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.

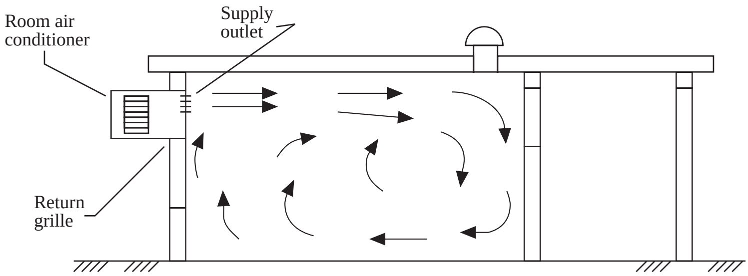


FIGURE 9.1.1 An individual room air-conditioning system.

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.

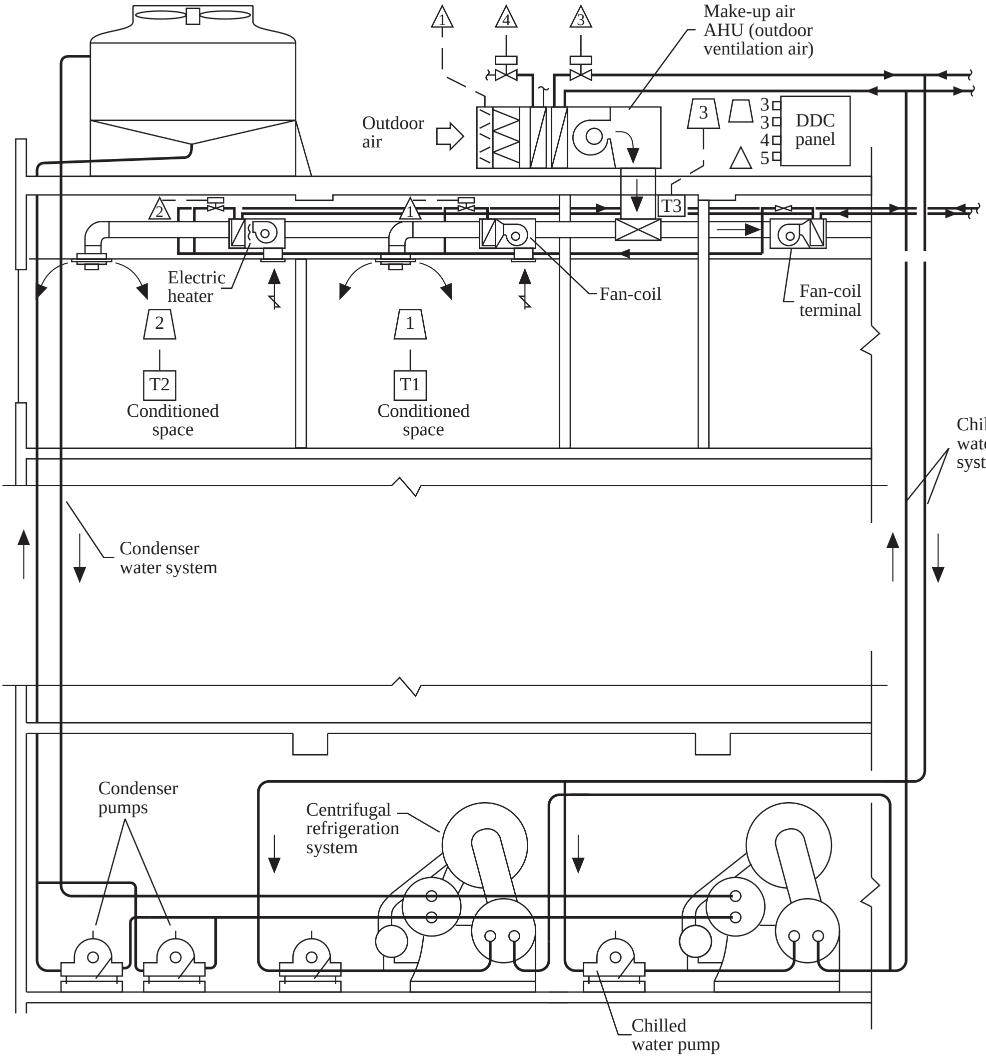


FIGURE 9.1.2 A space-conditioning air-conditioning system (fan-coil system).

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

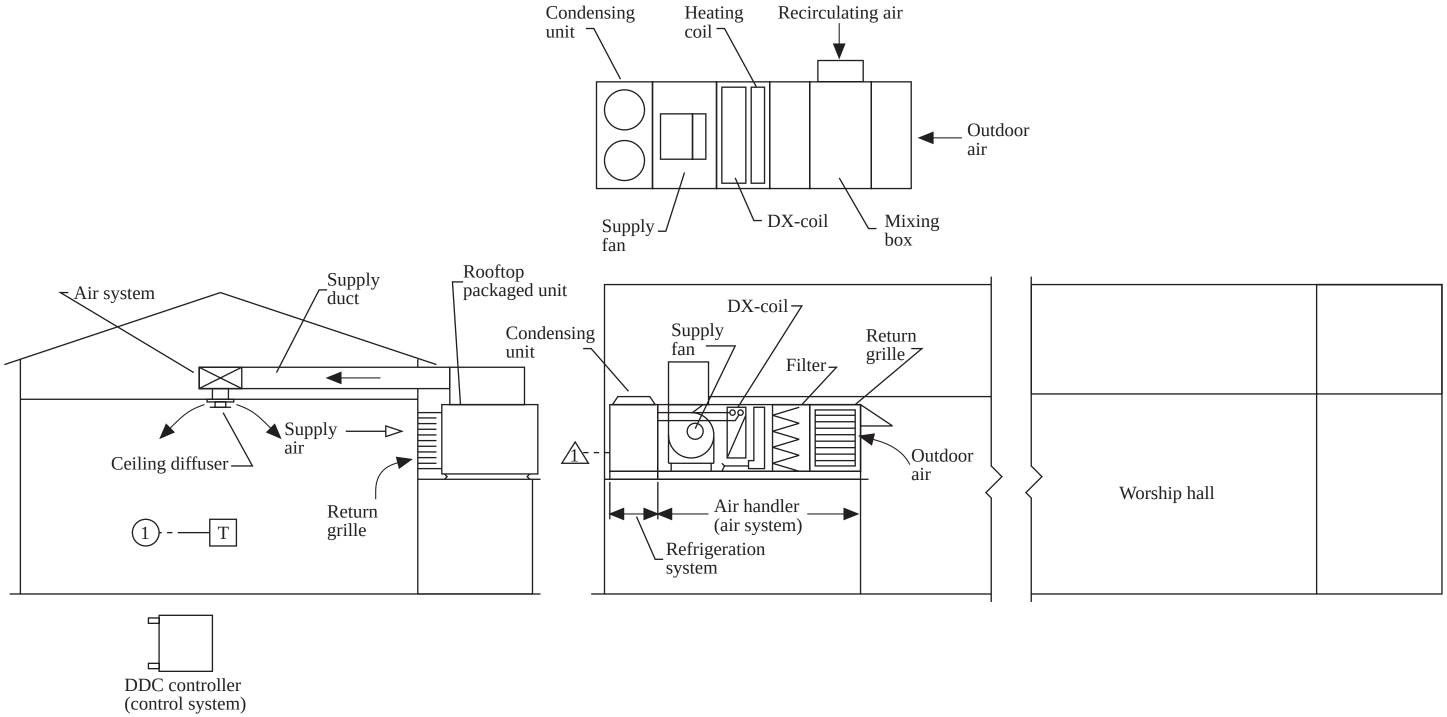


FIGURE 9.1.3 A packaged air-conditioning system.

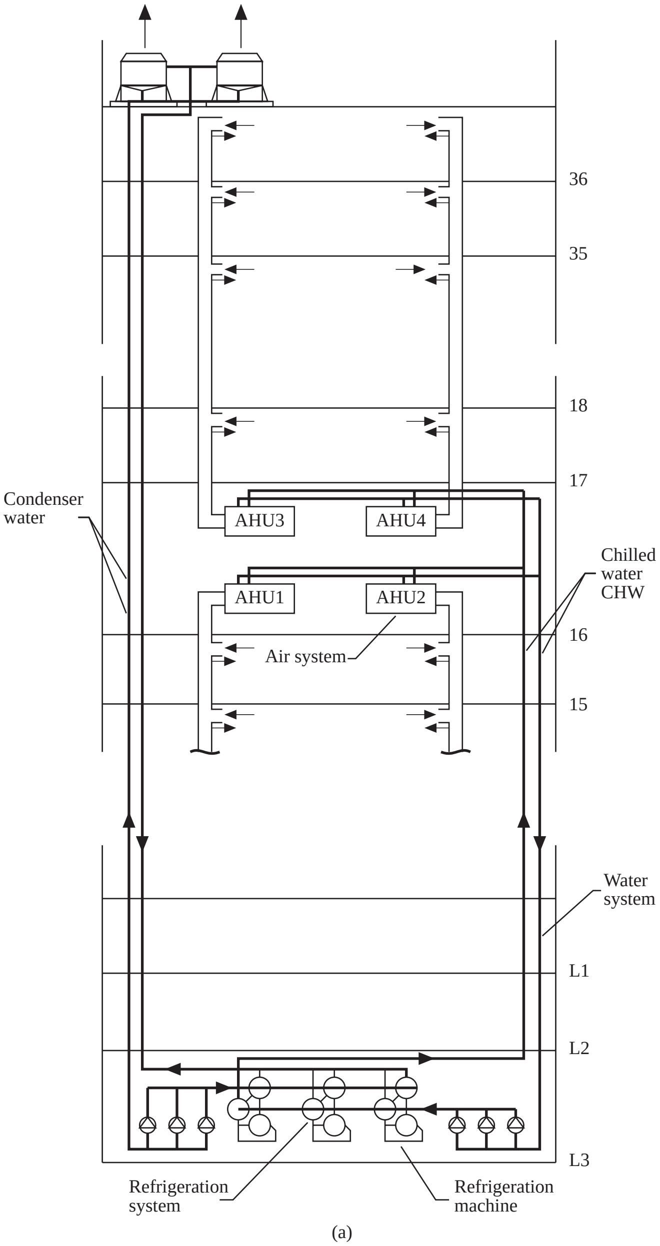


FIGURE 9.1.4a A central air-conditioning system: schematic diagram.

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 ft²	(25%)
Packaged systems	34,753 million ft²	(49%)
Central systems	14,048 million ft²	(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.

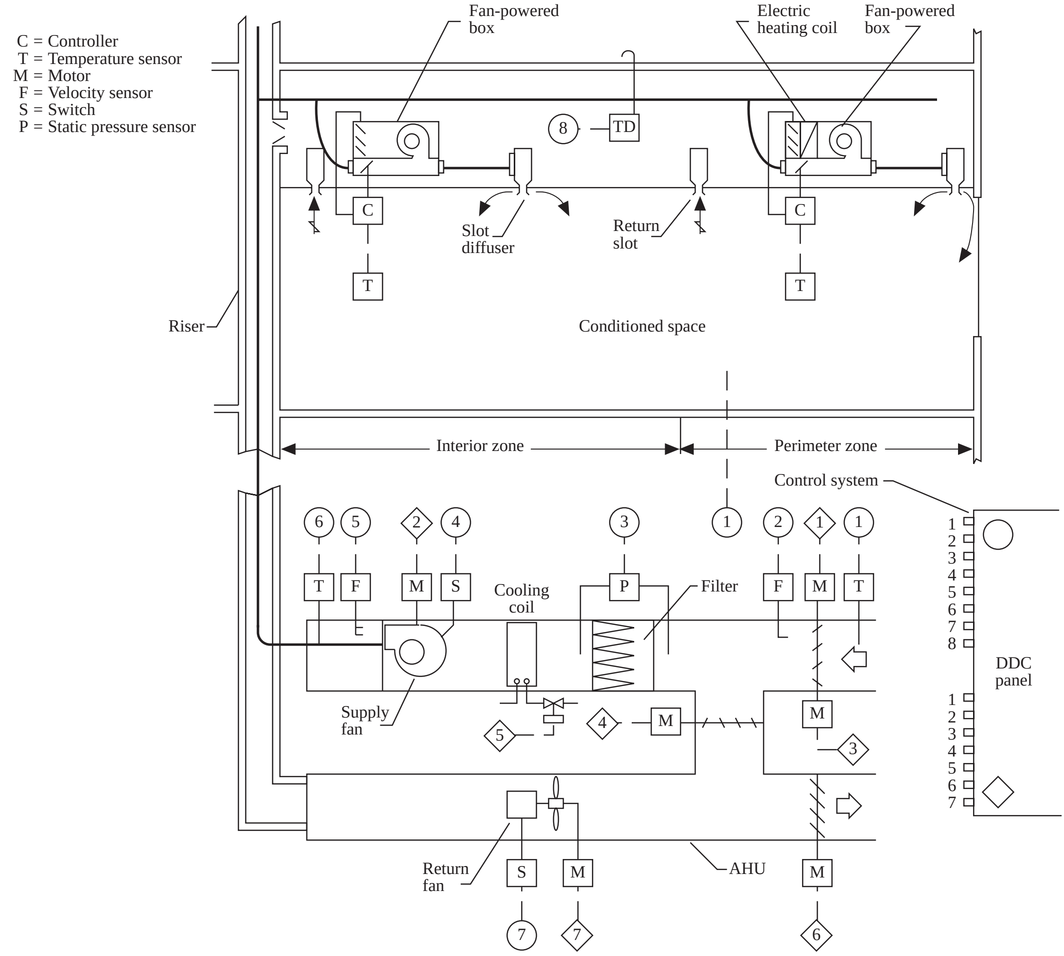


FIGURE 9.1.4b A central air-conditioning system: air and control systems for a typical floor.

