

Refrigeration

Cooling is a fundamental operation in food processing and preservation. Removal of heat could involve either the transfer of heat from one fluid to another or from a solid to a fluid, or it could be accomplished by vaporization of water from a material under adiabatic conditions. Knowledge of the principles of heat transfer is an essential prerequisite to the understanding of the design and operation of refrigeration systems.

Maintaining temperatures lower than ambient inside a system requires both the removal of heat and prevention of incursion of heat through the system's boundaries. The rate of heat removal from a system necessary to maintain the temperature is the refrigeration load. Refrigeration systems must be sized to adequately handle the refrigeration load. When heat has to be removed from a system continuously, at temperatures below ambient and for prolonged periods, a mechanical refrigeration system acts as a pump that extracts heat at low temperatures and transfers this heat to another part of the system where it is eventually dissipated to the surroundings at a higher temperature. The operation requires energy, and a well-designed system will allow the maximum removal of heat at minimum energy cost.

10.1 MECHANICAL REFRIGERATION SYSTEM

10.1.1 Principle of Operation: The Heat Pump

The second law of thermodynamics mandates that heat will flow only in the direction of decreasing temperature. In a system that must be maintained at a temperature below ambient, heat must be made to flow in the opposite direction. A refrigeration system may be considered as a pump that conveys heat from a region of low temperature to another region that is at a high temperature.

The low temperature side of a refrigeration system is maintained at a lower temperature than the system it is cooling to allow spontaneous heat flow into the refrigeration system. The high temperature side must have a temperature higher than ambient to allow dissipation of the absorbed heat to the surroundings. In some instances, this absorbed heat is utilized as a heat source for use in heating processes.

Maintaining a high and a low temperature in a refrigeration system is made possible by the use of a refrigerant fluid that is continuously recirculated through the system. A liquid's boiling or condensation temperature is a function of the absolute pressure. By reducing the pressure, a low boiling temperature is made possible, allowing for absorption of heat in the form of the heat of the refrigerant's vaporization

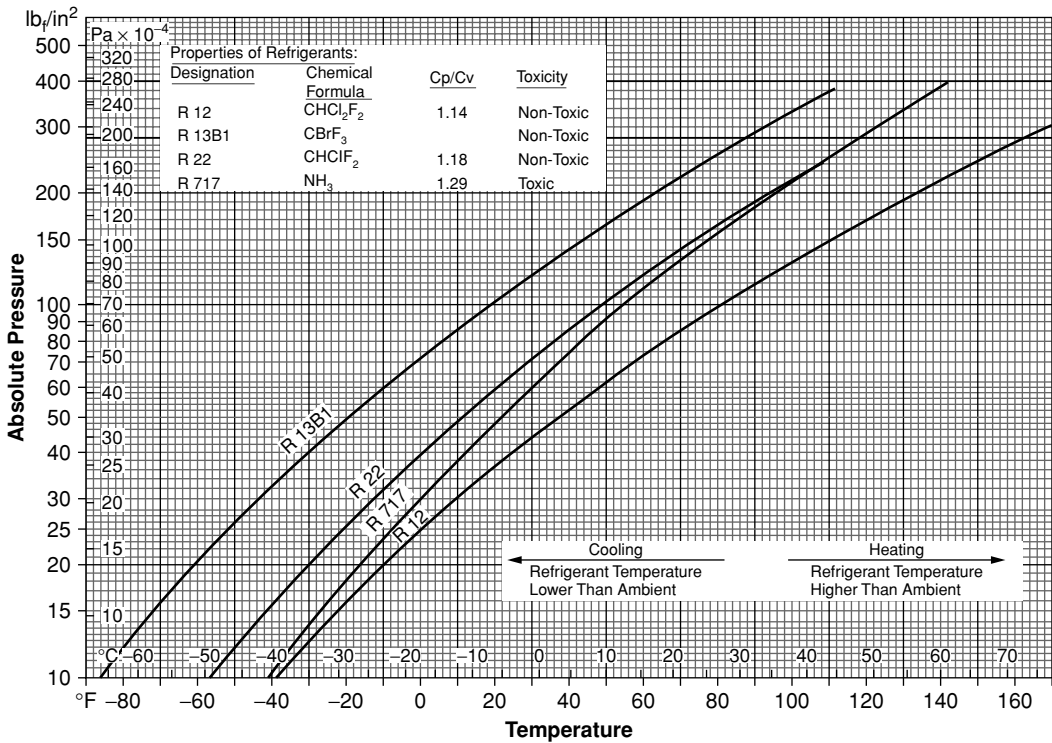


Figure 10.1 Vapor pressure of commonly used refrigerants as a function of temperature.

as it is vaporized at the low pressure and temperature. The vapors, when compressed to a high pressure, will condense at the high temperature and the absorbed heat will be released from the refrigerant as it condenses back into liquid at the high temperature and pressure. Figure 10.1 shows the vapor pressure versus temperature of commonly used refrigerants. The diagram also illustrates how this pressure and temperature relationship is utilized for cooling and heating.

10.1.2 Refrigerants

Atmospheric contamination of refrigerants that contain chlorine and fluorine in the molecule (chlorofluorocarbon; CFC) has been shown to contribute to global warming and cause ozone depletion in the upper atmosphere. Thus by international agreement, manufacturing of CFCs was stopped after 1996. CFCs have the highest ozone depletion potential (ODP) among the refrigerants and also has high global warming potential (GWP). Manufacturing of hydrochlorofluorocarbon (HCFC) refrigerants is to be phased out in 2030. HCFCs have lower ODP but also contribute to global warming. Because no new CFCs are being manufactured, replacement in existing refrigeration systems must come from recovered CFCs or the refrigeration unit must be recharged with a completely new refrigerant. Non-CFC refrigerants must be used in new refrigeration systems. Existing refrigeration systems may be

Table 10.1 Refrigerants Used in the Food Industry

<i>Refrigerant</i>	<i>Composition</i>	<i>Boiling Point at 1° C (° F)</i>
R12	CCl_2F_2	−30 (−22)
R22	CHClF_2	−41 (−41)
R32	$\text{C}_2\text{H}_2\text{F}_4$	−52 (−62)
R115	CClF_2CF_3	−39 (−38)
R124	CHClFCF_3	−12 (10)
R125	CHF_2CF_3	−49 (−56)
R134A	$\text{CH}_2\text{FCH}_2\text{F}$	−26 (−15)
R142B	$\text{CH}_3\text{CCl}_2\text{F}$	−10 (14)
R143A	CH_3CF_3	−25 (−13)
R152A	CH_3CHF_2	−25 (−13)
R290	$\text{CH}_3\text{CH}_2\text{CH}_3$	−42 (−44)
C318	CA (Cyclic)	−6 (23)
R401A	R22, R152A, R124(53:13:34)	
R401B	R22, R152A, R124(61:11:28)	
R402A	R125, R290, R22 (60:2:38)	
R402B	R125, R290, R22 (38:2:60)	
R404A	R125, R143A, R134A (44:52:4)	
R405A	R22, R152A, R142B, C318 (45:7:5.5:42.5)	
R407A	R32, R125, R134A (20:40:40)	
R407B	R32, R125, R134A (10:70:20)	
R407C	R32, R125, R134A (23:25:52)	
R410A	R32, R125 (50:50)	
R502	R22, R115 (48.8:51.2)	−45 (−49)
R507A	R125, R143A (50:50)	−46.7 (−52.1)

R 502 and R507A are azeotropes, while the R400 series are zeotropes

Source: ASHRAE Standards 2000. Standards for Designation and Safety of Refrigerants.

upgraded by utilizing a drop-in refrigerant (i.e., the new refrigerant is added after the original CFC refrigerant has been removed).

Refrigerants now used in the food industry include R502 for transport refrigeration, R502 and R22 for retail display cases and retail central storage, R502 for cold storage, R22 for refrigerated storage and refrigerated vending machines, and R717 for large freezers, frozen storage warehouses, and large refrigerated warehouses. R502 contains R12, which is no longer manufactured. R22 can still be used but will be phased out later. Thus, alternatives to these refrigerants must be selected for retrofits to existing installations or for new installations. For long-term retrofit, alternatives are R407A, R407B, and R507; R407C and R410A and R134A for R502, R22, and R12, respectively.

Table 10.1 lists the refrigerants used in the food industry and their boiling points. Also listed are alternative non-CFC refrigerants. Most alternatives are mixtures that are either azeotropes or zeotropes. Azeotropes are mixtures with a constant boiling point and where the vapor and liquid composition are the same. Zeotropes are mixtures where vapor and liquid compositions are different at different temperatures and there is a large gap between the dew point (the temperature where liquid first starts to form) and the bubble point (the temperature where the mixture is completely in the liquid phase).

The use of alternative refrigerants may require different compressor lubricant, so this has to be taken into consideration in retrofits.

10.1.3 The Refrigeration Cycle

Figure 10.2 shows a schematic diagram of a mechanical refrigeration system. The heart of the system is the *compressor*.

When the compressor is operating, refrigerant gas is drawn into the compressor continuously. Low pressure is maintained at the suction side and because of the low pressure, the refrigerant can vaporize at a low temperature. In the compressor, the refrigerant gas is compressed increasing in both pressure and temperature during the process. The hot refrigerant gas then flows into a heat exchange coil called the *condenser* where heat is released in the process of condensation at constant pressure and temperature. From the condenser, the liquid refrigerant flows into a liquid refrigerant holding tank. In small systems, there may be no holding tank, and the refrigerant just continuously cycles through the system.

When there is a demand for cooling, the liquid refrigerant flows from the holding tank to the low pressure side of the refrigeration system through an *expansion valve*. The drop in pressure that occurs as the refrigerant passes through the expansion valve does not change the heat content of the refrigerant. However, the temperature drops to the boiling temperature of the liquid at the low pressure. The cold liquid refrigerant then flows to another heat exchange coil called the *evaporator* where the system performs its cooling function, and heat is absorbed by the refrigerant in the process of vaporization at constant temperature and pressure. From the evaporator, the cold refrigerant gas is drawn into the suction side of the compressor thus completing the cycle.

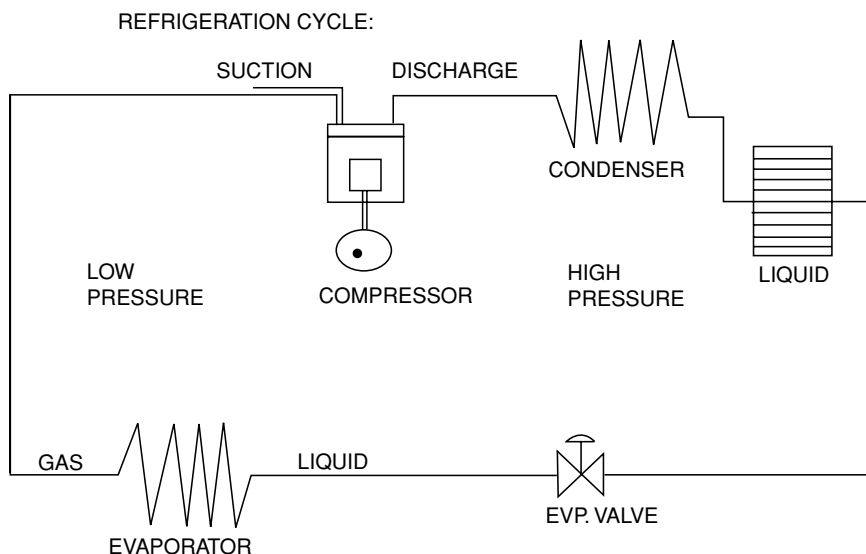


Figure 10.2 Schematic diagram of a refrigeration system.

