installed directly inside the conditioned space to add heat to the space through radiant and convective heat transfer. A fin-tube heater consists of a finned-tube element and an outer casing as shown in Figure 9.10.3(a). The tubes are often made of copper and steel. Copper tubes are generally 0.75, 1, and 1.25 in. in diameter and steel tubes 1.25 and 2 in. in diameter. The fins are usually made of aluminum for copper tubes and of steel for steel tubes. Fin density may vary from 24 to 60 fins per foot. A fin heater may have a maximum length of 12 ft. The outer casing of a finned-tube heater always has a bottom inlet and top outlet for better convection.

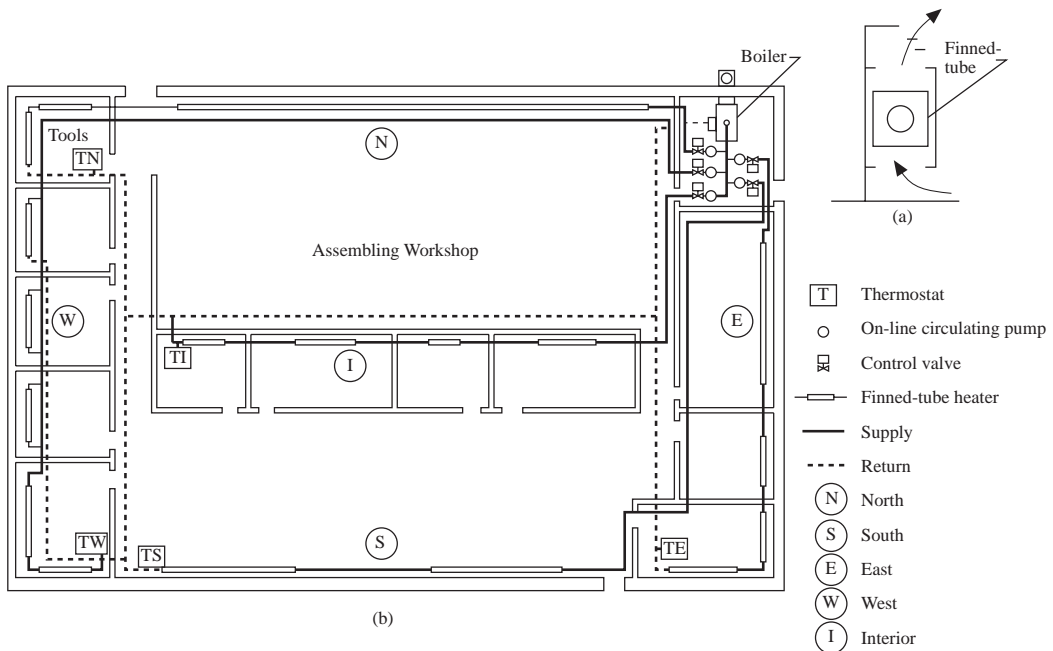


FIGURE 9.10.3 A two-pipe individual-loop low-temperature hot water heating system: (a) finned-tube heater and (b) piping layout.

The most widely used finned-tube heater is the *baseboard heater*, which is often mounted on cold walls at a level 7 to 10 in. from the floor. It is usually 3 in. deep and has one fin-tube row. A wall finned-tube heater has a greater height. A convector has a cabinet-type enclosure and is often installed under the windowsill.

Two-Pipe Individual-Loop Systems. Current low-temperature hot water heating systems using finned-tube heaters are often equipped with zone controls. Zone controls can avoid overheating rooms facing south and underheating rooms facing north because of the effects of solar radiation.

Figure 9.10.3(b) shows the piping layout of a *two-pipe individual-loop system* that is widely used in low-temperature hot water heating systems. Two-pipe means that there are a supply main and a return main pipe instead of one common main for both supply and return. Individual-loop means that there is an individual loop for each control zone. Several finned-tube heaters in a large room can be connected in series, while finned-tube heaters in several small rooms can be connected in reverse return arrangement.

The sizing of low-temperature hot water pipes is usually based on a pressure drop of 1 to 3 ft per 100 ft of pipe length. For a small low-temperature hot water heating system, an open-type expansion tank is often used. A diaphragm tank is often used for a large system. On-line circulating pumps with low head are often used.

Part Load and Control. Usually a hot water sensor located at the exit of the hot water boiler is used to control the firing rate of the boiler at part-load operation. Its set point is usually reset according to the outdoor temperature. Zone control is provided by sensing the return hot water temperature from each individual loop or zone and then varying the water volume flow rate supplied to that zone by modulating the speed of each corresponding on-line circulating pump or its control valve. For hot water heating systems using multiple boilers, on and off for each specific boiler depend not only on the heating demand, but also on minimizing the energy cost.

Infrared Heating

Infrared heating is a process that uses radiant heat transfer from a gas-fired or electrically heated high-temperature device to provide space heating on a localized area for the health and comfort of the occupants or to maintain a suitable indoor environment for a manufacturing process.

An *infrared heater* is a device used to provide infrared heating. Heat radiates from an infrared heater in the form of electromagnetic waves and scatters in all directions. Most infrared heaters have reflectors to focus the radiant beam onto a localized area. Therefore, they are often called *beam radiant heaters*. Infrared heaters are widely used in high-ceiling supermarkets, factories, warehouses, gymnasiums, skating rinks, and outdoor loading docks.

Gas Infrared Heaters. Infrared heaters can be divided into two categories: gas and electric infrared heaters. *Gas infrared heaters* are again divided into porous matrix gas infrared heaters and indirect gas infrared heaters. In a *porous matrix gas infrared heater*, a gas and air mixture is supplied and distributed evenly through a porous ceramic, a stainless steel panel, or a metallic screen, which is exposed to the ambient air and backed by a reflector. Combustion takes place at the exposed surface with a maximum temperature of about 1600°F. An indirect infrared heater consists of a burner, a radiating tube, and a reflector. Combustion takes place inside the radiating tube at a temperature not exceeding 1200°F.

Gas infrared heaters are usually vented and have a small conversion efficiency. Only 10 to 20% of the input energy of an open combustion gas infrared heater is radiated in the form of infrared radiant energy. Usually 4 cfm of combustion air is required for 1000 Btu/hr gas input. A thermostat often controls a gas valve in on-off mode. For standing pilot ignition, a sensor and a controller are used to cut off the gas supply if the flame is extinguished.

Electric Infrared Heaters. An *electric infrared heater* is usually made of nickel-chromium wire or tungsten filaments mounted inside an electrically insulated metal tube or quartz tube with or without inert gas. The heater also contains a reflector that directs the radiant beam to the localized area requiring heating. Nickel-chromium wires often operate at a temperature of 1200 to 1800°F. A thermostat is also used to switch on or cut off the electric current. An electric infrared heater has a far higher conversion efficiency and is cleaner and more easily managed.

Design Considerations. An acceptable radiative temperature increase ($T_{\text{rad}} - T_i$) of 20 to 25°F is often adopted for normal clothed occupants using infrared heating. The corresponding required watt density

for infrared heaters is 30 to 37 W/ft². At a mounting height of 11 ft, two heaters having a spacing of 6.5 ft can provide a watt density of 33 W/ft² and cover an area of 12 × 13 ft. The mounting height of the infrared heaters should not be lower than 10 ft. Otherwise the occupants may feel discomfort from the overhead radiant beam. Refer to Grimm and Rosaler (1990), *Handbook of HVAC Design*, for details.

Gas and electric infrared heaters should not be used in places where there is danger of ignitable gas or materials that may decompose into toxic gases.

9.11 Refrigeration Systems

Classifications of Refrigeration Systems

Most of the refrigeration systems used for air-conditioning are vapor compression systems. Because of the increase in the energy cost of natural gas in the 1980s, the application of absorption refrigeration systems has dropped sharply. According to Commercial Buildings Characteristics 1992, absorption refrigeration systems have a weight of less than 3% of the total amount of refrigeration used in commercial buildings in the United States. Air expansion refrigeration systems are used mainly in aircraft and cryogenics.

Refrigeration systems used for air-conditioning can be classified mainly in the following categories:

- Direct expansion (DX) systems and heat pumps
- Centrifugal chillers
- Screw chillers
- Absorption systems

Each can be either a single-stage or a multistage system.

Direct Expansion Refrigeration Systems

A *direct expansion refrigeration (DX) system*, or simply *DX system*, is part of the packaged air-conditioning system. The DX coil in the packaged unit is used to cool and dehumidify the air directly as shown in [Figure 9.11.1\(a\)](#). According to EIA Commercial Buildings Characteristics 1992, about 74% of the floor space of commercial buildings in the United States was cooled by DX refrigeration systems.

Refrigerants R-22 and R-134a are widely used. Azeotropics and near azeotropics are the refrigerants often used for low-evaporating-temperature systems like those in supermarkets. Because of the limitation of the size of the air system, the refrigeration capacity of DX systems is usually 3 to 100 tons.

Components and Accessories. In addition to the DX coil, a DX refrigeration system has the following components and accessories:

- *Compressor(s)* — Both reciprocating and scroll compressors are widely used in DX systems. Scroll compressors are gradually replacing reciprocating compressors because they have fewer parts and comparatively higher efficiency. For large DX systems, multiple compressors are adopted.
- *Condensers* — Most DX systems in rooftop packaged units are air cooled. Water-cooled condensers are adopted mainly for DX systems in indoor packaged units due to their compact volume. Evaporative-cooled condensers are also available.
- *Refrigeration feed* — Thermostatic expansion valves are widely used as the throttling and refrigerant flow control devices in medium and large DX systems, whereas capillary tubes are used in small and medium-sized systems.
- *Oil lubrication* — R-22 is partly miscible with mineral oil. Since R-134a is not miscible with mineral oil, synthetic polyolester oil should be used. For medium and large reciprocating compressors, an oil pump of vane, gear, or centrifugal type is used to force the lubricating oil to the bearings and moving surfaces via grooves. For small reciprocating compressors, splash lubrication using the rotation of the crankshaft and the connecting rod to splash oil onto the bearing surface and the cylinder walls is used.

A scroll compressor is often equipped with a centrifugal oil pump to force the oil to lubricate the orbiting scroll journal bearing and motor bearing. For the scroll contact surfaces, lubrication is provided by the small amount of oil entrained in the suction vapor.

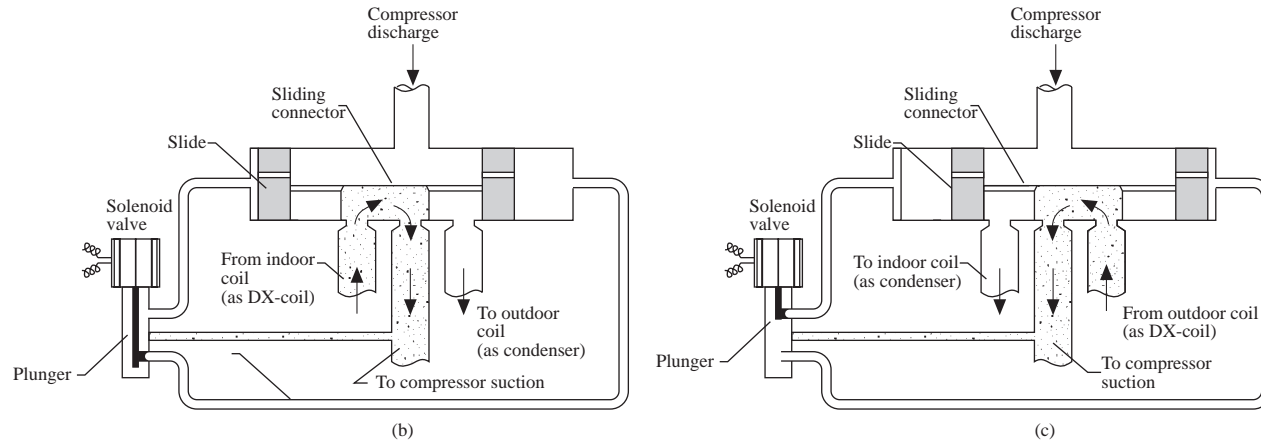
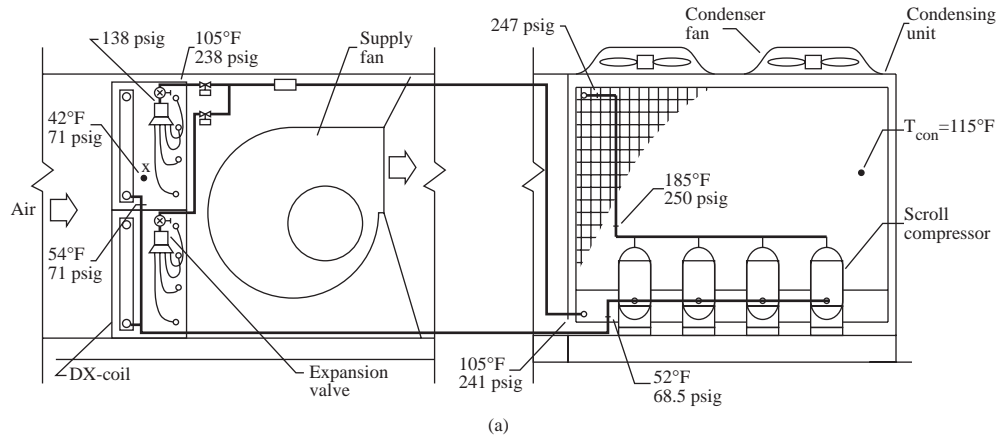


FIGURE 9.11.1 A DX refrigeration system: (a) schematic diagram; (b) four-way reversing valve, cooling mode; and (c) four-way reversing valve, heating mode.

- *Refrigerant piping* — Refrigerant piping transports refrigerant through the compressor, condenser, expansion valve, and DX coil to provide the required refrigeration effect. As shown in Figure 9.11.1(a), from the exit of the DX coil to the inlet of the compressor(s) is the *suction line*. From the outlet of the compressor to the inlet of the air-cooled condenser is the *discharge line*. From the exit of the condenser to the inlet of the expansion valve is the *liquid line*.

Halocarbon refrigerant pipes are mainly made of copper tubes of L type. In a packaged unit, refrigerant pipes are usually sized and connected in the factory. However, the refrigerant pipes in field-built and split DX systems for R-22 are sized on the basis of a pressure drop of 2.91 psi corresponding to a change of saturated temperature ΔT_{suc} of 2°F at 40°F for the suction line and a pressure drop of 3.05 psi corresponding to 1°F at 105°F for the discharge and liquid line. The pressure drop includes pressure losses of pipe and fittings. Refrigerant pipes should also be sized to bring back the entrained oil from the DX coil and condenser through the discharge and suction lines.

Accessories include a filter dryer to remove moisture from the refrigerant, strainer to remove foreign matter, and sight glass to observe the condition of refrigerant flow (whether bubbles are seen because of the presence of flash gas in the liquid line).

Capacity Controls. In DX systems, control of the mass flow rate of refrigerant through the compressor(s) is often used as the primary refrigeration capacity control. Row or intertwined face control at the DX coil is also used in conjunction with the capacity control of the compressor(s).

Three methods of capacity controls are widely used for reciprocating and scroll compressors in DX systems:

- *On-off control* — Starting or stopping the compressor is a kind of step control of the refrigerant flow to the compressor. It is simple and inexpensive, but there is a 100% variation in capacity for DX systems installed with only a single compressor. On-off control is widely used for small systems or DX systems with multiple compressors.
- *Cylinder unloader* — For a reciprocating compressor having multiple cylinders, a cylinder unloader mechanism bypasses the compressed gas from one, two, or three cylinders to the suction chamber to reduce the refrigeration capacity and compressing energy.
- *Speed modulation* — A two-speed motor is often used to drive scroll or reciprocating compressors so that the capacity can be reduced 50% at lower speed.

Safety Controls. In *low- and high-pressure control*, the compressor is stopped when suction pressure drops below a preset value, the cut-in pressure, or the discharge pressure of the hot gas approaches a dangerous level, the cut-out pressure.

In *low-temperature control*, a sensor is mounted on the outer pipe surface of the DX coil. When the surface temperature drops below 32°F, the controller stops the compressor to prevent frosting.

If the pressure of the oil discharged from the pump does not reach a predetermined level within a certain period, a mechanism in *oil pressure failure control* opens the circuit contact and stops the compressor.

In *motor overload control*, a sensor is used to measure the temperature of the electric winding or the electric current to protect the motor from overheating and overloading.

Pump-down control is an effective means of preventing the migration of the refrigerant from the DX coil (evaporator) to the crankcase of the reciprocating compressor. This prevents mixing of refrigerant and oil to form slugs, which may damage the compressor.

When a rise of suction pressure is sensed by a sensor, a DDC controller opens a solenoid valve and the liquid refrigerant enters the DX coil. As the buildup of vapor pressure exceeds the cut-in pressure, the compressor starts. When the DX system needs to shut down, the solenoid valve is closed first; the compressor still pumps the gaseous refrigerant to the condenser. As the suction pressure drops below the cut-in pressure, the compressor stops.

Full- and Part-Load Operations. Consider a DX system in a rooftop packaged unit using four scroll compressors, each with a refrigeration capacity of 10 tons at full load. Performance curves of the condensing unit and the DX coil of this DX system are shown in Figure 9.11.2(a). A DDC controller actuates on–off for these scroll compressors based on the signal from a discharge air temperature T_{dis} sensor.

On a hot summer day, when the rooftop packaged unit starts with a DX coil load or refrigeration capacity, $q_{rc} \approx 40$ tons, all four scroll compressors are operating. The operating point is at A' with a suction temperature T_{suc} of about 42°F, and the discharge air temperature T_{dis} is maintained around 53°F. As the space cooling load q_{ri} as well as T_{dis} decreases, the required DX coil load q_{rl} drops to 35 tons. Less evaporation in the DX coil causes a decrease of T_{suc} to about 40°F, and the operating point may move downward to point A with a DX coil refrigeration capacity of 39 tons. Since $q_{rc} > q_{rl}$, T_{dis} drops continually until it reaches 50°F, point A in Figure 9.11.2(b), and the DDC controller shuts down one of the scroll compressors. The operating point immediately shifts to B' on the three-compressor curve.

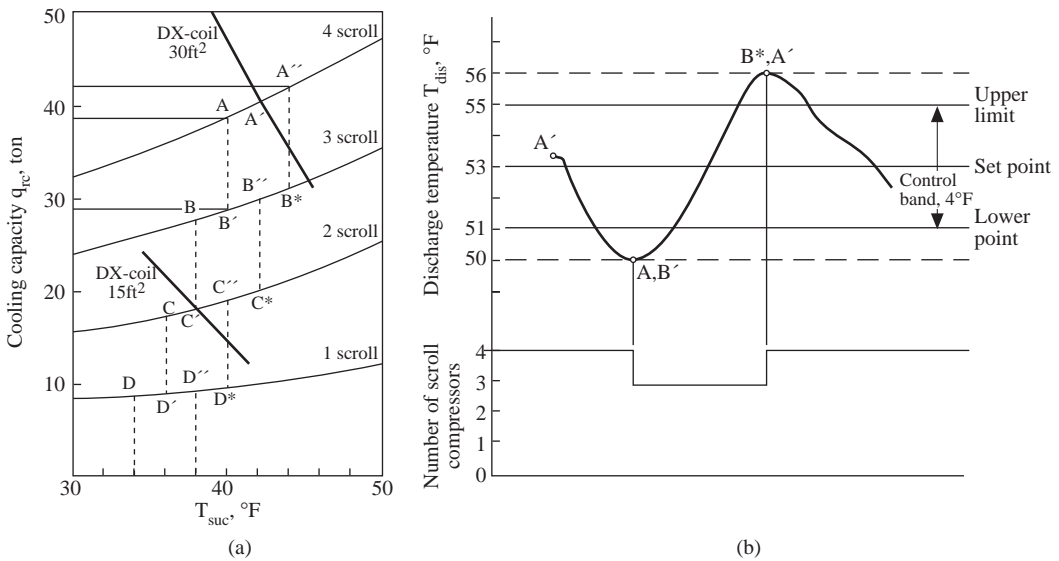


FIGURE 9.11.2 Capacity control of a DX refrigeration system: (a) performance curves and operating points and (b) locus of control point.

Because the refrigeration capacity at point B' q_{rc} is 29 tons, which is less than the required $q_{rl} = 35$ tons, both T_{dis} and T_{suc} rise. When the operating point moves up to B* and T_{dis} reaches 56°F, the DDC controller starts all four scroll compressors at operating point A'' with a refrigeration capacity of 42 tons. Since $q_{rc} > q_{rl}$, the operating point again moves downward along the four-compressor curve and forms an operating cycle A''AB' and B*. The timing of the operating period on four- or three-compressor performance curves balances any required q_{rl} between 29 and 42 tons. Less evaporation at part load in the DX coil results in a greater superheating region and therefore less refrigeration capacity to balance the reduction of refrigeration capacity of the compressor(s) as well as the condensing unit. The condition will be similar when $q_{rl} < 30$ tons, only three- or two-compressor, or two- or one-compressor, or even on–off of one compressor forms an operating cycle.

Main Problems in DX Systems

- *Liquid slugging* is formed by a mixture of liquid refrigerant and oil. It is formed because of the flooding back of liquid refrigerant from the DX coil to the crankcase of the reciprocating compressor due to insufficient superheating. It also may be caused by migration of liquid refrig-

erant from the warmer indoor DX coil to the colder outdoor compressor during the shut-down period in a split packaged unit. Liquid slugging dilutes the lubricating oil and causes serious loss of oil in the crankcase of the reciprocating compressor. Liquid slugging is incompressible. When it enters the compression chamber of a reciprocating compressor, it may damage the valve, piston, and other components. Pump-down control and installation of a crankcase heater are effective means of preventing liquid refrigerant migration and flooding back.

- *Compressor short cycling* — For on–off control, too short a cycle, such as less than 3 min, may pump oil away from the compressor or damage system components. It is due mainly to a too close low-pressure control differential or to reduced air flow.
- *Defrosting* — If the surface of a DX coil is 32°F or lower, frost accumulates on it. Frost blocks the air passage and reduces the rate of heat transfer. It should be removed periodically. The process of removing frost is called defrosting.

Air at a temperature above 36°F, hot gas inside the refrigerant tubes and an installed electric heating element can be used for defrosting. The defrosting cycle is often controlled by sensing the temperature or pressure difference of air entering the DX coil during a fixed time interval.

- *Refrigerant charge* — Insufficient refrigerant charge causes lower refrigeration capacity, lower suction temperature, and short on–off cycles. Overcharging refrigerant may cause a higher condensing pressure because part of the condensing surface becomes flooded with liquid refrigerant.

Heat Pumps

A *heat pump* in the form of a packaged unit is also a *heat pump system*. A heat pump can either extract heat from a heat source and reject heat to air and water at a higher temperature for heating, or provide refrigeration at a lower temperature and reject condensing heat at a higher temperature for cooling. During summer, the heat extraction, or refrigeration effect, is the useful effect for cooling in a heat pump. In winter, the rejected heat and the heat from a supplementary heater provide heating in a heat pump.

There are three types of heat pumps: air-source, water-source, and ground-coupled heat pumps. Ground-coupled heat pumps have limited applications. Water-source heat pump systems are covered in detail in a later section.