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

$$pv = RT_R \tag{9.2.1}$$

or

$$pV = mRT_R \tag{9.2.2}$$

where p = pressure of the gas, psf (1 psf = 144 psi)
v = specific volume of the gas, ft^3/lb
R = gas constant, ftlb$_f$/lb$_m$ °R
T_R = absolute temperature of the gas, °R
V = volume of the gas, ft^3
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:

$$p_{at} = p_a + p_w \tag{9.2.3}$$

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

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

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:

$$\varphi = x_w / x_{ws}\big|_{T,p} = p_w / p_{ws}\big|_{T,p} \tag{9.2.5}$$

and

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

$$x_a + x_w = 1 \tag{9.2.6}$$

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:

$$\mu = w / w_s\big|_{T,p} \tag{9.2.7}$$

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:

$$\begin{aligned} h &= c_{pd} T + w\left(h_{g0} + c_{ps} T\right) \\ &= 0.240T + w(1061 + 0.444T) \end{aligned} \tag{9.2.8}$$

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:

$$v = V/m_a \tag{9.2.9}$$

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

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

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:

$$\rho = m_a / V = 1/v \tag{9.2.11}$$

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), $(c_{pd} + wc_{ps})T$ 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_{fg0}, 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:

$$c_{pa} = c_{pd} + wc_{ps} = 0.240 + 0.0075 \times 0.444 = 0.243 \text{ Btu/lb °F} \tag{9.2.12}$$

The *dew point temperature* T_{dew}, in °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:

$$h_1 + \left(w_s^* - w_1\right) h_w^* = h_s^* \tag{9.2.13}$$

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'.

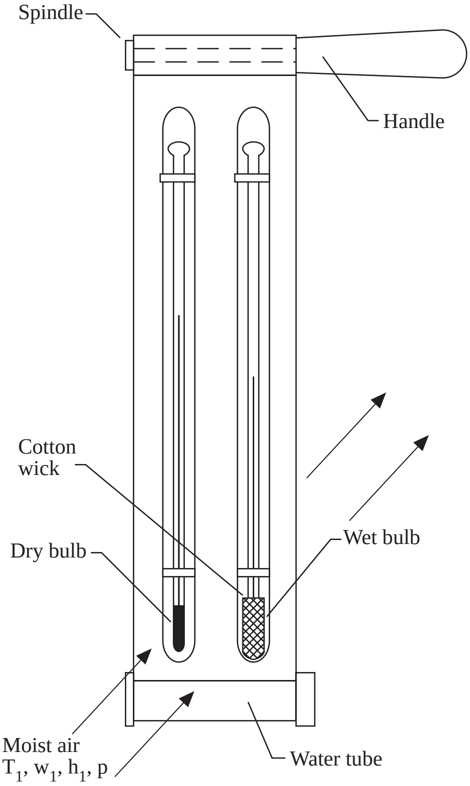


FIGURE 9.2.1 A sling psychrometer.

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:

$$h_c(T - T') + h_r(T_{ra} - T) = h_d(w_s' - w_1) \tag{9.2.14}$$

where h_c, h_r = mean conductive and radiative heat transfer coefficient, Btu/hr ft^2°F
h_d = mean convective mass transfer coefficient, lb/hr ft^2

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:

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

$$K' = c_{\text{pa}} Le^{0.6667} \big/ h'_{\text{fg}} \tag{9.2.15}$$

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 Charted 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, φ = 100%. On a saturation curve, temperature *T*, thermodynamic wet temperature bulb T^*, and dew point temperature T_{dew} have the same value.

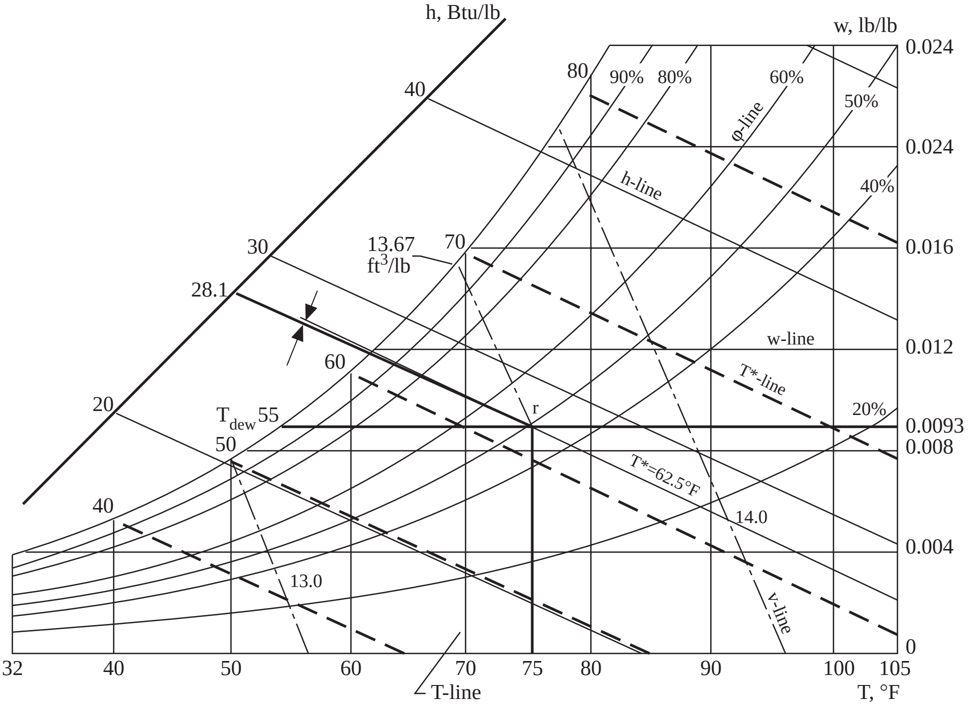


FIGURE 9.2.2 The abridged ASHRAE psychrometric chart and the determination of properties as in Example 9.2.1.

- 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 $w - p_w - T_{dew}$. 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 φ_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 ft^3/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_{dew} = 55°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°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:

$$\text{SHR} = \left|q_{\text{sen}}\right| \Big/ \left|q_{\text{total}}\right| = \left|q_{\text{sen}}\right| \Big/ \left(\left|q_{\text{sen}}\right| + \left|q_{1}\right|\right) \tag{9.3.1}$$

For any air-conditioning process, the sensible heat change

$$q_{\text{sen}} = 60\overset{\circ}{V}_{\text{s}}\,\rho_{\text{s}}c_{\text{pa}}\left(T_2 - T_1\right) = 60\overset{\circ}{m}_{\text{a}}\,c_{\text{pa}}\left(T_2 - T_1\right) \tag{9.3.2}$$

where $\overset{\circ}{V}_{\text{s}}$ = volume flow rate of supply air, cfm
ρ_{s} = density of supply air, lb/ft^3
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

$$\overset{\circ}{m}_{\text{s}} = \overset{\circ}{V}_{\text{s}}\,\rho_{\text{s}} \tag{9.3.3}$$

The latent heat change is

$$q_1 \approx 60\overset{\circ}{V}_{\text{s}}\,\rho_{\text{s}}\left(w_2 - w_1\right)h_{\text{fg.58}} = 1060 \times 60\overset{\circ}{V}_{\text{s}}\,\rho_{\text{s}}\left(w_2 - w_1\right) \tag{9.3.4}$$

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

In Equation (9.3.4), $h_{\text{fg58}} \approx 1060$ 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

$$\text{SHR} = \overset{\circ}{m}_{\text{a}}\,c_{\text{pa}}\left(T_2 - T_1\right) \Big/ \left[\overset{\circ}{m}_{\text{a}}\,c_{\text{pa}}\left(T_2 - T_1\right) + \overset{\circ}{m}_{\text{a}}\left(w_2 - w_1\right)h_{\text{fg.58}}\right] \tag{9.3.5}$$

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_{rc}, in Btu/hr, can be calculated as:

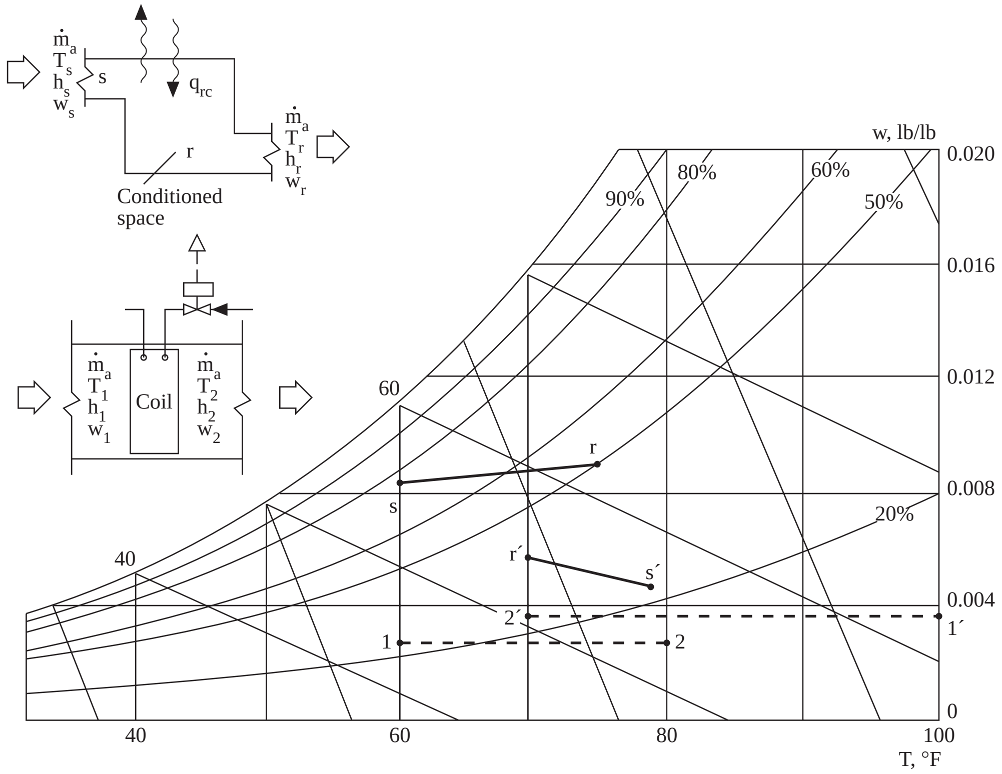


FIGURE 9.3.1 Supply conditioning, sensible heating, and sensible cooling processes.

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

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

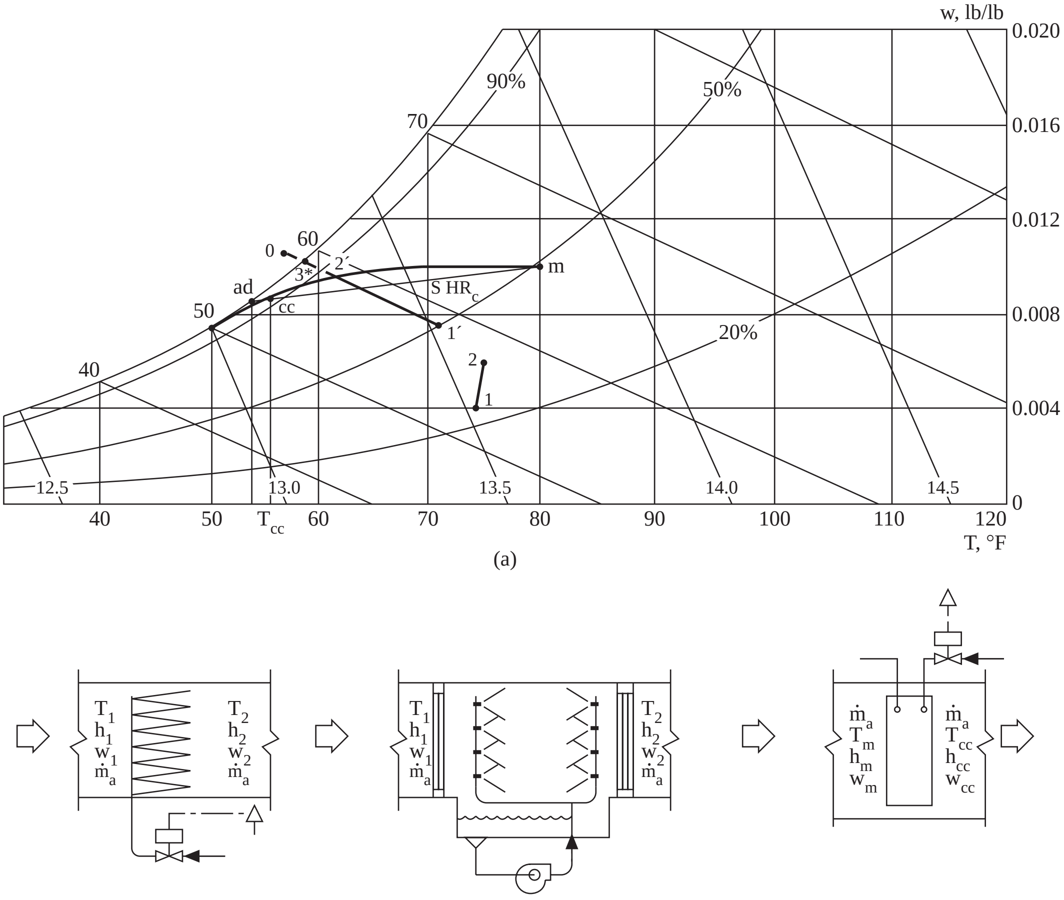
The space sensible cooling load q_{rs}, in Btu/hr, can be calculated from Equation (9.3.2) and the space latent load q_{rl}, in Btu/hr, from Equation (9.3.1). In Equation (9.3.4), T_2 should be replaced by T_r and T_1 by T_s. Also in Equation (9.3.1), w_2 should be replaced by w_r and w_1 by w_s. The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

$$q_{rh} = 60\mathring{m}_a c_{pa}\left(T_s - T_r\right) = 60\mathring{V}_s\rho_s c_{pa}\left(T_s - T_r\right) \tag{9.3.7}$$

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_{rh}, 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 $T_{2'}$.