A refrigeration system is usually equipped with low-pressure and high-pressure cut-off switches that interrupt power to the compressor when either the high-pressure set-point is exceeded (this occurs if cooling capacity of the condenser is inadequate) or when the pressure drops below the low-pressure set-point (this occurs when the compressor is running but the cooling demand is much less than the capacity of the refrigeration system). The low-pressure set-point can be used to control the evaporator temperature. In some systems, liquid refrigerant flow through the expansion valve is thermostatically controlled, interrupting flow when the evaporator temperature is lower than the set-point. In addition, refrigeration systems are equipped with thermostats that interrupt power to the compressor when the temperature of the refrigerated room reaches a set temperature.

A refrigeration system may also be used for heating. A system that alternates heating and cooling duty is called a *heat pump*. These units are used extensively for domestic heating or cooling in areas where winter temperatures are not too severe. In these units, either a low-pressure or a high-pressure refrigerant could go through the heat exchange coils that constitute the evaporator and condenser; thus, either could act as a heating or cooling coil depending upon the duty expected of the system. The ability of heat pumps to deliver heat with low power consumption is due to the fact that power is used only to pump energy from a low temperature to a higher temperature. The heat is not derived completely from the power supplied but rather it is extracted from cooler air from the surroundings. The efficiency of heat pumps in transforming applied power to heat varies inversely with the temperature differential between the temperature of the medium from which heat is extracted and the temperature of the system that is being heated. Heat pump systems can be used as a means of recovering energy from low temperature heat sources for use in low temperature heat applications such as dehydration.

10.1.4 The Refrigeration Cycle as a Series of Thermodynamic Processes

Starting from the compressor in Fig. 10.2, low-pressure gas is compressed adiabatically to high pressure, which should allow condensation at ambient temperature. Work is required to carry out this process, and this energy is supplied in the form of electrical energy to drive the compressor motor. The gas also gains in enthalpy during this compression process. At the condenser, the gas condenses and transfers the latent heat of condensation to the surroundings. There is a loss of enthalpy in a constant pressure process. At the expansion valve, the liquid expands to low pressure at constant enthalpy, while at the evaporator, the liquid evaporates at constant pressure and gains in enthalpy. The two processes crucial to the efficiency of the refrigeration system are the adiabatic compression process where energy is applied and the isobaric expansion process where energy is extracted by the refrigerant from the system. Work and enthalpy change in an adiabatic compression process has been derived in Chapter 4 in the section “Work and Enthalpy Change on Isothermal Expansion or Compression of an Ideal Gas.” The ratio of enthalpy change to work in adiabatic compression of an ideal gas is equal to the specific heat ratio.

$$\frac{\Delta H}{W} = -\gamma; \quad W = \frac{-\Delta H}{\gamma} \quad (10.1)$$

The negative sign indicates that energy is being used on the system.

10.1.5 The Refrigeration Cycle on the Pressure/Enthalpy Diagram for a Given Refrigerant

The thermodynamic properties of refrigerants when plotted on a pressure-enthalpy diagram result in a plot that is useful in determining capacities and power requirement for a refrigeration system.

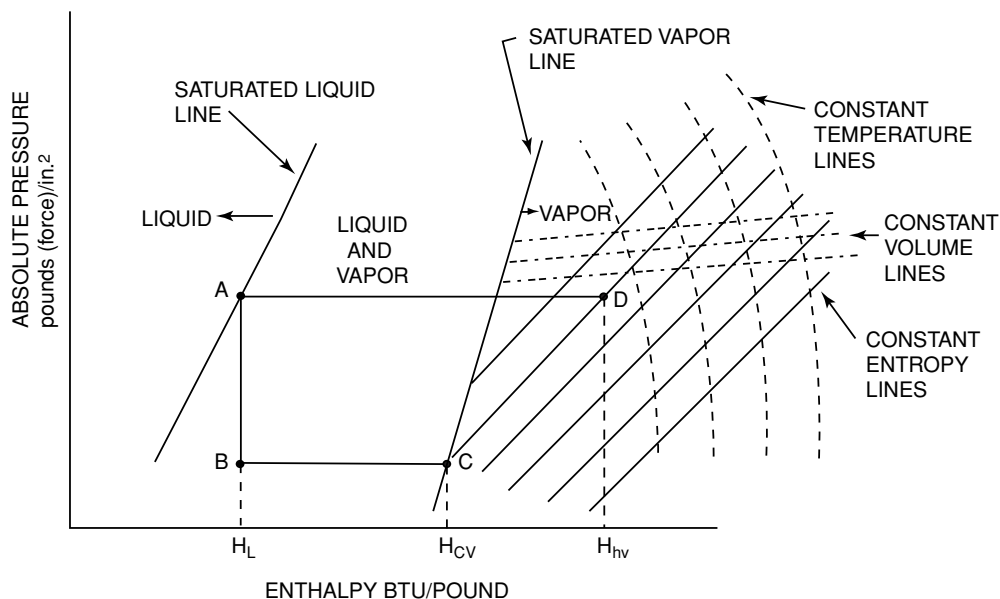


Figure 10.3 The refrigeration cycle on a pressure-enthalpy diagram.

Figure 10.3 is a schematic diagram of a refrigeration cycle on a pressure-enthalpy diagram. The diagram consists of lines representing the vapor and liquid pressure-enthalpy relationship, lines representing change in enthalpy with pressure during adiabatic compression (constant entropy), and in some charts, lines representing specific volumes at various pressures and enthalpies.

Point A in Fig. 10.3 represents the liquid refrigerant at high pressure entering the expansion valve. The refrigerant expands at constant enthalpy (H_L) as it goes through the expansion valve and leaves the valve as a mixture of liquid and vapor at a lower pressure, represented by point B. As the refrigerant absorbs heat in the evaporator, it gains in enthalpy represented by the line BC. The refrigerant leaves the evaporator as saturated vapor (represented by point C) having the enthalpy H_{cv} . The compressor raises the pressure and the change is represented by the line CD that parallels the lines of constant entropy. As the compressed refrigerant gas leaves the compressor at point D, it will have an enthalpy represented by H_{hv} . At the condenser, heat is dissipated resulting in a drop in enthalpy represented by line AD. The liquid refrigerant leaves the condenser with the pressure and enthalpy represented by point A.

The cooling capacity of the refrigeration system is represented by the length of line BC.

$$\text{Cooling capacity} = H_{cv} - H_L \quad (10.2)$$

The condenser heat exchange requirement or condenser duty is represented by the length of line AD.

$$\text{Condenser duty} = H_{hv} - H_L \quad (10.3)$$

The change in enthalpy due to compression ΔH_c is

$$\Delta H_c = H_{hv} - H_{cv} \quad (10.4)$$

From Equation (10.1), the work required for compression is

$$W = \frac{-\Delta H_c}{\gamma} = \frac{-(H_{hv} - H_{cv})}{\gamma} \quad (10.5)$$

If M is the mass of refrigerant recirculated through the system per unit time, the power requirement can be calculated as follows:

$$P = \frac{W}{\text{time}} = M \frac{-(H_{hv} - H_{cv})}{\gamma} \quad (10.6)$$

The negative sign on the work and power indicates that work is being added to the system.

The efficiency of a refrigeration system is also expressed in terms of a *coefficient of performance* (COP), which is a ratio of the cooling capacity over the gain in enthalpy due to compression.

$$\text{COP} = \frac{H_{cv} - H_L}{H_{hv} - H_{cv}} \quad (10.7)$$

The power requirement (P) can be expressed in terms of the coefficient of performance using Equations (10.6) and (10.7).

$$P = \frac{H_{cv} - H_L}{\gamma(\text{COP})} \cdot M \quad (10.8)$$

The refrigeration capacity is expressed in tons of refrigeration, the rate of heat removal sufficient to freeze 1 ton (2000 lb) of water in 24 hours. Because the heat of fusion of water is 144 BTU/lb, this rate of heat removal is equivalent to 12,000 BTU/h. The refrigeration capacity in tons is

$$(\text{tons})_r = \frac{(H_{cv} - H_L)(M)}{12,000} \quad (10.9)$$

Substituting Equation (10.9) in Equation (10.8):

$$P = \frac{(\text{tons})_r(12,000)}{\gamma(\text{COP})} \left[\frac{\text{BTU}}{h(\text{ton})_r} \right]$$

Expressing the power requirement in horsepower (HP)/(ton)_r:

$$\frac{\text{HP}}{(\text{tons})_r} = \frac{12,000}{\gamma(\text{COP})} \frac{\text{BTU}}{h(\text{ton})_r} \frac{1\text{HP}}{2545 \text{ BTU/h}} = \frac{4.715}{\gamma(\text{COP})} \quad (10.10)$$

Because there is a certain degree of slippage of refrigerant past the clearance between the cylinder and the piston, particularly at the high pressures and also because some frictional resistance occurs between the piston and the cylinder, the actual work expended would be higher than that determined using Equation (10.10). The ratio between the theoretical horsepower as calculated from Equation (10.10) and the actual horsepower expended is the efficiency of compression. The efficiency of compression depends upon the ratio of high side to low side pressure across the compressor and will be discussed later in the text.