Air-Source Heat Pump. An *air-source heat pump*, or *air-to-air heat pump*, is a DX system with an additional four-way reversing valve to change the refrigerant flow from cooling mode in summer to heating mode in winter and vice versa. The variation in connections between four means of refrigerant flow — compressor suction, compressor discharge, DX coil exit, and condenser inlet — causes the function of the indoor and outdoor coils to reverse. In an air-source heat pump, the coil used to cool or to heat the recirculating/outdoor air is called the *indoor coil*. The coil used to reject heat to or absorb heat from the outside atmosphere is called the *outdoor coil*. A short capillary or restrict tube is often used instead of a thermostatic expansion valve. Both reciprocating and scroll compressors are used in air-source heat pumps. R-22 is the refrigerant widely used. Currently available air-source heat pumps usually have a cooling capacity of 1½ to 40 tons.

Cooling and Heating Mode Operation. In *cooling mode operation*, as shown in Figure 9.11.1(b), the solenoid valve is deenergized and drops downward. The high-pressure hot gas pushes the sliding connector to the left end. The compressor discharge connects to the outdoor coil, and the indoor coil connects to the compressor inlet.

In *heating mode operation*, as shown in Figure 9.11.1(c), the solenoid plunger moves upward and pushes the slide connector to the right-hand side. The compressor discharge connects to the indoor coil, and the outdoor coil exit connects to the compressor suction.

System Performance. The performance of an air-source heat pump depends on the outdoor air temperature T_o , in °F as well as the required space heating load q_{th} . During cooling mode operation, both the refrigeration capacity q_{rc} , in Btu/hr, and EER for the heat pump EER_{hp} , in Btu/hr.W, increase as T_o drops.

During heating mode operation, the heating capacity q_{hp} , in Btu/hr, and COP_{hp} decrease, and q_{rh} increases as the T_o drops. When $q_{rh} > q_{hp}$, supplementary heating is required. If COP_{hp} drops below 1, electric heating may be more economical than a heat pump.

If on–off is used for compressor capacity control for an air-source heat pump in a split packaged unit, refrigerant tends to migrate from the warmer outdoor coil to the cooler indoor coil in summer and from the warmer indoor coil to the cooler outdoor coil in winter during the off period. When the compressor starts again, 2 to 5 min of reduced capacity is experienced before the heat pump can be operated at full capacity. Such a loss is called a *cycling loss*.

In winter, most air-source heat pumps switch from the heating mode to cooling mode operation and force the hot gas to the outdoor coil to melt frost. After the frost is melted, the heat pump is switched back to heating mode operation. During defrosting, supplementary electric heating is often necessary to prevent a cold air supply from the air-source heat pump.

Minimum Performance. ASHRAE/IES Standard 90.1-1989 specifies a minimum performance for air-cooled DX systems in packaged units as covered in Section 9.7. For air-cooled, electrically operated rooftop heat pumps (air-source heat pumps), minimum performance characteristics are:

Cooling EER	8.9
Heating, COP (at $T_oF = 47°F$)	3.0
Heating, COP (at $T_oF = 17°F$)	2.0

Centrifugal Chillers

A *chiller* is a refrigeration machine using a liquid cooler as an evaporator to produce chilled water as the cooling medium in a central air-conditioning system. A *centrifugal chiller*, as shown in Figure 9.11.3(a), is a refrigeration machine using a centrifugal compressor to produce chilled water. It is often a factory-assembled unit with an integrated DDC control system and sometimes may separate into pieces for transportation. A centrifugal chiller is also a *centrifugal vapor compression refrigeration system*.

Refrigerants. According to Hummel et al. (1991), in 1988 there were about 73,000 centrifugal chillers in the United States. Of these, 80% use R-11, 10% use R-12, and the remaining 10% use R-22 and others. As mentioned in Section 9.4, production of CFCs, including R-11 and R-12, ceased at the end of 1995 with limited exception for service. R-123 (HCFC) will replace R-11. The chiller's efficiency may drop 2 to 4%, and a capacity reduction of 5% is possible. R-123 has low toxicity. Its allowable exposure limit was raised to 50 ppm in 1997 from 30 ppm in 1993 by its manufacturers. A monitor and alarm device to detect R-123 in air should be installed in plant rooms and places where there may be refrigerant leaks.

R-134a (HFC) will replace R-12. According to Lowe and Ares (1995), as a result of the changeout from R-12 to R-134a for a 5000-hp centrifugal chiller in Sears Tower, Chicago, its speed increased from 4878 to 5300 rpm, its cooling capacity is 12 to 24% less, and its efficiency is 12 to 16% worse.

System Components. A centrifugal chiller consists of a centrifugal compressor, an evaporator or liquid cooler, a condenser, a flash cooler, throttling devices, piping connections, and controls. A purge unit is optional.

- Centrifugal compressor — According to the number of internally connected impellers, the centrifugal compressor could have a single, two, or more than two stages. A two-stage impeller with a flash cooler is most widely used because of its higher system performance and comparatively simple construction. Centrifugal compressors having a refrigeration capacity less than 1200 tons are often hermetic. Very large centrifugal compressors are of open type. A gear train is often required to raise the speed of the impeller except for very large impellers using direct drive.
- Evaporator — Usually a liquid cooler of flooded shell-and-tube type evaporator is adopted because of its compact size and high rate of heat transfer.
- Condenser — Water-cooled, horizontal shell-and-tube condensers are widely used.

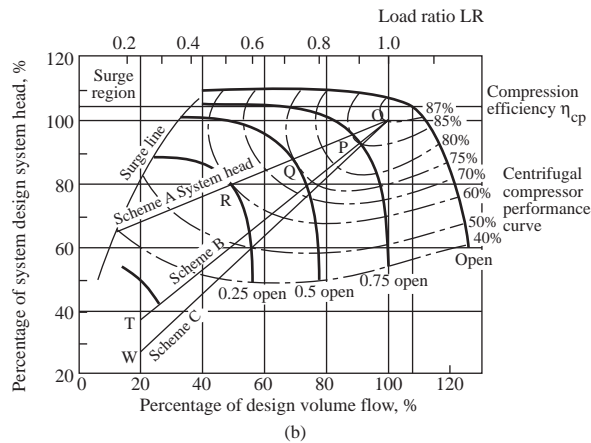
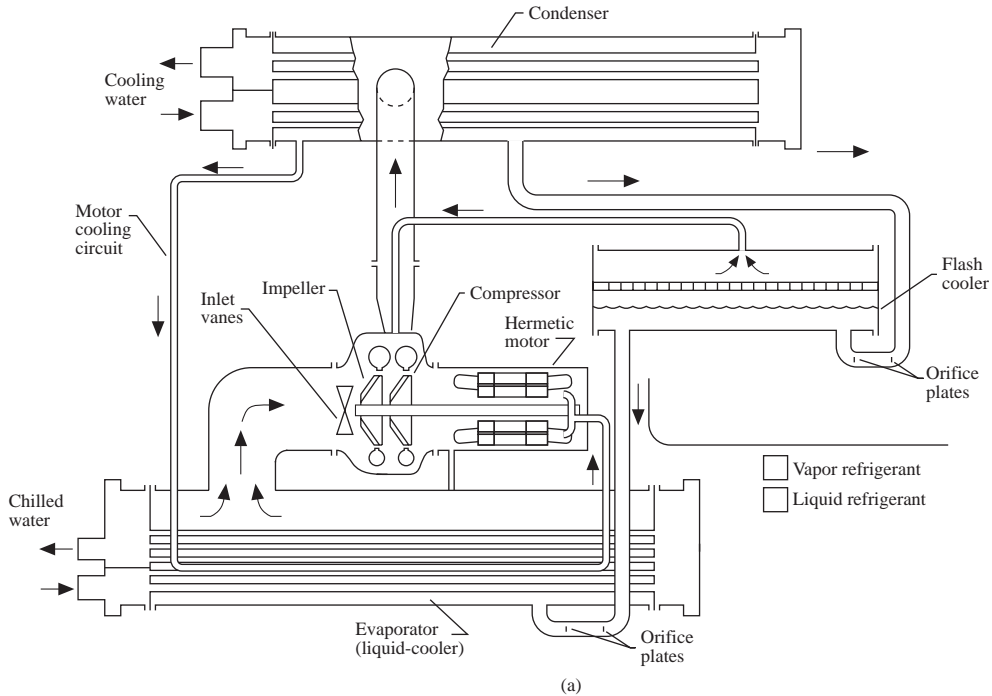


FIGURE 9.11.3 A two-stage water-cooled centrifugal chiller: (a) schematic diagram and (b) centrifugal compressor performance map at constant speed.

- Flash cooler — For a two-stage centrifugal compressor, a single-stage flash cooler is used. For a three-stage compressor, a two-stage flash cooler is used.
- Orifice plates and float valves — Both multiple-orifice plates such as that shown in Figure 9.11.3(a) and float valves are used as throttling devices in centrifugal chillers.
- Purge unit — R-123 has an evaporating pressure $p_{ev} = 5.8$ psia at 40°F, which is lower than atmospheric pressure. Air and other noncondensable gases may leak into the evaporator through cracks and gaps and usually accumulate in the upper part of the condenser. These noncondensable gases raise the condensing pressure, reduce the refrigerant flow, and lower the rate of heat transfer. A purge unit uses a cooling coil to separate the refrigerant and water from the noncondensable gases and purge the gases by using a vacuum pump.

Performance Ratings. The refrigeration cycle of a typical water-cooled, two-stage centrifugal chiller with a flash cooler was covered in Section 9.4. Centrifugal chillers have the same refrigeration capacity as centrifugal compressors, 100 to 10,000 tons. According to ARI Standard 550-88, the refrigeration capacity of a centrifugal chiller is rated as follows:

Chilled water temperature leaving evaporator T_{el} :	100% load 44°F 0% load 44°F
Chilled water flow rate:	2.4 gpm/ton
Condenser water temperature entering condenser T_{ce} :	100% load 85°F 0% load 60°F
Condenser water flow rate:	3.0 gpm/ton
Fouling factor in evaporator and condenser:	0.00025 hr.ft ² .°F/Btu

The integrated part-load value (IPLV) of a centrifugal chiller or other chillers at standard rating conditions can be calculated as:

$$\text{IPLV} = 0.1(A + B)/2 + 0.5(B + C)/2 + 0.3(C + D)/2 + 0.1D \quad (9.11.1)$$

where A , B , C , and D = kW/ton or COP at 100, 75, 50, and 25% load, respectively. If the operating conditions are different from the standard rating conditions, when T_{el} is 40 to 50°F, for each °F increase or decrease of T_{el} , there is roughly a 1.5% difference in refrigeration capacity and energy use; when T_{ce} is between 80 to 90°F, for each °F of increase or decrease of T_{ce} , there is roughly a 1% increase or decrease in refrigeration capacity and 0.6% in energy use.

ASHRAE/IES Standard 90.1-1989 and ARI Standard 550-88 specify the minimum performance for water-cooled water chillers from January 1, 1992:

	COP	IPLV
≥300 tons	5.2	5.3
≥150 tons < 300 tons	4.2	4.5
<150 tons	3.8	3.9

COP = 5.0 is equivalent to about 0.70 kW/ton. New, installed centrifugal chillers often have an energy consumption of 0.50 kW/ton.

Air-cooled centrifugal chillers have COPs from 2.5 to 2.8. Their energy performance is far poorer than that of water-cooled chillers. Their application is limited to locations where city water is not allowed to be used as makeup water for cooling towers.

Capacity Control. The refrigeration capacity of a centrifugal chiller is controlled by modulating the refrigerant flow at the centrifugal compressor. There are mainly two types of capacity controls: varying the opening and angle of the inlet vanes, and using an adjustable-frequency AC inverter to vary the rotating speed of the centrifugal compressor.

When the opening of the inlet vanes has been reduced, the refrigerant flow is throttled and imparted with a rotation. The result is a new performance curve at lower head and flow. If the rotating speed of a centrifugal compressor is reduced, it also has a new performance curve at lower volume flow and head. Inlet vanes are inexpensive, whereas the AC inverter speed modulation is more energy efficient at part-load operation.

Centrifugal Compressor Performance Map. Figure 9.11.3(b) shows a single-stage, water-cooled centrifugal compressor performance map for constant speed using inlet vanes for capacity modulation. A performance map consists of the compressor's performance curves at various operating conditions. The performance curve of a centrifugal compressor shows the relationship of volume flow of refrigerant \dot{V}_r

and its head lift Δp or compression efficiency η_{cp} at that volume flow. It is assumed that η_{cp} for a two-stage compressor is equal to the average of the two single-stage impellers having a head of their sum.

On the map, the required *system head curve* indicates the required system head lift at that volume flow of refrigerant. The intersection of the performance curve and the required system head curve is called the *operating point* O, P, Q, R, ... as shown in Figure 9.11.3(b). One of the important operating characteristics of a centrifugal chiller (a centrifugal vapor compression refrigeration system as well) is that the required system head lift is mainly determined according to the difference in condensing and evaporating pressure $\Delta p_{c-e} = (p_{con} - p_{ev})$. The pressure losses due to the refrigerant piping, fittings, and components are minor.

In Figure 9.11.3(b), the abscissa is the percentage of design volume flow of refrigerant, % \dot{V}_r , or load ratio; the ordinate is the percentage of design system head $\Delta H_{s,d}$, or percentage of design temperature lift $(T_{con} - T_{ev})$. Here load ratio LR is defined as the ratio of the refrigeration load to the design refrigeration load $q_r/q_{r,d}$. There are three schemes of required system head curves:

- Scheme A — $T_{ce} = \text{constant}$ and $T_{el} = \text{constant}$
- Scheme B — $T_{el} = \text{constant}$ and a drop of 2.5°F of T_{ce} for each 0.1 reduction of load ratio
- Scheme C — A reset of T_{el} of 1°F increase and a drop of 2.5°F of T_{ce} for each 0.1 reduction of load ratio

At design \dot{V}_r and system head $H_{s,d}$, $\eta_{cp} = 0.87$. As \dot{V}_r , load ratio, and required system head Δp decrease, η_{cp} drops accordingly.

Surge is a unstable operation of a centrifugal compressor or fan resulting in vibration and noise. In a centrifugal chiller, surge occurs when the centrifugal compressor is unable to develop a discharge pressure that satisfies the requirement at the condenser. A centrifugal compressor should never be operated in the surge region.

Part-Load Operation. During part-load operation, if T_{el} and T_{ce} remain constant, the evaporating temperature T_{ev} tends to increase from the design load value because there is a greater evaporating surface and a smaller temperature difference $(T_{el} - T_{ev})$. Similarly, T_{con} tends to decrease.

The ratio of actual compressor power input at part load to the power input at design load may be slightly higher or lower than the load ratios, depending on whether the outdoor wet bulb is constant or varying at part load or whether there is a T_{el} reset; it also depends on the degree of drop of η_{cp} at part load.

Specific Controls. In addition to generic controls, specific controls for a centrifugal chiller include:

- Chilled water leaving temperature T_{el} and reset
- Condenser water temperature T_{ce} control
- On and off of multiple chillers based on actual measured coil load
- Air purge control
- Safety controls like oil pressure, low-temperature freezing protection, high condensing pressure control, motor overheating, and time delaying

Centrifugal Chillers Incorporating Heat Recovery. A HVAC&R heat recovery system converts waste heat or waste cooling from any HVAC&R process into useful heat and cooling. A heat recovery system is often subordinate to a parent system, such as a heat recovery system to a centrifugal chiller.

A centrifugal chiller incorporating a heat recovery system often uses a double-bundle condenser in which water tubes are classified as tower bundles and heating bundles. Heat rejected in the condenser may be either discharged to the atmosphere through the tower bundle and cooling tower or used for heating through the heating bundle. A temperature sensor is installed to sense the temperature of return hot water from the heating coils in the perimeter zone. A DDC controller is used to modulate a bypass three-way valve which determines the amount of condenser water supplied to the heating bundle. The tower and heating bundles may be enclosed in the same shell, but baffle sheets are required to guide the water flows.

A centrifugal chiller incorporating a heat recovery system provides cooling for the interior zone and heating for the perimeter zone simultaneously in winter with a higher COP_{hr} . However, it needs a higher condenser water-leaving temperature T_{ci} of 105 to 110°F, compared with 95°F or even lower in a cooling-only centrifugal chiller. An increase of 10 to 12°F of the difference ($T_{con} - T_{ev}$) requires an additional 10 to 15% power input to the compressor. For a refrigeration plant equipped with multiple chillers, it is more energy efficient and lower in first cost to have only part of them equipped with double-bundle condensers.

Screw Chillers

A *screw chiller* or a *helical rotary chiller* is a refrigeration machine using a screw compressor to produce chilled water. A factory-fabricated and assembled screw chiller itself is also a screw vapor compression refrigeration system.

Twin-screw chillers are more widely used than single-screw chillers. A twin-screw chiller consists of mainly a twin-screw compressor, a flooded shell-and-tube liquid cooler as evaporator, a water-cooled condenser, throttling devices, an oil separator, an oil cooler, piping, and controls as shown in [Figure 9.11.4\(a\)](#). The construction of twin-screw compressors has already been covered. For evaporator, condenser, and throttling devices, they are similar to those in centrifugal chillers. Most twin-screw chillers have a refrigeration capacity of 100 to 1000 tons.

Following are the systems characteristics of screw chillers.

Variable Volume Ratio. The ratio of vapor refrigerant trapped within the interlobe space during the intake process V_{in} to the volume of trapped hot gas discharged V_{dis} is called the *built-in volume ratio* of the twin-screw compressor $V_1 = V_{in}/V_{dis}$, or simply *volume ratio*, all in ft^3 .

There are two types of twin-screw chiller: fixed and variable volume ratio. For a twin-screw chiller of *fixed volume ratio*, the isentropic efficiency η_{isen} becomes maximum when the system required compression ratio $R_{s,com} \approx V_1$. Here $R_{s,com} = p_{con}/p_{ev}$. If $p_{dis} > p_{con}$, overcompression occurs, as shown in [Figure 9.11.4\(b\)](#). The discharged hot gas reexpands to match the condensing pressure. If $p_{dis} < p_{con}$, undercompression occurs ([Figure 9.11.4\(c\)](#)). A volume of gas at condensing pressure reenters the trapped volume at the beginning of the discharge process. Both over- and undercompression cause a reduction of η_{isen} .

For a twin-screw chiller of *variable volume ratio*, there are two slides: a sliding valve is used for capacity control and a second slide. By moving the second slide back and forth, the radial discharge port can be relocated. This allows variation of suction and discharge pressure levels and still maintains maximum efficiency.

Economizer. The hermetic motor shell is connected to an intermediate point of the compression process and maintains an intermediate pressure p_i between p_{con} and p_{ev} . Liquid refrigerant at condensing pressure p_{con} is throttled to p_i , and a portion of the liquid is flashed into vapor. This causes a drop in the temperature of the remaining liquid refrigerant down to the saturated temperature corresponding to p_i . Although the compression in a twin-screw compressor is in continuous progression, the mixing of flashed gas with the compressed gas at the intermediate point actually divides the compression process into two stages. The resulting economizing effect is similar to that of a two-stage compound refrigeration system with a flash cooler: an increase of the refrigeration effect and a saving of the compression power from $(p_{con} - p_{ev})$ to $(p_{con} - p_i)$.

Oil Separation, Oil Cooling, and Oil Injection. Oil entrained in the discharged hot gas enters an oil separator. In the separator, oil impinges on an internal perforated surface and is collected because of its inertia. Oil drops to an oil sump through perforation. It is then cooled by condenser water in a heat exchanger. A heater is often used to vaporize the liquid refrigerant in the oil sump to prevent dilution of the oil. Since the oil sump is on the high-pressure side of the refrigeration system, oil is forced to the rotor bearings and injected to the rotors for lubrication.

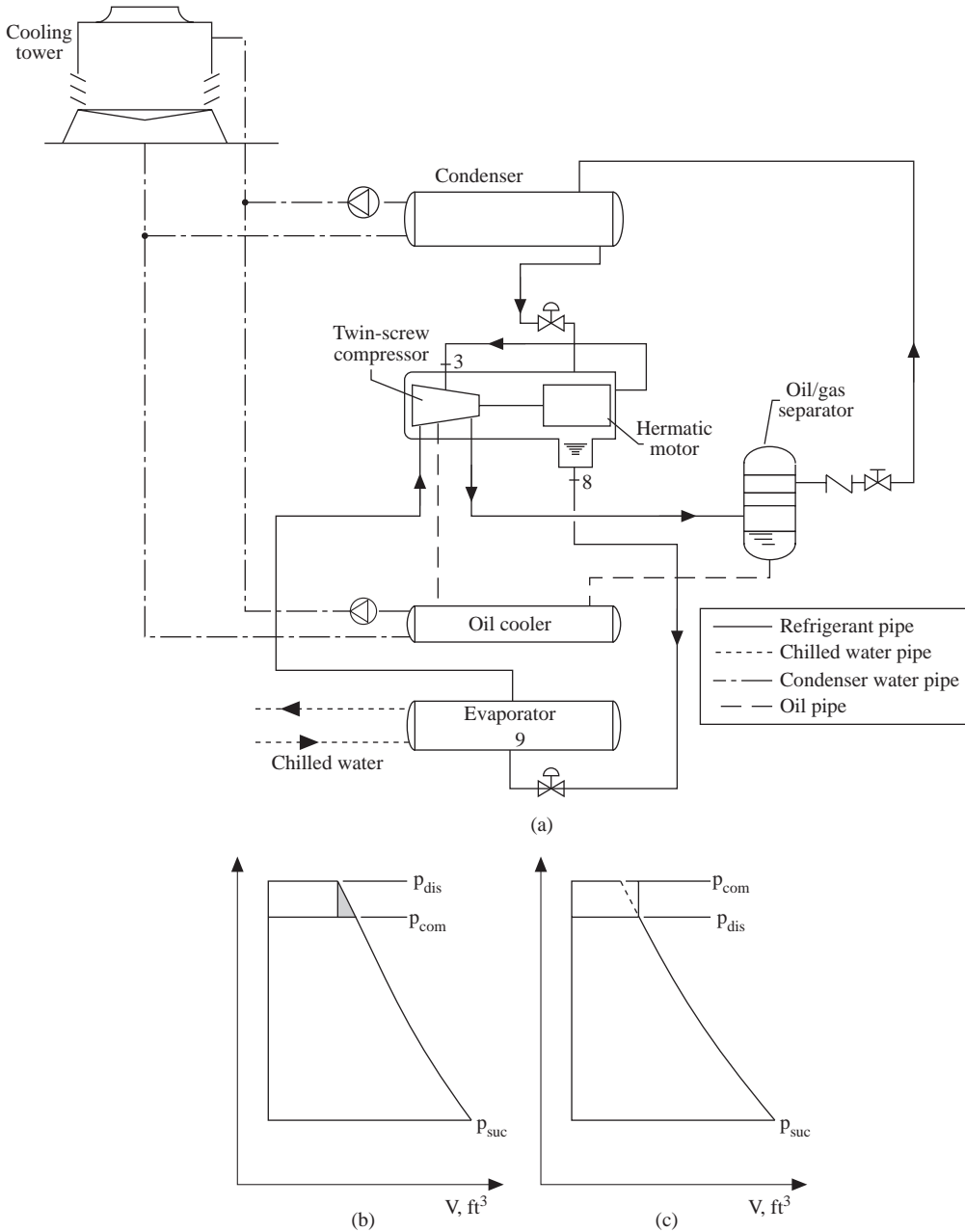


FIGURE 9.11.4 A typical twin-screw chiller: (a) schematic diagram, (b) over-compression, and (c) under-compression.