(a)

(b)

(c)

(d)

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 $\overset{\circ}{m}_{\mathrm{hu}}$, in lb/min, is given by:

$$\overset{\circ}{m}_{\mathrm{hu}} = \overset{\circ}{V}_{\mathrm{s}} \rho_{\mathrm{s}} \left(w_{\mathrm{hl}} - w_{\mathrm{he}} \right) \tag{9.3.8}$$

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:

$$\left(T_2 - T_1\right) = w_{\mathrm{sm}} c_{\mathrm{ps}} T_{\mathrm{s}} \Big/ \left(c_{\mathrm{pd}} + w_{12} c_{\mathrm{ps}}\right) \tag{9.3.9}$$

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, $\mathring{m}_{\mathrm{s}} / \mathring{m}_{\mathrm{a}}$
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:

$$w_{\mathrm{os}} = w_{\mathrm{o}} - w_{\mathrm{s}}^* = \left(w_{2'} + w_{\mathrm{w}}\right) - w_{\mathrm{s}}^* \tag{9.3.10}$$

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 $T_{\mathrm{s.t}} < T''_{\mathrm{ae}}$, it is a cooling and dehumidifying process. If $T_{\mathrm{we}} \geq T''_{\mathrm{ae}}$, or $T_{\mathrm{s.t}} \geq T''_{\mathrm{ae}}$, sensible cooling occurs. The cooling coil's load or the cooling capacity of the air washer q_{cc}, in Btu/hr, is

$$q_{\mathrm{cc}} = 60 \mathring{V}_{\mathrm{s}} \rho_{\mathrm{s}} \left(h_{\mathrm{ae}} - h_{\mathrm{cc}}\right) - 60 \mathring{m}_{\mathrm{w}} h_{\mathrm{w}} \tag{9.3.11a}$$

where $h_{\mathrm{ae}}, h_{\mathrm{cc}}$ = enthalpy of moist air entering and leaving the coil or washer, Btu/lb
$\mathring{m}_{\mathrm{w}}$ = 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 $60\overset{\circ}{m}_w h_w$ is often neglected, and

$$q_{cc} = 60\overset{\circ}{V}_s \rho_s (h_{ae} - h_{cc}) \tag{9.3.11b}$$

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

$$\text{SHR}_c = q_{cs}/q_{cc} \tag{9.3.12}$$

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 m1 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:

$$\begin{aligned} \overset{\circ}{m}_1 h_1 + \overset{\circ}{m}_2 h_2 &= \overset{\circ}{m}_m h_m \\ \overset{\circ}{m}_1 w_1 + \overset{\circ}{m}_2 w_2 &= \overset{\circ}{m}_m w_m \\ \overset{\circ}{m}_1 T_1 + \overset{\circ}{m}_2 T_2 &= \overset{\circ}{m}_m T_m \end{aligned} \tag{9.3.13}$$

In Equation (9.3.13), $\overset{\circ}{m}$ 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,

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

Similarly,

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

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,

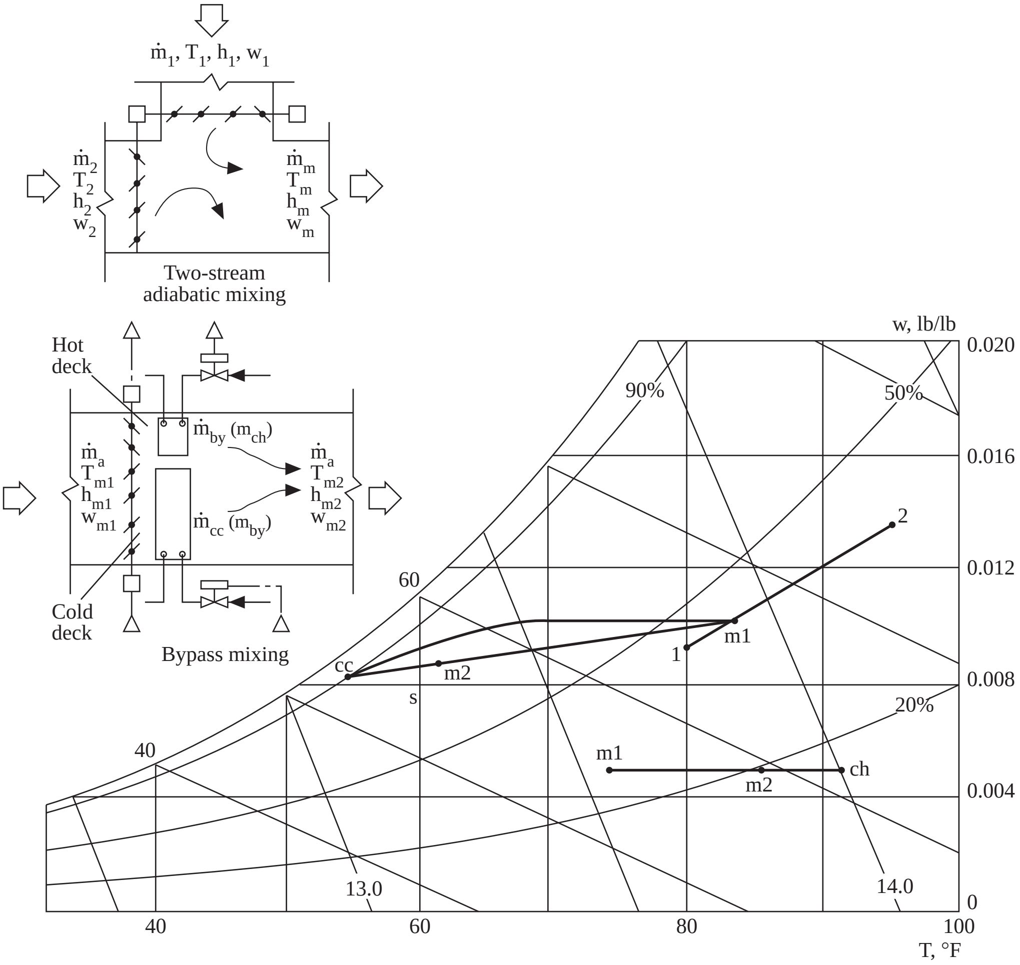


FIGURE 9.3.3 Mixing processes.

$$\mathring{V}_1 h_1 + \mathring{V}_2 h_2 = \mathring{V}_m h_m$$

$$\mathring{V}_1 w_1 + \mathring{V}_2 w_2 = \mathring{V}_m w_m \tag{9.3.16}$$

$$\mathring{V}_1 T_1 + \mathring{V}_2 T_2 = \mathring{V}_m T_m$$

$$\mathring{V}_1 + \mathring{V}_2 = \mathring{V}_m \tag{9.3.17}$$

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 $K_{cc} = \mathring{m}_{cc}/\mathring{m}_m$ and $K_{ch} = \mathring{m}_{ch}/\mathring{m}_m$; 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

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$$q_{\mathrm{cc}} = K_{\mathrm{cc}} \mathring{V}_{\mathrm{s}} \rho_{\mathrm{s}} \left(h_{\mathrm{m}} - h_{\mathrm{cc}}\right)$$
$$q_{\mathrm{ch}} = K_{\mathrm{ch}} \mathring{V}_{\mathrm{s}} \rho_{\mathrm{s}} \left(h_2 - h_1\right) \tag{9.3.18}$$

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.

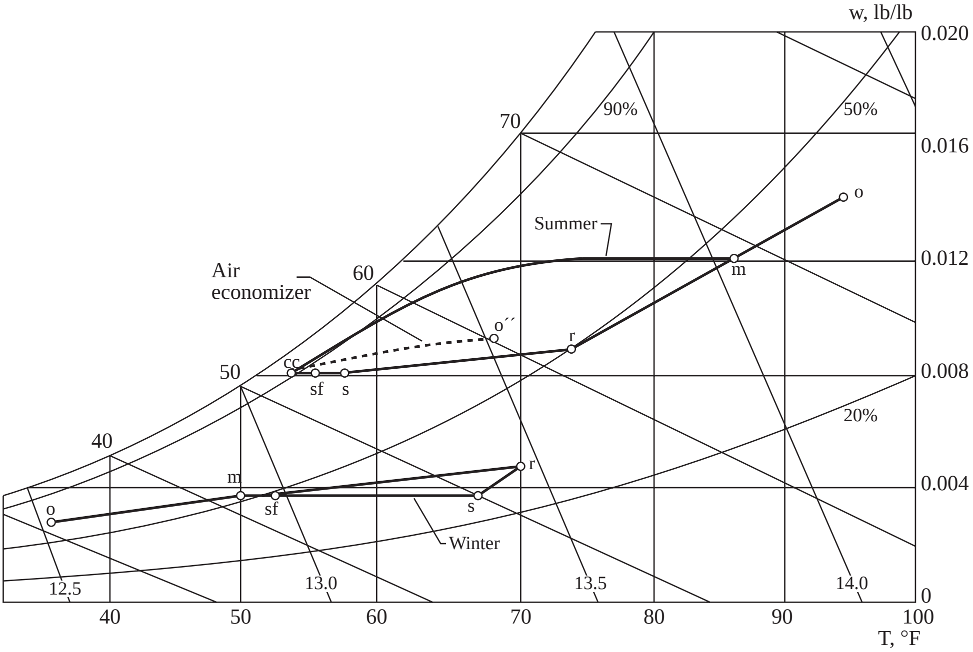


FIGURE 9.3.4 Basic air-conditioning cycle — summer, winter, and air economizer mode.

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:

$$q_{\mathrm{s.s}} = q_{\mathrm{sf}} + q_{\mathrm{sd}} = \overset{\circ}{V}_{\mathrm{s}} \rho_{\mathrm{s}} c_{\mathrm{pa}} \Delta T_{\mathrm{s.s}} \tag{9.3.19}$$

It is more convenient to use the temperature rise of the supply system $\Delta T_{\mathrm{s.s}}$ 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* $\overset{\circ}{V}_{\mathrm{s.d}}$, 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_{r}:

$$\overset{\circ}{V}_{\mathrm{s.d}} = q_{\mathrm{rc.d}} \Big/ \left[60 \rho_{\mathrm{s}} \left(h_{\mathrm{r}} - h_{\mathrm{s}}\right)\right] = q_{\mathrm{rs.d}} \Big/ \left[60 \rho_{\mathrm{s}} c_{\mathrm{pa}} \left(T_{\mathrm{r}} - T_{\mathrm{s}}\right)\right] \tag{9.3.20}$$

where $q_{\mathrm{rc.d}}$, $q_{\mathrm{rs.d}}$ = design space cooling load and design sensible cooling load, Btu/hr. In Equation (9.3.20), the greater the $q_{\mathrm{rs.d}}$, the higher $\overset{\circ}{V}_{\mathrm{s}}$ 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 $\overset{\circ}{V}_{\mathrm{s.d}}$ for a given supply mass flow rate. For a given $q_{\mathrm{rs.d}}$, the supply temperature differential $\Delta T_{\mathrm{s}} = (T_{\mathrm{r}} - T_{\mathrm{s}})$ is an important parameter that affects $\overset{\circ}{V}_{\mathrm{s.d}}$. 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 $\overset{\circ}{V}_{\mathrm{s}}$ 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 $q_{\mathrm{rs.d}}$ is used to determine $\overset{\circ}{V}_{\mathrm{s.d}}$, 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/m^3, the design supply volume flow rate is

$$\overset{\circ}{V}_{\mathrm{s.d}} = 2118 \overset{\circ}{m}_{\mathrm{par}} \Big/ \left(C_1 - C_{\mathrm{s}}\right) \tag{9.3.21}$$

where C_{s} = concentration of air contaminants in supply air, mg/m^3
$\overset{\circ}{m}_{\mathrm{par}}$ = 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

$$\overset{\circ}{V}_{\mathrm{s.d}} = q_{\mathrm{rl.d}} \Big/ \left[60 \rho_{\mathrm{s}} \left(w_{\mathrm{r}} - w_{\mathrm{s}}\right) h_{\mathrm{fg.58}}\right] \tag{9.3.22}$$

where $q_{\mathrm{rl.d}}$ = 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

$$\overset{\circ}{V}_{\mathrm{s.d}} = A_{\mathrm{r}} v_{\mathrm{r}} \tag{9.3.23a}$$

where A_{r} = cross-sectional area perpendicular to the air flow in the working area, ft^2.