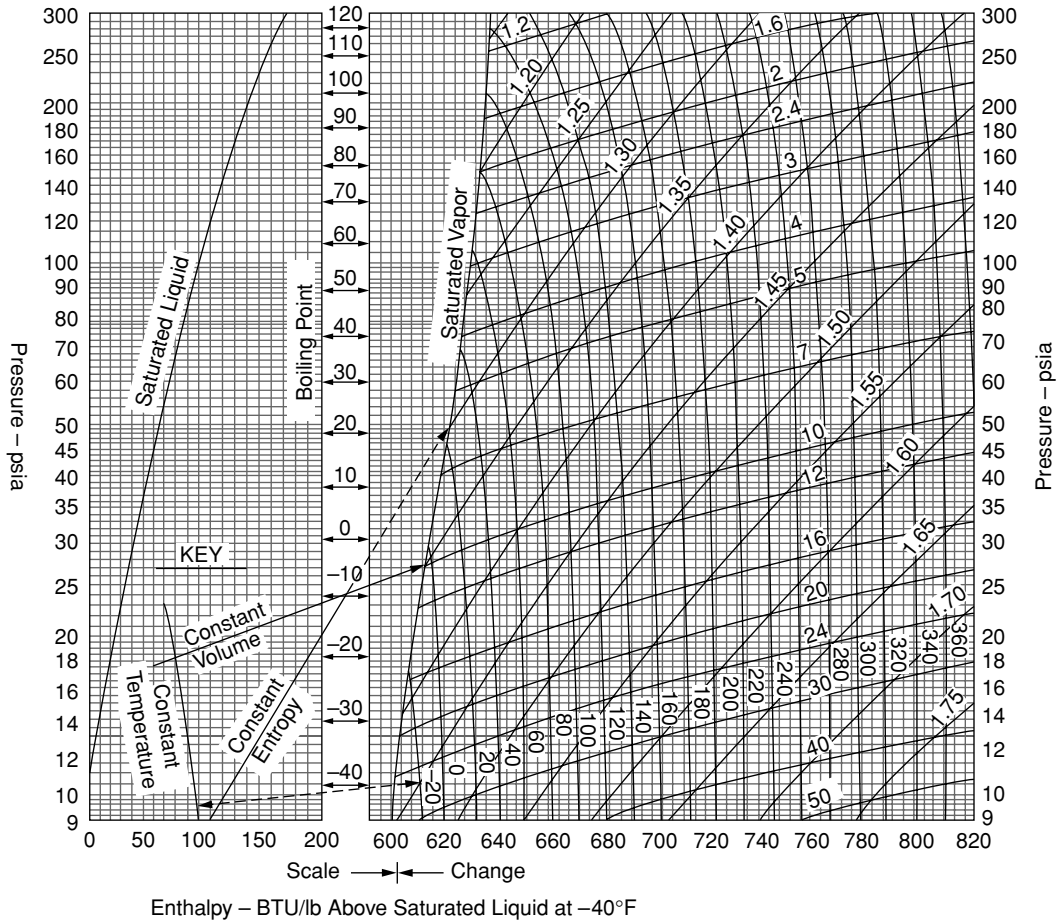


Figure 10.4 Pressure-enthalpy diagram for refrigerant 717 (NH₃). (Adapted from ASHRAE Guide and Data Book, Fundamentals and Equipment, 1965.)

Pressure-enthalpy diagrams for ammonia, Freon 12 and Freon 22 are shown in Figs. 10.4, 10.5, and 10.6.

10.1.5.1 Example Problems on the Use of Refrigerant Charts

Example 10.1. A refrigerated room is to be maintained at 40°F (4.44°C) and 80% relative humidity. The contents of the room vaporize moisture requiring removal of moisture from the air to maintain the desired relative humidity. Assuming that air flow over the evaporator coil is such that the bulk mean air temperature reaches to within 2°F (1.11°C) of the coil temperature, determine the low side pressure for an ammonia refrigeration system such that the desired humidity will be maintained.

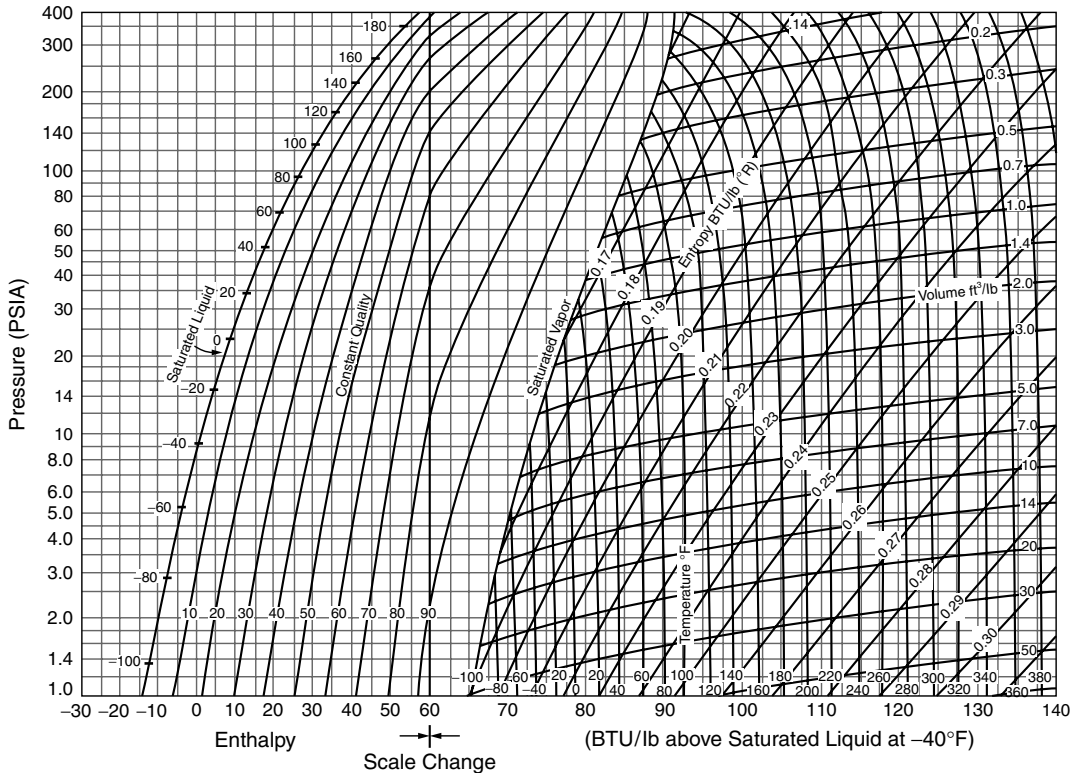


Figure 10.5 Pressure-enthalpy diagram for refrigerant 12 (CCl_2F_2). (Courtesy of E. I. Du Pont de Nemours Inc.)

Solution:

From a psychrometric chart, the dew point of air at 40°F and 80% RH is 33°F (0.56°C). When room air is cooled to 33°F passing over the evaporator coils, any moisture present in excess of the saturation humidity at this temperature will condense. If air flow over the coil is sufficient to allow enough moisture removal through condensation to equal the rate of moisture vaporization into the air, then the desired relative humidity will be maintained. The coil temperature should be 31°F (-0.56°C). From Fig. 10.1 for ammonia, the pressure corresponding to a temperature of 31°F is 61 psia (420 kPa absolute).

Example 10.2. A heat pump is proposed as a means of heating a cabinet drier. Air is continuously recycled through the system passing through the condenser coil at the air inlet to the drying chamber where the air is heated. Prior to recycling, the hot moist air leaving the drying chamber is passed through the evaporator coil where moisture is removed by condensation. Inlet air to the drying chamber is at 140°F (60°C) and 4% relative humidity. Assuming that the evaporator and condenser coil design and air flow are such that air temperature approaches 5°F (2.78°C) of the evaporator coil temperature and log mean ΔT between condenser coil and air is 20°F (5.56°C), determine the high and low side pressure for the refrigeration system using refrigerant 22 as the refrigerant.

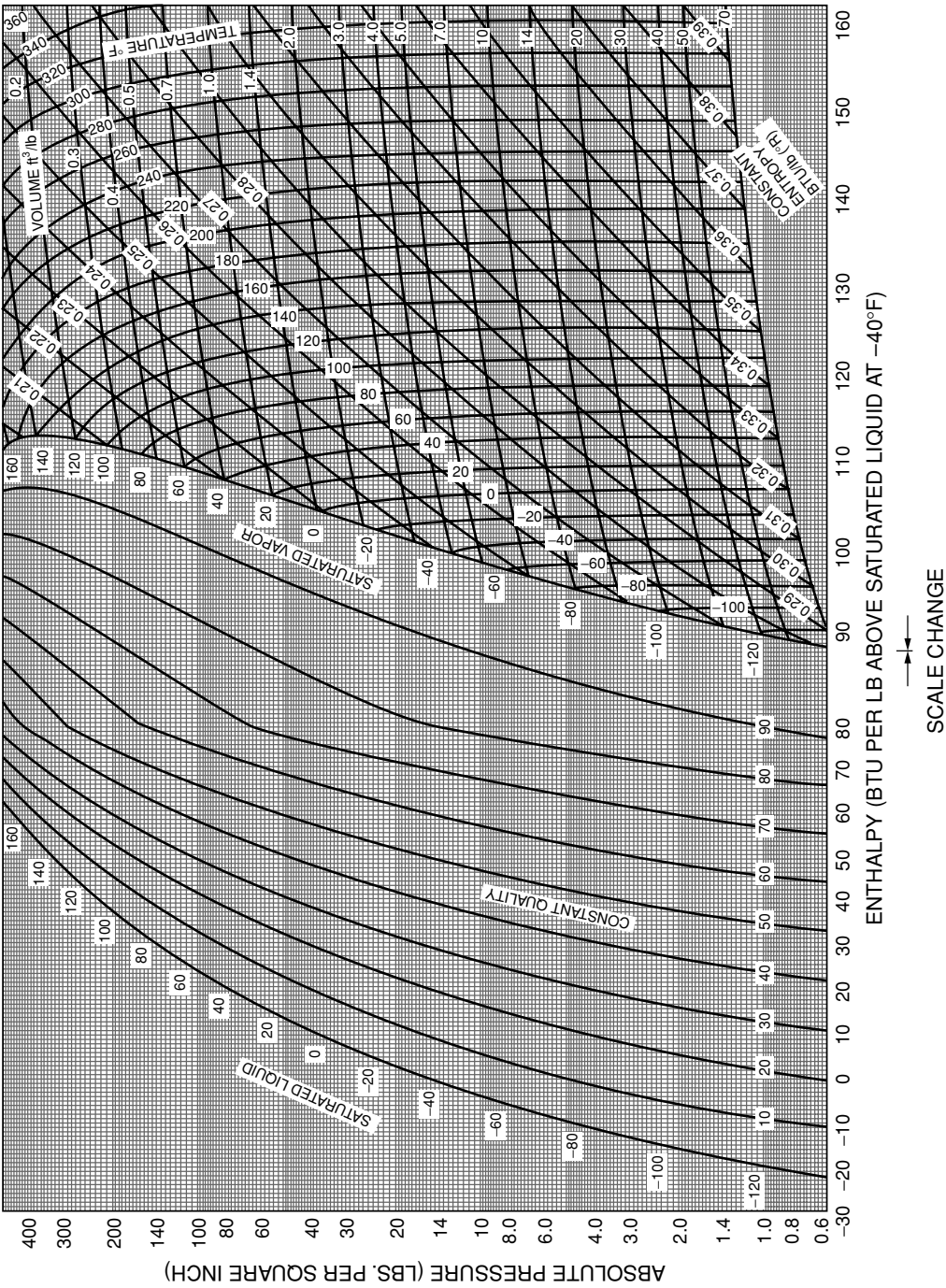


Figure 10.6 Pressure-enthalpy diagram for refrigerant 22 (CHClF₂). (Courtesy of E. I. Du Pont de Nemours Inc.)