Oil slugging is not a problem for twin-screw compressors. When suction vapor contains a small amount of liquid refrigerant that carries over from the oil separator, often called *wet suction*, it often has the benefit of scavenging the oil from the evaporator.

Twin-screw compressors are positive displacement compressors. They are critical in oil lubrication, sealing, and cooling. They are also more energy efficient than reciprocating compressors. Twin-screw chillers are gaining more applications, especially for ice-storage systems with cold air distribution.

9.12 Thermal Storage Systems

Thermal Storage Systems and Off-Peak Air-Conditioning Systems

Many electric utilities in the United States have their *on-peak hours* between noon and 8 p.m. during summer weekdays, which include the peak-load hours of air-conditioning. Because the capital cost of a new power plant is so high, from \$1200 to \$4000 per kW, electric utilities tend to increase their power output by using customers' thermal energy storage (TES) systems, or simply thermal storage systems, which are much less expensive.

A *thermal storage system* as shown in Figure 9.12.1(a) may have the same refrigeration equipment, like chillers, additional storage tank(s), additional piping, pumps, and controls. The electric-driven compressors are operated during off-peak, partial-peak, and on-peak hours. *Off-peak hours* are often nighttime hours. *Partial-peak hours* are hours between on-peak and off-peak hours in a weekday's 24-hr day-and-night cycle. Chilled water or ice is stored in tanks to provide cooling for buildings during on-peak hours when higher electric demand and electric rates are effective. Although thermal storage systems operate during nighttime when outdoor dry and wet bulbs are lower, they are not necessarily energy saving due to lower evaporating temperature, additional pump power, and energy losses. Thermal storage systems significantly reduce the electric energy cost.

Utilities in the United States often use higher electric demand and rates as well as incentive bonus to encourage the shift of electric load from on-peak to off-peak hours by using thermal storage systems and others. Such a shift not only saves expensive capital cost, but also increases the use of base-load high-efficiency coal and nuclear plants instead of inefficient diesel and gas turbine plants.

The air-conditioning systems that operate during off-peak and partial-peak hours for thermal storage, or those that use mainly natural gas to provide cooling to avoid higher electric demand and rates during on-peak hours, are called *off-peak air-conditioning systems*. These systems include ice-storage and chilled-water storage systems, desiccant cooling systems, absorption systems, and gas engine chillers.

Absorption chillers and desiccant cooling systems are covered in other sections. Gas engine-driven reciprocating chillers are often a cogeneration plant with heat recovery from engine jacket and exhaust gas, and will not be covered here.

Full Storage and Partial Storage

Ice and chilled-water storage systems are the two most common thermal energy storage systems today. Knebel (1995) estimated that more than 4000 cool storage systems are operated in various commercial buildings.

The unit of stored thermal energy for cooling is ton-hour, or ton.hr. One *ton.hr* is the refrigeration capacity of one refrigeration ton during a 1-hr period, or 12,000 Btu.

In order to achieve minimum life-cycle cost, thermal storage systems could be either full storage or partial storage. For a *full-storage*, or *load shift*, thermal storage system, all refrigeration compressors cease to operate during on-peak hours. Building refrigeration loads are entirely offset by the chilled water or brine from the thermal storage system within an on-peak period. In a *partial storage*, or *load-leveling*, thermal storage system as shown in Figure 9.12.1(b) all or some refrigeration compressor(s) are operated during on-peak hours.

Direct cooling is the process in which refrigeration compressors produce refrigeration to cool the building directly. During a specific time interval, if the cost of direct cooling is lower than the stored energy, the operation of a thermal storage system is said to be in *chiller priority*. On the contrary, if the cost of direct cooling is higher than the cost of stored energy, the operation is said to be at *storage priority*.

The optimum size of a thermal storage system is mainly determined according to the utility's electric rate structure, especially a time-of-day structure whose electric rates are different between on-peak, partial-peak, and off-peak hours. Not only the design day's instantaneous building peak cooling load is important, but also an hour-by-hour cooling load profile of the design day is required for thermal storage design. A simple payback or a life-cycle cost analysis is usually necessary.

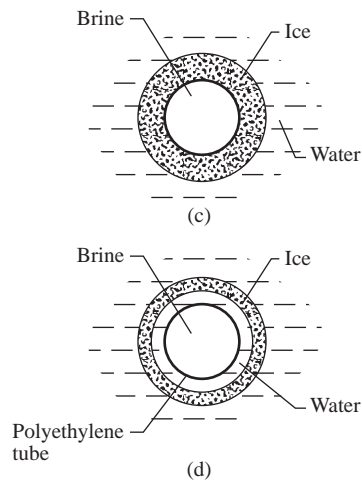
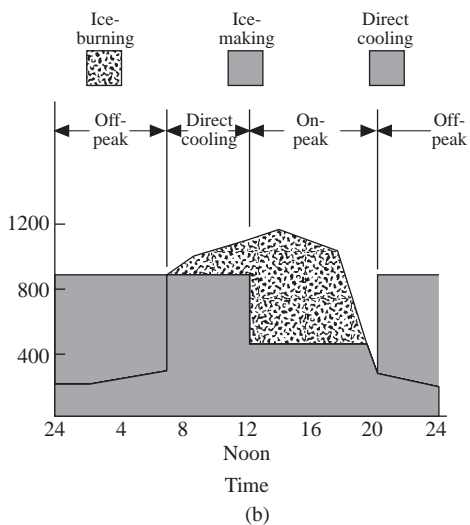
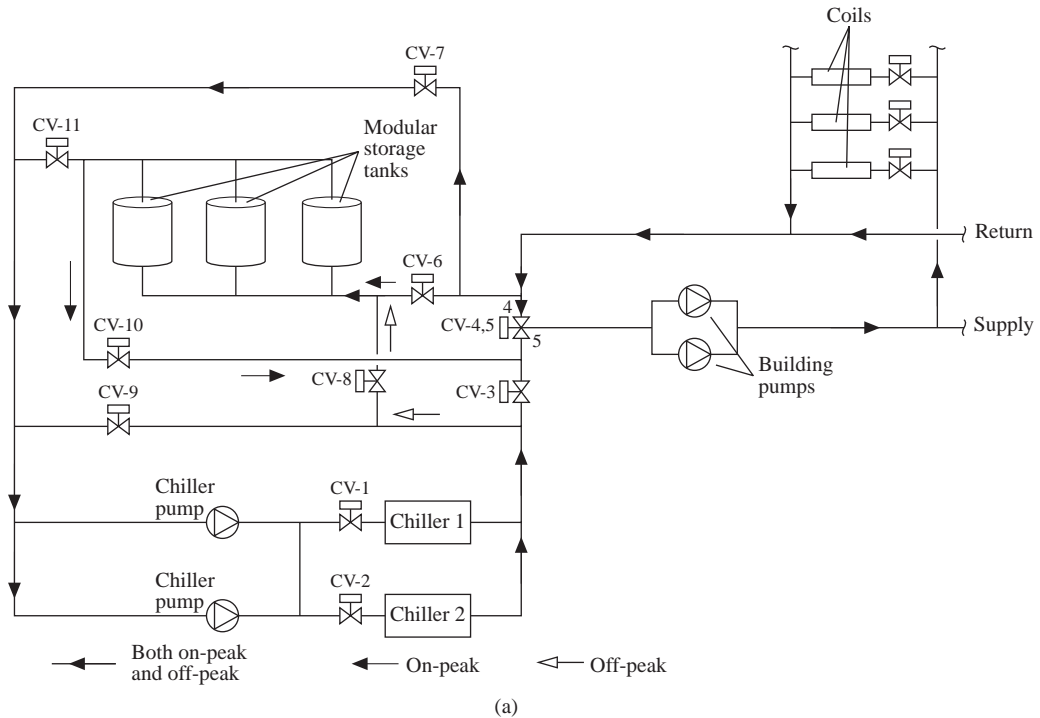


FIGURE 9.12.1 A brine-coil ice-storage system: (a) schematic diagram, (b) partial-storage time schedule, (c) ice making, and (d) ice burning.

Ice-Storage Systems

System Characteristics

In an *ice-thermal-storage* system, or simply an *ice-storage* system, ice is stored in a tank or other containers to provide cooling for buildings in on-peak hours or on- and partial-peak hours. One pound of ice can store $[(1 \times 144) + (55 - 35)] = 164$ Btu instead of $(60 - 44) = 16$ Btu for chilled water storage. For the same cooling capacity, the storage volume of ice is only about 12% of chilled water. In addition,

an air-conditioning system using ice storage often adopts cold air distribution to supply conditioned air at a temperature typically at 44°F. Cold air distribution reduces the size of air-side equipment, ducts, and investment as well as fan energy use. It also improves the indoor air quality of the conditioned space. Since the late 1980s, ice storage has gained more applications than chilled water storage.

Brine is a coolant without freezing and flashing during normal operation. The freezing point of brine, which has a mass fraction of ethylene glycol of 25%, drops to 10°F, and a mass fraction of propylene glycol of 25% drops to 15°F. Glycol-water, when glycol is dissolved in water, is another coolant widely used in ice-storage systems. Ice crystals are formed in glycol-water when its temperature drops below its freezing point during normal operation.

In an ice-storage system, *ice making* or *charging* is a process in which compressors are operated to produce ice. *Ice burning*, or *discharging*, is a process in which ice is melted to cool the brine or glycol-water to offset refrigeration load.

Brine-Coil Ice-Storage Systems. Currently used ice-storage systems include brine-coil, ice-harvester, and ice-on-coil systems. According to Knebel (1995), the brine-coil ice-storage system is most widely used today because of its simplicity, flexibility, and reliability as well as using modular ice-storage tanks.

In a typical brine-coil ice-storage system, ice is charged in multiple modular factory-fabricated storage tanks as shown in [Figure 9.12.1\(a\)](#). In each storage tank, closely spaced polyethylene or plastic tubes are surrounded by water. Brine, a mixture of 25 to 30% of ethylene glycol by mass and 70 to 75% water, circulates inside the tubes at an entering temperature of 24°F during the ice-making process. The water surrounding the tubes freezes into ice up to a thickness of about 1/2 in. as shown in [Figure 9.12.1\(c\)](#). Brine leaves the storage tank at about 30°F. Water usually at atmospheric pressure is separated from brine by a plastic tube wall. Plastic tubes occupy about one tenth of the tank volume, and another one tenth remains empty for the expansion of ice. Multiple modular storage tanks are always connected in parallel.

During the ice-melting or -burning process, brine returns from the cooling coils in the air-handling units (AHUs) at a temperature of 46°F or higher. It melts the ice on the outer surface of the tubes and is thus cooled to a temperature of 34 to 36°F, as shown in [Figure 9.12.1\(d\)](#). Brine is then pumped to the AHUs to cool and dehumidify the air again.

Case Study of a Brine-Coil Ice-Storage System. In Tackett (1989), a brine-coil ice-storage system cools a 550,000-ft² office building near Dallas, TX, as shown in [Figure 9.12.1\(a\)](#). Ethylene glycol water is used as the brine. There are two centrifugal chillers, each of them having a refrigeration capacity of 568 tons when 34°F brine is produced with a power consumption of 0.77 kW/ton for direct cooling. The refrigeration capacity drops to 425 tons if 24°F brine leaves the chiller with a power consumption of 0.85 kW/ton. A demand-limited partial storage is used, as shown in [Figure 9.12.1\(b\)](#). During on-peak hours, ice is melted; at the same time one chiller is also operating. The system uses 90 brine-coil modular storage tanks with a full-charged ice-stored capacity of 7500 ton.hr.

For summer cooling, the weekdays' 24-hr day-and-night cycle is divided into three periods:

- Off peak lasts from 8 p.m. to AHU's start the next morning. Ice is charged at a maximum capacity of 650 tons. The chillers also provide 200 tons of direct cooling for refrigeration loads that operate 24 hr continuously.
- Direct cooling lasts from AHU's start until noon on weekdays. Chillers are operated for direct cooling. If required refrigeration load exceeds the chillers' capacity, some ice storage will be melted to supplement the direct cooling.
- On peak lasts from noon to 8 p.m. Ice is burning with one chiller in operation. During ice burning, water separates the tube and ice. Water has a much lower thermal conductivity (0.35 Btu/hr.ft.°F) than ice (1.3 Btu/hr.ft.°F). Therefore, the capacity of a brine-coil ice-storage system is dominated by the ice burning.

Ice-Harvester Ice-Storage Systems. In an *ice-harvester* system, glycol-water flows on the outside surface of the evaporator and forms ice sheets with a thickness of 0.25 to 0.40 in. within 20 to 30 min. Ice is harvested in the form of flakes when hot gas is flowing inside the tubes of the evaporator during a time interval of 20 to 30 sec. Ice flakes fall to the glycol-water in the storage tank below. The glycol-water at a temperature of 34°F is then supplied to the cooling coils in AHUs for conditioning. After cooling and dehumidifying the air, glycol-water returns to the storage tank at a temperature of 50 to 60°F and is again cooled to 34°F again.

Ice harvesting is an intermittent process. It has a cycle loss due to harvesting of about 15%. In addition, because of its operating complexity and maintenance, its applications are more suitable for large ice-storage systems.

Ice-on-Coil Ice-Storage Systems. In an *ice-on-coil* system, refrigerant is evaporated in the coils submerged in water in a storage tank. Ice of a thickness not exceeding 0.25 in. builds up on the outer surface of the coils. The remaining water in the storage tank is used for cooling in AHUs. Ice-on-coil systems need large amounts of refrigerant charge and are less flexible in operation. They are usually used in industrial applications.

Ice-in-Containers Ice-Storage Systems. Ice-in-containers ice-storage systems store ice in enclosed containers. Brine circulating over the containers produces the ice inside containers. Complexity in control of the ice inventory inside the containers limits the application of the ice-in-containers systems.

Chilled-Water Storage Systems

Basics

Chilled-water storage uses the same water chiller and a similar coolant distribution system, except for additional water storage tanks and corresponding piping, additional storage pumps, and controls. The larger the chilled-water storage system, the lower the installation cost per ton.hr storage capacity.

Various types of storage tanks had been used in chilled-water storage systems during the 1970s. A diaphragm tank uses a rubber diaphragm to separate the colder and warmer water. Baffles divide the tank into cells and compartments. Today, stratified tanks have become the most widely used chilled-water storage systems because of their simplicity, low cost, and negligible difference in loss of cooling capacity between stratified tanks and other types of storage tanks.

During the storage of chilled water, the loss in cooling capacity includes direct mixing, heat transfer between warmer return chilled water and colder stored chilled water, and also heat transfer between warmer ambient air and colder water inside the tank. An enthalpy-based easily measured index called *figure of merit (FOM)* is often used to indicate the loss in cooling capacity during chilled-water storage. FOM is defined as:

$$\text{FOM} = q_{\text{dis}}/q_{\text{ch}} \quad (9.12.1)$$

where q_{dis} = cooling capacity available in the discharge process, Btu/hr
 q_{ch} = theoretical cooling capacity available during charging process, Btu/hr

Charging is the process of filling the storage tank with colder chilled water from the chiller. At the same time, warmer return chilled water is extracted from the storage tank and pumped to the chiller for cooling.

Discharging is the process of discharging the colder stored chilled water from the storage tank to AHUs and terminals. Meanwhile, the warmer return chilled water from the AHUs and terminals fills the tank.

Stratified Tanks. *Stratified tanks* utilize the buoyancy of warmer return chilled water to separate it from the colder stored chilled water during charging and discharging, as shown in [Figure 9.12.2\(a\)](#). Colder stored chilled water is always charged and discharged from bottom diffusers, and the warmer return chilled water is introduced to and withdrawn from the top diffusers.

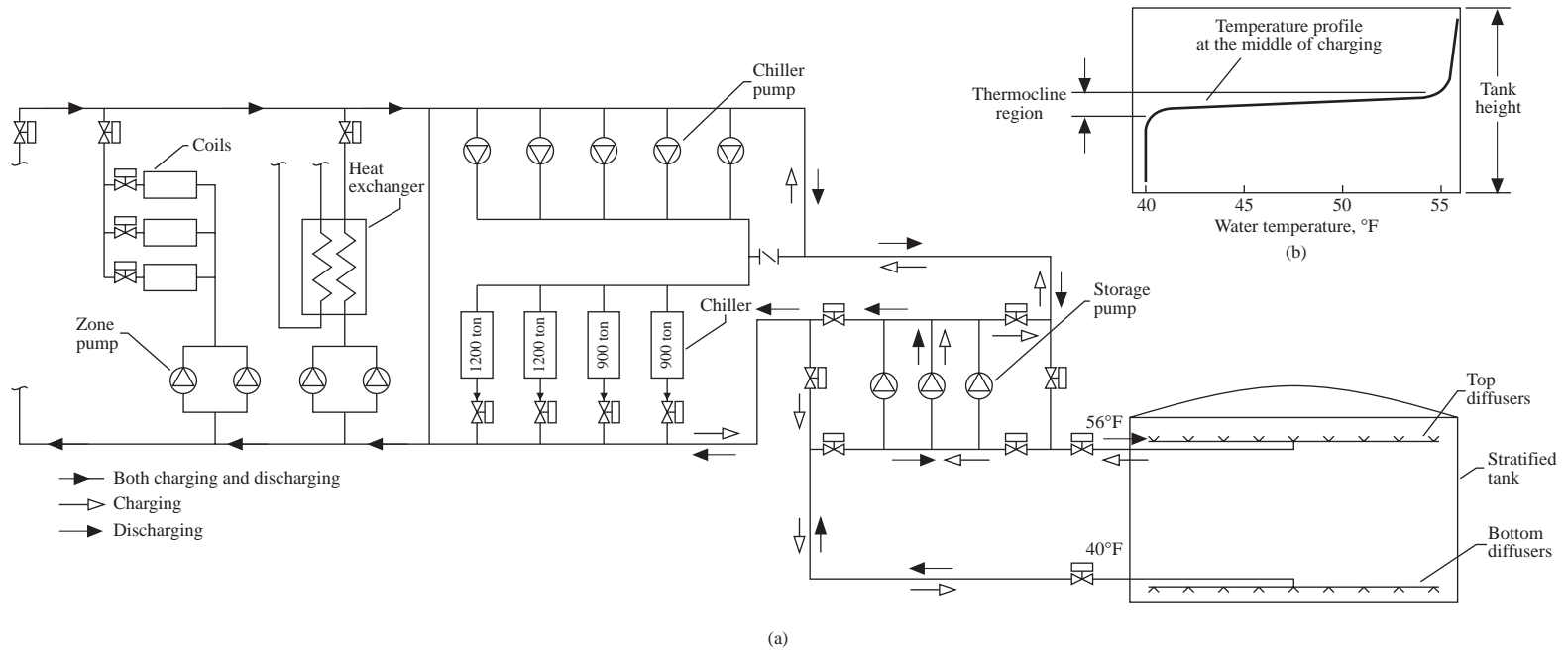


FIGURE 9.12.2 A chilled-water storage system using stratified tanks: (a) schematic diagram of a chilled-water storage system and (b) thermocline at the middle of charging process.

Chilled-water storage tanks are usually vertical cylinders and often have a height-to-diameter ratio of 0.25:0.35. Steel is the most commonly used material for above-grade tanks, with a 2-in.-thick spray-on polyurethane foam, a vapor barrier, and a highly reflective coating. Concrete, sometimes precast, prestressed tanks are widely used for underground tanks.

A key factor to reduce loss in cooling capacity during chilled water storage is to reduce mixing of colder and warmer water streams at the inlet. If the velocity pressure of the inlet stream is less than the buoyancy pressure, the entering colder stored chilled water at the bottom of tank will stratify. Refer to Wang's handbook (1993) and Knebel (1995) for details.

A *thermocline* is a stratified region in a chilled-water storage tank of which there is a steep temperature gradient as shown in [Figure 9.12.2\(b\)](#). Water temperature often varies from 42°F to about 60°F. Thermocline separates the bottom colder stored chilled water from the top warmer return chilled water. The thinner the thermocline, the lower the mixing loss.

Diffusers and symmetrical connected piping are used to evenly distribute the incoming water streams with sufficient low velocity, usually lower than 0.9 ft/sec. Inlet stream from bottom diffusers should be downward and from the top diffusers should be upward or horizontal.

Field measurements indicate that stratified tanks have a FOM between 0.85 to 0.9.

9.13 Air System Basics

Fan-Duct Systems

Flow Resistance

Flow resistance is a property of fluid flow which measures the characteristics of a flow passage resisting the fluid flow with a corresponding total pressure loss Δp , in in. WG, at a specific volume flow rate \dot{V} , in cfm:

$$\Delta p = R \dot{V}^2 \quad (9.13.1)$$

where R = flow resistance (in. WG/(cfm)²).

For a duct system that consists of several duct sections connected in series, its flow resistance R_s , in in. WG/(cfm)², can be calculated as

$$R_s = R_1 + R_2 + \dots + R_n \quad (9.13.2)$$

where R_1, R_2, \dots, R_n = flow resistance of duct section 1, 2, ... n in the duct system (in. WG/(cfm)²).

For a duct system that consists of several duct sections connected in parallel, its flow resistance R_p , in in. WG/(cfm)², is:

$$1/\sqrt{R_p} = 1/\sqrt{R_1} + 1/\sqrt{R_2} + \dots + 1/\sqrt{R_n} \quad (9.13.3)$$

Fan-Duct System

In a *fan-duct system*, a fan or several fans are connected to ductwork or ductwork and equipment. The volume flow and pressure loss characteristics of a duct system can be described by its performance curve, called *system curve*, and is described by $\Delta p = R \dot{V}^2$.

An *air system* or an air handling system is a kind of fan-duct system. In addition, an outdoor ventilation air system to supply outdoor ventilation air, an exhaust system to exhaust contaminated air, and a smoke control system to provide fire protection are all air systems, that is, fan-duct systems.

Primary, Secondary, and Transfer Air

Primary air is the conditioned air or makeup air. Secondary air is often the induced space air, plenum air, or recirculating air. Transfer air is the indoor air that moves to a conditioned space from an adjacent area.

System-Operating Point

A *system-operating point* indicates the operating condition of an air system or fan-duct system. Since the operating point of a fan must lie on the fan performance curve, and the operating point of a duct system on the system curve, the system operating point of an air system must be the intersection point P_s of the fan performance curve and system curve as shown in [Figure 9.13.1\(a\)](#).

Fan-Laws

For the same air system operated at speeds n_1 and n_2 , both in rpm, their relationship of \dot{V} volume flow rate, in cfm, system total pressure loss, in in. WG, and fan power input, in hp, can be expressed as

$$\begin{aligned} \dot{V}_2/\dot{V}_1 &= n_2/n_1 \\ \Delta p_{t2}/\Delta p_{t1} &= (n_2/n_1)^2 (\rho_2/\rho_1) \\ P_2/P_1 &= (n_2/n_1)^3 (\rho_2/\rho_1) \end{aligned} \quad (9.13.4)$$

where ρ = air density (lb/ft³). Subscripts 1 and 2 indicate the original and the changed operating conditions. For air systems that are geometrically and dynamically similar:

$$\begin{aligned}\dot{V}_2/\dot{V}_1 &= (D_2/D_1)^3 (n_2/n_1) \\ \Delta p_{t2}/\Delta p_{t1} &= (D_2/D_1)^2 (n_2/n_1)^2 (\rho_2/\rho_1) \\ P_2/P_1 &= (D_2/D_1)^5 (n_2/n_1)^3 (\rho_2/\rho_1)\end{aligned}\quad (9.13.5)$$

where D = diameter of the impeller (ft).