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

$$\mathring{V}_{\mathrm{s}} \ge n\mathring{V}_{\mathrm{oc}} \tag{9.3.23b}$$

where n = number of occupants
$\mathring{V}_{\mathrm{oc}}$ = outdoor air requirement per person, cfm/person

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

$$\mathring{V}_{\mathrm{s}} \ge \mathring{V}_{\mathrm{ex}} + \mathring{V}_{\mathrm{ef}} \tag{9.3.24}$$

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,

$$\begin{aligned} \mathring{m}_{\mathrm{a}} &= \mathring{V}_{\mathrm{cc}}\,\rho_{\mathrm{cc}} = \mathring{V}_{\mathrm{s}}\,\rho_{\mathrm{s}} = \mathring{V}_{\mathrm{sf}}\,\rho_{\mathrm{sf}} \\ \mathring{V}_{\mathrm{sf}} &= \mathring{V}_{\mathrm{s}}\,\rho_{\mathrm{s}} \Big/ \rho_{\mathrm{sf}} \end{aligned} \tag{9.3.25}$$

where $\mathring{V}_{\mathrm{sf}}$ = volume flow rate at supply fan outlet, cfm
ρ_{sf} = air density at supply fan outlet, lb/ft^3

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/ft^3. However, a fan is a constant-volume machine at a given fan size and speed; that is, $\mathring{V}_{\mathrm{sf}} = \mathring{V}_{\mathrm{sf.r}}$. Here $\mathring{V}_{\mathrm{sf.r}}$ represents the rated volume flow rate of a fan at standard air conditions. Therefore,

$$\mathring{V}_{\mathrm{sf.r}} = \mathring{V}_{\mathrm{sf}} = q_{\mathrm{rs.d}} \Big/ \left[60\rho_{\mathrm{sf}} c_{\mathrm{pa}} \left(T_{\mathrm{r}} - T_{\mathrm{s}}\right)\right] \tag{9.3.26}$$

- For conditioned air leaving the cooling coil at T_{cc} = 55°F with a relative humidity of 92% and T_{sf} of 57°F, $\rho_{\mathrm{sf.r}} = 1/v_{\mathrm{sf}} = 1/13.20 = 0.0758$ lb/ft^3. From Equation (9.3.26):

$$\mathring{V}_{\mathrm{sf.r}} = q_{\mathrm{rs.d}} \Big/ \left[60 \times 0.0758 \times 0.243\left(T_{\mathrm{r}} - T_{\mathrm{s}}\right)\right] = q_{\mathrm{rs.d}} \Big/ \left[1.1\left(T_{\mathrm{r}} - T_{\mathrm{s}}\right)\right] \tag{9.3.26a}$$

  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°F and φ_{cc} = 98%, if T_{sf} = 42°F, then v_{sf} = 12.80 ft^3/lb, and the rated supply volume flow rate:

$$\mathring{V}_{\mathrm{sf.r}} = 12.80 q_{\mathrm{rs.d}} \Big/ \left[60 \times 0.243\left(T_{\mathrm{r}} - T_{\mathrm{s}}\right)\right] = q_{\mathrm{rs.d}} \Big/ \left[1.14\left(T_{\mathrm{r}} - T_{\mathrm{s}}\right)\right] \tag{9.3.26b}$$

- For a blow-through fan in which the fan is located upstream of the coil, if T_{sf} = 82°F and φ_{sf} = 43%, then v_{sf} = 13.87 ft^3/lb, and the rated supply volume flow rate:

$$\overset{\circ}{V}_{\text{sf.r}} = 13.87 q_{\text{rs.d}} \Big/ \left[60 \times 0.243\left(T_{\text{r}} - T_{\text{s}}\right)\right] = q_{\text{rs.d}} \Big/ \left[1.05\left(T_{\text{r}} - T_{\text{s}}\right)\right] \qquad (9.3.26c)$$

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 $\overset{\circ}{V}_{\text{sf.r}}$ is needed. For an air temperature of 70°F:

$$\overset{\circ}{V}_{\text{x.ft}} = \overset{\circ}{V}_{\text{sf.r}}\left(p_{\text{sea}}/p_{\text{x.ft}}\right) = \overset{\circ}{V}_{\text{sf.r}}\left(\rho_{\text{sea}}/\rho_{\text{x.ft}}\right) \qquad (9.3.27)$$

where $\overset{\circ}{V}_{\text{x.ft}}$ = supply volume flow rate at an altitude of *x* ft, cfm
p_{sea}, $p_{\text{x.ft}}$ = atmospheric pressure at sea level and an altitude of *x* ft, psia
ρ_{sea}, $\rho_{\text{x.ft}}$ = 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 $\overset{\circ}{V}_{\text{r.2000}} = \overset{\circ}{V}_{\text{sf.r}}$ $(\text{p}_{\text{sea}}/\text{p}_{\text{x.ft}}) = 1.076$ $\overset{\circ}{V}_{\text{sf.r}}$ cfm instead of $\overset{\circ}{V}_{\text{sf.r}}$ cfm at sea level.

Altitude, ft	p_{at}**, psia**	**ρ, lb/ft³**	**($p_{\text{sea}}/p_{\text{x,ft}}$)**
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 $\text{q}_{\text{s.s}}$, 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 $\overset{\circ}{V}_{\text{s}}$ 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 $T_{\text{s}} = (T_{\text{cc}} + 2 + 3) \leq 55$°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.

3. 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

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

Solution

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

$$\mathrm{SHR_s} = |q_{\mathrm{rs}}| / \left(|q_{\mathrm{rs}}| + |q_{\mathrm{rl}}|\right) = 60{,}000/(60{,}000 + 15{,}000) = 0.8$$

On the psychrometric chart, from given T_r = 75°F and φ_r = 50%, plot space point r. Draw a space conditioning line sr from point r with $\mathrm{SHR_s}$ = 0.8.
Since $\Delta T_{\mathrm{s.s}}$ = 2 + 2 = 4°F, move line segment cc s (4°F) up and down until point s lies on line sr and point cc lies on the φ_{cc} = 93% line. The state points s and cc are then determined as shown in Figure 9.3.4:

T_s = 57.5°F, φ_s = 82%, and w_s = 0.0082 lb/lb
T_{cc} = 53.5°F, φ_{cc} = 93%, h_{cc} = 21.8 Btu/lb, and w_{cc} = 0.0082 lb/lb

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

$$\overset{\circ}{V}_{\mathrm{sf.r}} = q_{\mathrm{rs.d}} / \left[60\rho_{\mathrm{sf}} c_{\mathrm{pa}} \left(T_r - T_s\right)\right]$$

$$= 60{,}000 / \left[60 \times 0.076 \times 0.243(75 - 57.5)\right] = 3094 \text{ cfm}$$

3. 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

$$\mathrm{rm/ro} = 1800/3094 = 0.58$$

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:

$$T_\mathrm{m} = 86.7°\mathrm{F}, \qquad h_\mathrm{m} = 35 \text{ Btu/lb}$$

From Equation (9.3.11), the cooling coil load is

$$q_\mathrm{cc} = 60 \overset{\circ}{V}_\mathrm{s} \rho_\mathrm{s} \left(h_\mathrm{m} - h_\mathrm{cc}\right) = 60 \times 3094 \times 0.076(35 - 21.8) = 186{,}234 \text{ Btu/lb}$$

4. From the psychrometric chart, since the dew point of the space air T_r'' = 55°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°F which is higher than T_r'' = 55°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:

$$\begin{gathered} \left(w_\mathrm{r} - w_\mathrm{m}\right) \big/ \left(w_\mathrm{r} - w_\mathrm{o}\right) = \overset{\circ}{V}_\mathrm{o} \big/ \overset{\circ}{V}_\mathrm{s} \\ \left(w_\mathrm{r} - w_\mathrm{s}\right) = q_\mathrm{rl} \Big/ \left(60 \overset{\circ}{V}_\mathrm{s} \rho_\mathrm{s} h_\mathrm{fg.58}\right) \\ w_\mathrm{s} = w_\mathrm{m} \end{gathered} \tag{9.3.28}$$

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:

$$\begin{gathered} \left(T_\mathrm{r} - T_\mathrm{s}\right) = q_\mathrm{rs} \Big/ \left(60 \overset{\circ}{V}_\mathrm{s} \rho_\mathrm{s} c_\mathrm{pa}\right) \\ \left(T_\mathrm{r} - T_\mathrm{s}\right) \big/ \left(T_\mathrm{r} - T_\mathrm{o}\right) = \overset{\circ}{V}_\mathrm{o} \big/ \overset{\circ}{V}_\mathrm{s} \end{gathered} \tag{9.3.29}$$

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/ft^3, 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

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

From Equation 9.3.29, the supply air temperature differential is

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

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

$$\left(w_r - w_s\right)/\left(w_r - w_o\right) = 0.00079/\left(w_r - w_o\right) = \overset{\circ}{V}_o / \overset{\circ}{V}_s = 0.58$$

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

And from given information,

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

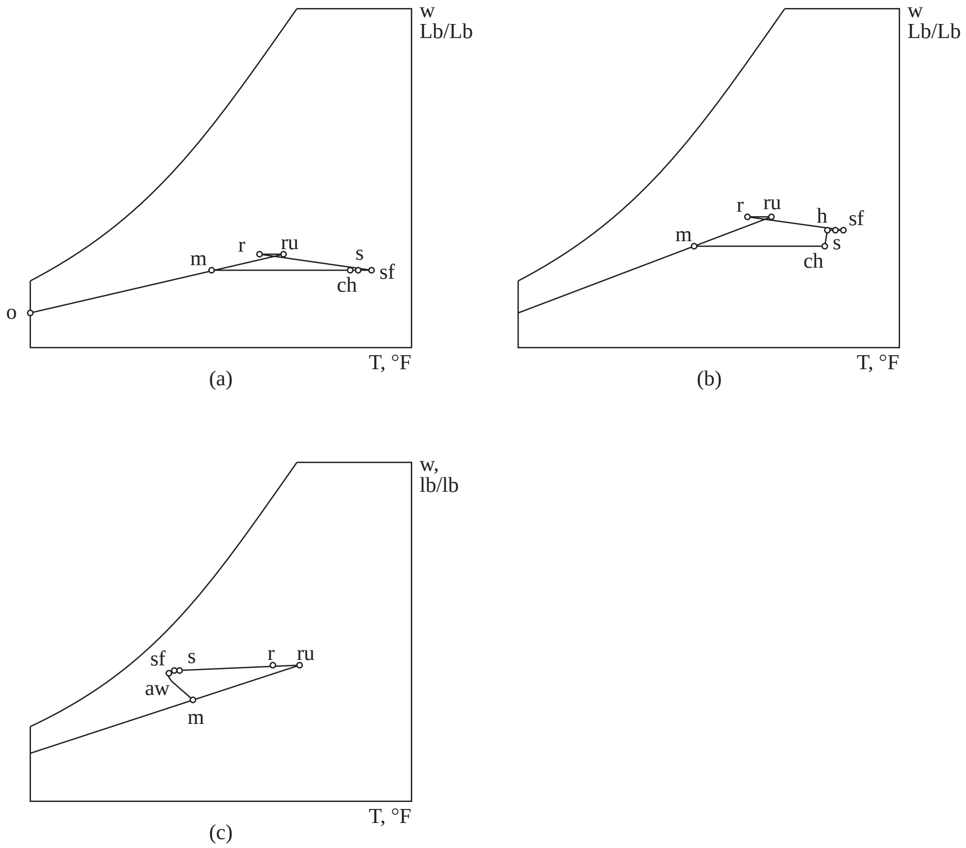
From the psychrometric chart, for T_r = 70°F and w_r = 0.00486 lb/lb, point r can be plotted, and φ_r is about 32% (see Figure 9.3.4).

2. Since mr/or = 0.58, point m can be determined, and from the psychrometric chart T_m = 50.0°F. As T_s = 70 – 2.88 = 67.12°F and $T_{sf} = T_m$ + 2 = 50.0 + 2 = 52.0°F, from Equation 9.3.7 the heating coil's load is

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

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



(a)

(b)

(c)

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, $(T_s - T_r) > 20°F$ 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 Condensin

		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_2F_2	52.02	0.0	0.14	135.6	340.2	2.51	
R-125	Pentafluoroethane	CHF_2CF_3	120.03	0.0	0.84	111.9	276.2	2.47	37.1
R-134a	Tetrafluoroethane	CF_3CH_2F	102.03	0.0	0.26	49.7	138.8	2.79	65.2
R-143a	Trifluoroethane	CH_3CF_3	84.0	0.0					
R-152a	Difluoroethane	CH_3CHF_2	66.05	0.0		44.8	124.3	2.77	
R-245ca	Pentafluoropropane	$CF_3CF_2CH_3$	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	$CHCIF_2$	86.48	0.05	0.40	82.09	201.5	2.46	69.0
R-123	Dichlorotrifluoroethane	$CHCl_2CF_3$	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_2O	18.02	0					
R-729	Air		28.97	0					
Chlorofluorocarbons CFCs, halons BFCs and their azeotropic									
R-11	Trichlorofluoromethane	CCl_3F	137.38	1.00	1.00	6.92	23.06	3.33	68.5
R-12	Dichlorodifluoromethane	CCl_2F_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_2FCClF_2	187.39	0.80	1.4	2.64	10.21	3.87	54.1
R-114	Dichlorotetrafluoroethane	CCl_2FCF_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 ft³/lb	Compresssor Displacement cfm/ton	Power Consumption hp/ton	Critical Temperature °F	Discharge Temperature °F	Flammability	Safety
Hydrofluorocarbons HFCs									
R-32			0.63			173.1			
R-125			0.33			150.9	103	Nonflammable	A1
R134a	R-12		0.95			213.9		Nonflammable	A1
R143a									
R-152a			1.64			235.9		Lower flammable	A2
R-245ca									
HFC's azeotropics									
R-507	R-502	Genetron AZ-50							
HFC's near azeotropic									
R-404A	R-22	SUVA HP-62							A1/A1[a]
R-407A	R-22	KLEA 60							A1/A1[a]
R-407C	R-22	KLEA 66							A1/A1[a]
Hydrochlorofluorocarbons HCFC's and their azeotropics									
R-22			0.66	1.91	0.696	204.8	127	Nonflammable	A1
R-123	R-11		5.88	18.87	0.663	362.6		Nonflammable	B1
R-124			1.30	5.06	0.698	252.5			
HCFC's near azeotropics									
R-402A	R-502	SUVA HP-80							A1/A1[a]
HCFC's azeotropics									
R-401A	R-12	MP 39							A1/A1[a]
R-401B	R-12	MP 66							A1/A1[a]
Inorganic compounds									
R-717			3.98	1.70	0.653	271.4	207	Lower flammability	B2
R-718								Nonflammable	
R-729								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 ft³/lb	Compresssor 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