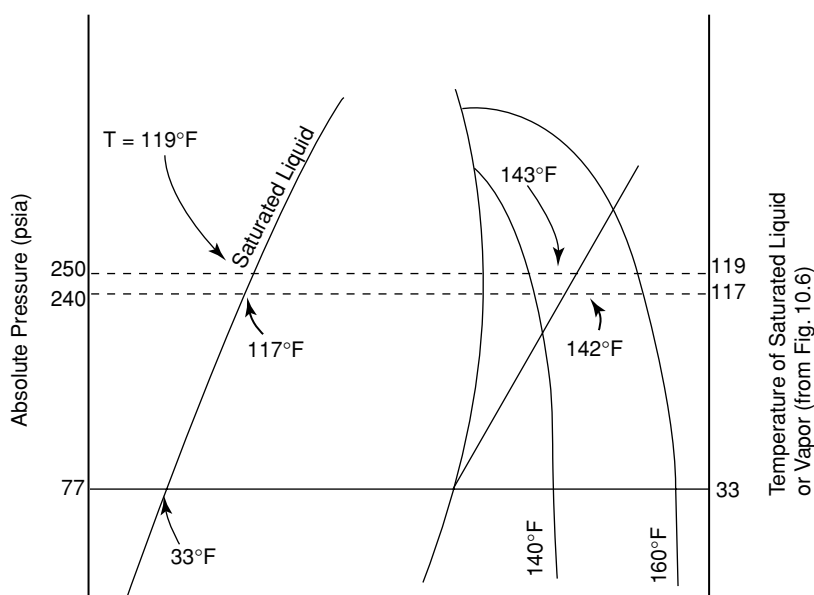


Figure 10.7 Pressure-enthalpy diagram of the heat pump system in Example 10.2.

Solution:

From a psychrometric chart, the dew point of air at 140°F (60°C) and 4% RH is 38°F (3.33°C). Thus, the evaporator coil temperature should be at 33°F (0.56°C). The condenser coil temperature and pressure will be calculated from the specified log mean ΔT .

From Fig. 10.1, for refrigerant 22, the low side pressure corresponding to 33°F (0.56°C) is 77 psia (530 kPa absolute).

At the condenser, air enters at 38°F and leaves at 140°F. If the system is set-up in countercurrent flow, the log mean ΔT can be calculated as follows:

$$\overline{\Delta T} = \frac{(T_1 - 38) - (T_g - 140)}{\ln(T_1 - 38)/(T_g - 140)}$$

where T_1 and T_g refer to the temperature of refrigerant liquid and gas, respectively.

Figure 10.7 shows the pressure-enthalpy diagram for the system under the conditions given.

The high pressure will be determined by trial and error from Fig. 10.6. Following the constant entropy line from a pressure of 77 psia, a high pressure of 250 psia will give a hot refrigerant gas temperature of 143°F and a liquid temperature of 119°F. The log mean ΔT is

$$\overline{\Delta T}_1 = \frac{(119 - 38) - (143 - 140)}{\ln 92/3} = 23.6$$

This is higher than the 20°F ΔT specified. At a pressure of 240 psia, the hot refrigerant gas temperature is 142°F and the liquid temperature is 117°F.

$$\overline{\Delta T}_1 = \frac{(117 - 38) - (142 - 140)}{\ln 79/2} = 20.9$$

This calculated value for the log mean ΔT is almost equal to the specified 20°F ; therefore, the high side pressure for the system should be at 240 psia (1650 kPa).

Example 10.3. A refrigeration system is to be operated at an evaporator coil temperature of -30°F (-34.4°C), and a condenser temperature of 100°F (37.8°C) for the liquid refrigerant. For (a) refrigerant 12 and (b) refrigerant 717, determine:

1. the high side pressure,
2. the low side pressure,
3. refrigeration capacity per unit weight of refrigerant,
4. the coefficient of performance,
5. the theoretical horsepower of compressor per ton of refrigeration, and
6. the quantity of refrigerant circulated through the system per ton of refrigeration.

Solution:

(a) For refrigerant 12:

The high and low side pressures (1 and 2) are determined from Fig. 10.1. At -30°F (-34.4°C), pressure is 12.3 psia (85 kPa). The high side pressure corresponding to a refrigerant liquid temperature of 100°F (37.8°C) is 133 psia (910 kPa).

Knowing the high and low side pressures, Fig. 10.5 is used to construct the pressure-enthalpy diagram for the refrigeration cycle. This diagram is shown in Fig. 10.8.

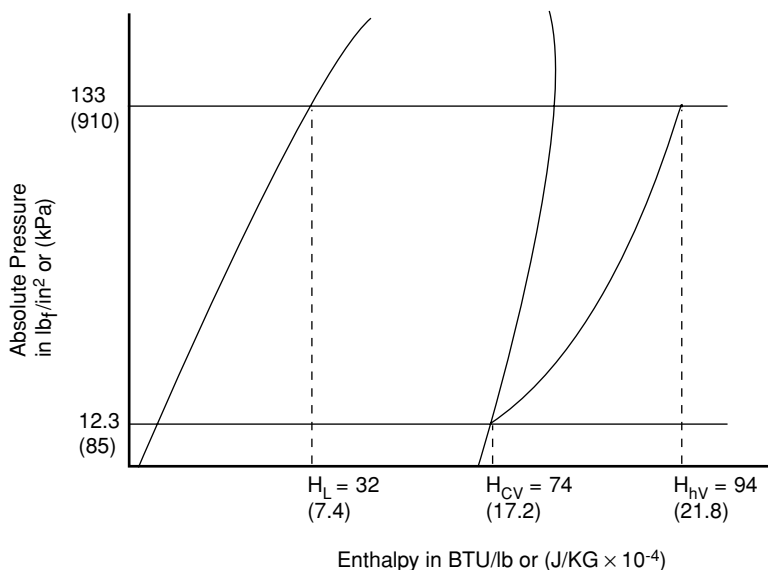


Figure 10.8 Pressure-enthalpy diagram of refrigerant 12 for the problem in Example 10.3.

The refrigeration capacity (Question 3) is calculated using Equation (10.2):

$$\text{Refrigeration capacity} = (74 - 32) = 42 \text{ BTU/lb} = (17.2 - 7.4) \times 10^4 = 98,000 \text{ J/kg}$$

The coefficient of performance (4) is calculated using Equation (10.7):

$$\text{COP} = \frac{74 - 32}{94 - 74} = 2.1$$

The theoretical horsepower per ton of refrigeration (5) is calculated using Equation (10.10):

$$\begin{aligned} \frac{C_p}{C_v} \text{ for Freon 12} &= \gamma = 1.14 \\ \frac{\text{HP}}{(\text{ton})_r} &= \frac{4.715}{(\gamma)(\text{COP})} = \frac{4.715}{(1.14)(2.1)} = 1.97 \end{aligned}$$

One ton of refrigeration (6) is equivalent to 12,000 BTU/h or 3517 W. Weight of refrigerant circulated/h is

$$\begin{aligned} \text{Weight} &= \frac{\text{coolong capacity/ton of refrigeration}}{\text{coolong capacity/unit weight iof refrigerant}} \\ &= \frac{12,000 \text{ BTU/h}}{42 \text{ BTU/lb}} = 286 \text{ lb refrigerant per hour} \\ &= \frac{3517 \text{ J/s}}{98,000 \text{ J/kg}} = 0.0359 \text{ kg/s} = 129 \text{ kg/h} \end{aligned}$$

(b) For refrigerant 717

The high and low side pressures (1 and 2) corresponding to -30°F (-34.4°C) and 100°F (37.8°C) are determined from Fig. 10.1. The low side pressure is 13.7 psia (94 kPa). The high side pressure is 207 psia (1440 kPa).

The pressure-enthalpy diagram (Question 3) is then constructed knowing the high and low side pressures using Fig. 10.4. Figure 10.9 shows this diagram.

$$\text{Refrigeration capacity} = 600 - 155 = 445 \text{ BTU/lb} = (139 - 36) \times 10^4 = 1,030,000 \text{ J/kg}$$

The coefficient of performance (4):

$$\text{COP} = \frac{600 - 155}{784 - 600} = 2.42$$

The theoretical horsepower per ton of refrigeration (5) is calculated using a C_p/C_v ratio of 1.29.

$$\frac{\text{HP}}{(\text{ton})_r} = \frac{4.715}{1.29(2.42)} = 1.51$$

Weight of refrigerant circulated per hour $(\text{ton})_r$ (6) is

$$\begin{aligned} \text{Weight} &= \frac{12,000 \text{ BTU}/(\text{h})(\text{ton})_r}{445 \text{ BTU/lb}} = 26.97 \text{ lb/h}(\text{ton})_r = \frac{1317 \text{ J/s}(\text{ton})_r}{1,030,000 \text{ J/kg}} \\ &= 0.00341 \text{ kg/s}(\text{ton})_r = 12.3 \text{ kg/h}(\text{ton})_r \end{aligned}$$

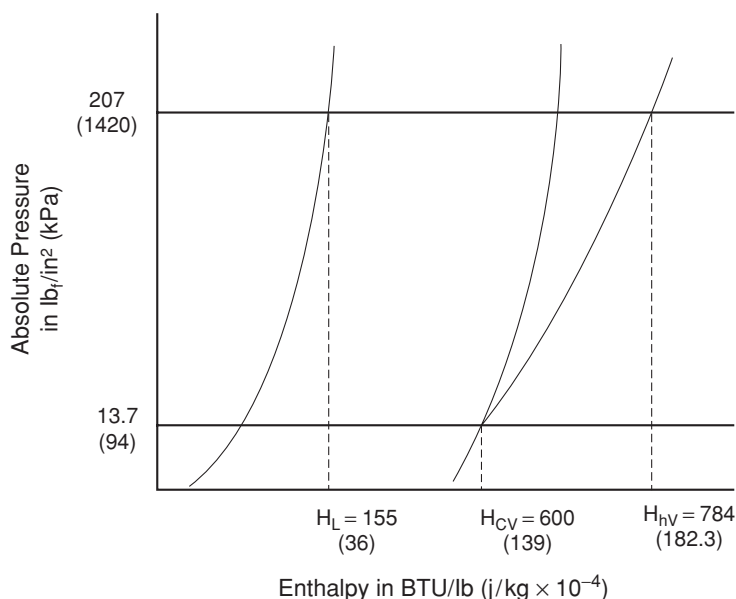


Figure 10.9 Pressure-enthalpy diagram of ammonia for the problem in Example 10.3.

10.1.6 The Condenser and Evaporator

Most refrigeration systems transfer heat between the refrigerant and air. Because heat transfer coefficients to air are usually very low, the air film resistance controls the rate of heat transfer. Very large heat transfer areas would be required to achieve the necessary heat transfer rate. To accomplish the necessary heat transfer rate and still have equipment that is reasonably sized, the heat transfer surface area of the tubes that comprise the evaporator or condenser coil is increased by the use of fins.