Geometrically similar means that two systems are similar in shape and construction. For two systems that are dynamically similar, they must be geometrically similar, and in addition, their velocity distribution or profile of fluid flow should also be similar. When fluid flows in the air systems are at high Reynolds number, such as $Re > 10,000$, their velocity profiles can be considered similar to each other.

System Effect

The system effect Δp_{se} , in in. WG, of an air system is its additional total pressure loss caused by uneven or nonuniform velocity profile at the fan inlet, or at duct fittings after fan outlet, due to the actual inlet and outlet connections as compared with the total pressure loss of the fan test unit during laboratory ratings. The selected fan total pressure which includes the system effect Δp_{ts} , in in. WG, as shown in Figure 9.13.1(a), can be calculated as

$$\begin{aligned}\Delta p_{ts} &= \Delta p_{sy} + \Delta p_{se} = \Delta p_{sy} + \Delta p_{s,i} + \Delta p_{s,o} \\ &= \Delta p_{sy} + C_{s,i} (v_{fi}/4005)^2 + C_{s,o} (v_{fo}/4005)^2\end{aligned}\quad (9.13.6)$$

where Δp_{sy} = calculated total pressure loss of the air system, in WG

$\Delta p_{s,i}$, $\Delta p_{s,o}$ = fan inlet and outlet system effect loss, in WG

$C_{s,i}$, $C_{s,o}$ = local loss coefficient of inlet and outlet system effect, in WG

v_{fi} , v_{fo} = velocity at fan inlet (fan collar) and fan outlet, fpm

Both $C_{s,i}$ and $C_{s,o}$ are mainly affected by the length of connected duct between the duct fitting and fan inlet or outlet, by the configuration of the duct fitting, and by the air velocity at the inlet or outlet. Because v_{fi} and v_{fo} are often the maximum velocity of the air system, system effect should not be overlooked. According to AMCA Fan and Systems (1973), a square elbow (height to turning radius ratio $R/H = 0.75$) connected to a fan inlet with a connected duct length of $2 D_e$ (equivalent diameter) and $v_{fi} = 3000$ fpm may have a 0.67 in. WG $\Delta p_{i,o}$ loss. Refer to AMCA Fan and Systems (1973) or Wang's handbook (1993) for details.

Modulation of Air Systems

Air systems can be classified into two categories according to their operating volume flow: constant volume and variable-air-volume systems. The volume flow rate of a *constant volume system* remains constant during all the operating time. Its supply temperature is raised during part load. For a *variable-air-volume (VAV) system*, its volume flow rate is reduced to match the reduction of space load at part-load operation. The system pressure loss of a VAV system can be divided into two parts: variable part Δp_{var} and fixed part Δp_{fix} , which is the set point of the duct static pressure control as shown in Figure 9.13.1(b) and (c). The *modulation curve* of a VAV air system its its operating curve, or the locus of system operating points when its volume flow rate is modulated at full- and part-load operation.

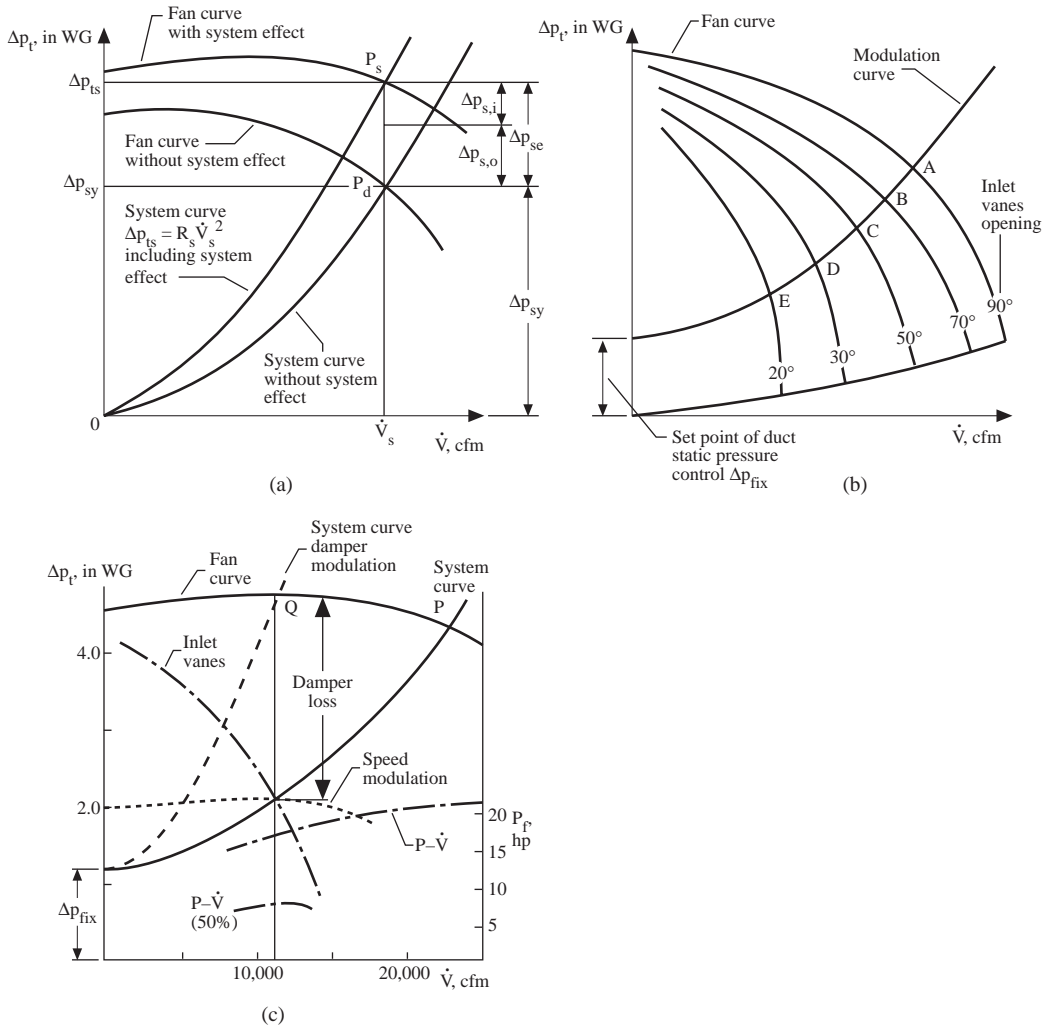


FIGURE 9.13.1 Air system \dot{V} - Δp_t performance: (a) system operating point and system effect, (b) modulation curve, and (c) damper, inlet vanes, and fan speed modulation.

The volume flow and system pressure loss of an air system can be modulated either by changing its fan characteristics or by varying its flow resistance of the duct system. Currently used types of modulation of volume flow rate of VAV air systems are

1. *Damper modulation* uses an air damper to vary the opening of the air flow passage and therefore its flow resistance.
2. *Inlet vanes modulation* varies the opening and the angle of inlet vanes at the centrifugal fan inlet and then gives different fan performance curves.
3. *Inlet cone modulation* varies the peripheral area of the fan impeller and therefore its performance curve.
4. *Blade pitch modulation* varies the blade angle of the axial fan and its performance curve.
5. *Fan speed modulation* using *adjustable frequency AC drives* varies the fan speed by supplying a variable-frequency and variable-voltage power source. There are three types of AC drives: adjustable voltage, adjustable current, and pulse width modulation (PWM). The PWM is universally applicable.

Damper modulation wastes energy. Inlet vanes are low in cost and are not so energy efficient compared with AC drives and inlet cones. Inlet cone is not expensive and is suitable for backward curved centrifugal fans. Blade pitch modulation is energy efficient and is mainly used for vane and tubular axial fans. AC drive is the most energy-efficient type of modulation; however, it is expensive and often considered cost effective for air systems using large centrifugal fans.

Example 9.13.1

A multizone VAV system equipped with a centrifugal fan has the following characteristics:

\dot{V} (cfm)	5,000	10,000	15,000	20,000	25,000
Δp_t , in. WG	4.75	4.85	4.83	4.60	4.20
P, hp		17.0	18.6	20.5	21.2

At design condition, it has a volume flow rate of 22,500 cfm and a fan total pressure of 4.45 in. WG. The set point of duct static pressure control is 1.20 in. WG.

When this VAV system is modulated by inlet vanes to 50% of design flow, its fan performance curves show the following characteristics:

\dot{V} (cfm)	5,000	10,000	11,250
Δp_t , in. WG	3.6	2.5	2.1
P, hp		7.5	7.8

Determine the fan power input when damper, inlet vanes, or AC drive fan speed modulation is used. Assume that the fan total efficiency remains the same at design condition when the fan speed is reduced.

Solutions

- At 50% design flow, the volume flow of this VAV system is $0.5 \times 22,500 = 11,250$ cfm. The flow resistance of the variable part of this VAV system is

$$R_{\text{var}} = \Delta p_{\text{va}} = R \dot{V}^2 = (4.45 - 1.20) / (22,500)^2 = 6.42 \times 10^{-9} \text{ in. WG}/(\text{cfm})^2$$

When damper modulation is used, the system operating point Q must be the intersection of the fan curve and the system curve that has a higher flow resistance and a $\dot{V} = 11,250$ cfm. From Figure 9.13.1(c), at point Q, the fan power input is 17.0 hp.

- From the given information, when inlet vane modulation is used, the fan power input is 7.8 hp.
- The total pressure loss of the variable part of the VAV system at 50% volume flow is

$$\Delta p_{\text{var}} = R_{\text{var}} \dot{V}^2 = 6.42 \times 10^{-9} (11,250)^2 = 0.81 \text{ in. WG}$$

From Figure 9.13.1(c), since the fan power input at design condition is 21.2 hp, then its fan total efficiency is:

$$\eta_f = \dot{V} \Delta p_{\text{tf}} / (6356 P_f) = 22,500 \times 4.45 / (6356 \times 21.2) = 74.3\%$$

The fan power input at 50% design volume flow is:

$$P = \dot{V} \Delta p_{\text{tf}} / (6356 \eta_f) = 11,250(0.81 + 1.20) / (6356 \times 0.743) = 4.8 \text{ hp}$$

Damper modulation has a energy waste of $(17 - 4.8) = 12.2$ hp

Fan Combinations in Air-Handling Units and Packaged Units

Currently used fan combinations in air-handling units (AHUs) and packaged units (PUs) (except dual-duct VAV systems) are shown in Figure 9.13.2(a), (b), and (c):

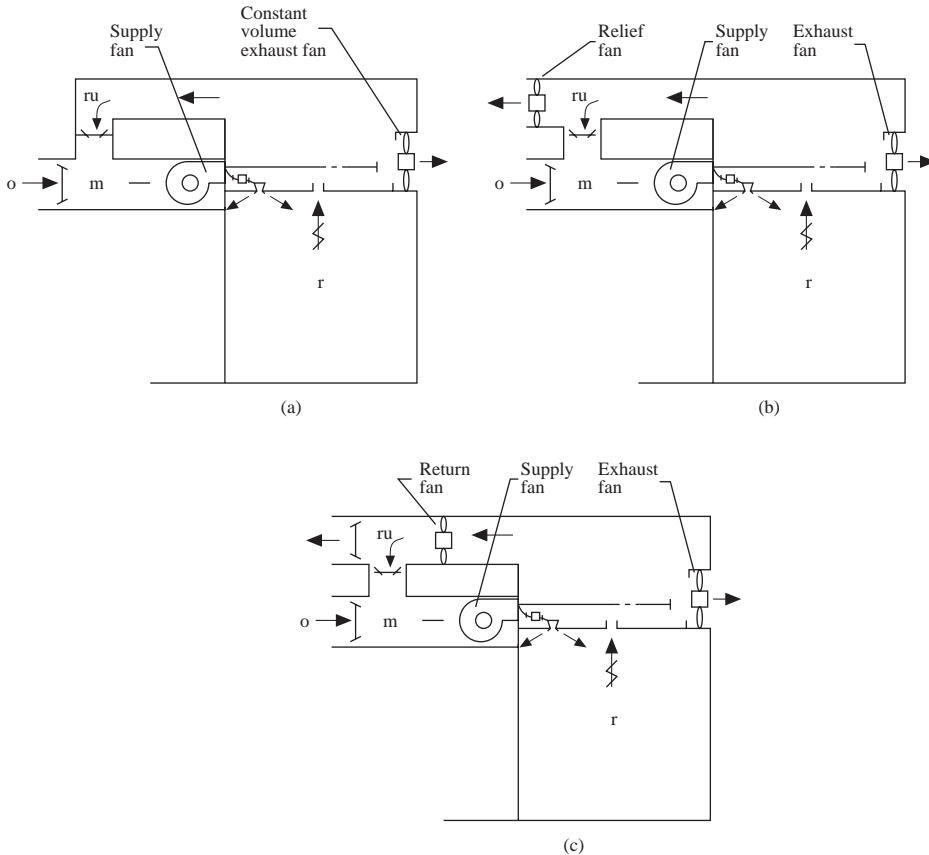


FIGURE 9.13.2 Fan combinations: (a) supply and exhaust fan, (b) supply and relief fan, and (c) supply and return fan.

Supply and Exhaust Fan/Barometric Damper Combination

An air system equipped with a single supply fan and a constant-volume exhaust fan, or a supply fan and a barometric damper combination as shown in Figure 9.13.2(a), is often used in buildings where there is no return duct or the pressure loss of the return duct is low. An all-outdoor air economizer cycle is usually not adopted due to the extremely high space pressure. A barometric relief damper is often installed in or connected to the conditioned space to prevent excessively high space pressure. When the space-positive pressure exerted on the barometric damper is greater than the weight of its damper and/or a spring, the damper is opened and the excessive space pressure is relieved.

Consider a VAV rooftop packaged system using a supply fan and a constant-volume exhaust system to serve a typical floor in an office building. This air system has the following design parameters:

During minimum outdoor ventilation air *recirculating mode* at summer design volume flow and at 50% of design volume flow, the outdoor damper is at its minimum opening. The recirculating damper is fully opened. The outdoor air intake at the PU must be approximately equal to the exfiltration at the conditioned space due to the positive space pressure p_r , in in. WG. By using the iteration method, the calculated pressure characteristics and the corresponding volume flow rates are shown below:

Supply volume flow rate:		20,000 cfm
Total pressure loss:	Across the recirculating damper	0.1 in. WG
	Filter and coils	2.5 in. WG
	Supply main duct	0.85 in. WG
	Return system between point r and ru	0.2 in. WG
Effective leakage area on the building shell		3.35 ft ²
Minimum outdoor ventilation air		3000 cfm
Space pressure at design flow		+0.05 in. WG

Point	o	m	r	ru
Design flow:				
p_s in. WG	0	-0.20	+0.05	-0.15
V , cfm	3,000	20,000	20,000	17,000
50% design:				
p_s in. WG	0	-0.092	+0.0225	-0.052
V , cfm	2,025	10,000	10,000	7,975

Here, o represents outdoor, m the mixing box, r the space, and ru the recirculating air inlet to the PU.

When the supply volume flow is reduced from the design volume flow to 50% of design flow during the recirculating mode, the total pressure in the mixing box increases from -0.20 to -0.092 in. WG and the outdoor air intake reduces from 3000 to 2025 cfm. Refer to Wang's (1993) *Handbook of Air Conditioning and Refrigeration* for details.

Supply and Relief Fan Combination

Figure 9.13.2(b) shows the schematic diagrams of an air system of supply fan and relief fan combination. A relief fan is used to relieve undesirable high positive space pressure by extracting space air and relieving it to the outside atmosphere. A relief fan is always installed in the relief flow passage after the junction of return flow, relief flow, and recirculating flow passage, point ru. It is usually energized only when the air system is operated in air economizer mode. A relief fan is often an axial fan. Since the relief fan is not energized during recirculating mode operation, the volume flow and pressure characteristics of a supply fan and relief fan combination are the same as that in a single supply fan and barometric damper combination when they have the same design parameters.

During air economizer mode, the outdoor air damper(s) are fully opened and the recirculating damper closed. The space pressure $p_r = +0.05$ in. WG is maintained by modulating the relief fan speed or relief damper opening. The pressure and volume flow characteristics of a supply and relief fan combination at design volume flow and 50% of design flow are as follows:

Point	o	m	r	ru
Design flow:				
p_s in. WG	0	-0.20	+0.05	-0.15
V , cfm	20,000	20,000	20,000	17,000
50% design:				
p_s in. WG	0	-0.057	+0.05	+0.007
V , cfm	10,000	10,000	10,000	7,000

Supply Fan and Return Fan Combination

A return fan is always installed at the upstream of the junction of return, recirculating, and exhaust flow passage, point ru as shown in Figure 9.13.2(c). A supply and return fan combination has similar pressure and volume flow characteristics as that of a supply and relief fan combination, except a higher total

pressure at point ru. If the return fan is improperly selected and has an excessive fan total pressure, total pressure at point m may be positive. There will be no outdoor intake at the PU or AHU, and at the same time there will also be a negative space pressure and an infiltration to the space.

Comparison of These Three Fan Combination Systems

A supply fan and barometric damper combination is simpler and less expensive. It is suitable for an air system which does not operate at air economizer mode and has a low pressure drop in the return system.

For those air systems whose pressure drop of return system is not exceeding 0.3 in. WG, or there is a considerable pressure drop in relief or exhaust flow passage, a supply and relief fan combination is recommended. For air systems whose return system has a pressure drop exceeding 0.6 in. WG, or those requiring a negative space pressure, a supply and return fan combination seems more appropriate.

Year-Round Operation and Economizers

Consider a typical single-duct VAV reheat system to serve a typical floor whose indoor temperature is 75°F with a relative humidity of 50%, as shown in Figure 9.13.3(a). During summer, the off-coil temperature is 55°F. The year-round operation of this air system can be divided into four regions on the psychrometric chart, as shown in Figure 9.13.3(b):

- *Region I — Refrigeration/evaporative cooling.* In this region, the enthalpy of the outdoor air h_o is higher than the enthalpy of the recirculating air h_{ru} , $h_o > h_{ru}$. It is more energy efficient to condition the mixture of recirculating air and minimum outdoor air.
- *Region II — Free cooling and refrigeration.* In this region, $h_o \leq h_{ru}$. It is more energy efficient and also provides better indoor air quality to extract 100% outdoor air.
- *Region III — Free cooling evaporative cooling, and refrigeration.* In this region, extract 100% outdoor air for free cooling because $h_o \leq h_{ru}$. Use evaporative cooling and refrigeration to cool and humidify if necessary.
- *Region IV — Winter heating.* Maintain a 55°F supply temperature by mixing the recirculating air with the outdoor air until the outdoor air is reduced to a minimum value. Provide heating if necessary.

An economizer is a device consisting of dampers and control that uses the free cooling capacity of either outdoor air or evaporatively cooled water from the cooling tower instead of mechanical refrigeration. An air economizer uses outdoor air for free cooling. There are two kinds of air economizers: enthalpy-based, in which the enthalpy of outdoor and recirculating air is compared, and temperature-based, in which temperature is compared. A water economizer uses evaporatively cooled water.

Fan Energy Use

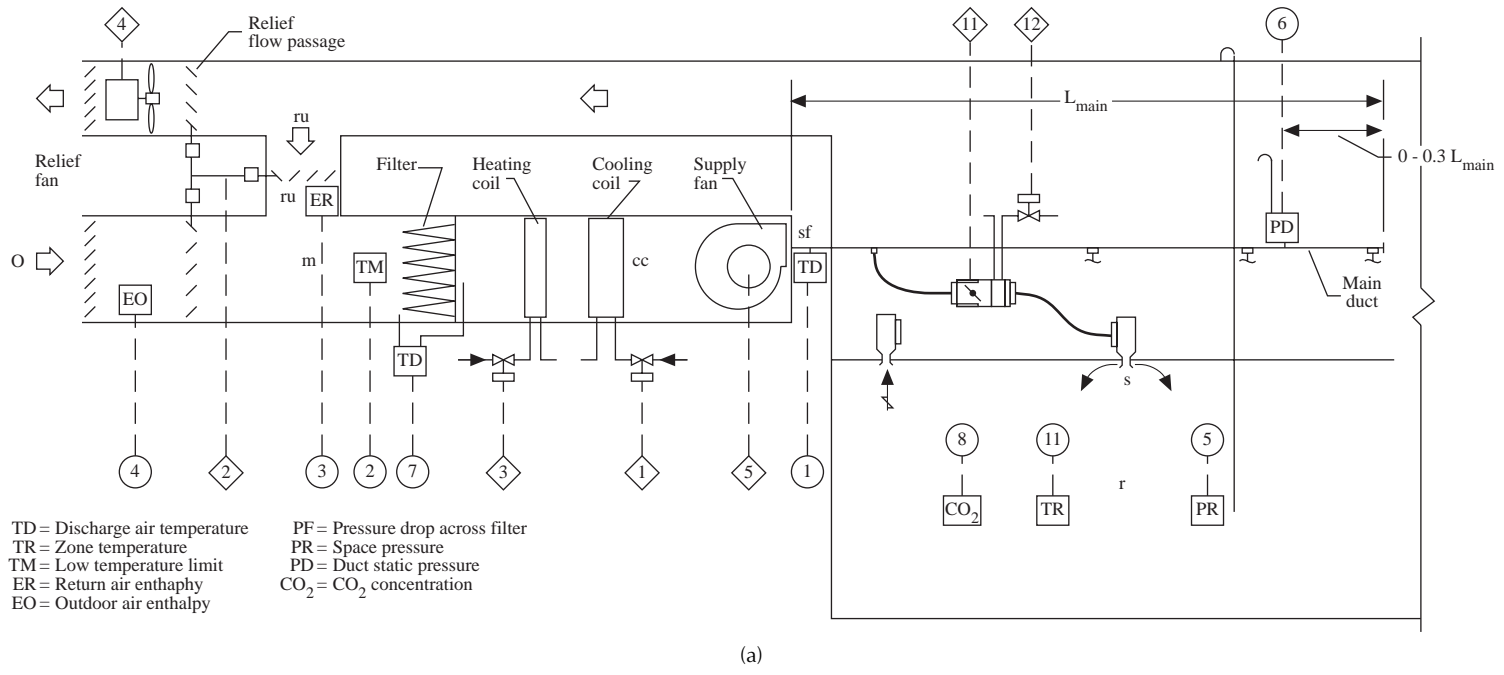
For an air system, fan energy use for each cfm of conditioned air supplied from the AHUs and PUs to the conditioned space within a certain time period, in W/cfm, can be calculated as

$$W/cfm = 0.1175\Delta p_{sy} / (\eta_f \eta_m) \quad (9.13.7)$$

where Δp_{sy} = mean system total pressure loss during a certain time period, in. WG
 η_f = fan total efficiency
 η_m = combined motor and drive (direct drive or belt drive) efficiency

For an air system using a separate outdoor ventilation system, its fan energy use, in W/cfm, is then calculated as

$$W/cfm = (1 + R_{o,s}) \left[0.1175\Delta p_{sy} / (\eta_f \eta_m) \right] \quad (9.13.8)$$



(a)

FIGURE 9.13.3 Year-round operation, discharge air temperature, and duct static pressure control for a VAV reheat system: (a) control diagram, (b) year-round operation, and (c) discharge air temperature control output diagram.