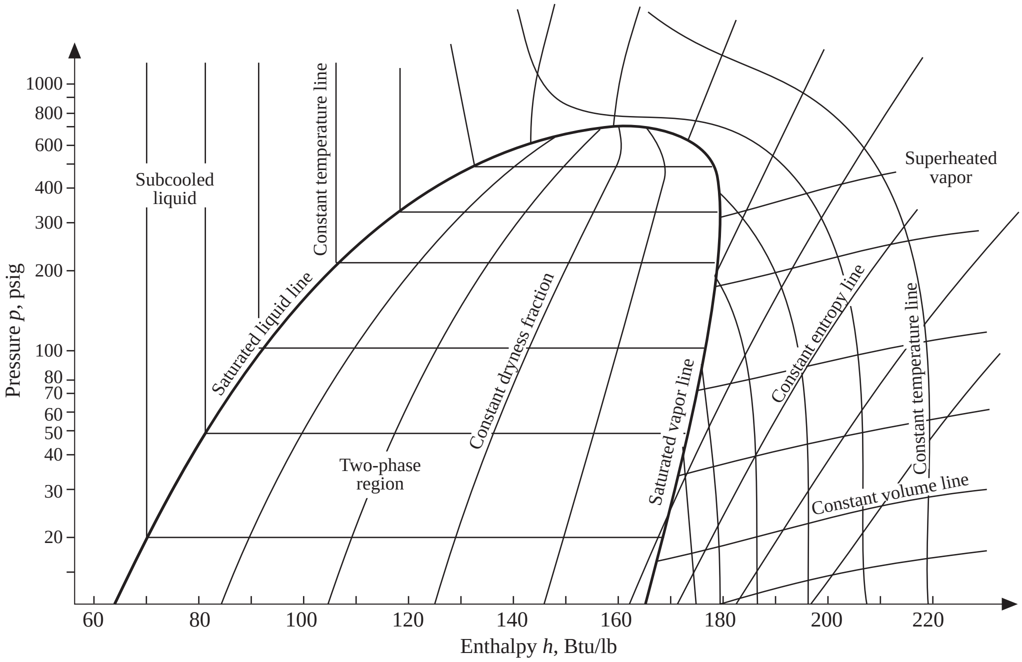


FIGURE 9.4.1 Skeleton of pressure-enthalpy diagram for R-22.

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:

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

   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

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

   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.

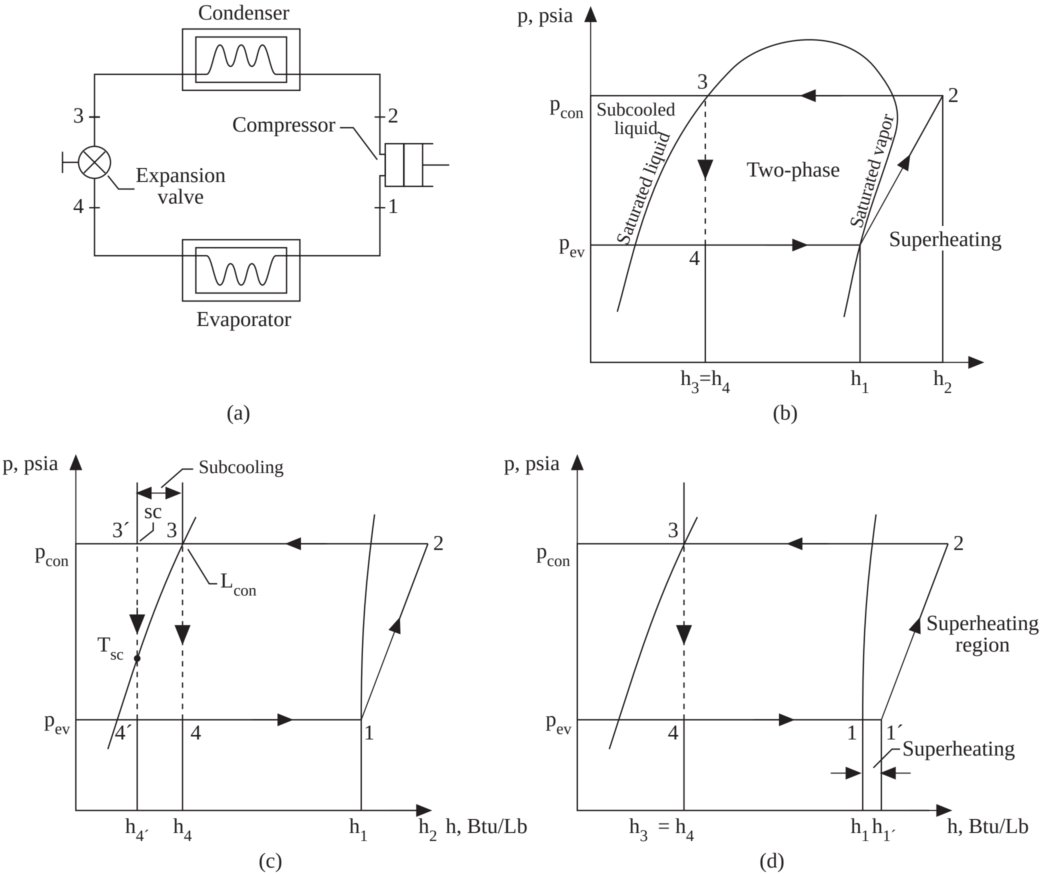


FIGURE 9.4.2 A single-stage ideal vapor compression refrigeration cycle: (a) schematic diagram, (b) *p*-*h* diagram, (c) subcooling, and (d) superheating.

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

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

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 $\mathring{m}_r$, in lb/min, is

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

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

$$\mathrm{COP}_{\mathrm{ref}} = q_{\mathrm{rf}}/W_{\mathrm{in}} \tag{9.4.6}$$

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

$$\mathrm{COP}_{\mathrm{hp}} = q_{2-3}/W_{\mathrm{in}} \tag{9.4.7}$$

- 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 $q_{2\text{-}3}$ to the work input, or

$$\mathrm{COP}_{\mathrm{hr}} = \left(\left|q_{\mathrm{rf}}\right| + \left|q_{2-3}\right|\right)/W_{\mathrm{in}} \tag{9.4.8}$$

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):

$$q_{\mathrm{rf.sc}} = \left(h_{4'} - h_1\right) > \left(h_4 - h_1\right) \tag{9.4.9}$$

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:

$$h_{\mathrm{sc}} = h_{3'} = h_{4'} = h_{\mathrm{l.con}} - c_{pr}\left(T_{\mathrm{con}} - T_{\mathrm{sc}}\right) \approx h_{\mathrm{s.sc}} \tag{9.4.10}$$

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}:

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

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

$$p_{\text{i}} = \sqrt{\left(p_{\text{con}} p_{\text{ev}}\right)} \tag{9.4.12}$$

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

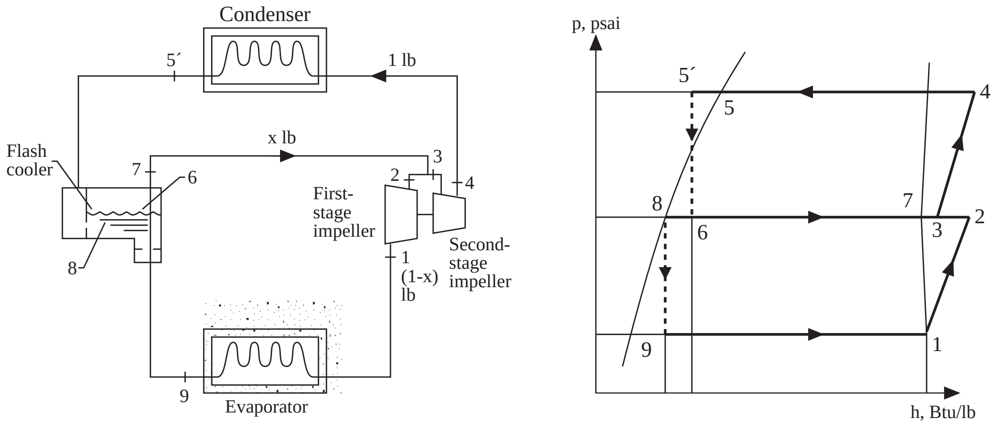
For a multistage system of n stages, then, the compression ratio of each stage is

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

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

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



(a) (b)

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

$$\text{COP}_{\text{ref}} = q_{\text{rf}}/W_{\text{in}} = (1-x)\left(h_1 - h_9\right)\Big/\left[(1-x)\left(h_2 - h_1\right) + \left(h_4 - h_3\right)\right] \tag{9.4.15}$$

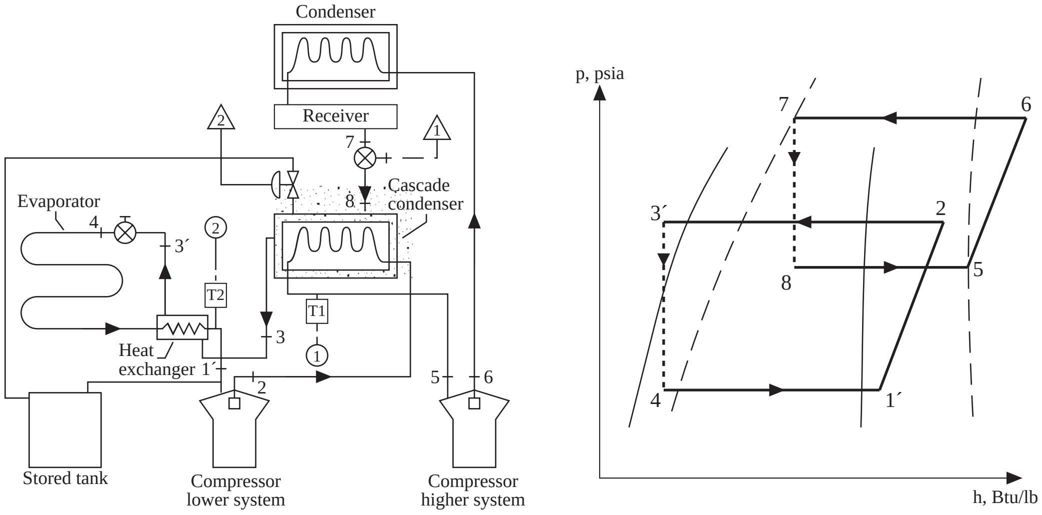
where h_1, h_2, h_3, h_4, h_9 = enthalpy of refrigerant at state points 1, 2, 3, 4, and 9, respectively, Btu/lb. The mass flow rate of refrigerant flowing through the condenser, $\mathring{m}_{\text{r}}$, in lb/min, can be calculated as

$$\mathring{m}_{\text{r}} = q_{\text{rc}}/60q_{\text{rf}} \tag{9.4.16}$$

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.



(a) (b)

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

$$q_{\mathrm{rf}} = \left(h_1 - h_4\right) \tag{9.4.17}$$

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

$$W_{\mathrm{in}} = \left(h_2 - h_{1'}\right) + \mathring{m}_{\mathrm{h}}\left(h_6 - h_5\right) \Big/ \mathring{m}_1 \tag{9.4.18}$$

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
$\mathring{m}_{\mathrm{h}}, \mathring{m}_1$ = mass flow rate of the refrigerant of the higher and lower systems, respectively

The coefficient of performance of a cascade system $\mathrm{COP}_{\mathrm{ref}}$ is

$$\mathrm{COP}_{\mathrm{ref}} = q_{\mathrm{rf}} / W_{\mathrm{in}} = \mathring{m}_1\left(h_1 - h_4\right) \Big/ \left[\mathring{m}_1\left(h_2 - h_{1'}\right) + \mathring{m}_{\mathrm{h}}\left(h_6 - h_5\right)\right] \tag{9.4.19}$$

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.ft^2, 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

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

where n = number of days for which $T_{o.m} < 65$°F
m = number of days for which $T_{o.m} > 65$°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.ft^2). 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.ft^2.°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 Occcupational 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 $\mu g/m^3$ (0.000022 $grain/ft^3$), 1 year
   150 $\mu g/m^3$ (0.000066 $grain/ft^3$), 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 (NO_2) 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 CO_2 as the representative human bioeffluent to an allowable indoor concentration of 1000 ppm. Field measurements of daily maximum CO_2 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:

	Long-term concentration			**Short-term concentration**		
Pollutants	**μg/m³**	**ppm**	**Exposure**	**μg/m³**	**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 ft^3:

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 ± 2°F and space humidity at 45 ± 5%. Here, ±2°F and ±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 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 exited 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:

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

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:

$$L_p = 20 \log\left(p/2 \times 10^{-5}\ \mathrm{Pa}\right) \text{ re 20 μPa} \tag{9.5.3}$$