Finned heat exchange units are sized in terms of an effective heat transfer surface area which is the sum of the area of the bare tube A_t and the effective area of the fin, which is the product of the fin surface area A_s and the fin efficiency η .

$$A_{\text{eff}} = \eta A_f + A_t$$

The fin efficiency, η , decreases as the base area of the fin decreases and as the height of the fin increases. Furthermore, as the ratio of the heat transfer coefficient to air over the thermal conductivity of the metal that constitutes the fin (h_o/k) approaches zero, η approaches unity. Figure 10.10 shows fin efficiency for the most common systems. The fins in these units often consist of a stack of very thin sheets of metal pierced by a bundle of tubes. The equivalent fin surface area, A_f , is calculated as follows:

$$A_f = 2\pi(r_f^2 - r^2)(n_t)(n_f)$$

where n_t is number of tubes, and n_f is number of fins along the length of each tube.

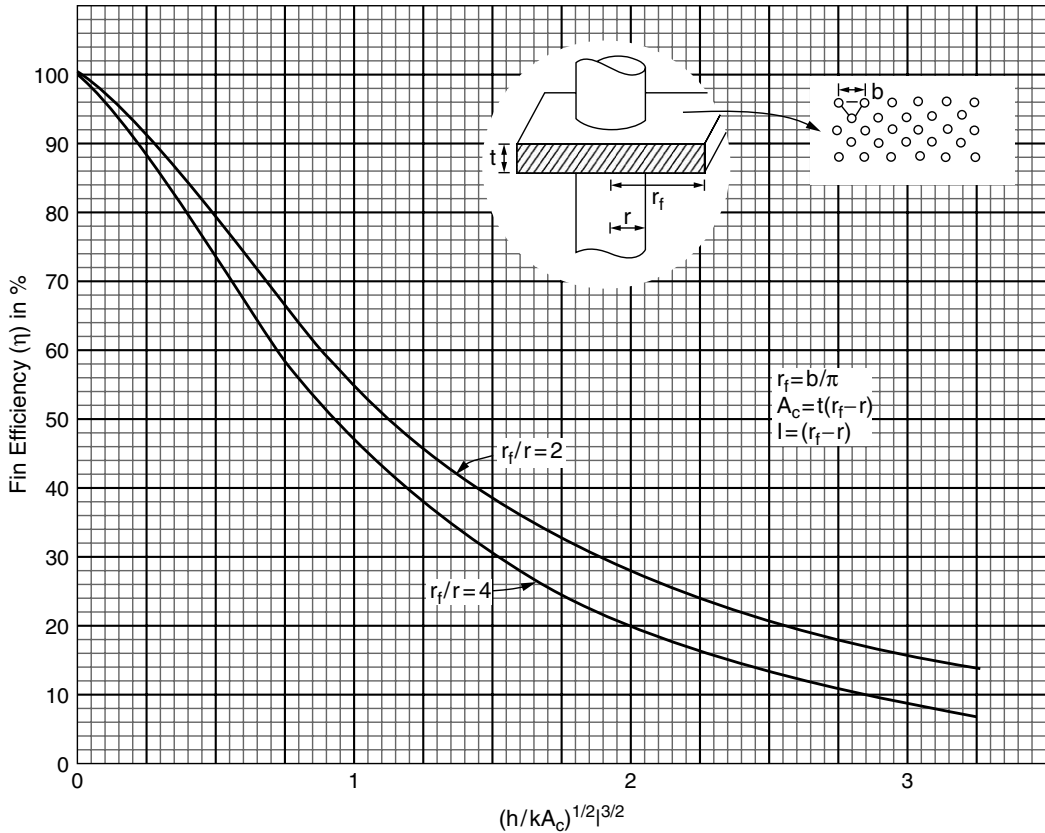


Figure 10.10 Fin efficiency of fins consisting of a stack of sheet metal each sheet with thickness l , pierced by tubes of radius r . The tubes are arranged with centers on an equilateral triangle having distance b between centers.

The capacity of a condensing or evaporating unit depends upon the surface area available for heat transfer and the mean temperature difference between the surfaces of the unit and the air going through the unit.

10.1.6.1 Problems with Heat Exchange in Systems Using Zeotropic Refrigerants

The following examples are based on systems using a single compound or an azeotrope as the refrigerant. When an azeotrope is used as refrigerant, provision must be given to the fact that local heat transfer coefficient will be reduced on the refrigerant side of the heat exchanger because the composition of liquid refrigerant on the heat exchange surface will not be the same at different points in the heat exchanger. The thermal properties of the condensate will vary at different points in the heat exchanger and the vapor phase of the other refrigerant components may also interfere with the heat transfer. Thus, the usual method of calculating the log mean ΔT in the heat transfer calculations is no longer correct. Depending upon the refrigerant, errors could go as high as 11% in the determined heat transfer surface area compared to using the standard heat transfer coefficient from condensing

liquids and conventional log mean ΔT calculations. This error could result in undersized evaporator or condenser units.

Example 10.4. A condensing unit of a refrigeration system consists of 13 rows of 38.1-cm-long, 1.27-cm outside diameter copper tubes with 1.651-mm wall stacked four deep on 2.54-cm centers. The fin consists of 0.0254-cm thick aluminum sheets spaced 10 per 2.54 cm. If the rate of air flow through this condenser is such that the outside heat transfer coefficient, h , is 9 BTU/(h \cong ft² \cong °F) or 51.1 W/(m² \cong K), calculate the effective heat transfer surface area for this unit.

Solution:

Refer to Fig. 10.10. Because the tubes are spaced on 2.54-cm centers, $b = 2.54$ cm. The thermal conductivity of aluminum (from Appendix Table A.9) is 206 W/(m \cong K). Figure 10.10 can now be used to determine the fin efficiency η .

The tube radius, $r = 0.00635$ m

$$\text{Equivalent fin radius, } r_f = \frac{b}{\sqrt{\pi}} = \frac{0.0254}{1.772} = 0.014334 \text{ m}$$

Fin cross-sectional area $A_c = t(r_f - r)$

$$= 0.000254(0.014334 - 0.00635) = 2.0279 \times 10^{-6}$$

The height of the fin, $P = (r_f - r)$

$$= (0.014334 - 0.00635) = 0.007984 - \text{m}$$

Values for the abscissa in Fig. 10.10 can now be calculated:

$$\left[\frac{h}{kA_c} \right]^{0.5} (\ell)^{1.5} = \left[\frac{51.1}{206(2.0279 \times 10^{-6})} \right]^{0.5} (0.007984)^{1.5} = (349.7)(0.000713) = 0.2495$$

$$r_f/r = 0.014334/0.00635 = 2.26$$

From Fig. 10.10, $\eta = 0.9$. The equivalent fin area is calculated as follows:

$$n_t = 13(4) = 52 \text{ tubes}$$

$$n_f = 38.1 \text{ cm} \left[\frac{10 \text{ fins}}{2.54 \text{ cm}} \right] = 150$$

$$A_f = 2\pi(r_f^2 - r^2)(n_t/n_f) = 2\pi[(0.014334)^2 - (0.00635)^2](52)(150) = 8.094 \text{ m}^2$$

The area of the bare tube A_t is the total tube area minus the area covered by the fins. Let P_t = length of the tube.

$$A_t = (2\pi r l_t)(n_t) - (2\pi r)(t)(n_t)(n_f)$$

t = fin thickness = 0.000254 m

$$A_t = 2\pi(0.00635)(0.381)(52) - 2\pi(0.00635)(0.000254)(52)(150)$$

$$= 0.790 - 0.0790 = 0.711 \text{ m}^2$$

$$\text{Effective area } A_{\text{eff}} = 0.91(8.094) + 0.711 = 8.0765 \text{ m}^2$$

Example 10.5. If the conditions are such that a mean temperature difference ΔT of 45°F (25°C) exists in the condensing system and the refrigeration unit has a coefficient of performance of 2, how many tons of refrigeration can be supplied by the refrigeration system fitted with the condensing unit in Example 10.4? Assume the outside coefficient of heat transfer controls the heat transfer rate, $U = h_0 = 51.1 \text{ W/(m}^2 \cong \text{K)}$.

Solution:

The rate of heat transferred through the unit is

$$q = U A_{\text{eff}} \Delta T = 51.1(8.0765)(25) = 10,318 \text{ W}$$

From Equation (10.7), let R_r = refrigeration capacity.

$$\text{COP} = \frac{R_r}{H_{\text{hv}} - H_{\text{cv}}}$$

Where:

$$R_r = H_{\text{cv}} - H_L$$

$$H_L = H_{\text{cv}} - R_r$$

Condenser load:

$$C = H_{\text{hv}} - H_L = H_{\text{hv}} - H_{\text{cv}} + R_r$$

$$(H_{\text{hv}} - H_{\text{cv}}) = C - R_r$$

$$(\text{tons})_r = \frac{6878 \text{ W}}{3515.88 \text{ W/ton}} = 1.956 \text{ tons}$$

The coefficient of performance can be expressed in terms of the condenser load C and the refrigeration capacity R_r .

$$\text{COP} = \frac{R_r}{C - R_r}$$

Solving for R_r :

$$R_r = \frac{C(\text{COP})}{1 + \text{COP}}$$

Substituting values for COP and C:

$$R_r = \frac{10,318(2)}{3} = 6878 \text{ W}$$

A similar procedure may be used to evaluate evaporators. The examples illustrate the dependence of the capacity of a refrigeration system on the heat exchange capacity of the condensing or evaporator unit in the system.

10.1.7 The Compressor

The compressor must be adequate to compress the required amount of refrigerant per unit time between the high side and low side pressures, to provide the necessary refrigeration capacity. Compressor capacities are determined by the displacement and the volumetric efficiency of the unit. The

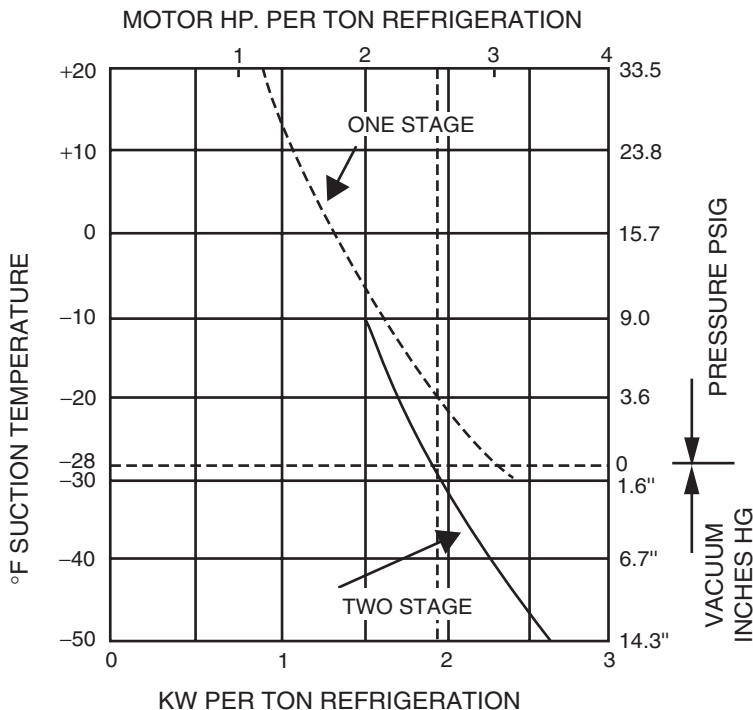


Figure 10.11 Motor horsepower and kW/Ton of refrigeration for one and two stage compressors. (From Food Eng. 41(11):91, 1969.)

displacement is the volume displaced by the piston or rotary unit per unit time. For a reciprocating compressor, the displacement is calculated as follows:

$$\text{Displacement} = (\text{area of bore}) (\text{stroke}) (\text{cycles/min})$$

Theoretically, the quantity of gas passing through the unit per unit time should be independent of the pressure and should be a function only of the displacement of the unit. However, at high pressures, rotary units allow a certain amount of slip and reciprocating units leave a certain amount of gas at the space between the cylinder head and the piston at the end of each stroke. Thus, the actual amount of gas passed through the unit per cycle is less than the displacement. The volumetric efficiency of a unit is the ratio of the actual volume delivered to the displacement. The volumetric efficiency decreases with increasing ratios of high side to low side pressures.