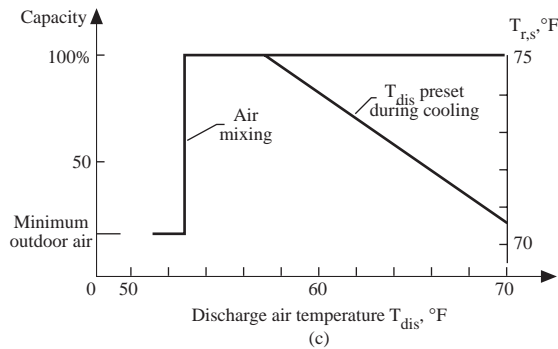
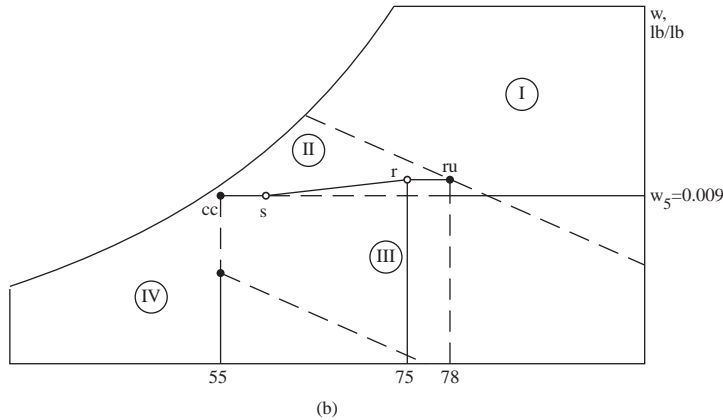


FIGURE 9.13.3bc

where $R_{o,s}$ = ratio of outdoor air volume flow rate to supply volume flow rate.

Outdoor Ventilation Air Supply

Basics

- An adequate amount of outdoor ventilation air supply is the key factor to provide acceptable indoor air quality (IAQ) for a conditioned space. Although an inadequate amount of outdoor ventilation air supply causes poor IAQ, an oversupply of outdoor ventilation air other than in an air economizer cycle is often a waste of energy.
- According to local codes and ANSI/ASHRAE Standard 62-1989, the minimum outdoor ventilation rate for each person must be provided at the outdoor air intake of AHU or PU, or by an outdoor air ventilation system. If the minimum outdoor ventilation air rate is reduced by using higher efficiency filters to remove air contaminants in the recirculating air, then indoor air contaminant concentration must be lower than the specified level in ANSI/ASHRAE Standard 62-1989.
- For a multizone air system, although the ratio of outdoor ventilation air rate to supply air volume flow rate required may be varied from zone to zone, the excessive outdoor air supply to a specified zone will increase the content of unused outdoor air in the recirculating air in AHU or PU. This helps to solve the problem in any zone that needs more outdoor air.
- Since the occupancy in many buildings is often variable and intermittent, a demand-based variable amount of outdoor ventilation air control should be used instead of time-based constant volume outdoor ventilation air control, except during the air economizer cycle.

- Carbon dioxide (CO₂) is a gaseous body effluent. CO₂ is an indicator of representative odor and an indicator of adequate outdoor ventilation rate at specific outdoor and indoor air concentration in a control zone at steady state. For most of the comfort air-conditioning systems, it is suitable to use CO₂ as a key parameter to control the intake volume flow rate of outdoor ventilation air to maintain an indoor CO₂ concentration not exceeding 800 to 1000 ppm in a critical or representative zone. As mentioned in Section 9.5, Persily (1993) showed that the actual measured indoor daily maximum CO₂ concentration levels in five buildings were all within 400 to 820 ppm.

If a field survey finds that a specific indoor air contaminant exceeds a specified indoor concentration, then a gas sensor for this specific contaminant or a mixed gas sensor should be used to control this specific indoor concentration level.

Types of Minimum Outdoor Ventilation Air Control. There are four types of minimum outdoor ventilation air control that are currently used:

- Type I uses a CO₂ sensor or a mixed gas sensor and a DDC controller to control the volume flow rate of outdoor ventilation air for a separate outdoor ventilation air system on the demand-based principle.
- Type II uses a CO₂ or mixed gas sensor and a DDC controller to control the ratio of the openings between outdoor and recirculating dampers and, therefore, the volume flow rates of outdoor air and recirculating air in AHUs or PUs on the demand-based principle.
- Type III uses a flow sensor or a pressure sensor and a DDC controller to control the openings of outdoor and recirculating dampers to provide a nearly constant volume outdoor air intake in VAV AHUs or VAV PUs.
- Type IV adjusts the opening of the outdoor damper manually to provide a constant volume of outdoor air in constant-volume AHUs and PUs. If the outdoor intake is mounted on the external wall without a windshield, the volume flow of outdoor ventilation air intake will be affected by wind force and direction.

Type I is the best minimum outdoor ventilation air control for the air system. For a VAV system, it is expensive. Type II is a better choice. Type III is more complicated and may cause energy waste. Type IV has the same result as Type III and is mainly used in constant-volume systems.

Outdoor intake must be located in a position away from the influence of exhaust outlets. Fans, control dampers, and filters should be properly operated and maintained in order to provide a proper amount of outdoor ventilation air as well as an acceptable IAQ.

9.14 Absorption System

Absorption systems use heat energy to produce refrigeration as well as heating if it is required. Water is the refrigerant and aqueous lithium bromide (LiBr) is widely used as the carrier to absorb the refrigerant and provide a higher coefficient of performance.

The mixture of water and anhydrous LiBr is called *solution*. The composition of a solution is usually expressed by its mass fraction, or percentage of LiBr, often called *concentration*. When the water vapor has boiled off from the solution, it is called *concentration solution*. If the solution has absorbed the water vapor, it is called *diluted solution*.

Absorption systems can be divided into the following categories:

- *Absorption chillers* use heat energy to produce refrigeration.
- *Absorption chiller/heaters* use direct-fired heat input to provide cooling or heating separately.
- *Absorption heat pumps* extract heat energy from the evaporator, add to the heat input, and release them both to the hot water for heating.
- *Absorption heat transformers* raise the temperature of the waste heat source to a required level.

Most recently installed absorption chillers use direct-fired natural gas as the heat source in many locations in the United States where there are high electric demand and electric rate at on-peak hours. Absorption chillers also are free from CFC and HCFC. An energy cost analysis should be done to determine whether an electric chiller or a gas-fired absorption chiller is the suitable choice.

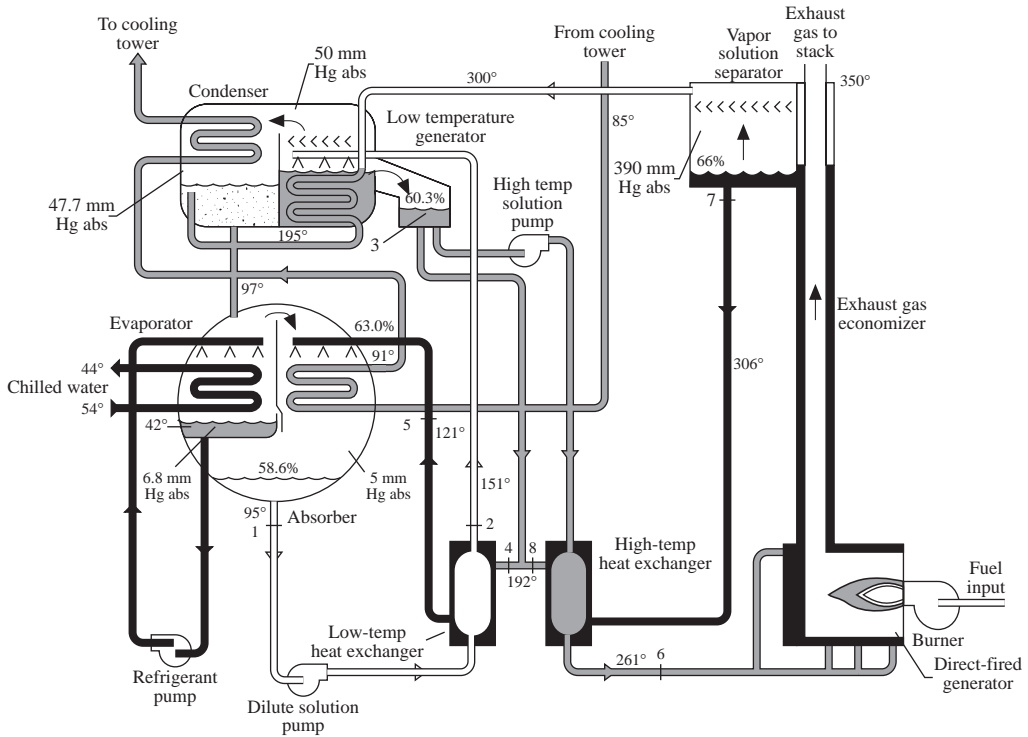
Absorption heat pumps have only limited applications in district heating. Most absorption heat transformers need industrial waste heat. Both of them will not be covered here.

Double-Effect Direct-Fired Absorption Chillers

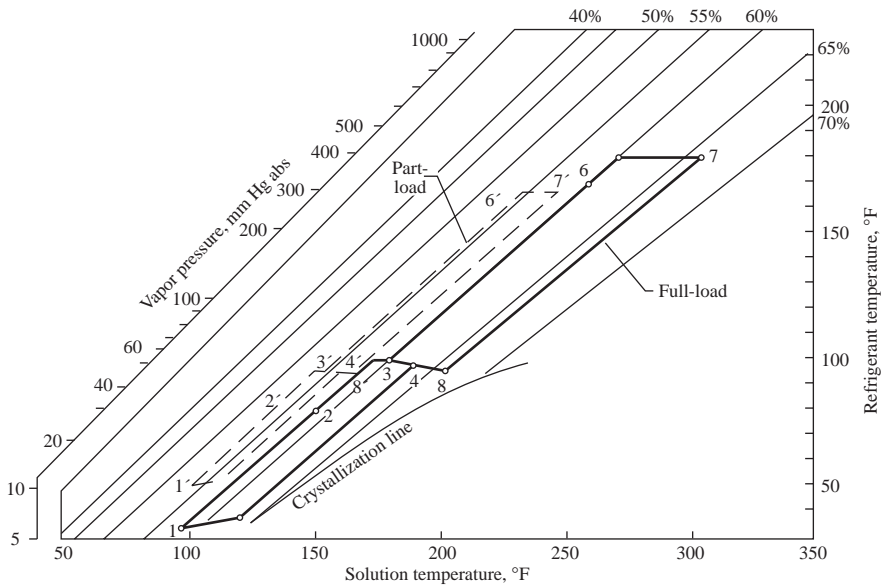
Figure 9.14.1(a) shows a double-effect direct-fired absorption chiller. *Double effect* means that there are two generators. *Direct fired* means that gas is directly fired at the generator instead of using steam or hot water. A single-effect absorption chiller using steam as the heat input to its single generator has a COP only from 0.7 to 0.8, whereas a double-effect direct-fired absorption chiller has a COP approximately equal to 1 and therefore is the most widely used absorption chiller in the United States for new and retrofit projects today. The refrigeration capacity of double-effect direct-fired absorption chillers varies from 100 to 1500 tons.

A double-effect direct-fired absorption chiller mainly consists of the following components and controls:

- **Evaporator** — An *evaporator* is comprised of a tube bundle, spray nozzles, a water trough, a refrigerant pump, and an outer shell. Chilled water flows inside the tubes. A refrigerant pump sprays the liquid refrigerant over the outer surface of the tube bundle for a higher rate of evaporation. A water trough is located at the bottom to maintain a water level for recirculation.
- **Absorber** — In an *absorber*, there are tube bundles in which cooling water flows inside the tubes. Solution is sprayed over the outer surface of the tube bundle to absorb the water vapor. A solution pump is used to pump the diluted solution to the heat exchanger and low-temperature generator.
- **Heat exchangers** — There are two heat exchangers: *low-temperature heat exchanger* in which the temperature of hot concentrated solution is lower, and *high-temperature heat exchanger* in which the temperature of hot concentrated solution is higher. In both heat exchangers, heat is transferred from the hot concentrated solution to the cold diluted solution. Shell-and-tube or plate-and-frame heat exchangers are most widely used for their higher effectiveness.
- **Generators** — *Generators* are also called *desorbers*. In the *direct-fired generator*, there are the fire tube, flue tube, vapor/liquid separator, and flue-gas economizer. Heat is supplied from the gas burner or other waste heat source. The *low-temperature generator* is often of the shell-and-tube type. The water vapor vaporized in the direct-fired generator is condensed inside the tubes. The



(a)



(b)

FIGURE 9.14.1 A double-effect direct-fired reverse-parallel-flow absorption chiller: (a) schematic diagram (reprinted by permission from the Trane catalog) and (b) absorption cycle.

latent heat of condensation thus released is used to vaporize the dilute solution in the low-temperature generator.

- Condenser — A condenser is usually also of the shell-and-tube type. Cooling water from the absorber flows inside the tubes.
- Throttling devices — Orifices and valves are often used as throttling devices to reduce the pressure of refrigerant and solution to the required values.
- Air purge unit — Since the pressure inside the absorption chiller is below atmospheric pressure, air and other noncondensable gases will leak into it from the ambient air. An *air purge unit* is used to remove these noncondensable gases from the chiller. A typical air purge unit is comprised of a pickup tube, a purge chamber, a solution spray, cooling water tubes, a vacuum pump, a solenoid valve, and a manual shut-off valve.

When noncondensable gases leak into the system, they tend to migrate to the absorber where pressure is lowest. Noncondensable gases and water vapor are picked from the absorber through the pickup tube. Water vapor is absorbed by the solution spray and returned to the absorber through a liquid trap at the bottom of the purge chamber. Heat of absorption is removed by the cooling water inside the tubes. Noncondensable gases are then evacuated from the chamber periodically by a vacuum pump to the outdoor atmosphere.

Palladium cells are used to continuously remove a small amount of hydrogen that is produced due to corrosion. Corrosion inhibitors like lithium chromate are needed to protect the machine parts from the corrosive effect of the absorbent when air is present.

Absorption Cycles, Parallel-, Series-, and Reverse-Parallel Flow

An *absorption cycle* shows the properties of the solution and its variation in concentrations, temperature, and pressure during absorbing, heat exchanging, and concentration processes on an equilibrium chart as shown in [Figure 9.14.1\(b\)](#). The ordinate of the equilibrium chart is the saturated temperature and pressure of water vapor, in °F and mm Hg abs. The abscissa is the temperature of the solution, in °F. Concentration lines are incline lines. At the bottom of the concentration lines, there is a *crystallization line* or *saturation line*. If the mass of fraction of LiBr in a solution which remains at constant temperature is higher than the saturated condition, that part of LiBr exceeding the saturation condition tends to form solid crystals.

Because there are two generators, the flow of solution from the absorber to generators can be in series flow, parallel flow, or reverse-parallel flow. In a series-flow system, the diluted solution from the absorber is first pumped to the direct-fired generator and then to the low-temperature generator. In a parallel-flow system, diluted solution is pumped to both direct-fired and low-temperature generators in parallel. In a reverse-parallel-flow system as shown in [Figure 9.14.1\(a\)](#), diluted solution is first pumped to the low-temperature generator. After that, the partly concentrated solution is then sent to the direct-fired generator as well as to the intermediate point 4 between high- and low-temperature heat exchangers in parallel. At point 4, partly concentrated solution mixes with concentrated solution from a direct-fired generator. A reverse-parallel-flow system is more energy efficient.

Solution and Refrigerant Flow

In a typical double-effect direct-fired reverse-parallel-flow absorption chiller operated at design full load, water is usually evaporated at a temperature of 42°F and a saturated pressure of 6.8 mm Hg abs in the evaporator. Chilled water returns from the AHUs or fan coils at a temperature typically 54°F, cools, and leaves the evaporator at 44°F. A refrigeration effect is produced due to the vaporization of water vapor and the removal of latent heat of vaporization from the chilled water.

Water vapor in the evaporator is then extracted to the absorber due to its lower vapor pressure. It is absorbed by the concentrated LiBr solution at a pressure of about 5 mm Hg abs. After absorption, the solution is diluted to a concentration of 58.6% and its temperature increases to 95°F (point 1). Most of the heat of absorption and the sensible heat of the solution is removed by the cooling water inside the

tube bundle. Diluted solution is then pumped by a solution pump to the low-temperature generator through a low-temperature heat exchanger.

In the low-temperature generator, the dilute solution is partly concentrated to 60.3% at a solution temperature of 180°F (point 3). It then divides into two streams: one of them is pumped to the direct-fired generator through a high-temperature heat exchanger, and the other stream having a slightly greater mass flow rate is sent to the intermediate point 4. In the direct-fired generator, the concentrated solution leaves at a concentration of 66% and a solution temperature of 306°F (point 7).

The mixture of concentrated and partly concentrated solution at point 4 has a concentration of 63% and a temperature of 192°F. It enters the low-temperature heat exchanger. Its temperature drops to 121°F before entering the absorber (point 5).

In the direct-fired generator, water is boiled off at a pressure of about 390 mm Hg abs. The boiled-off water vapor flows through the submerged tube in the low-temperature generator. The release of latent heat of condensation causes the evaporation of water from the dilution solution at a vapor pressure of about 50 mm Hg abs. The boiled-off water vapor in the low-temperature generator flows to the condenser through the top passage and is condensed into liquid water at a temperature of about 99°F and a vapor pressure of 47.7 mm Hg abs. This condensed liquid water is combined with the condensed water from the submerged tube at the trough. Both of them return to the evaporator after its pressure is throttled by an orifice plate.

Part-Load Operation and Capacity Control

During part-load operation, a double-effect direct-fired reverse-parallel-flow absorption chiller adjusts its capacity by reducing the heat input to the direct-fired generator through the burner. Lower heat input results at less water vapor boiled off from the solution in the generators. This causes the drop in solution concentration, the amount of water vapor extracted, the rate of evaporation, and the refrigeration capacity. Due to less water vapor being extracted, both evaporating pressure and temperature will rise. Since the amount of water vapor to be condensed is greater than that boiled off from the generators, both the condensing pressure and condensing temperature decrease.

Coefficient of Performance (COP)

The COP of an absorption chiller can be calculated as

$$\text{COP} = 12,000/q_{1g} \quad (9.14.1)$$

where q_{1g} = heat input to the direct-fired generator per ton of refrigeration output (Btu/hr.ton).

Safety Controls

Safety controls in an absorption chiller include the following:

- Crystallization controls are devices available to prevent crystallization and dissolve crystals. Absorption chillers are now designed to operate in a region away from the crystallization line. It is no longer a serious problem in newly developed absorption systems. One such device uses a bypass valve to permit refrigerant to flow to the concentration solution line when crystallization is detected. Condenser water temperature is controlled by using a three-way bypass valve to mix the recirculating water with the evaporated cooled water from the tower to avoid the sudden drop of the temperature of concentrated solution in the absorber.
- Low-temperature cut-out control shuts down the absorption chiller if the temperature of the refrigerant in the evaporator falls below a preset limit to protect the evaporator from freezing.
- Chilled and cooling water flow switches stop the absorption chiller when the mass flow rate of chilled water or the supply of cooling water falls below a preset value.
- A high-pressure relief valve is often installed on the shell of the direct-fired generator to prevent its pressure from exceeding a predetermined value.

- Monitoring of low and high pressure of gas supply and flame ignition are required for direct-fired burner(s).
- Interlocked controls between absorption chiller and chilled water pumps, cooling water pumps, and cooling tower fans are used to guarantee that they are in normal operation before the absorption chiller starts.

Absorption Chiller/Heater

A double-effect direct-fired reverse-parallel-flow absorption chiller/heater has approximately the same system components, construction, and flow process as the absorption chiller. The cooling mode operation is the same as in an absorption chiller. During the heating mode of an absorption chiller/heater, its evaporator becomes the condenser and is used to condense the water vapor that has been boiled off from the direct-fired generator. At design condition, hot water is supplied at a temperature of 130 to 140°F. The condenser and the low-temperature generator are not in operation. (See [Figure 9.14.1](#).)

In order to increase the coefficient of performance of the absorption chiller, a triple-effect cycle with three condensers and three desorbers has been proposed and is under development. A triple-effect absorption chiller is predicted to have a coefficient of performance of around 1.5. Its initial cost is also considerably increased due to a greater number of condensers and desorbers.

9.15 Air-Conditioning Systems and Selection

Basics in Classification

The purpose of classifying air-conditioning or HVAC&R systems is to distinguish one type from another so that an optimum air-conditioning system can be selected according to the requirements. Proper classification of air-conditioning systems also will provide a background for using knowledge-based expert systems to help the designer to select an air-conditioning system and its subsystems.

Since air system characteristics directly affect the space indoor environmental parameters and the indoor air quality, the characteristics of an air system should be clearly designated in the classification.

The system and equipment should be compatible with each other. Each system has its own characteristics which are significantly different from others.

Individual Systems

As described in Section 9.1, air conditioning or HVAC&R systems can be classified as individual, space, packaged, and central systems.

Individual systems usually have no duct and are installed only in rooms that have external walls and external windows. Individual systems can again be subdivided into the following.

Room Air-Conditioner Systems

A room air conditioner is the sole factory-fabricated self-contained equipment used in the room air-conditioning system. It is often mounted on or under the window sill or on a window frame as shown in [Figure 9.1.1](#). A room air-conditioner consists mainly of an indoor coil, a small forward-curved centrifugal fan for indoor coil, a capillary tube, a low-efficiency dry and reusable filter, grilles, a thermostat or other controls located in the indoor compartment, and a rotary, scroll, or reciprocating compressor, an outdoor coil, and a propeller fan for the outdoor coil located in the outdoor compartment. There is an outdoor ventilation air intake opening and a manually operated damper on the casing that divides the indoor and outdoor compartments. Room air-conditioners have a cooling capacity between 1/2 to 2 tons.

The following are system characteristics of a room air-conditioner system:

Room heat pump system is a room air-conditioner plus a four-way reversing valve which provides both the summer cooling and winter heating.

Air system: single supply fan

Fan, motor, and drive combined efficiency: 25%

Fan energy use: 0.3 to 0.4 W/cfm

Fan speed: HI-LO 2-speed or HI-MED-LO 3-speed

Outdoor ventilation air system: type IV

Cooling system: DX system, air-cooled condenser

EER 7.5 to 9.5 Btu/hr.W

Evaporating temperature T_{ev} at design load: typically 45°F

Heating system: electric heating (if any)

Part-load: on-off of refrigeration compressor

Sound level: indoor NC 45 to 50

Maintenance: More maintenance work is required.

Summer and winter mode air-conditioning cycles of a room air-conditioning system are similar to that shown in [Figure 9.3.4](#).

All fan, motor, and drive combined efficiencies for various air-conditioning systems are from data in ASHRAE Standard 90.1-1989.

Packaged Terminal Air-Conditioner (PTAC) Systems

A packaged terminal air-conditioner is the primary equipment in a PTAC system. A PTAC system is similar to a room air-conditioner system. Their main differences are

- A PTAC uses a wall sleeve and is intended to be mounted through the wall.
- Heating is available from hot water, steam, heat pump, electric heater, and sometimes even direct-fired gas heaters.