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_e, 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_e, 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_{rh}, 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:

$$q_{rl} = q_{cc} + q_{pi} + q_{pu} + q_{s.t} \tag{9.6.1}$$

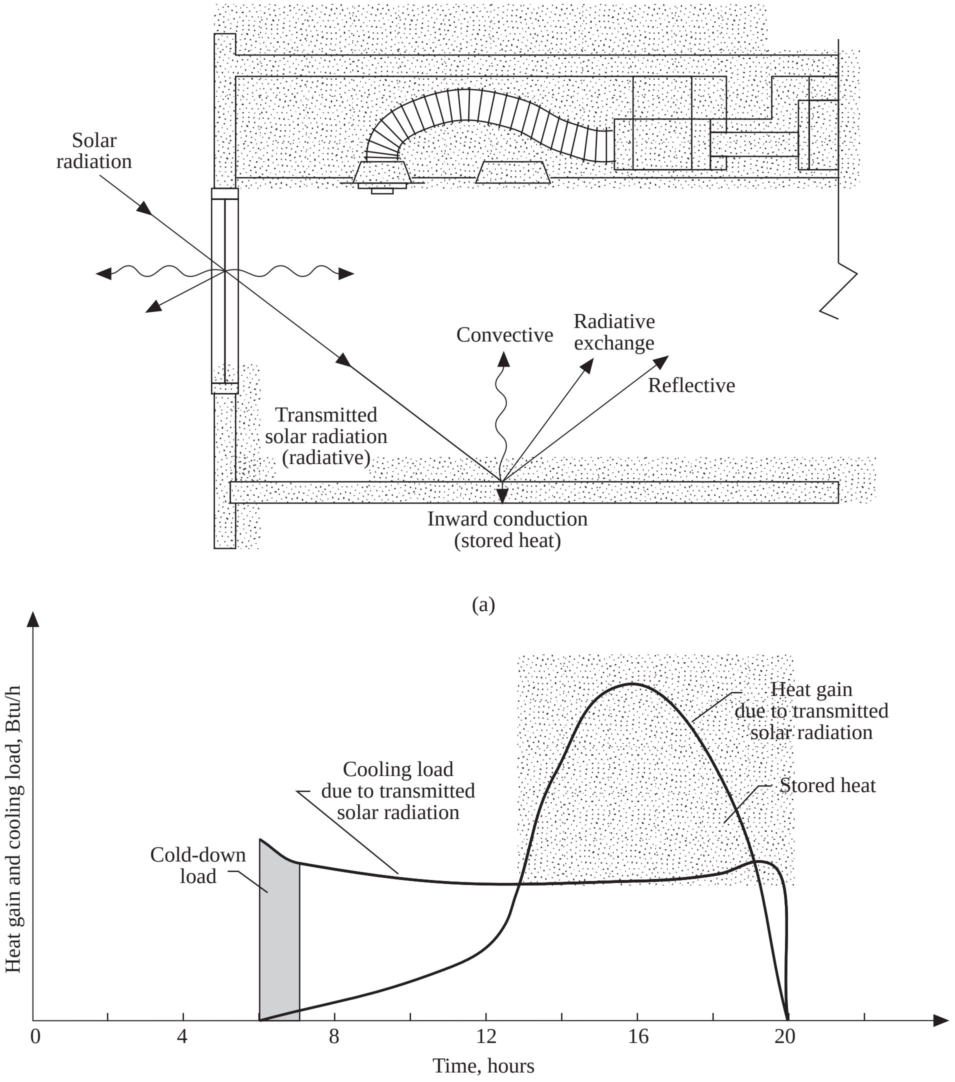
where q_{pi} = chilled water piping heat gain, Btu/hr
q_{pu} = pump power heat gain, Btu/hr
$q_{s.t}$ = storage tank heat gain, if any, Btu/hr

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.



(a)

(b)

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:

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

where h_{ci} = convective heat transfer coefficient, Btu/hr.ft^2.°F
g_{ij} = radiative heat transfer factor between inside surface *i* and inside surface *j*, Btu/hr.ft^2.°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*, ft^2
$S_{i,t}$, $L_{i,t}$, $P_{i,t}$, $E_{i,t}$ = 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:

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

where $\overset{\circ}{V}_{if}$ = volume flow rate of infiltrated air, cfm
ρ_o = air density of outdoor air, lb/ft^3
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,

$$K = Y/G = \left(v_o + v_1 z^{-1} + v_2 z^{-2} + \ldots\right) / \left(1 + w_1 z^{-1} + w_2 z^{-2} + \ldots\right)$$
$$z = e^{s\Delta} \tag{9.6.4}$$

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/ft^2 of floor area; medium construction, 70 lb/ft^2; and heavy construction, 130 lb/ft^2. 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.ft^2.°F and a constant *inside heat transfer coefficient* h_i = 1.46 Btu/hr.ft^2.°F during cooling load calculations. However, many software programs use the following empirical formula (TRACE 600 *Engineering Manual*):

$$h_{o.t} = 2.00 + 0.27 v_{wind.t} \tag{9.6.5}$$

where $h_{o.t}$ = h_o at time t, Btu/hr.ft^2.°F
$v_{wind.t}$ = wind velocity at time t, ft/sec

Coefficient h_i may be around 2.1 Btu/hr.ft^2.°F when the space air system is operating, and h_i drops to only about 1.4 Btu/hr.ft^2.°F when the air system is shut down. The *R value*, expressed in hr.ft^2.°F/Btu, is defined as the reciprocal of the overall heat transfer coefficient U value, in Btu/hr.ft^2.°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

$$T_{sol} = T_o + \alpha I_t / h_o - \varepsilon \Delta R / h_o \tag{9.6.6}$$

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.ft^2

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.ft^2.°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 $t\text{-}n\delta$, $T_{sol,t-n\delta}$, in °F, as the outdoor air temperature and a constant indoor space air temperature T_r, in °F, as

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

where A = surface area of wall or roof, ft^2
$q_{e,t-n\delta}$ = heat gain through wall or roof at time $t\text{-}n\delta$, 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

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

where U = overall heat transfer coefficient of the ceiling, floor, and partition (Btu/hr.ft^2.°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

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

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.ft^2, are the average solar heat gains during cloudless days through DSA glass. $\mathrm{SHGF_{max}}$ 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:

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

where A_s, A_{sh} = sunlit and shaded areas of the glass (ft^2).

In Equation (9.6.10), $\mathrm{SHGF_{sh}}$ 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 $\mathrm{SHGF_{sh}}$. Because the mass of window glass is small, its heat storage effect is often neglected; then the conduction heat gain at time t is

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

where U_g = overall heat transfer coefficient of window glass, Btu/hr.ft^2.°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, $q_{\mathrm{es,t}}$ and $q_{\mathrm{el,t}}$, both in Btu/hr, are given as

$$\begin{aligned} q_{\mathrm{es},t} &= N_t q_{\mathrm{os}} \\ q_{\mathrm{el},t} &= N_t q_{\mathrm{ol}} \end{aligned} \tag{9.6.12}$$

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, $q_{\mathrm{e.li}}$, in Btu/hr, is calculated as

$$q_{\mathrm{e.li}} = \sum_{n=1} \left[3.413 W_{\mathrm{li}} N_{\mathrm{li}} F_{\mathrm{ul}} F_{\mathrm{al}} \left(1 - F_{\mathrm{lp}}\right)\right] = 3.413 W_{\mathrm{lA}} A_{\mathrm{fl}} \tag{9.6.13}$$

where W_{li} = watt input to each lamp, W
W_{lA} = lighting power density, W/ft^2
A_{fl} = floor area of conditioned space, ft^2
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
$\mathrm{F_{lp}}$ = 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, $q_{\mathrm{e.ap}}$ and $q_{\mathrm{l.ap}}$, both in Btu/hr, can be calculated as

$$\begin{aligned} q_{\mathrm{e.ap}} &= 3.413 W_{\mathrm{ap}} F_{\mathrm{ua}} F_{\mathrm{load}} F_{\mathrm{ra}} = q_{\mathrm{is}} F_{\mathrm{ua}} F_{\mathrm{ra}} \\ q_{\mathrm{l.ap}} &= q_{\mathrm{il}} F_{\mathrm{ua}} \end{aligned} \tag{9.6.14}$$

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/ft^2 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 $q_{\mathrm{s.if}}$ and latent heat gain due to infiltration $q_{\mathrm{l.if}}$, in Btu/hr, are

$$\begin{aligned} q_{\mathrm{s.if}} &= 60\mathring{V}_{\mathrm{if}}\,\rho_{\mathrm{o}} c_{pa}\left(T_{\mathrm{o}} - T_{\mathrm{r}}\right) \\ q_{\mathrm{l.if}} &= 60 \times 1060\mathring{V}_{\mathrm{if}}\,\rho_{\mathrm{o}}\left(w_{\mathrm{o}} - w_{\mathrm{r}}\right) \end{aligned} \tag{9.6.15}$$

where $\mathring{V}_{\mathrm{if}}$ = volume flow rate of infiltration, cfm
ρ_{o} = air density of outdoor air, lb/ft^3
c_{pa} = specific heat of moist air, 0.243 Btu/lb.°F
$w_{\mathrm{o}}, w_{\mathrm{r}}$ = humidity ratio of outdoor and space air, lb/lb
$h_{\mathrm{fg.58}}$ = 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 $q_{\mathrm{rs,t}}$, in Btu/hr, is calculated as

$$q_{\mathrm{rs},t} = q_{\mathrm{s-e},t} + q_{\mathrm{s-c},t} \tag{9.6.16}$$

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

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

$$q_{\mathrm{s-e},t} = \sum_{i=1}\left(v_{\mathrm{o}} q_{\mathrm{e},t} + v_1 q_{\mathrm{e},t-\delta} + v_2 q_{\mathrm{e},t-2\delta} + \ldots\right) - \left(w_1 q_{\mathrm{r},t-\delta} + w_2 q_{\mathrm{r},t-2\delta} + \ldots\right) \tag{9.6.17}$$

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

$$q_{\mathrm{s-c},t} = \sum_{k=1} q_{\mathrm{ec},t} \tag{9.6.18}$$

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

$$q_{\mathrm{rl},t} = \sum_{m=1} q_{\mathrm{el},t} \tag{9.6.19}$$

where $q_{\mathrm{el},t}$ = 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, $q_{\mathrm{xs},t}$, 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, $q_{\mathrm{xs},t}$, 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 $q_{\mathrm{xs},t}$ can be expressed as

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

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

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

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

$$\begin{aligned} q_{\mathrm{cs}} &= 60\overset{\circ}{V}_{\mathrm{s}} \rho_{\mathrm{s}} c_{pa} \left(T_{\mathrm{m}} - T_{\mathrm{cc}}\right) \\ q_{\mathrm{cl}} &= 60\overset{\circ}{V}_{\mathrm{s}} \rho_{\mathrm{s}} \left(w_{\mathrm{m}} - w_{\mathrm{cc}}\right) \end{aligned} \tag{9.6.21}$$

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_{rh}, in Btu/hr, can be calculated as

$$\begin{aligned} q_{\text{rl}} &= q_{\text{trans}} + q_{\text{if.s}} + q_{\text{ma}} + q_{\text{hu}} \\ &= \left[\left(\sum_{n=1} AU\right) + 60\,\mathring{V}_{\text{if}}\,\rho_{\text{o}} c_{pa}\,\mathring{m}_{\text{m}}\,c_{\text{m}}\right]\left(T_{\text{r}} - T_{\text{o}}\right) + \mathring{m}_{\text{w}}\,h_{\text{fg.58}} \end{aligned} \tag{9.6.22}$$

where A = area of the external walls, roofs, glasses, and floors, ft^2

U = overall heat transfer coefficient of the walls, roofs, glasses, and floors, Btu/hr.ft^2.°F

c_{m}, $\mathring{m}_{\text{m}}$ = specific heat and mass flow rate of the cold product entering the space per hour, Btu/lb.°F and lb/hr

$\mathring{m}_{\text{w}}$ = mass flow rate of water evaporated for humidification, lb/hr

$h_{\text{fg.58}}$ = 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, $q_{\text{if.s}}$ 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.

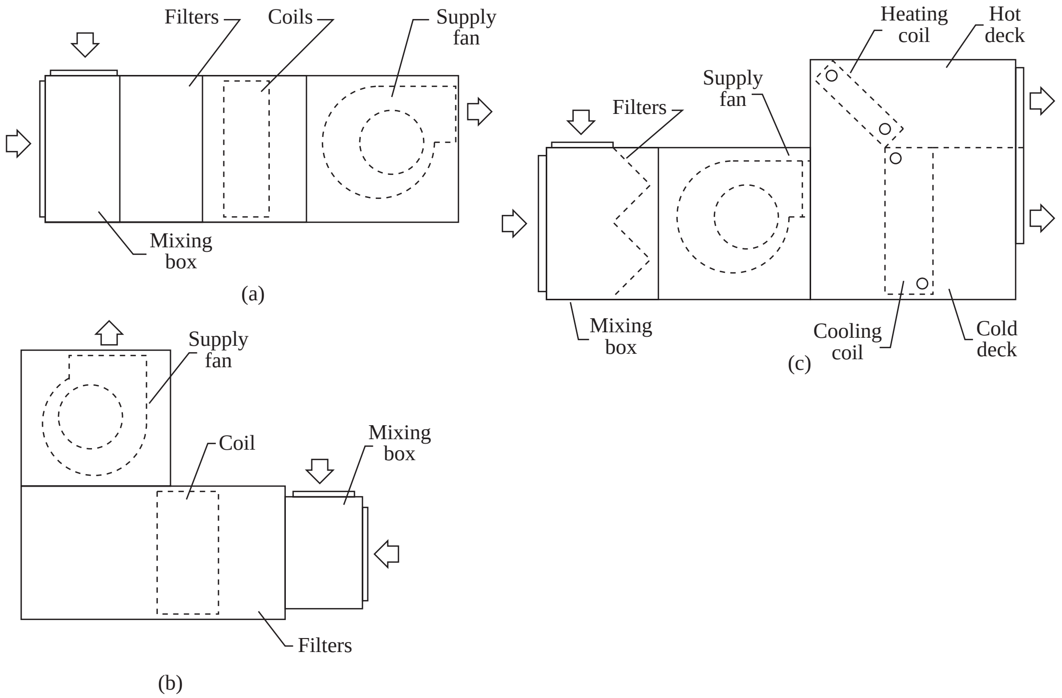


FIGURE 9.7.1 Type of air handling units: (a) horizontal draw-through unit, (b) vertical draw-through unit, and (c) multizone blow-through unit.