Large units that operate at very low pressures at the evaporator (e.g., units used for freezing) often are of a multistage type, where the refrigerant is compressed to an intermediate pressure, then partially cooled before it enters a second stage where it is compressed to the next higher pressure, and so forth, until the desired high side pressure is reached. The staging is designed such that equal work is accomplished in each stage and the volumetric efficiency remains high because of the low ratio of high side to low side pressure in each of the stages.

Figure 10.11 shows a comparison between the power requirement per ton of refrigeration between one stage and two stage units operating at various suction side pressures. For an ammonia compressor

operating at a suction side pressure equivalent to a temperature of -30°F , the motor horsepower per ton of refrigeration is 3.2 for a one stage unit and is only 2.5 for a two stage unit. In Example 10.3, the theoretical horsepower per ton of refrigeration under these conditions is only 1.51. Thus, the efficiency of the compressors represented by the graph in Fig. 10.11 is only 47% for a one-stage unit and 60% for a two-stage unit.

The presence of air in the refrigerant line reduces the efficiency of the compressor if expressed in terms of horsepower per ton of refrigeration as the same amount of power will be required to compress the gases to the required pressure. Yet, less refrigerant is passed through because part of the gas compressed is air.

10.2 REFRIGERATION LOAD

The amount of heat that must be removed by a refrigeration system per unit of time is the refrigeration load. The load may be subdivided into two categories: the *unsteady-state* load, which is the rate of heat removal necessary to reduce the temperature of the material being refrigerated to the storage temperature within a specific period of time, and the *steady-state* load, which is the amount of heat removal necessary to maintain the storage temperature. For food products, the temperature has to be reduced to the storage temperature in the shortest time possible to prevent microbiological spoilage and quality deterioration. Therefore, if large quantities need to be introduced into a refrigerated storage room intermittently, these materials are usually precooled to the storage temperature in a smaller precool or chill room or by other means prior to introduction into the large storage warehouse. This practice prevents the necessity of having to install oversized refrigeration units on the large storage warehouse that would operate at full capacity only during the short periods that materials are being introduced into the warehouse.

The unsteady-state load includes the sensible heat of the product, the heat of respiration if the product is fresh produce and the heat of fusion of the water in the product if the product is to be frozen. The steady-state load includes heat incursion through the walls of the enclosure, through cracks and crevices and through doors, latent heat of condensation of moisture infiltrating into the room, and heat generated inside the room.

10.2.1 Heat Incursion Through Enclosures

Heat transfer rates through composite solids and techniques for determination of heat transfer coefficients were discussed thoroughly in Chapter 7. The ASHRAE guide and data book for 1965 recommends the design heat transfer coefficients to be used for various surfaces as shown in Table 10.2.

10.2.2 Heat Incursion Through Cracks and Crevices

A majority of heat transferred is due to fluctuations in pressure caused by cycling of temperature in the room. At the high temperature point in the cycle, cold air will be expelled from the room. When the pressure drops at the low temperature point in the cycle, warm air will be drawn into the room. The amount of air admitted into the room because of temperature cycling can be calculated using the principles discussed in Chapter 4.

Table 10.2 Heat Transfer Coefficient to Air from Various Surfaces under Various Conditions

Surface	Heat Transfer Coefficients	
	BTU	W
	$h(\text{ft}^2)(^\circ\text{F})$	$\text{m}^2(\text{K})$
Inside wall (still air)	1.5	8.5
Outside wall or roof 15 miles/hr or 24 km/h wind	5.9	33.5
Outside wall or roof 7.5 miles/hr or 12 km/h wind	4.0	22.7
Horizontal surface, still air, upward heat flow	1.7	9.7
Horizontal surface, still air, downward heat flow	1.1	6.25

Source: Adapted from ASHRAE, 1965. Design heat transmission coefficient. *ASHRAE Guide and Data Book. Fundamentals and Equipment for 1985 and 1986*. American Society of Heating, Refrigeration, and Air Conditioning Engineers, Atlanta, GA.

10.2.3 Heat Incursion Through Open Doors

Opening doors allow entry of warm outside air and expulsion of cold air. The rate of heat incursion is dependent upon the size of the door and the temperature differential between the inside and outside. Data on rate of heat transfer through doors of refrigerated rooms have been determined empirically. The equation for rate of heat loss calculated from data published in *Food Engineering Magazine* [Food Eng. 41(11):91, 1969] for values of ΔT between 40°F and 120°F (22.2°C and 66.7°C) is

$$q = 2126 W[e]^{0.0484(\Delta T)}(h)^{1.71} \quad (10.11)$$

where q is rate of heat incursion into the room in watts, W is the width of the door in m, ΔT is the temperature difference in $^\circ\text{C}$ inside and outside the room, and h is the height of the door in m.

Example 10.6. The door to a refrigerated room is 3.048 m high and 1.83 m wide. It is opened and closed at least five times each hour with the door remaining open for at least 1 min at each opening. Calculate the refrigeration load due to the door opening if the room is maintained at 0°C and ambient temperature is 29.4°C .

Solution:

$h = 3.048$ m, $W = 1.83$ m, $\Delta T = 29.4^\circ\text{C}$. Substituting into Equation (10.11):

$$q = 2126(1.83)(e)^{0.0484(29.4)}(3.048)^{1.71} = 108.6 \text{ KW}$$

The total time the door was opened in 1 hour is 300 seconds. Refrigeration load = 108.6 KW (300 s) = 35.58 MJ .

10.2.4 Heat Generation

Motors inside a refrigerated room generate heat at the rate of 1025.5 W/hp. This rate drops to 732.48 W/hp if only the motor is inside and the load it drives is outside the refrigerated room. Workers inside a room generate approximately 293 W per person. The rate of heat generation by personnel increases with decreasing room temperatures. Heat dissipated by light bulbs is the same as the wattage of the lamp.

Fruits and vegetables respire and the heat of respiration adds to the refrigeration load. The heat of respiration of fruits and vegetables is a function of temperature and can be calculated using the expression:

$$q = a(e)^{bT} \quad (10.12)$$

q is the rate of heat generation (mW/kg in SI). Values for a and b for various fruits and vegetables are shown in Table 10.3.

During cooling, heat of respiration will decrease as temperature decreases. An average heat of respiration may be calculated between two temperatures T_1 and T_2 knowing the time Δt for the temperature to change between T_1 and T_2 and assuming that the temperature change during this period is linear.

$$q = \frac{1}{\Delta t} \int_0^{\Delta t} a(e)^{bT} dt \quad (10.13)$$

Assuming a linear temperature change:

$$T_1 - T = \frac{T_1 - T_2}{\Delta t} t \quad (10.14)$$

Substituting Equation (10.14) for T in Equation (10.13):

$$q = \frac{a}{\Delta t} \int_0^{\Delta t} [e]^{bT_1 - \frac{T_1 - T_2}{\Delta t} bt} dt$$

Integrating and substituting limits:

$$q = \frac{ae^{bT_1}}{b(T_1 - T_2)} [1 - (e)^{-b(T_1 - T_2)}] \quad (10.15)$$

Equation (10.15) is the average rate of heat generation over the time Δt . The units will be the same as the units of a , which in SI, is mW/kg as tabulated in Table 10.3. Multiplying q in Equation (10.15) by Δt will give the total heat generated for the time period under consideration.

Although the temperature at any given point in a material during the process of cooling usually changes exponentially with time, materials that are cooled in bulk (e.g., head lettuce or boxed products) would have a temperature gradient with the interior parts at a higher temperature than the external sections exposed to the cooling medium. Thus, if the bulk mean temperature is considered, the deviation from the linear temperature change assumption would be minimal.

Example 10.7. It is desired to cool cabbage from 32.2°C to 4.44°C in 4 hours. Calculate the heat of respiration during this cooling period. Assume a linear temperature change.

Solution:

$T_1 = 32.2^\circ\text{C}$; $T_2 = 4.44^\circ\text{C}$; $\Delta t = 4 \text{ h} = 14,400 \text{ seconds}$. From Table 10.3 for cabbage, $a = 16.8$ and $b = 0.074$ in SI units. Examination of Equation (10.12) will reveal that a will have the same units

Table 10.3 Heat of Respiration of Fruits and Vegetables in Air Values of the Constants a and b in the Equation $q = a(e)^{bT}$

Product	$T \text{ in } ^\circ\text{F}$		$T \text{ in } ^\circ\text{C}$	
	$q \text{ in } \frac{\text{BTU}}{\text{ton (24 hr)}}$		$q \text{ in } \frac{\text{mW}}{\text{kg}}$	
	a	b	a	b
Apples	213	0.06	19.4	0.108
Asparagus	2779	0.048	173.0	0.086
Beans (green or snap)	829	0.064	86.1	0.115
Beans (lima)	376	0.071	48.9	0.128
Beets (topped)	1054	0.031	38.1	0.056
Broccoli	854	0.067	97.7	0.121
Brussels sprouts	1845	0.045	104.0	0.081
Cabbage	337	0.041	16.8	0.074
Cantaloupes	128	0.07	16.1	0.126
Carrots (topped)	498	0.046	29.1	0.083
Celery	237	0.58	20.3	0.104
Corn (sweet)	2465	0.043	131.0	0.077
Grapefruit	171	0.051	11.7	0.092
Lettuce (head)	416	0.049	26.7	0.088
Lettuce (leaf)	1188	0.041	59.1	0.074
Onions	89	0.055	6.92	0.099
Oranges	151	0.059	13.4	0.106
Peaches	104	0.074	14.8	0.133
Pears	42	0.096	12.1	0.173
Peas (green)	1249	0.059	111.0	0.106
Peppers (sweet)	796	0.040	33.4	0.072
Spinach	473	0.073	65.6	0.131
Strawberries	568	0.059	50.1	0.106
Sweet potatoes	796	0.034	31.7	0.061
Tomatoes	159	0.057	13.2	0.103
Turnips	590	0.037	25.8	0.067

Source: ASHRAE, 1974. Approximate rates of evolution of heat by certain fruits and vegetables when stored at temperatures indicated. In: *ASHRAE Handbook and Product Directory—1974 Applications*. American Society of Heating, Refrigeration, and Air Conditioning Engineers, Atlanta, GA.