PTACs are available in cooling capacity between 1/2 to 1 1/2 tons and a heating capacity of 2500 to 35,000 Btu/hr.

Space (Space-Conditioning) Systems

Most space conditioning air-conditioning systems cool, heat, and filtrate their recirculating space air above or in the conditioned space. Space conditioning systems often incorporate heat recovery by transferring the heat rejected from the interior zone to the perimeter zone through the condenser(s). Space systems often have a separate outdoor ventilation air system to supply the required outdoor ventilation air.

Space systems can be subdivided into four-pipe fan-coil systems and water-source heat pump systems.

Four-Pipe Fan-Coil Systems

In a four-pipe fan-coil unit system, space recirculating air is cooled and heated at a fan coil by using four pipes: one chilled water supply, one hot water supply, one chilled water return, and one hot water return. Outdoor ventilation air is conditioned at a make-up AHU or primary AHU. It is then supplied to the fan coil where it mixes with the recirculating air, as shown in [Figure 9.15.1\(a\)](#), or is supplied to the conditioned space directly.

A *fan-coil unit* or a *fan coil* is a *terminal* as shown in [Figure 9.15.1\(b\)](#). Fan-coil units are available in standard sizes 02, 03, 04, 06, 08, 10, and 12 which correspond to 200 cfm, 400 cfm, and so on in volume flow.

The following are system characteristics of a four-pipe fan-coil system:

A *two-pipe fan-coil system* has a supply and a return pipe only. Because of the problems of changeover from chilled water to hot water and vice versa, its applications are limited.

A *water-cooling electric heating fan-coil system* uses chilled water for cooling and an electric heater for heating as shown in [Figure 9.1.2](#). This system is often used in a location that has a mild winter.

Air system:

Fan-coil, space air recirculating

Fan, motor, and drive combined efficiency: 25%

Fan speed: HI-LO 2-speed and HI-MED-LO 3-speed

External pressure for fan coil: 0.06 to 0.2 in. WG

System fan(s) energy use: 0.45 to 0.5 W/cfm

No return air and return air duct

Outdoor ventilation air system: type I

An exhaust system to exhaust part of the outdoor ventilation air

Cooling system: chilled water from centrifugal or screw chiller

Water-cooled chiller energy use: 0.4 to 0.65 kW/ton

Heating system: hot water from boiler, electric heater

Part load: control the flow rate of chilled and hot water supplied to the coil. Since air leaving coil temperature T_{cc} rises during summer mode part load, space relative humidity will be higher.

Sound level: indoor NC 40 to 45

Maintenance: higher maintenance cost

System fan(s) energy use: 0.45 to 0.55 W/cfm (includes all fans in the four-pipe fan-coil system)

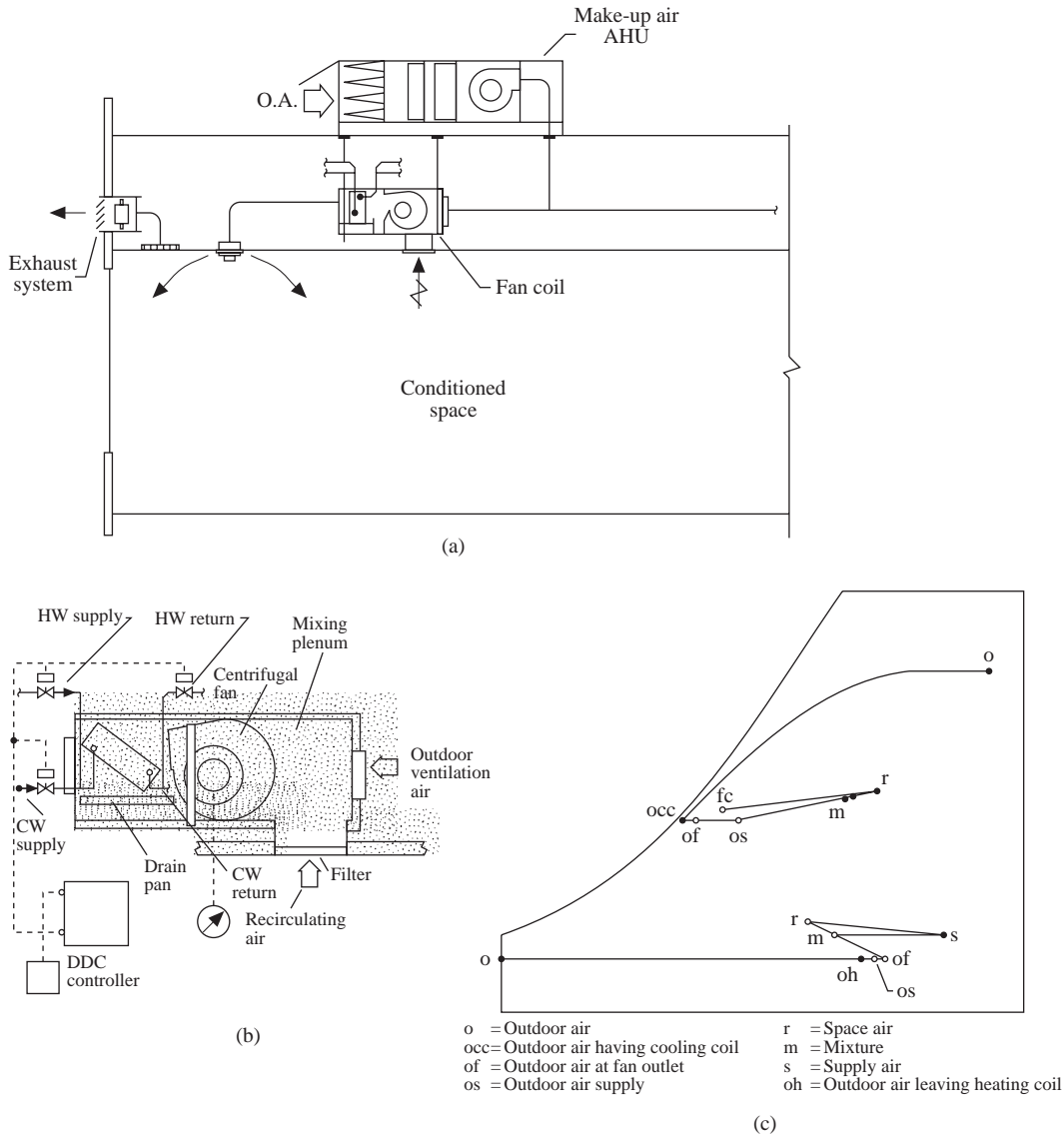


FIGURE 9.15.1 A four-pipe fan-coil system: (a) schematic diagram, (b) fan-coil unit, and (c) air-conditioning cycle.

An air-conditioning cycle for a four-pipe fan-coil system with outdoor ventilation air delivered to the suction side of the fan coil is shown in Figure 9.15.1(c). A part of the space cooling and dehumidifying load is usually taken care by the conditioned outdoor ventilation air from the make-up AHU. A double-bundle condenser is often adopted in a centrifugal chiller to incorporate heat recovery for providing winter heating.

Water-Source Heat Pump Systems

Water-source heat pumps (WSHPs) are the primary equipment in a water-source heat pump system as shown in Figure 9.15.2(a). A *water-source heat pump* usually consists of an air coil to cool and heat the air; a water coil to reject and extract heat from the condenser water; a forward-curved centrifugal fan; reciprocating, rotary, or scroll compressor(s); a short capillary tube; a reversing valve; controls; and an outer casing. WSHPs could be either a horizontal or vertical unit. WSHPs usually have cooling

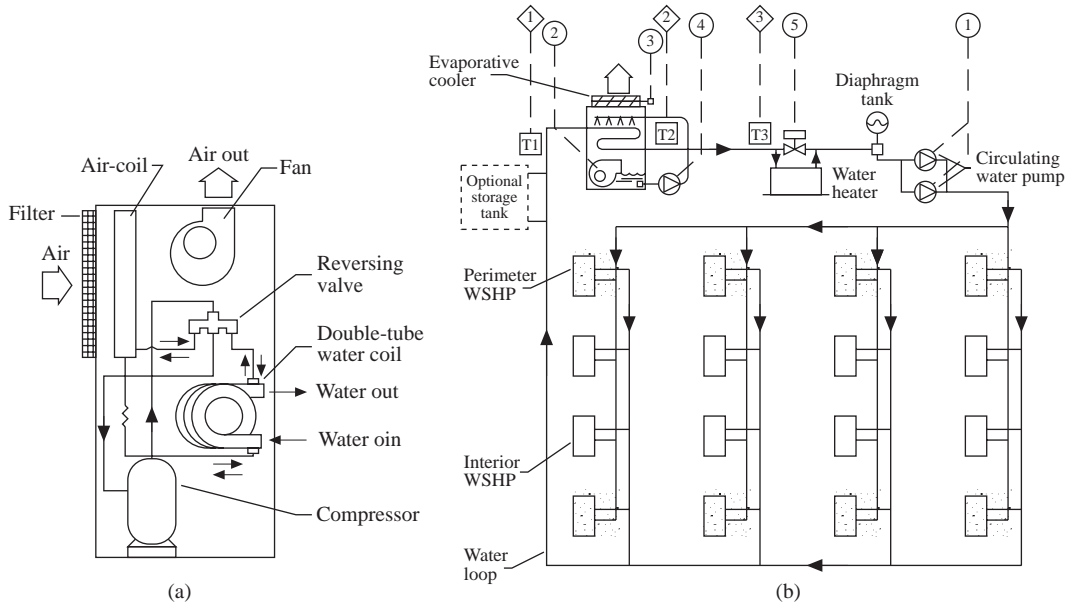


FIGURE 9.15.2 A water-source heat pump system: (a) vertical system and (b) system schematic diagram..

capacities between 1/2 to 26 tons. Small-capacity WSHPs of 3 tons or less without ducts are used in perimeter zones, whereas large-capacity WSHPs with ductwork are used only in interior zones.

In addition to the WSHPs, a WSHP system usually is also comprised of an evaporative cooler or cooling tower to cool the condenser water; a boiler to provide the supplementary heat for the condenser water if necessary; two circulating pumps, one of them being standby; and controls, as shown in Figure 9.15.2(b). A separate outdoor ventilation air system is required to supply outdoor air to the WSHP or directly to the space.

During hot weather, such as outdoor wet bulb at 78°F, all the WSHPs are operated in cooling mode. Condenser water leaves the evaporative cooler at a temperature typically 92°F and absorbs condensing heat rejected from the condensers — the water coils in WSHPs. Condenser water is then raised to 104°F and enters the evaporative cooler. In an evaporative cooler, condenser water is evaporatively cooled indirectly by atmospheric air, so that it would not foul the inner surface of water coils in WSHPs.

During moderate weather, the WSHPs serving the shady side of a building may be in heating mode, and while serving the sunny side of the building and the interior space in cooling mode. During cold weather, most of the WSHPs serving perimeter zones are in heating mode, while serving interior spaces are in cooling mode except morning warm-up. Cooling WSHPs reject heat to the condenser water loop; meanwhile heating WSHPs absorb heat from the condenser water loop. The condenser water is usually maintained at 60 to 90°F. If its temperature rises above 90°F, the evaporative cooler is energized. If it drops below 60°F, the boiler or electric heater is energized. A WSHP system itself is a combination of WSHP and a heat recovery system to transfer the heat from the interior space and sunny side of the building to the perimeter zone and shady side of building for heating in winter, spring, and fall.

System characteristics of air, cooling, and heating in a WSHP system are similar to a room conditioner heat pump system. In addition:

- Outdoor ventilating air system: type I and IV
- Water system: two-pipe, close circuit
- Centrifugal water circulating pump
- Water storage tank is optional

To prevent freezing in locations where outdoor temperature may drop below 32°F, isolate the outdoor portion of the water loop, outdoor evaporative cooler, and the pipe work from the indoor portion by using a plate-and-frame heat exchanger. Add ethylene or propylene glycol to the outdoor water loop for freezing protection.

There is another space system called a panel heating and cooling system. Because of its higher installation cost and dehumidification must be performed in the separate ventilation systems, its applications are very limited.

A space conditioning system has the benefit of a separate demand-based outdoor ventilation air system. A WSHP system incorporates heat recovery automatically. However, its indoor sound level is higher; only a low-efficiency air filter is used for recirculating air, and more space maintenance is required than central and packaged systems. Because of the increase of the minimum outdoor ventilation air rate to 15 cfm/person recently, it may gain more applications in the future.

Packaged Systems

In packaged systems, air is cooled directly by a DX coil and heated by direct-fired gas furnace or electric heater in a packaged unit (PU) instead of chilled and hot water from a central plant in a central system. Packaged systems are different from space conditioning systems since variable-air-volume supply and air economizer could be features in a packaged system. Packaged systems are often used to serve two or more rooms with supply and return ducts instead of serving individual rooms only in an individual system.

As mentioned in Section 9.7, packaged units are divided according to their locations into rooftop, split, or indoor units. Based on their operating characteristics, packaged systems can be subdivided into the following systems:

Single-Zone Constant-Volume (CV) Packaged Systems

Although a single-zone CV packaged system may have duct supplies to and returns from two or more rooms, there is only a single zone sensor located in the representative room or space. A CV system has a constant supply volume flow rate during operation except the undesirable reduction of volume flow due to the increase of pressure drop across the filter.

A single-zone CV packaged system consists mainly of a centrifugal fan, a DX coil, a direct-fired gas furnace or an electric heater, a low or medium efficiency filter, mixing box, dampers, DDC controls, and an outer casing. A relief or a return fan is equipped for larger systems.

A single-zone CV packaged system serving a church is shown in [Figure 9.1.3](#). This system operates on basic air-conditioning cycles as shown in [Figure 9.3.4](#) during cooling and heating modes.

The system characteristics of a single-zone CV packaged system are

Air system: single supply fan, a relief or return fan for a large system

Fan, motor, and drive combined efficiency: 40 to 45%

Fan total pressure: 1.5 to 3 in. WG

Fan(s) energy use: 0.45 to 0.8 W/cfm

Outdoor ventilation air system: type IV and II

Enthalpy or temperature air economizer

Cooling systems: DX system, air cooled

Compressor: reciprocating or scroll

EER: 8.9 to 10.0 Btu/hr.W

Heating system: direct-fired gas furnace, air-source heat pump, or electric heating

Part load: on-off or step control of the compressor capacity, DX-coil effective area, and the gas flow to the burner

Sound level: indoor NC 35 to 45

Maintenance: higher maintenance cost than central systems

Single-zone, CV packaged systems are widely used in residences, small retail stores, and other commercial buildings.

Constant-Volume Zone-Reheat Packaged Systems

System construction and system characteristics of a CV zone-reheat system are similar to the single-zone CV packaged systems except:

1. It serves multizones and has a sensor and a DDC controller for each zone.
2. There is a reheating coil or electric heater in the branch duct for each zone.

A CV zone-reheat packaged system cools and heats simultaneously and therefore wastes energy. It is usually used for the manufacturing process and space needs control of temperature and humidity simultaneously.

Variable-Air-Volume Packaged Systems

A variable-air-volume (VAV) system varies its volume flow rate to match the reduction of space load at part load. A VAV packaged system, also called a *VAV cooling packaged system*, is a multizone system and uses a VAV box in each zone to control the zone temperature at part load during summer cooling mode operation, as shown in [Figure 9.15.3\(a\)](#).

A *VAV box* is a terminal in which the supply volume flow rate of the conditioned supply air is modulated by varying the opening of the air passage by means of a single blade damper, as shown in [Figure 9.15.3\(b\)](#), or a moving disc against a cone-shaped casing.

The following are the system characteristics of a VAV packaged system:

Single-zone VAV packaged system which serves a single zone without VAV boxes. A DDC controller modulates the position of the inlet vanes or the fan speed according to the signal of the space temperature sensor.

Air system: a supply/relief fan or supply/return fan combination. Space pressurization control by a relief/return fan

Fan, motor, and drive combined efficiency: 45%

Supply fan total pressure: 3.75 to 4.5 in. WG

Fan(s) energy use at design condition: 1 to 1.25 W/cfm

VAV box minimum setting: 30% of peak supply volume flow

Outdoor ventilation air system: type II and III

Economizer: enthalpy air economizer or water economizer

Cooling system: DX coil, air-, water-, or evaporative-cooled condenser

Compressor: reciprocating, scroll, and screw

EER: 8.9 to 12 Btu/hr.W

Capacity: 20 to 100 tons

Part load: zone volume flow modulation by VAV box; step control of compressor capacity; modulation of gas flow to burner; and discharge air temperature reset

Smoke control: exhausts smoke on the fire floor, and supplies air and pressurizes the floors immediately above or below the fire floor

Diagnostics: a diagnostic module displays the status and readings of various switches, dampers, sensors, etc. and the operative problems by means of expert system

Maintenance: higher than central system

Sound level: indoor NC 30 to 45

Heating system characteristics as well as the air-conditioning cycles are similar as that in a single-zone CV packaged system.

VAV Reheat Packaged Systems

A VAV reheat packaged system has its system construction and characteristics similar to that in a VAV packaged system except in each VAV box there is an additional reheating coil. Such a VAV box is called

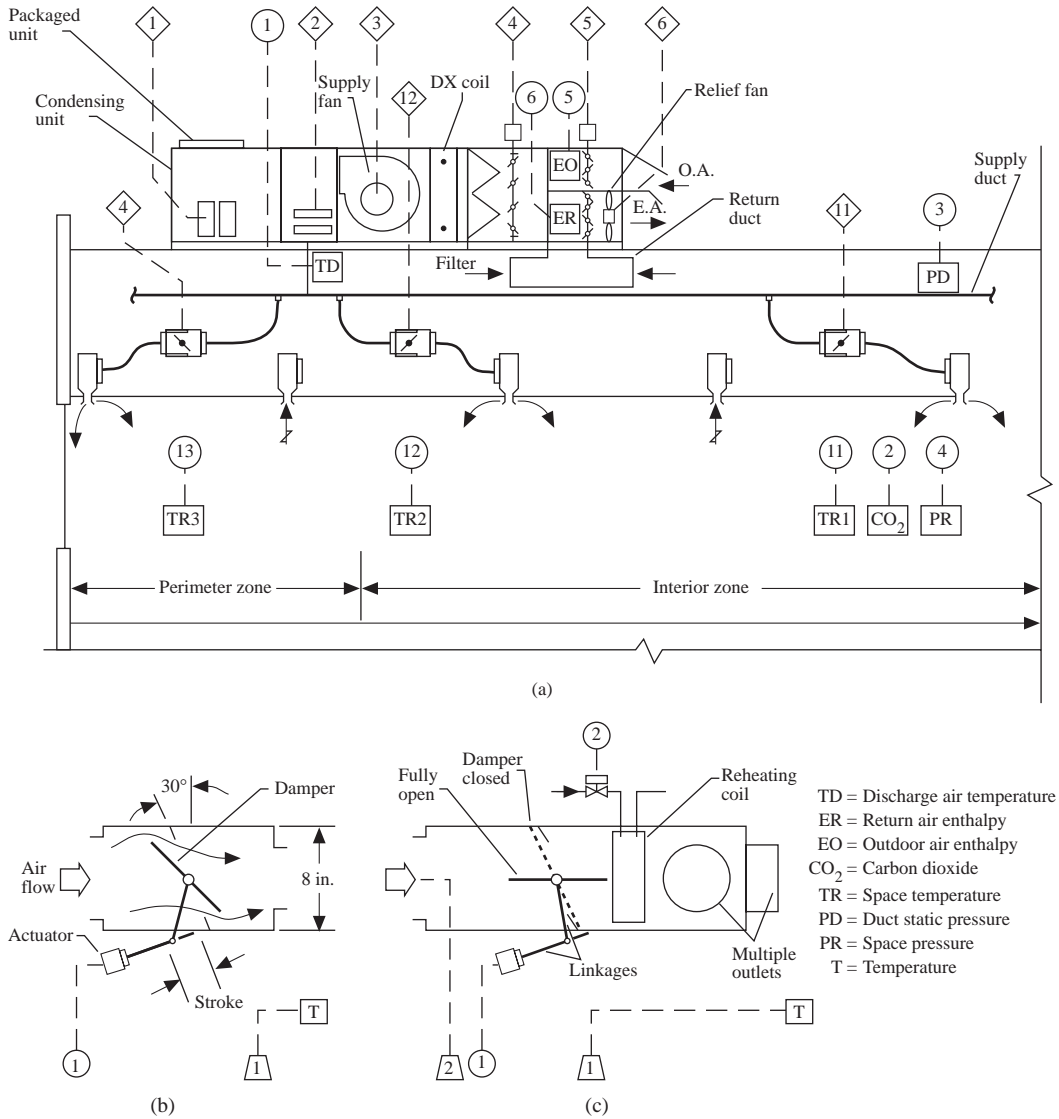


FIGURE 9.15.3 A variable-air-volume (VAV) package system: (a) schematic diagram, (b) VAV box, (c) reheating box, (d) parallel-flow fan-powered VAV box.

a *reheating VAV box*, as shown in [Figure 9.15.2\(a\)](#) and [9.15.3\(c\)](#). VAV reheat packaged systems are used to serve perimeter zones where winter heating is required.

Fan-Powered VAV Packaged Systems

A fan-powered VAV packaged system is similar to that of a VAV packaged system except *fan-powered VAV boxes* as shown in [Figure 9.15.3\(d\)](#) are used instead of VAV boxes.

There are two types of fan-powered VAV boxes: parallel-flow and series-flow boxes. In a *parallel-flow* fan-powered box, the plenum air flow induced by the fan is parallel with the cold primary air flow through the VAV box. These two air streams are then combined and mixed together. In a *series-flow* box, cold primarily from the VAV box is mixed with the induced plenum air and then flows through the small fan. The parallel-flow fan-powered VAV box is more widely used.

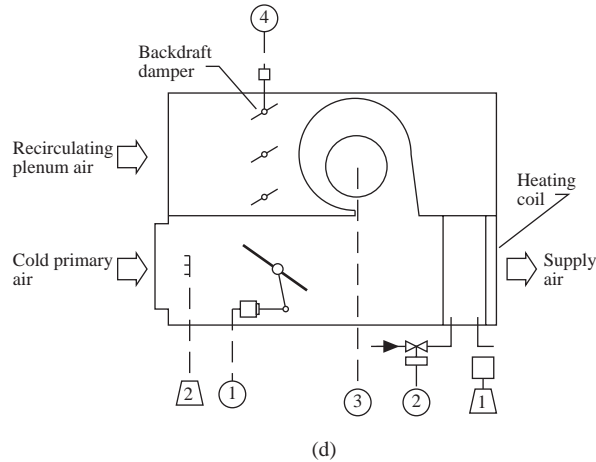


FIGURE 9.15.3d

In a fan-powered VAV box, volume flow dropping to minimum setting, extracting of ceiling plenum air, and energizing of reheating coil will actuate in sequence to maintain the space temperature during part-load/heating mode operation. A fan-powered VAV box can also mix the cold primary air from cold air distribution with the ceiling plenum air and provides greater space air movements during minimum space load.

Packaged systems are lower in installation cost and occupy less space than central systems. During the past two decades, DDC-controlled packaged systems have evolved into sophisticated equipment and provide many features that only a built-up central system could provide before.