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

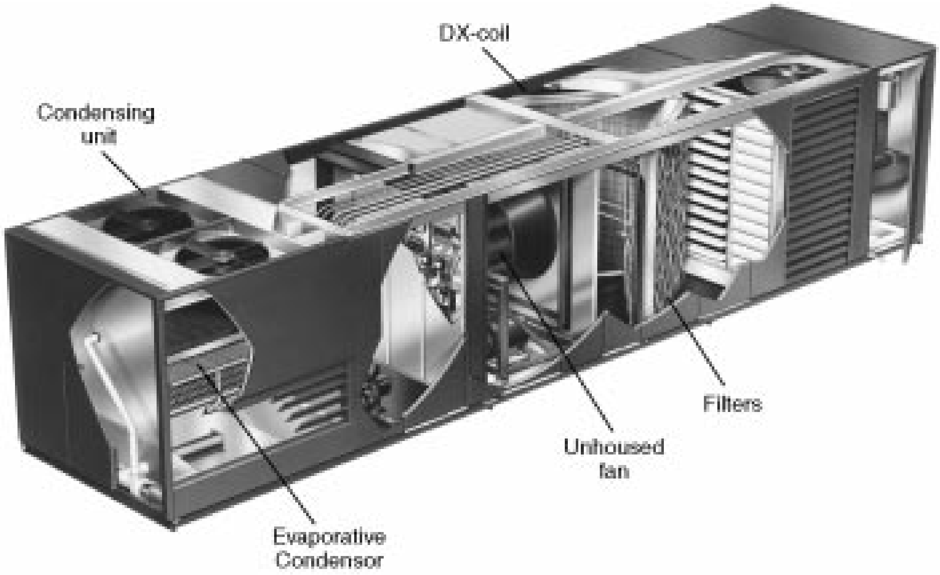


FIGURE 9.7.2 A cut view of a rooftop package unit. (Source: Mammoth, Inc. Reprinted by permission.)

- 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.

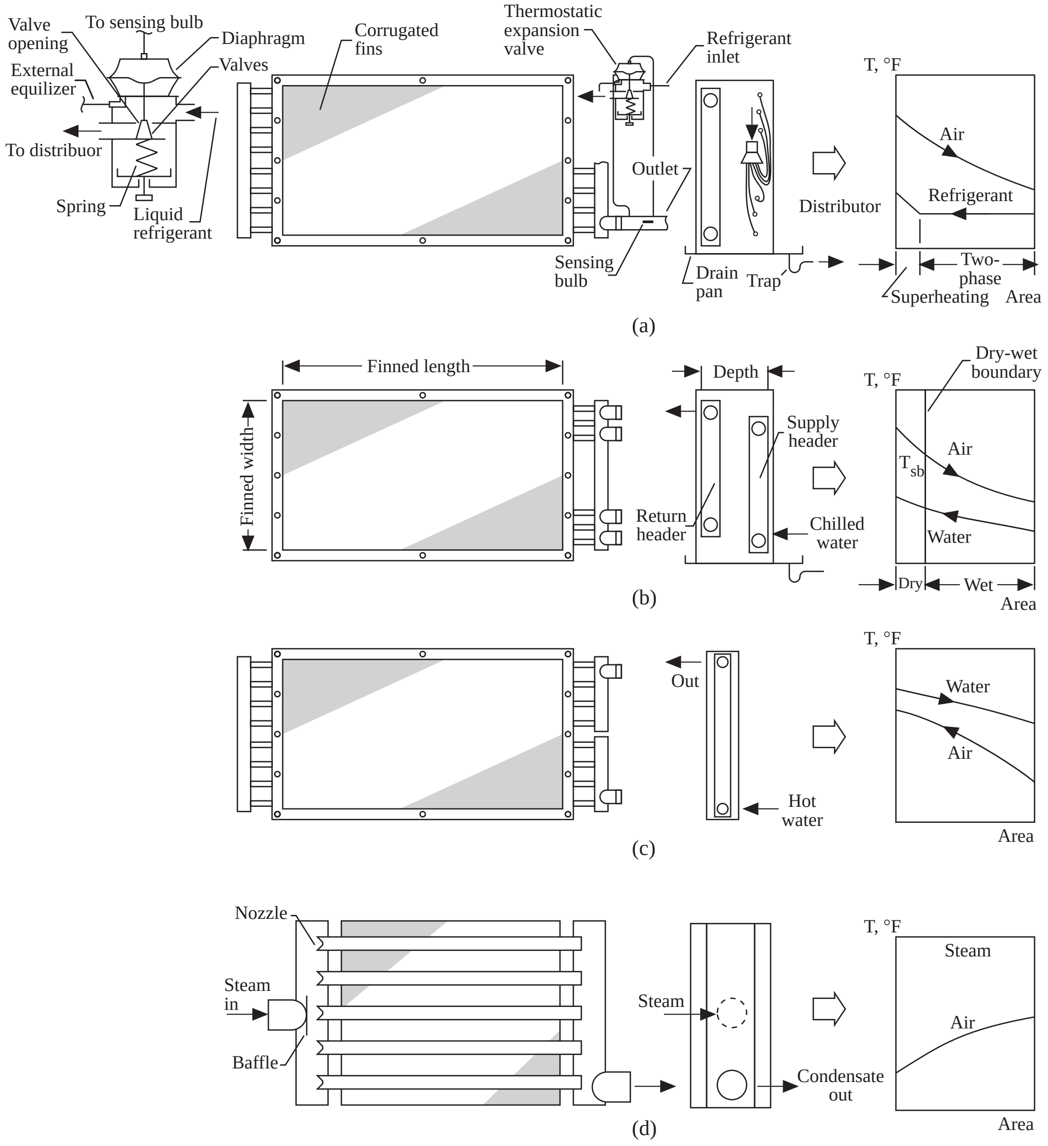


FIGURE 9.7.3 Types of coils: (a) direct expansion coil, (b) water cooling coil, (c) water heating coil, and (d) steam heating coil.

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 $\in_{dry}$ 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 serving 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/ft^2
- *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 μm that make up 10% of the total weight; 0.1% of atmospheric dusts is particles >1 μm 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 µm. According to U.S. Military Standard MIL-STD-282 (1956), a smoke cloud of uniform dioctyl phthalate (DOP) droplets 0.18 µm 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.

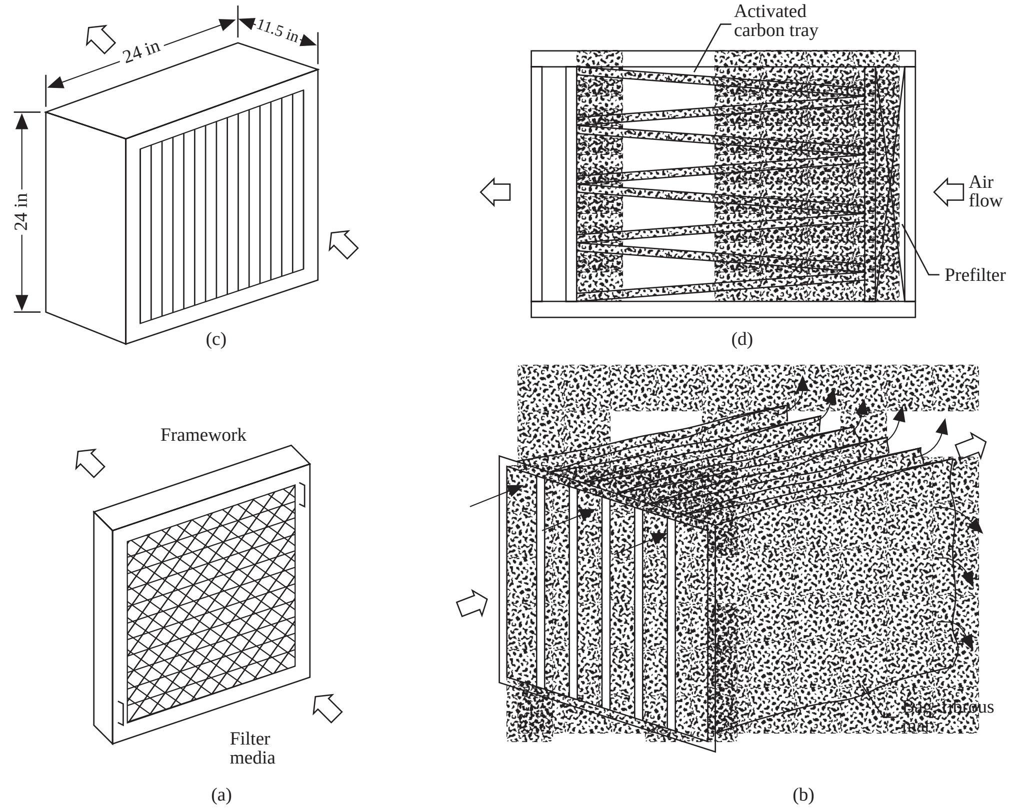


FIGURE 9.7.4 Various types of air filters: (a) low efficiency, (b) medium efficiency, (c) HEPA and ULPA filters, and (d) activated carbon filter.

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 ≥0.3 µm in diameter. *ULPA filters* have a DOP test efficiency of 99.999% for dust particles ≥0.12 µm in diameter.

A typical HEPA filter, shown in Figure 9.7.4(d), has dimensions of 24 × 24 × 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

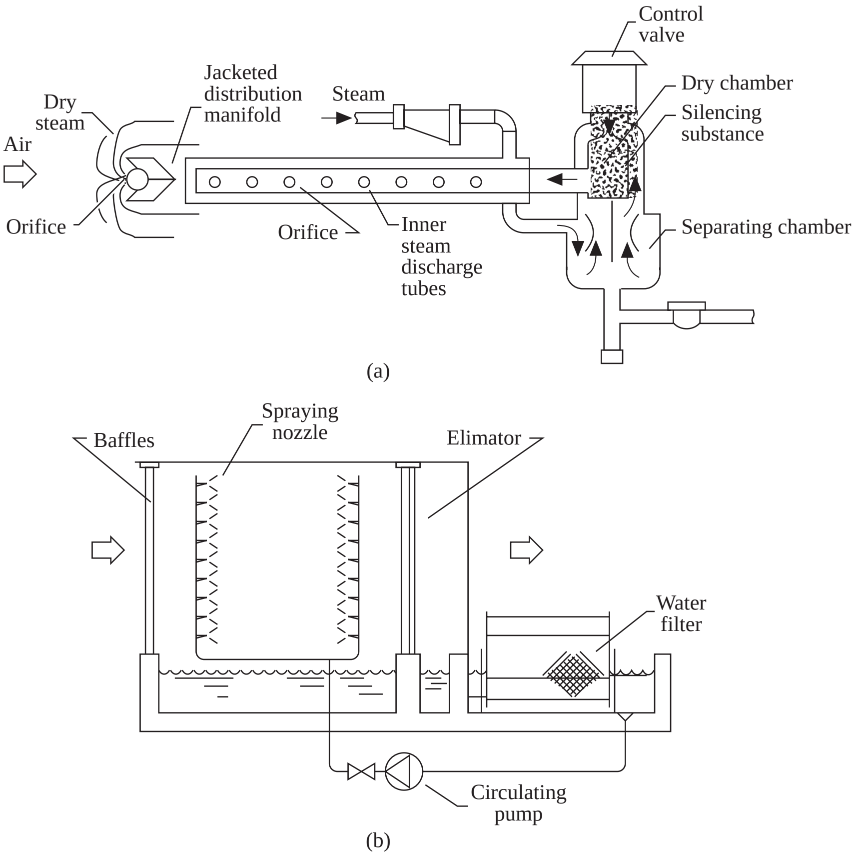
These filters are widely used to remove objectional odors and irritating gaseous airborne particulates, typically 0.003 to 0.006 µm in size, from the air stream by adsorption. *Adsorption* is physical condensation of gas or vapor on the surface of an activated substance like activated carbon. Activated 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 × 23 × 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.



(a)

(b)

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 ≥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
- *Semihermetic 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

$$\eta_v = \mathring{V}_{a.v} \big/ \mathring{V}_p \tag{9.8.1}$$

where $\mathring{V}_{a.v}$ = actual induced volume of the suction vapor at suction pressure, cfm
$\mathring{V}_p$ = 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

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

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

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

where $\overset{\circ}{m}_r$ = mass flow rate of refrigerant, lb/min
ρ_{suc} = density of suction vapor, lb/ft^3
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 $R_{com} = 4$ and $\eta_{isen} = 80\%$. 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.