¹ q calculated using the constants a and b are maximum values in the range reported. Minimum values average 67% of the maximum.

as q , and b will have units of $1/EC$.

$$q = \frac{16.8 e^{0.074(32.2)}}{(0.074)(32.2 - 4.44)} \left[1 - (e)^{-0.074(32.2 - 4.44)} \right] = 77.253 \text{ mW/kg}$$

Heat generated over the time period of 14400 s is Q :

$$Q = (77.253 \times 10^{-3} \text{ W/kg})(14,400 \text{ s}) = 1112 \text{ J/kg}$$

Example 10.8. Calculate the refrigeration load due to the heat of respiration of spinach at a constant temperature of 3.33°C.

Solution:

$T = 3.33^\circ\text{C}$. From Table 10.3, $a = 65.6 \text{ mW / kg}$ and $b = 0.131^\circ\text{C}^{-1}$.

$$q = 65.6 e^{0.131(3.35)} = 101.74 \text{ mW/kg}$$

10.2.5 The Unsteady-State Refrigeration Load

Procedures for calculating the heat capacity and sensible heat gain or loss are discussed Chapter 5. When a change in phase is involved, the latent heat of fusion of the water must be considered. The heat of fusion of ice is 144 BTU/lb, 80 cal/g, or 0.334860 MJ/kg.

Example 10.9. Calculate the refrigeration load when 100 kg/h of peas needs to be frozen from 30°C to -40°C. The peas have a moisture content of 74%. The freezing point is -0.6°C.

Solution:

The refrigeration load above freezing will be calculated using Siebel's equation. From Equation (5.9), Chapter 5, the specific heat above freezing for a material that contains 74% water and 26% solids non fat is

$$C_p = 837.36(0.26) + 4186.8(0.74) = 3315.9 \text{ J/(kg K)}.$$

The refrigeration load required for cooling 1 kg peas from 30°C to -0.6°C = Q_1

$$Q_1 = 1(3315.9)(30.6) = 101,467 \text{ J}$$

The refrigeration load below freezing will be calculated using the concepts and equations discussed in the section "Enthalpy Changes in Food Freezing" in Chapter 5. Molality of solutes in unfrozen water at different temperatures below freezing will be used as the basis for the calculations.

$$T_f = -0.6^\circ\text{C}; T_2 = -40^\circ\text{C}; K_f = 1.86; \Delta H_f = 334,860 \text{ J}$$

The specific heat of ice = 2093.4 J/(kg K)

The specific heat of water is 4186.8 J/(kg K)

The specific heat of solids non-fat = 837.36 J/(kg K)

The initial moisture before freezing = $W_o = 0.740 \text{ kg}$

The liquid water at -40°C = $W = W_o(-T_f / -T_2) = 740(0.6/40) = 0.0111 \text{ kg}$

Ice at -40°C = $0.740 - 0.0111 = 0.7289 \text{ kg}$

Change in sensible heat for ice from -0.6°C to -40°C

(From Eq. 5.23, Chapter 5)

$$Q_{\text{ice}} = I C_{pi} W_o [(T_f - T_2) - (-T_f \ln(-T_2 / -T_f))] = 0.0111 (2093.4)[39.4 - 0.6 \ln(40/0.6)] = 856.9 \text{ J}$$

Change in sensible heat of water from -0.6°C to -40°C

(From Eq. 5.22, Chapter 5)

$$Q_{\text{water}} = C_{pw} W_o (-T_f) \ln (-T_2 / -T_f) = 4186.8(0.7289)[0.6 \ln(40/0.6)] = 7689.9 \text{ J}$$

$$Q_{\text{snf}} = 0.26 (837.36)[-0.6 - (-40)] = 8577.9 \text{ J}$$

The amount of ice that is formed = 0.7289 kg

$$\Delta H_f = 0.7289 (334,860) = 244,079 \text{ J}$$

Total refrigeration load/kg = $101,467 + 856.9 + 8577.9 + 7689.9 + 244,079 = 362,670 \text{ J/kg}$

For 100 kg, the refrigeration load will be 36.267 MJ.

10.3 COMMODITY STORAGE REQUIREMENTS

Most food products would benefit from a reduction in the storage temperature provided that no freezing occurs. Reduced temperature reduces the rate of chemical reactions that deteriorate the product and also reduce microbiological activity. Freezing damages the cellular structure of fruits and vegetables and severely affects acceptability. Meat pigments darken irreversibly upon freezing. Acceleration of the development of a strong fishy flavor has been observed when fish were frozen at or near the freezing point and thawed. Thus, for perishable foods not specifically prepared for frozen storage, the freezing temperature should be the lowest acceptable limit for storage. Some fruits and vegetables are susceptible to chill injury at temperatures above the freezing point. The lowest safe temperatures have been defined for these commodities. Table 10.4 shows the temperature and humidity recommended for storage of various fruits and vegetables.

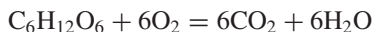
10.4 CONTROLLED ATMOSPHERE STORAGE

10.4.1 Respiration

Fruits and vegetables continue their metabolic activity after harvest. Maintenance of this metabolic activity is essential in the preservation of quality. Metabolic activity is manifested by respiration, a process where the material consumes oxygen and evolves carbon dioxide. A major nutrient metabolized during respiration is carbohydrate. Because respiration depletes nutrients in a product after harvest, the key to prolonging shelf life of fruits and vegetables is in reducing the rate of respiration. Early studies on the relationship between respiration and quality have established that storage life is inversely related to the rate of respiration and that for two products respiring at different rates, when the total quantity of CO_2 evolved by each product is equal, the products would have reached a comparable stage in their storage life.

Reduction of temperature is an effective means of reducing the rate of respiration. However, for some products subject to chill injury, respiration rate may still be quite high even at the lowest safe storage temperature. Controlled atmosphere storage has been developed as a supplement to refrigeration in prolonging the storage life of actively respiring fruits and vegetables.

The reaction involved in respiration is primarily the oxidation of carbohydrates:



The respiratory quotient (RQ) defined as the ratio of CO_2 produced to O_2 consumed is 1 for the reaction. Most products show respiratory quotients close to 1. The heat evolved during respiration has also been shown to be related to the quantity of CO_2 produced in the same manner as that released during the combustion of glucose (10.7 J/mg CO_2). The rate of oxygen consumption and CO_2 evolution can be calculated from the heat of respiration and vice versa.

Table 10.4 Recommended Storage Conditions for Fruits and Vegetables

(1) Storage temperatures 30°–32°F (–1.11°–0°C). Highest Freezing point 28°F (–2.22°C). 85–90% RH.

apricots pears
cherries peaches
grapes plums

(2) Storage temperatures 32°F (0°C)

75%RH	90% RH	95% RH	95% RH	95% RH
-------	--------	--------	--------	--------

garlic				
onions	mushrooms	artichokes	sweet corn	carrots
	oranges	asparagus	endives ¹	lettuce ¹
	tangerines	lima beans	escarole ¹	parsnips
		beets	leafy greens	rutabagas
		broccoli	parsley	turnips
		brussels sprouts	green peas	
		cabbage	radishes	
		cauliflower	rhubarb	
		celery ²	spinach ¹	

(3) Storage temperature 36°F (2.22°C). 95% RH.

apples

(4) Storage temperature 45°F (7.22°C). 90% RH. Subject to chill injury at temperatures below 45°F

green or snap beans

ripe tomatoes

(5) Storage temperature 50°F (10°C). Subject to chill injury at temperatures below 50°F

85% RH	90–95% RH
--------	-----------

melons	
potatoes	cucumbers
pumpkin	eggplants
squash	sweet peppers
green tomatoes	okra

(6) Storage temperatures 58°–60°F (14.4–15.6°C). 85–90% RH

bananas
grapefruit
lemons

Source: ASHRAE. 1974 ASHRAE Handbook and Product Directory—1974. Applications. American Society of Heating, Refrigeration, and Air Conditioning Engineers, New York.

¹Highest freezing point 31.9°F (–40.06°C).

²Highest freezing point 31.1°F (–0.5°C).

All others have freezing points below 31°F (–0.56°C).

Controlled atmosphere (CA) storage is based on the premise that increasing the CO₂ level and decreasing the O₂ level in the storage atmosphere will result in a reduction of the rate of respiration. Indeed, calorimetric determination of heat of respiration of products in continuously flowing CA of the optimum composition for the given product has shown that CA could reduce the respiration rate to approximately 1/3 of the respiration rate in air at the same temperature.

Example 10.10. One pound (0.454 kg) of head lettuce is packaged in an air-tight container with a volume of 4 L. The product occupies 80% of the volume, the rest being air. If the product is at a constant temperature of 4°C, calculate how long it will take for the oxygen content in the package to drop to 2.5%. Assume a RQ of 1.

Solution:

From Table 10.3, the constants a and b for the heat of respiration of head lettuce (in mW/kg) is 26.7 and 0.088, respectively.

$$q = 26.7(e)^{0.088(4)} = 38 \text{ mW/kg}$$

$$\frac{\text{mg CO}_2}{\text{h}} = (38 \times 10^{-3}) \cdot \frac{\text{J}}{\text{s} \cdot \text{kg}} \cdot \frac{1 \text{ mg CO}_2}{10.7 \text{ J}} \cdot 3600 \frac{\text{s}}{\text{h}} (0.454 \text{ kg}) = 5.8$$

$$\frac{\text{g moles CO}_2}{\text{h}} = \frac{\text{g moles O}_2 \text{ depleted}}{\text{h}} = \frac{5.8 \times 10^{-3}}{44} = 1.318 \times 10^{-4}$$

Air is approximately 21% O₂ and 79% N₂. The total number of moles of air originally in the container is:

$$n_{\text{air}} = \frac{PV}{RT} = \frac{(1 \text{ atm})(4)(0.2)1}{0.08206(1 \text{ atm} / \text{g mole K})(273 + 4)\text{K}}$$

$$n_{\text{air}} = 0.0352 \text{ moles}$$

Because the RQ is 1, there will be no net change in the total number of moles of gases inside the container. The number of moles of oxygen when the concentration is 2.5% is

$$n_{\text{O}_2 \text{ final}} = 0.025(0.0352) = 0.00088$$

The original number of moles of O₂ is

$$n_{\text{O}_2 \text{ initial}} = 0.21(0.0352) = 0.007392 \text{ moles}$$

The number of moles of O₂ that must be depleted by respiration is

$$n_{\text{O}_2 \text{ depleted}} = 0.007392 - 0.00088 = 0.006512$$

The time required to deplete O₂ to the desired level is

$$\text{Time} = \frac{0.006512 \text{ mole}}{0.0001318 \text{ mole / h}} = 49.4 \text{ h}$$

10.4.2 CA Gas Composition

CA storage has been used successfully with apples and pears. Experimental results have been very encouraging on CA storage of cabbage, head lettuce, broccoli, and brussels sprouts. Atmospheric modification of product containers during transit of vegetables is also gaining wider acceptance. CA is recommended when transit time is 5 days or more.