Central Systems

Central systems use chilled and hot water that comes from the central plant to cool and heat the air in the air-handling units (AHUs). Central systems are built-up systems. The most clean, most quiet thermal-storage systems, and the systems which offer the most sophisticated features, are always central systems. Central systems can be subdivided into the following.

Single-Zone Constant-Volume Central Systems

A single-zone CV central system uses a single controller to control the flow of chilled water, hot water, or the opening of dampers to maintain a predetermined indoor temperature, relative humidity, or air contaminants. They are often used in manufacturing factories. The system characteristics of a single-zone CV central system are

Single-zone CV air washer central system uses air washer to control both space relative humidity and temperature. This system is widely used in textile mills. The reason to use constant volume is to dilute the fiber dusts produced during manufacturing. A rotary filter is often used for high dust-collecting capacity.

Air system: supply and return fan combination

Fan, motor, and drive combined efficiency: 45 to 50%

Outdoor ventilation air system: type II and IV

Economizer: air or water economizer

Smoke control: exhaust smoke on the fire floor, and pressurize adjacent floor(s) or area

Cooling system: centrifugal or screw chiller, water-cooled condenser

Cooling energy use: 0.4 to 0.65 kW/ton

Heating system: hot water from boiler or from heat recovery system

Part load: modulate the water mass flow to cooling and heating coils in AHUs, and discharge air temperature reset

Sound level: indoor NC 30 to 45. Silencers are used both in supply and return air systems if they are required

Maintenance: in central plant and fan rooms, lower maintenance cost

Single-Zone CV Clean Room Systems

This is the central system which controls the air cleanliness, temperature, and relative humidity in Class 1, 10, 100, 1000, and 10,000 clean rooms for electronic, pharmaceutical, and precision manufacturing and other industries. [Figure 9.15.4\(a\)](#) shows a schematic diagram of this system. The recirculating air unit (RAU) uses prefilter, HEPA filters, and a water cooling coil to control the space air cleanliness and required space temperature, whereas a make-up air unit (MAU) supplies conditioned outdoor air, always within narrow dew point limits to the RAU at any outside climate, as shown in [Figure 9.15.4\(b\)](#). A unidirectional air flow of 90 fpm is maintained at the working area. For details, refer to *ASHRAE Handbook 1991 HVAC Applications* and Wang's *Handbook of Air Conditioning and Refrigeration*.

CV Zone-Reheat Central Systems

These systems have their system construction and characteristics similar to that for a single-zone CV central system, except they serve multizone spaces and there is a reheating coil, hot water, or electric heating in each zone. CV zone-reheat central systems are often used for health care facilities and in industrial applications.

VAV Central Systems

A VAV central system is used to serve multizone space and is also called *VAV cooling central system*. Its schematic diagram is similar to that of a VAV packaged system ([Figure 9.15.3](#)) except air will be cooled or heated by water cooling or heating coils in the AHUs. The same VAV box shown in [Figure 9.15.3\(b\)](#) will be used in a VAV central system. The system characteristics of VAV central systems are as follows:

Single-zone VAV central system differs from a VAV central system only because it serves a single zone, and therefore there is no VAV box in the branch ducts. Supply volume flow is modulated by inlet vanes and AC inverter.

Air system: supply and relief/return fan combination

Fan, motor, and drive combined efficiency for airfoil centrifugal fan with AC inverter fan speed modulation: 55%

Fan(s) energy use: 0.9 to 1.2 W/cfm

VAV box: minimum setting 30% of peak supply volume flow

Outdoor ventilation air system: type I, II, and III

Cooling system: centrifugal, screw, and reciprocating chillers, water-cooled condenser, with energy use 0.4 to 0.65 kW/ton; or sometimes absorption chiller

Heating system: hot water from boiler or electric heating at the terminals

Economizer: air and water economizer

Part load: zone volume flow modulation, discharge air temperature reset, and chilled water temperature reset

Smoke control: exhausts smoke from the fire floor and pressurizes the immediate floors above and below

Sound level: indoor NC 20 to 40. Silencers are often used both in supply and return systems.

Maintenance: less space maintenance

VAV central systems are widely used for interior zone in buildings.

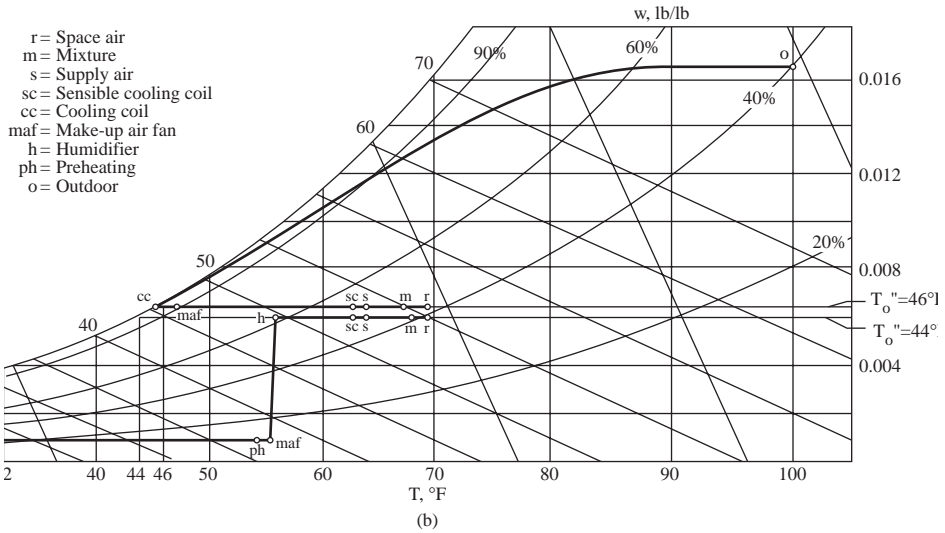
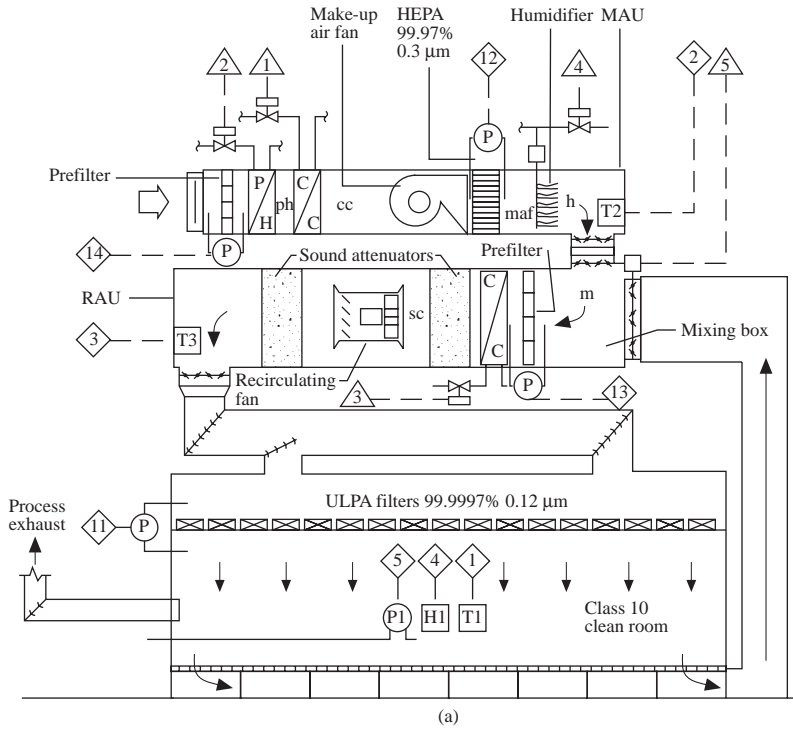


FIGURE 9.15.4 A single-zone CV clean room system: (a) schematic diagram and (b) air-conditioning cycle.

VAV Reheat Central Systems

A VAV reheat system is similar in system construction and characteristics to that in a VAV central system except that reheating boxes are used instead of VAV boxes in a VAV central system.

Fan-Powered VAV Central Systems

A fan-powered VAV central system is similar in system construction and characteristics to that in a VAV central system except that fan-powered VAV boxes are used instead of VAV boxes.

Dual-Duct VAV Central Systems

A dual-duct VAV system uses a warm air duct and a cold air duct to supply both warm and cold air to each zone in a multizone space, as shown in [Figure 9.15.5\(a\)](#). Warm and cold air are first mixed in a mixing VAV box, and are then supplied to the conditioned space. Warm air duct is only used for perimeter zones.

A *mixing VAV box* consists of two equal air passages, one for warm air and one for cold air, arranged in parallel. Each of them has a single blade damper and its volume flow rate is modulated. Warm and cold air are then combined, mixed, and supplied to the space.

A dual-duct VAV system is usually either a single supply fan and a relief/return fan combination, or a warm air supply fan, a cold air supply fan, and a relief/return fan. A separate warm air fan and cold air supply fan are beneficial in discharge air temperature reset and fan energy use.

During summer cooling mode operation, the mixture of recirculating air and outdoor air is used as the warm air supply. The heating coil is not energized. During winter heating mode operation, mixture of outdoor and recirculating air or 100% outdoor air is used as the cold air supply; the cooling coil is not energized.

Because there is often air leakage at the dampers in the mixing VAV box, more cold air supply is needed to compensate for the leaked warm air or leaked cold air.

Other system characteristics of a dual-duct VAV central system are similar to a VAV central system.

Dual-Duct CV Central System

This is another version of a dual-duct VAV central system and is similar in construction to a dual-duct VAV system, except that a mixing box is used instead of a mixing VAV box. The supply volume flow rate from a mixing box is nearly constant. Dual-duct CV central systems have only limited applications, like health care facilities, etc.

An ice- or chilled-water storage system is always a central system plus a thermal storage system. The thermal storage system does not influence the system characteristics of the air distribution, and air cooling and heating — except for a greater head lift for a refrigeration compressor — is needed for ice-storage systems. Therefore, the following central systems should be added:

- VAV ice-storage or chilled-water systems
- VAV reheat ice-storage or chilled-water storage systems
- Fan-powered VAV ice-storage systems

Some of the air-conditioning systems are not listed because they are not effective or are a waste of energy, and therefore rarely used in new and retrofit projects such as:

- High-velocity induction space conditioning systems which need a higher pressure drop primary air to induce recirculating air in the induction unit and use more energy than fan-coil systems
- Multizone central systems which mix warm and cool air at the fan room and use a supply duct from fan room to each control zone
- Air skin central systems which use a warm air heating system to offset transmission loss in the perimeter zone and overlook the effect of the solar radiation from variation building orientations

In the future, there will be newly developed systems added to this classification list.

Air-Conditioning System Selection

As described in Section 9.1, the goal of an air-conditioning or HVAC&R system is to provide a healthy, comfortable, manufacturable indoor environment at acceptable indoor air quality, keeping the system

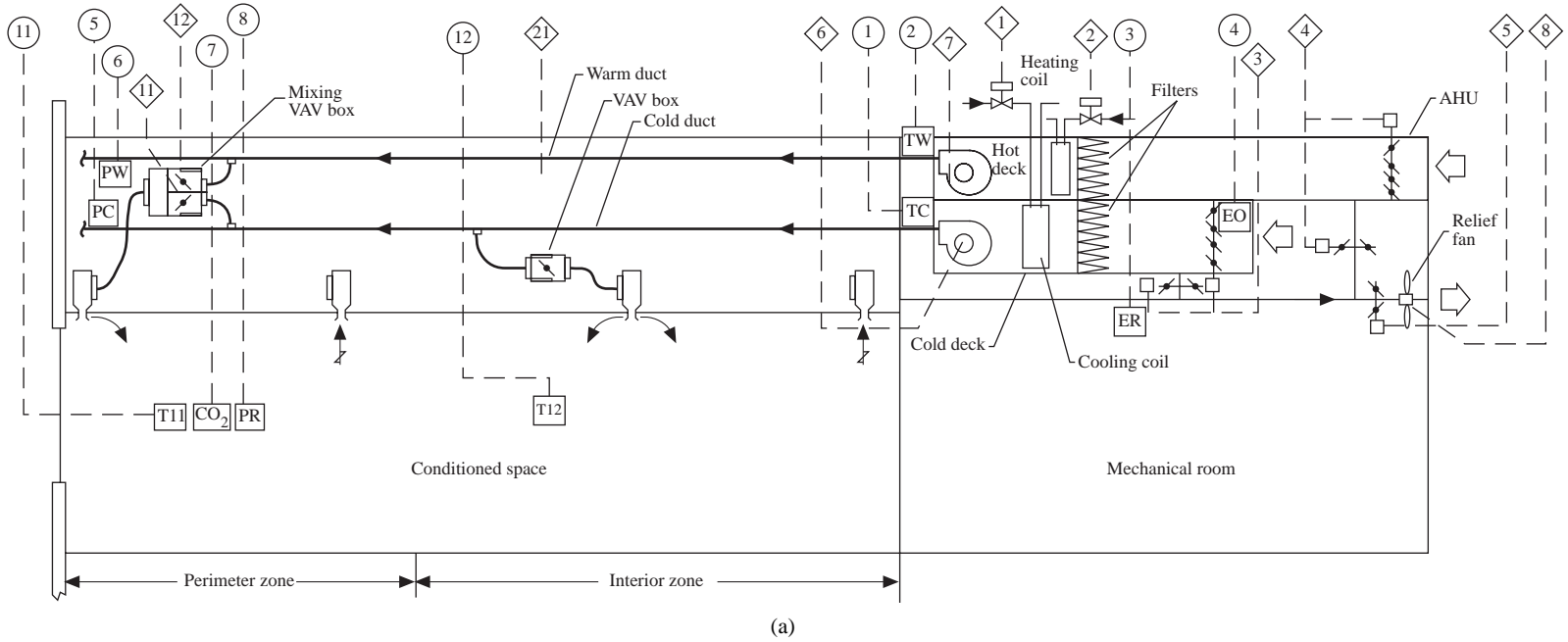


FIGURE 9.15.5 A dual-duct VAV central system: (a) schematic diagram.

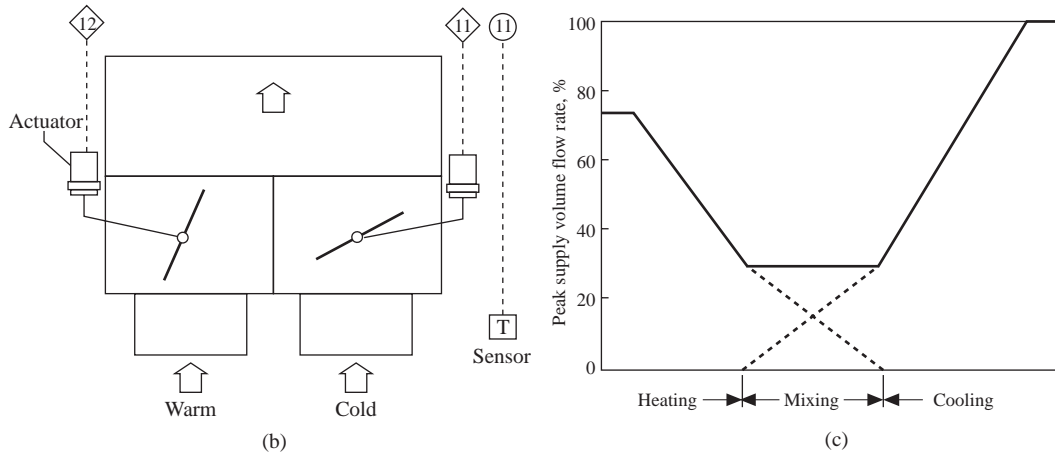


FIGURE 9.15.5 A dual-duct VAV central system: (b) mixing VAV box and (c) volume flow-operating mode diagram.

energy efficient. Various occupancies have their own requirements for their indoor environment. The basic considerations to select an air-conditioning system include:

1. The selection of an air-conditioning system must satisfy the required space temperature, relative humidity, air cleanliness, sound level, and pressurization. For a Class 100 clean room, a single-zone CV clean room system is always selected. A four-pipe fan-coil space conditioning system is usually considered suitable for guest rooms in hotels for operative convenience, better privacy, and a guaranteed outdoor ventilation air system. A concert hall needs a very quiet single-zone VAV central system for its main hall and balcony.
2. The size of the project has a considerable influence on the selection. For a small-size residential air-conditioning system, a single-zone constant-volume packaged system is often the first choice.
3. Energy-efficient measures are specified by local codes. Comparison of alternatives by annual energy-use computer programs for medium and large projects is often necessary. Selection of energy source includes electricity or gas, and also using electrical energy at off-peak hours, like thermal storage systems is important to achieve minimum energy cost.

For a building whose sound level requirement is not critical and conditioned space is comprised of both perimeter and interior zones, a WSHP system incorporating heat recovery is especially suitable for energy saving.

4. First cost or investment is another critical factor that often determines the selection.
5. Selection of an air-conditioning system is the result of synthetical assessment. It is difficult to combine the effect of comfort, reliability, safety, and cost. Experience and detailed computer program comparisons are both important.

The selection procedure usually begins whether an individual, space conditioning, packaged, central system, or CV, VAV, VAV reheat, fan-powered VAV, dual-duct VAV, or thermal storage system is selected. Then the air, refrigeration, heating, and control subsystems will be determined. After that, choose the option, the feature, the construction, etc. in each subsystem.

Comparison of Various Systems

The sequential order of system performance — excellent, very good, good, satisfactory — regarding temperature and relative humidity control (T&HC), outdoor ventilation air (OA), sound level, energy use, first cost, and maintenance for individual, space conditioning (SC), packaged, and central systems is as follows:

	Excellent (low or less)	Very good	Good	Satisfactory
T&HC	Central	Packaged	Space	Individual
IAQ	Space	Central	Packaged	Individual
Sound	Central	Packaged	Space	Individual
Energy use	Individual	Space	Packaged	Central
First cost	Individual	Packaged	Space	Central
Maintenance	Central	Packaged	Space	Individual

Among the packaged and central systems, VAV cooling systems are used only for interior zones. VAV reheat, fan-powered VAV, and dual-duct VAV central systems are all for perimeter zones. VAV reheat systems are simple and effective, but have a certain degree of simultaneous cooling and heating when their volume flow has been reduced to minimum setting. Fan-powered VAV systems have the function of mixing cold primary air with ceiling plenum air. They are widely used in ice-storage systems with cold air distribution. Fan-powered VAV is also helpful to create a greater air movement at minimum cold primary air flow. Dual-duct VAV systems are effective and more flexible in operation. They are also more complicated and expensive.

Subsystems

Air Systems

The economical size of an air system is often 10,000 to 25,000 cfm. A very large air system always has higher duct pressure loss and is more difficult to balance. For highrise buildings of four stories and higher, floor-by-floor AHU(s) or PU(s) (one or more AHU or PU per floor) are often adopted. Such an arrangement is beneficial for the balance of supply and return volume flow in VAV systems and also for fire protection. A fan-powered VAV system using a riser to supply less cold primary to the fan-powered VAV box at various floors may have a larger air system. Its risers can be used as supply and exhaust ducts for a smoke-control system during a building fire.

In air systems, constant-volume systems are widely used in small systems or to dilute air contaminants in health care facilities and manufacturing applications. VAV systems save fan energy and have better operating characteristics. They are widely used in commercial buildings and in many factories.

Refrigeration Systems

For comfort air-conditioning systems, the amounts of required cooling capacity and energy saving are dominant factors in the selection of the refrigeration system. For packaged systems having cooling capacity less than 100 tons, reciprocating and scroll vapor compression systems with air-cooled condensers are most widely used. Evaporative-cooled condensers are available in many packaged units manufactured for their lower energy use. Scroll compressors are gradually replacing the reciprocating compressors for their simple construction and energy saving. For chillers of cooling capacity of 100 tons and greater, centrifugal chillers are still most widely used for effective operation, reliability, and energy efficiency. Screw chillers have become more popular in many applications, especially for ice-storage systems.

Heating Systems

For locations where there is a cold and long winter, a perimeter baseboard hot water heating system or dual-duct VAV systems are often a suitable choice. For perimeter zones in locations where winter is mild, winter heating is often provided by using warm air supply from AHU or PU from terminals with electric or hot water heaters. Direct-fired furnace warm air supply may be used for morning warm-up. For interior or conditioned zones, a cold air supply during occupied periods in winter and a warm air supply from the PUs or AHUs during morning warm-up period is often used.

Control Systems

Today, DDC microprocessor-based control with open data communication protocol is often the choice for medium- and large-size HVAC&R projects. For each of the air, cooling, and heating systems, carefully select the required generic and specific control systems. If a simple control system and a more complicated control system can provide the same required results, the simple one is always the choice.

Energy Conservation Recommendations

1. Turn off electric lights, personal computers, and office appliances when they are not needed. Shut down AHUs, PUs, fan coils, VAV boxes, compressors, fans, and pumps when the space or zone they serve is not occupied or working.
2. Provide optimum start and stop for the AHUs and PUs and terminals daily.
3. Temperature set point should be at its optimum value. For comfort systems, provide a dead band between summer and winter mode operation. Temperature of discharged air from the AHU or PU and chilled water leaving the chiller should be reset according to space or outdoor temperature or the system load.
4. Reduce air leakages from ducts and dampers. Reduce the number of duct fittings and pipe fittings and their pressure loss along the design path if this does not affect the effectiveness of the duct system. The maximum design velocity in ducts for comfort systems should not exceed 3000 fpm, except that a still higher velocity is extremely necessary.
5. Adopt first the energy-efficient cooling methods: air and water economizer, evaporative cooler, or ground water instead of refrigeration.
6. Use cost-effective high-efficiency compressors, fans, pumps, and motors as well as evaporative-cooled condensers in PUs. Use adjustable-frequency fan speed modulation for large centrifugal fans. Equipment should be properly sized. Over-sized equipment will not be energy efficient.
7. Use heat recovery systems and waste heat for winter heating or reheating. Use a heat-pump system whenever its COP_{hp} is greater than 1.
8. For medium- and large-size air-conditioning systems, use VAV systems instead of CV systems except for health care or applications where dilution of air contaminant is needed. Use variable flow for building-loop and distribution-loop water systems.
9. Use double- and triple-pane windows with low emissive coatings. Construct low U-value roofs and external walls.

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9.16 Desiccant Dehumidification and Air-Conditioning

Zalman Lavan

Introduction

Desiccant air-conditioning is a promising emerging technology to supplement electrically driven vapor compression systems that rely almost exclusively on R22 refrigerant that causes depletion of the ozone layer. To date, this technology has only a limited market, e.g., in supermarkets where the latent heat loads are very high, in specialized manufacturing facilities that require very dry air, and in hospitals where maximum clean air is required. However, recent emphasis on increased air change requirements (see ASHRAE standards, ANSI 62-1989), improved indoor air quality, and restriction on use of CFC refrigerants (see The Montreal Protocol Agreement, as amended in Copenhagen in 1992, United Nations Environmental Programme, 1992) may stimulate wider penetration of desiccant-based air-conditioning which can be used as stand-alone systems or in combination with conventional systems. (See Table 9.4.1 for properties of some refrigerants.)