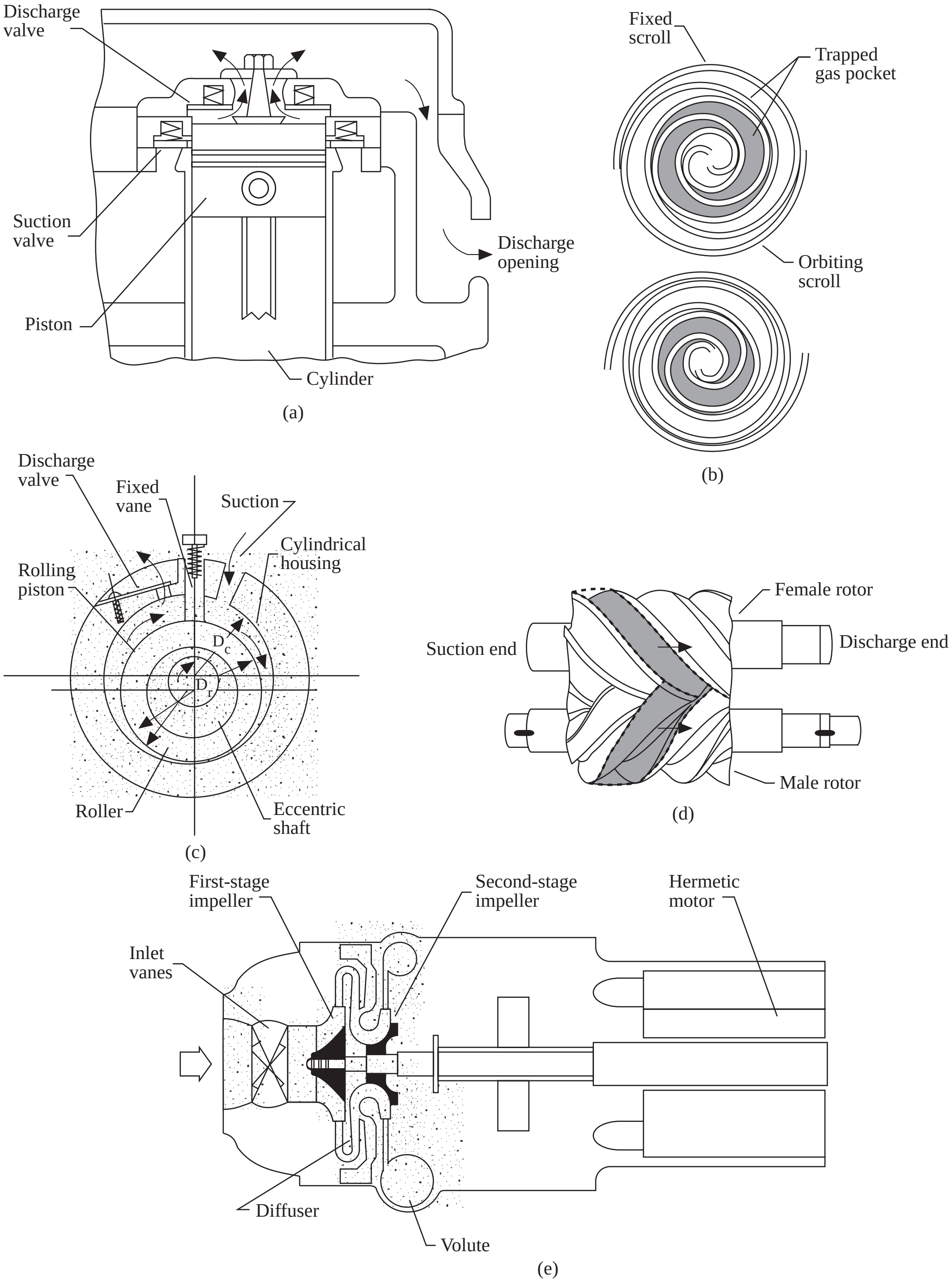


FIGURE 9.8.1 Various types of refrigeration compressors: (a) reciprocating, (b) scroll, (c) rotary, (d) twin-screw, and (e) centrifugal.

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

$$Q_{\text{rej}} = U_{\text{con}} A_{\text{con}} \Delta T_{\text{m}} = 60\, \overset{\circ}{m}_{\text{r}} \left(h_2 - h_3'\right) = q_{\text{rl}} + \left(2545 P_{\text{com}}\right) / \eta_{\text{mo}} \tag{9.8.4}$$

where U_{con} = overall heat transfer coefficient across the tube wall in the condenser, Btu/hr.ft^2.°F
A_{con} = condensing area in the condenser, ft^2
ΔT_{m} = logarithmic temperature difference, °F
$\overset{\circ}{m}_{\text{r}}$ = 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 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

$$F_{\text{rej}} = Q_{\text{rej}} / q_{\text{rl}} = 1 + \left(2545 P_{\text{com}}\right) / \left(q_{\text{rl}} \eta_{\text{mo}}\right) \tag{9.8.5}$$

Fouling factor R_{f}, in hr.ft^2.°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 $\overset{\circ}{V}_{\text{ca}} / Q_{\text{u.rej}}$ 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 $(T_{\text{ca.l}} - T_{\text{o}})$ — 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 $(T_{\text{con.i}} - T_{\text{o}})$. 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.

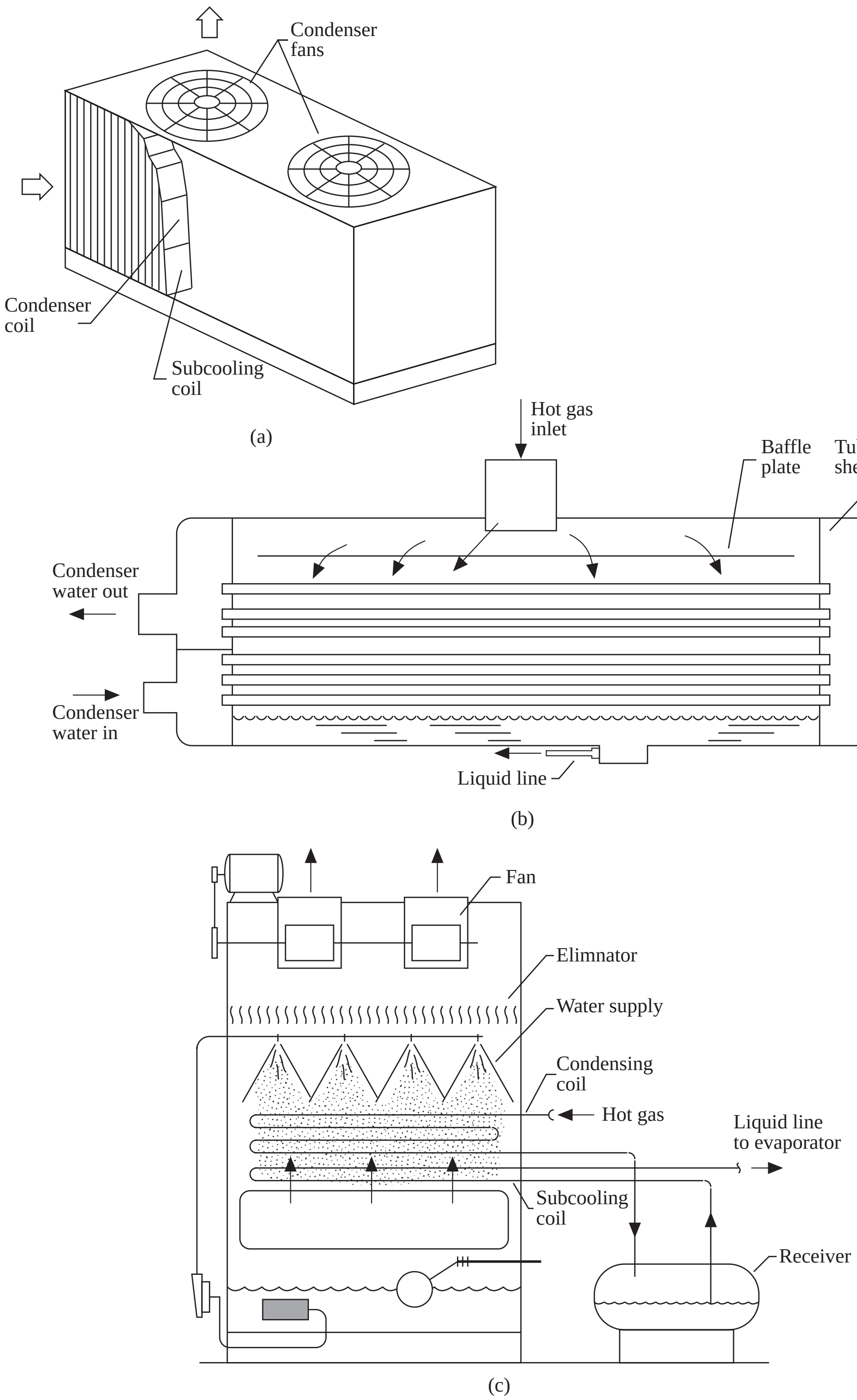


FIGURE 9.8.2 Various types of refrigeration condensers: (a) air-cooled, (b) shell-and-tube water-cooled, and (c) evaporative cooled.

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 $\overset{\circ}{V}_{\mathrm{ca}}/Q_{\mathrm{u.rej}}$ 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°F and some specify a minimum operating temperature at T_{o} = 0°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 °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.ft^2.°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_{con} = 110°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.

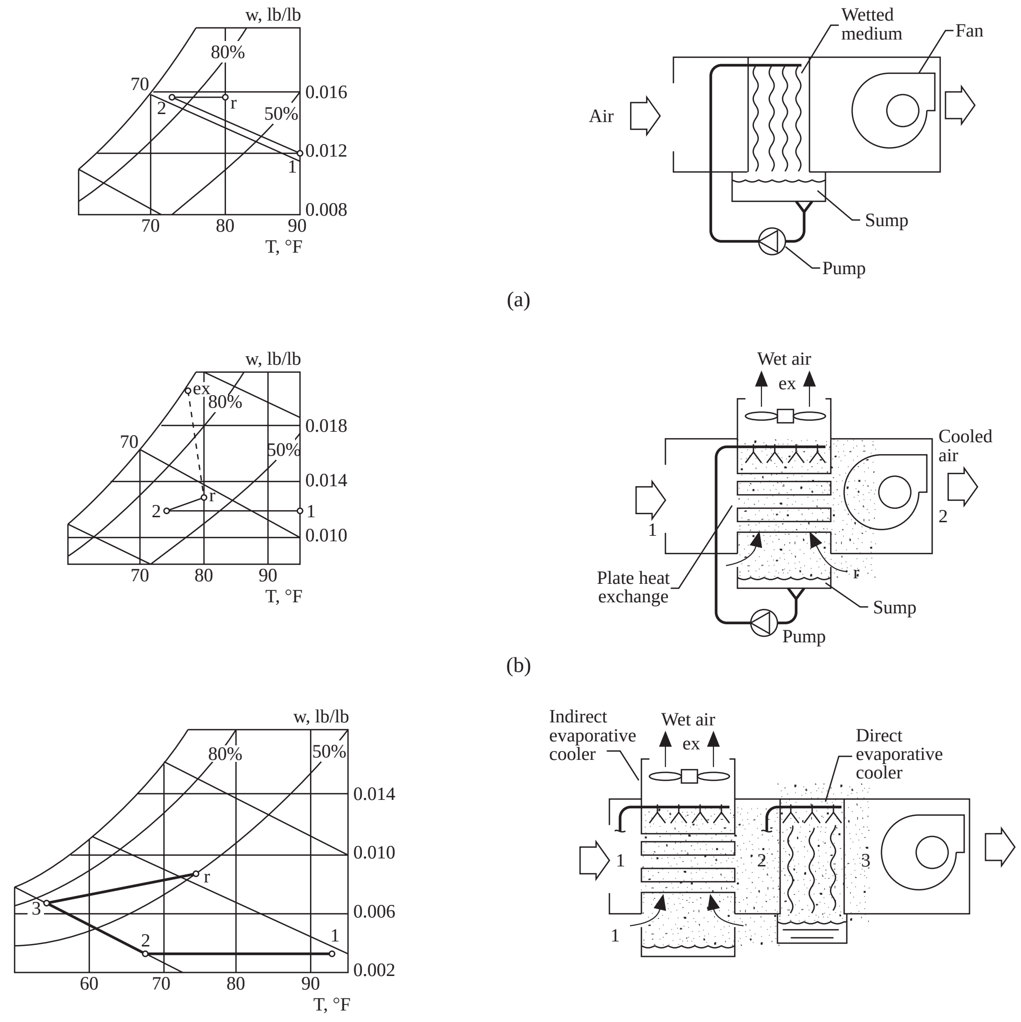
Direct Evaporative Cooler

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:

$$\varepsilon_{\text{sat}} = \left(T_{\text{ae}} - T_{\text{al}}\right) \big/ \left(T_{\text{ae}} - T_{\text{ae}}^{*}\right) \tag{9.8.6}$$

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.



(a)

(b)

(c)

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:

$$e_{\mathrm{in}} = \left(T_{\mathrm{ca.e}} - T_{\mathrm{ca.l}}\right)/\left(T_{\mathrm{ca.e}} - T_{\mathrm{s.a}}\right) \tag{9.8.7}$$

where $T_{\mathrm{ca.e}}$, $T_{\mathrm{ca.l}}$ = temperature of cooled air entering and leaving the indirect evaporative cooler, °F, and $T_{\mathrm{s.a}}$ = 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 $\varepsilon_{\mathrm{sat}}$ 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_{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 $\varepsilon_{\mathrm{sat}}$ = 0.8.

In locations where outdoor wet bulb $T'_o \leq 60°F$, a direct evaporative can often provide an indoor environment of 78°F and a relative humidity of 60%. In locations $T'_o \leq 68°F$, an indirect–direct evaporative cooler can maintain a comfortable indoor environment. In locations $T'_o \geq 72°F$, 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 $T'_o \leq 60°F$.