Sorbents and Desiccants

Sorbents are materials which attract and hold certain vapor or liquid substances. The process is referred to **absorption** if a chemical change takes place and as **adsorption** if no chemical change occurs. **Desiccants**, in both liquid and solid forms, are a subset of sorbents that have a high affinity to water molecules. Liquid desiccants *absorb* water molecules, while solid desiccants *adsorb* water molecules and hold them on their vast surfaces (specific surface areas are typically hundreds of square meters per gram).

While desiccants can sorb water in both liquid and vapor forms, the present discussion is limited to **sorption** of water vapor from adjacent air streams. The sorption driving force for both liquid and solid desiccants is a vapor pressure gradient. Adsorption (in solid desiccants) and absorption (in liquid desiccants) occur when the water vapor partial pressure of the surrounding air is larger than that at the desiccant surface. When an air stream is brought in contact with a desiccant, water vapor from the air is attracted by the desiccant, the air is **dehumidified**, and the water content of the desiccant rises. As the water sorbed by the desiccant increases, the sorption rate decreases and finally stops when *sorption equilibrium* is reached. For dehumidification to be resumed, water must be removed from the desiccant by heating. This process is referred to as **desorption**, **reactivation**, or **regeneration**. The heat of sorption (or desorption) is generally higher than the latent heat of vaporization of water; it approaches the latter as sorption equilibrium is reached.

Some typical *liquid desiccants* are water solutions of calcium chloride (CaCl), lithium chloride (LiCl), lithium bromide (LiBr), and triethylene glycol. The equilibrium water vapor pressure at the solution surface as a function of temperature and water content is shown in Figure 9.16.1 for water-lithium chloride solution. The surface vapor pressure (and dew point) increases with increasing solution temperature and decreases with increasing moisture content.

Common *solid desiccants* are silica gel, molecular sieves (zeolites), activated alumina, and activated carbon. The equilibrium sorption capacity (or moisture content) at a constant temperature, referred to as an **isotherm**, is usually presented as percent water (mass of water divided by mass of dry desiccant) vs. percent relative humidity (vapor pressure divided by saturation vapor pressure). Sorption capacity decreases with increasing temperature, but the spread of isotherms is relatively small (especially for concave down isotherms). Figure 9.16.2 shows normalized loading (sorption capacity divided by sorption capacity at 100% relative humidity) vs. relative humidity for silica gel, molecular sieve, and a generic desiccant, type 1 (modified) or simply 1-M (Collier et al., 1986).

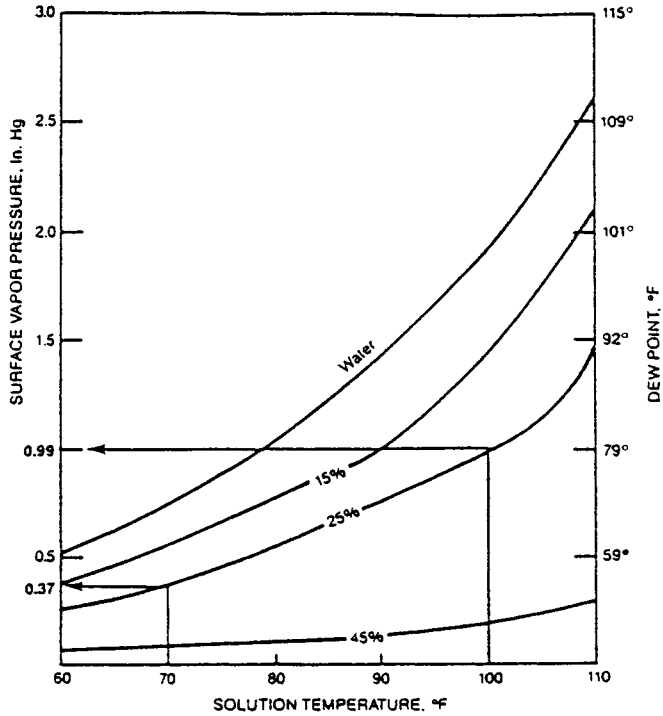


FIGURE 9.16.1 Surface vapor pressure of water-lithium chloride solutions. (Source: ASHRAE 1993, *Fundamentals Handbook*, chap. 19. With permission.)

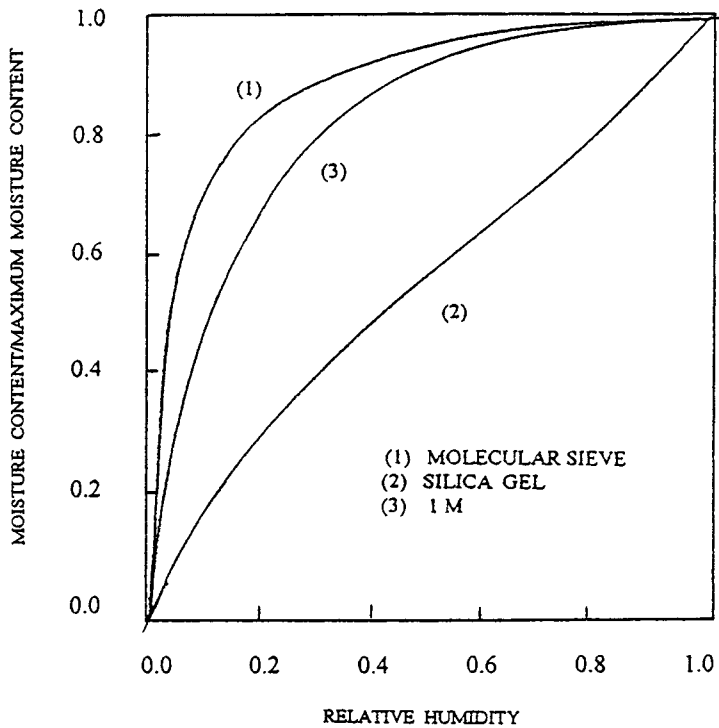


FIGURE 9.16.2 Normalized solid desiccant isotherms.

Dehumidification

Dehumidification by vapor compression systems is accomplished by cooling the air below the dew point and then reheating it. The performance is greatly hindered when the desired outlet dew point is below 40°F due to frost formation on the cooling coils (*ASHRAE, Systems and Equipment Handbook, 1992*).

Desiccant dehumidification is accomplished by direct exchange of water vapor between an air stream and a desiccant material due to water vapor pressure difference. [Figure 9.16.3](#) shows the cyclic operation of a desiccant dehumidification system.

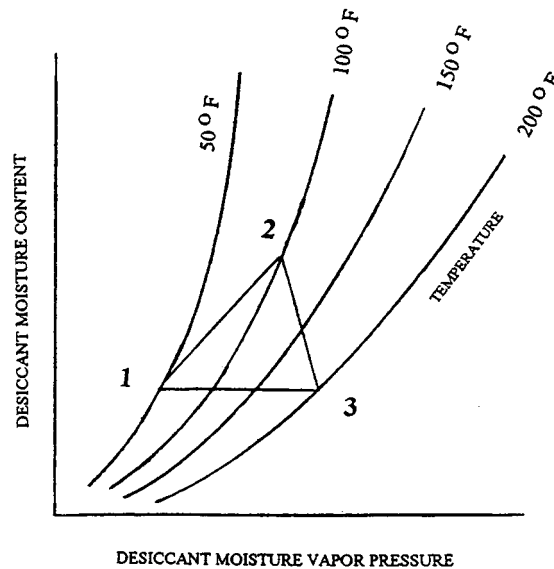


FIGURE 9.16.3 Cyclic dehumidification processes.

In *sorption* (1–2), dry and cold desiccant (point 1) sorbs moisture since the vapor pressure at the surface is lower than that of the air stream. During this process the moisture content (loading or uptake) increases, the surface vapor pressure increases, and the liberated heat of sorption raises the desiccant temperature. During *desorption* (2–3), the desiccant is subjected to a hot air stream, and moisture is removed and transferred to the surrounding air. The surface vapor pressure is increased and the desiccant temperature rises due to the added heat. The cycle is closed by *cooling* (3–1). The desiccant is cooled while its moisture content is constant and the surface vapor pressure is lowered. The above cycle of sorption, desorption, and cooling can be modified by combining the sorption process with cooling to approach isothermal rather than adiabatic sorption.

Desirable Characteristics for High-Performance Liquid and Solid Desiccant Dehumidifiers

- High equilibrium moisture sorption capacity
- High heat and mass transfer rates
- Low heat input for regeneration
- Low pressure drop
- Large contact transfer surface area per unit volume
- Compatible desiccant/contact materials
- Inexpensive materials and manufacturing techniques
- Minimum deterioration and maintenance

Additional Requirements for Liquid Desiccant Dehumidifiers

- Small liquid side resistance to moisture diffusion

Minimum crystallization

Additional Requirements for Solid Desiccant Dehumidifiers

The desiccant should not deliquesce even at 100% relative humidity.

The airflow channels should be uniform.

The desiccant should be bonded well to the matrix.

The material should not be carcinogenic or combustible.

Liquid Spray Tower

Figure 9.16.4 is a schematic of a liquid spray tower. A desiccant solution from the sump is continuously sprayed downward in the absorber, while air, the process stream, moves upward. The air is dehumidified and the desiccant solution absorbs moisture and is weakened. In order to maintain the desired solution concentration, a fraction of the solution from the sump is passed through the regenerator, where it is heated by the heating coil and gives up moisture to the desorbing air stream. The strong, concentrated solution is then returned to the sump. The heat liberated in the absorber during dehumidification is removed by the cooling coil to facilitate continuous absorption (see Figures 9.16.1 and 9.16.3). The process air stream exits at a relatively low temperature. If sufficiently low water temperature is available (an underground well, for example), the process stream could provide both sensible and latent cooling.

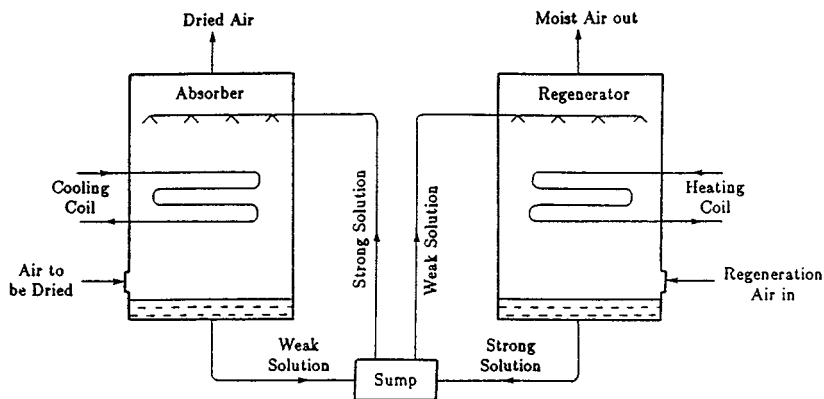


FIGURE 9.16.4 Liquid desiccant dehumidifier with heating and cooling coils.

The heating and cooling coils, shown in Figure 9.16.4, are often eliminated and the liquid solutions are passed through heating and cooling heat exchangers before entering the spray towers.

Advantages

The system is controlled to deliver the desired level of dry air by adjusting the solution concentration.

Uniform exit process stream conditions can be maintained.

A concentrated solution can be economically stored for subsequent drying use.

The system can serve as a humidifier when required by simply weakening the solution.

When used in conjunction with conventional A/C systems, humidity control is improved and energy is conserved.

Disadvantages

Some desiccants are corrosive.

Response time is relatively large.

Maintenance can be extensive.

Crystallization may be a problem.

Solid Packed Tower

The dehumidification system, shown in [Figure 9.16.5](#), consists of two side-by-side cylindrical containers filled with solid desiccant and a heat exchanger acting as a desiccant cooler. The air stream to be processed is passed through dry desiccant in one of the containers, while a heated air stream is passed over the moist desiccant in the other. Adsorption (1–2) takes place in the first container, desorption (2–3) in the other container, and cooling (3–1) occurs in the desiccant cooler. The function of the two containers is periodically switched by redirecting the two air streams.

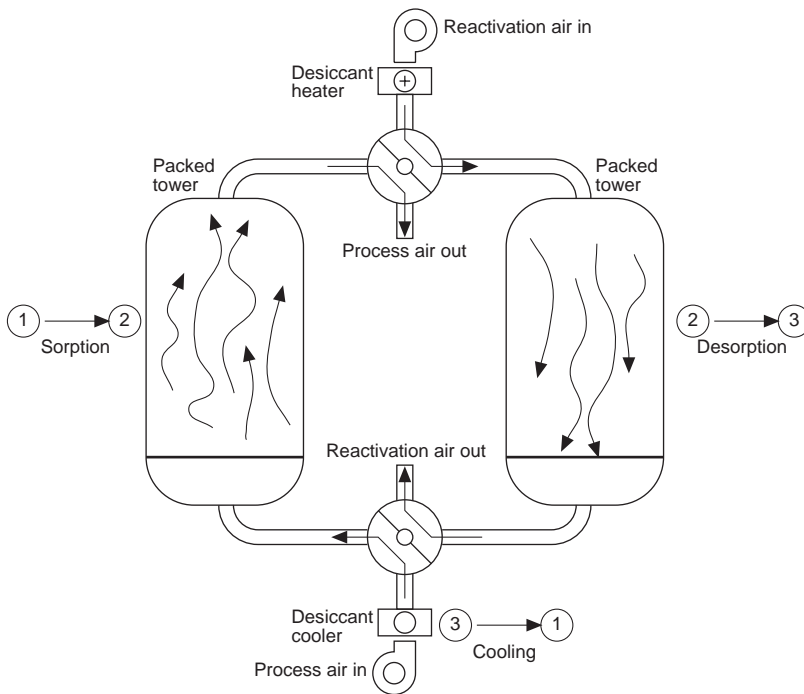


FIGURE 9.16.5 Solid packed tower dehumidification.

Advantages

- No corrosion or crystallization
- Low maintenance
- Very low dew point can be achieved

Disadvantages

- The air flow velocity must be low in order to maintain uniform velocity through the containers and to avoid dusting.
- Uniform exit process stream dew point cannot be maintained due to changing moisture content in the adsorbing desiccant.

Rotary Desiccant Dehumidifiers

A typical rotary solid desiccant dehumidifier is shown in [Figure 9.16.6](#). Unlike the intermittent operation of packed towers, rotary desiccant dehumidifiers use a wheel (or drum) that rotates continuously and delivers air at constant humidity levels.

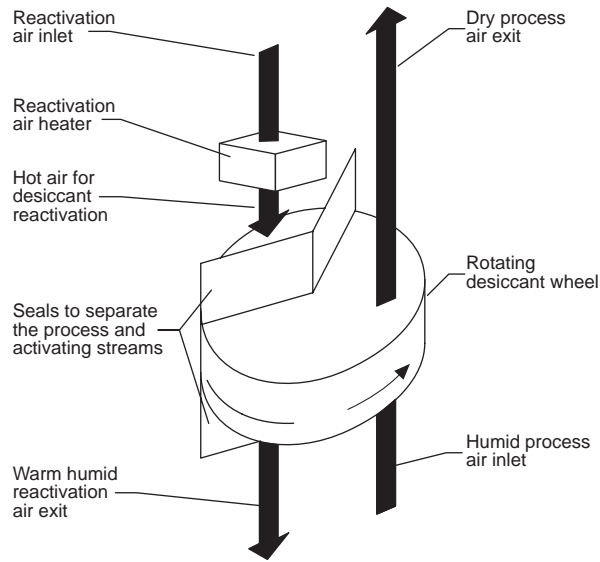


FIGURE 9.16.6 Rotary desiccant dehumidification wheel.

Desiccant wheels typically consist of very fine desiccant particles dispersed and impregnated with a fibrous or ceramic medium shaped like a honeycomb or fluted corrugated paper. The wheel is divided into two segments. The process stream flows through the channels in one segment, while the regenerating (or reactivating) stream flows through the other segment.

Desiccant Material

The desired desiccant properties for optimum dehumidification performance are a suitable isotherm shape and a large moisture sorption capacity. The isotherms of silica gel are almost linear. The moisture sorption capacity is high; the desiccant is reactivated at relatively low temperatures and is suitable for moderate dehumidification. Molecular sieves have very steep isotherms at low relative humidity. The desiccant is reactivated at relatively high temperatures and is used for deep dehumidification. The isotherm of the type 1-M yields optimum dehumidification performance (Collier et al., 1986), especially when used in conjunction with high regeneration temperatures.

The Desiccant Wheel

Some considerations for selection of desiccant wheels are:

- Appropriate desiccant materials
- Large desiccant content
- Wheel depth and flute size (for large contact surface area and low pressure drop)
- Size and cost

The actual performance depends on several additional factors that must be addressed. These include:

- Inlet process air temperature and humidity
- Desired exit process air humidity
- Inlet reactivating air temperature and humidity
- Face velocity of the two air streams
- Size of reactivation segment

It should be noted that:

Higher inlet process air humidity results in higher exit humidity and temperature (more heat of sorption is released).

Lower face velocity of the process stream results in lower exit humidity and higher temperature.

Higher regeneration temperatures result in deeper drying, hence lower exit process air humidity and higher temperature.

When lower exit air temperature is required, the exit process air should be cooled by a heat exchanger. Final cooling of the exit process air can be achieved by partial humidification (this counteracts in part previous dehumidification).

The following is a range of typical parameters for rotary desiccant wheels:

Rotation speed: 4 to 10 rpm

Desiccant fraction: 70 to 80%

Flute size: 1 to 2 mm

Reactivation segment: 25 to 30% of wheel

Face velocity: 300 to 700 fpm

Reactivating temperature: 100 to 300°F

Hybrid Cycles

A limited number of hybrid systems consisting of desiccant dehumidifiers and electrically driven vapor compression air-conditioners are presently in use in supermarkets. This application is uniquely suited for this purpose since the latent heat loads are high due to the large number of people and frequent traffic through doors. Also, low relative humidity air is advantageous for open-case displays.

Vapor compression systems are inefficient below a dew point of 45 to 50°F. When used in supermarkets, they require high airflow rates, the air must be reheated for comfort, and the evaporator coils must be defrosted frequently. Hybrid systems offer improved performance and lower energy cost in these cases.

Figure 9.16.7 shows a typical hybrid air-conditioning system for supermarkets. A mixture of outdoor and recirculated air is first passed through the desiccant and sensible heat exchanger wheels, where it is dehumidified and precooled. It then enters the conventional chiller before it is introduced to the interior of the supermarket. The sensible heat exchanger wheel is cooled by outdoor air and the desiccant wheel is regenerated by air heated with natural gas. Energy cost can be further reduced by preheating the reactivating air stream with waste heat rejected from the condenser of the refrigeration and/or air-conditioning systems.

The advantages of these hybrid systems are

Air-conditioning requirement is reduced by up to 20%.

The vapor compression system operates at a higher coefficient of performance (COP) since the evaporator coils are at a higher temperature.

Airflow requirements are reduced; electric fan energy is saved and duct sizes are reduced.

The refrigeration cases run more efficiently since the frequency of defrost cycles is greatly reduced.

Solid Desiccant Air-Conditioning

Several stand-alone desiccant air-conditioning systems were suggested and extensively studied. These systems consist of a desiccant wheel, a sensible heat exchanger wheel, and evaporating pads. Sorption can be adiabatic or cooled (if cooling is combined with sorption). When room air is dehumidified and recirculated, the system is said to operate in the **recirculating** mode. When 100% outside air is used as the process stream, the system operates in the **ventilating mode**.

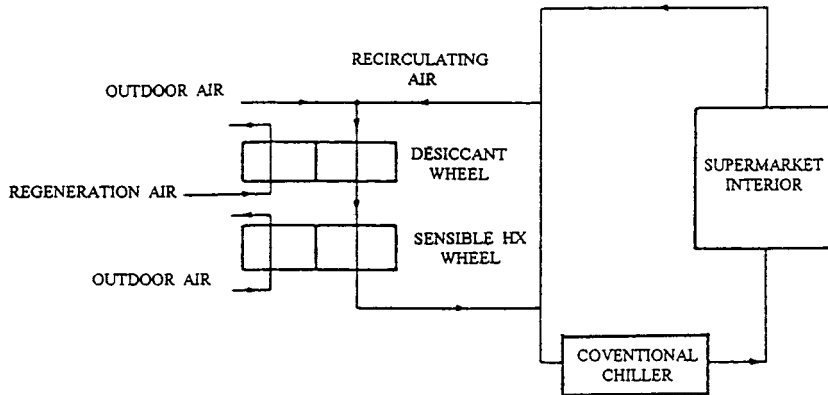


FIGURE 9.16.7 Hybrid air-conditioning system for supermarkets.

Ventilation Mode

In the *adsorption path* the process air stream drawn from the outdoors is passed through the dry section of the desiccant wheel where it is dehumidified and heated by the liberated heat of sorption. It then passes through the sensible heat exchanger wheel and exits as dry but slightly warm air. The hot and dry air leaving the dehumidifier enters the heat exchanger, where it is sensibly cooled down to near room temperature. It is then passed through the evaporative cooler, where it is further cooled and slightly humidified as it enters the conditioned space.

In the *desorption path*, air is drawn from the conditioned space; it is humidified (and thus cooled) in the evaporative cooler. The air stream enters the sensible heat exchanger, where it is preheated, and it is then heated to the desired regeneration temperature by a suitable heat source (natural gas, waste heat, or solar energy), passed through the desiccant wheel (regenerating the desiccant material), and discharged out of doors.

Performance. In order to achieve high performance, the maximum moisture content of the desiccant should be high and the isotherm should have the optimum shape (1 M). In addition, Zheng et al. (1993) showed that the optimum performance is very sensitive to the rotational speed of the desiccant wheel. Glav (1966) introduced stage regeneration. He showed that performance is improved when the reactivation segment of the wheel is at a temperature which increases in the direction of rotation. Collier (Collier et al., 1986) showed that well-designed open-cycle desiccant cooling systems can have a thermal COP of 1.3. This, however, would require the use of high-effectiveness sensible heat exchangers, which would be large and expensive. Smaller and more affordable heat exchangers should yield system COPs in the order of unity. An extensive review of the state-of-the-art assessment of desiccant cooling is given by Pesaran et al. (1992).

Conclusions

Desiccant-based air-conditioning offers significant advantages over conventional systems. Desiccant systems are already successfully used in some supermarkets. It is expected that these systems will gradually attain wider market penetration due to environmental requirements and potential energy savings.

The advantages of desiccant air-conditioning are summarized below:

- No CFC refrigerants are used.
- Indoor air quality is improved.
- Large latent heat loads and dry air requirements are conveniently handled.
- Individual control of temperature and humidity is possible.

The energy source may be natural gas and/or waste heat.
 Less circulated air is required.
 Summer electric peak is reduced.

Defining Terms

Absorb, absorption: When a chemical change takes place during sorption.

Adsorb, adsorption: When no chemical change occurs during sorption.

Dehumidification: Process of removing water vapor from air.

Desiccant: A subset of sorbents that has a particular affinity to water.

Desorb, desorption: Process of removing the sorbed material from the sorbent.

Isotherm: Sorbed material vs. relative humidity at a constant temperature.

Reactivation: Process of removing the sorbed material from the sorbent.

Recirculation: Indoor air only is continuously processed.

Regeneration: Process of removing the sorbed material from the sorbent.

Sorbent: A material that attracts and holds other gases or liquids.

Sorption: Binding of one substance to another.

Staged regeneration: When the temperature of the regeneration segment of the desiccant wheel is not uniform.

Ventilation mode: 100% of outdoor air is processed.

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