Reduction of oxygen concentration in the storage atmosphere to 3% or below has been shown to be most effective in reducing respiration rate, with or without CO₂. Too low an oxygen concentration, however, leads to anaerobic respiration resulting in the development of off-flavors in the product. The O₂ concentration for onset of anaerobic respiration ranges from 0.8% for spinach to 2.3% for

Table 10.5 CA Storage Conditions Suitable for Use with Various Products

<i>Product</i>	<i>% CO₂</i>	<i>% O₂</i>
Apples	2–5	3
Asparagus	5–10	2.9
Brussels sprouts	2.5–5	2.5–5
Beans (green or snap)	5	2
Broccoli	10	2.5
Cabbage	2.5–5	2.5–5
Lettuce (head or leaf)	5–10	2
Pears	5	1
Spinach	11	1
Tomatoes (green)	0	3

Percentages are volume or mole percent. Balance is nitrogen.
 Reprinted from Toledo, R. T. 1980. *Fundamentals of Food Process Engineering*. AVI Pub. Co., Westport, Conn.

asparagus. The optimum CA composition varies from one product to another and between varieties of a given product. CA compositions found to be effective for some products are shown in Table 10.5.

To be effective, CA storage rooms must be reasonably air-tight. To test for air-tightness, the room should be pressurized to a positive pressure of 1 in. water gauge (wg) (249 Pa gauge). A room is considered air-tight if at the end of 1 hour, the pressure does not drop below 0.2 in. wg (49.8 Pa). This requirement is equivalent to an air incursion rate of 0.2% of the gas volume in the room per hour at a constant pressure differential between inside and outside of 0.5 in. wg (124 Pa). Current standards for refrigerated tractor trailers call for a maximum air incursion rate of 2 ft³/min (3.397 m³/h) at a pressure differential of 0.5 in. water (124 Pa) between the inside and outside of the trailer. This rate is approximately 15 times what would meet the specification for air-tightness.

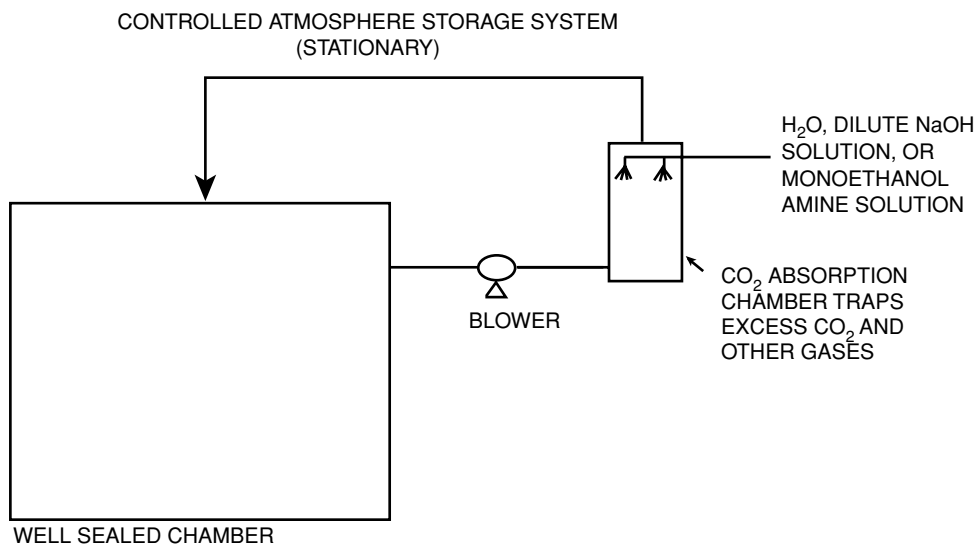
CA in storage warehouses are maintained at the proper composition by introduction of CO₂ and/or N₂ if product respiration is insufficient to raise CO₂ levels or lower O₂ levels. In a full, tightly sealed warehouse, ventilation with fresh air and scrubbing of CO₂ from the storage atmosphere may be necessary to maintain the correct CA composition and prevent anaerobic respiration. Successful operation of a CA storage facility necessitates regular monitoring and analysis of gas composition and making appropriate corrections when necessary. Figures 10.12 and 10.13 are schematic diagrams of CA storage systems for warehouses and for in-transit systems.

Example 10.11. The rate of flow of fluids through narrow openings is directly proportional to the square root of the pressure differential across the opening. If the pressure inside a chamber of volume V changes exponentially with time from 1 in. wg (249 Pa) at time 0 to 0.2 in. wg (49.8 Pa) at 1 hour, calculate the mean volumetric rate of flow of gases out of the chamber and the mean pressure at which this rate of flow would be expected if the pressure is held constant.

Solution:

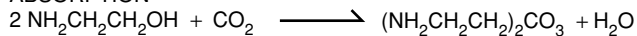
The expression for the volumetric rate of flow Q is

$$Q = k(\Delta P)^{1/2}$$



MONOETHANOL AMINE REACTION:

ABSORPTION



REGENERATION



Figure 10.12 Schematic diagram of a controlled atmosphere storage system for warehouses.

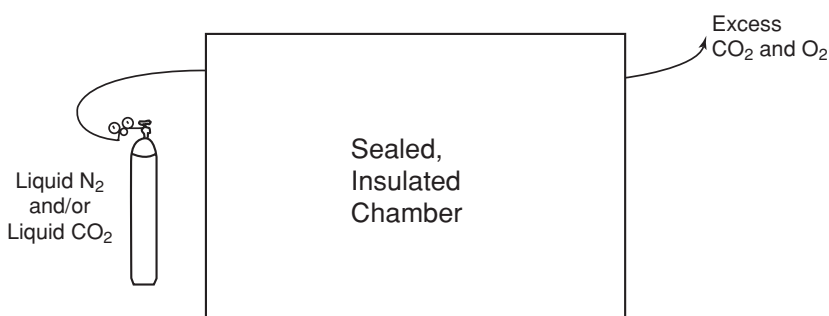


Figure 10.13 Schematic diagram of a transportable system for generating and maintaining controlled atmospheres.

The volume dQ expelled over a period of time dt is

$$dQ = k(\Delta P)^{1/2} dt$$

Because P is exponential with time, substituting the boundary conditions, $\Delta P = 1$ at $t = 0$, and $\Delta P = 0.2$ at $t = 1$, will give the equation for ΔP as a function of t as follows:

$$\Delta P = (e)^{-1.6094t}$$

Substituting P in the differential equation and integrating:

$$Q = \int_0^1 [k(e)^{0.5(-1.6094)t}] dt$$

The mean rate of flow in one hour, $\bar{q} = Q/t$

$$\bar{q} = \frac{1}{1} \left[\left(\frac{k}{-0.8047} \right) (e)^{-0.8047t} \right]_0^1 = 0.687 k$$

Because $\bar{q} = k(\overline{\Delta P})^{1/2}$, the mean pressure $\overline{\Delta P}$ that would give the equivalent rate of flow as \bar{q} is $(0.687)^2$ or 0.472.

One atmosphere is 406.668 in. of water. Using the ideal gas equation, the number of moles gas in the chamber at pressure P_1 and P_2 are

$$n_1 = \frac{P_1 V}{RT} \text{ and } n_2 = \frac{P_2 V}{RT}$$

The number of moles lost with a drop in pressure from P_1 to P_2 would be

$$n_1 = \frac{V}{RT} (P_1 - P_2)$$

Expressing the quantity expelled as volume at standard atmospheric pressure:

$$V_1 = n_1 \frac{RT}{P} = \frac{(P_1 - P_2)V}{P}$$

Substituting $P_1 = 1$ in.; $P_2 = 0.2$ in.; and $P = 406.668$:

$$V_1 = \frac{0.8V}{406.668} = 0.00196 V$$

The specified conditions of a drop in pressure from 1 in. wg to 0.2 in. wg in one hour is equivalent to a rate of loss of approximately 0.2% of the chamber volume per hour at an average pressure drop of approximately 0.5 in. wg.

Example 10.12. A refrigerated tractor-trailer is loaded with head lettuce such that 20% of its total volume is air space. The trailer has a total volume of 80 m³. Assuming that the trailer meets the specifications for the maximum air incursion rate of 2 ft³/min (3.397 m³/h) at a pressure differential of 0.5 in. wg (124 Pa), what would be the rate of addition of N₂ and CO₂ necessary to maintain the atmosphere inside the truck at 2% O₂ and 5% CO₂ when the truck is traveling at the rate of 50 miles/h (80 km/h). The average temperature of the product is 1.5°C and the trailer contains 18,000 kg of lettuce.

Solution:

If the atmosphere inside the truck has been pre-equilibrated at the point of origin to the desired CA composition, the amount of nitrogen and CO₂ that must be added during transit would be that necessary to displace the air infiltrating into the trailer. Let x = rate of N₂ addition and y = rate of CO₂ addition. If n_{ai} = the rate of air infiltration, n_{rc} = the rate of CO₂ generation by the product, and n_{ro} = the rate of oxygen consumption by the product during respiration, the following material balance may be set up for CO₂ and O₂ assuming that the expelled gases are of the same composition as those in the interior of the trailer.

O₂ balance (air is 21% oxygen):

$$n_{ai}(0.21) = (n_{ai} + x + y)(0.02) + n_{ro}$$

CO₂ balance:

$$y + n_{rc} = (n_{ai} + x + y)(0.05)$$

The heat of respiration of head lettuce in mW/Kg from values of $a = 26.7$ and $b = 0.088$ from Table 10.3 and for $T = 1.5^\circ\text{C}$:

$$\begin{aligned} q &= 26.7(e)^{1.5(0.088)} = 30.47 \text{ mW/kg} \\ \frac{\text{mg CO}_2}{\text{kg} \cdot \text{h}} &= \frac{30.47 \times 10^{-3} \text{ J}}{\text{kg} \cdot \text{s}} \frac{\text{mg CO}_2}{10.7 \text{ J}} \frac{3600 \text{ s}}{\text{h}} = 10.25 \\ \frac{\text{g moles CO}_2}{\text{kg} \cdot \text{h}} &= \frac{\text{g moles O}_2}{\text{kg} \cdot \text{h}} = \frac{10.25 \times 10^{-3}}{44} = 0.233 \times 10^{-3} \\ n_{rc} &= (18,000 \text{ kg}) \frac{0.233 \times 10^{-3} \text{ g moles}}{\text{kg} \cdot \text{h}} = 4.194 \frac{\text{g moles}}{\text{h}} \\ n_{ro} &= 4.194 \text{ g moles/h} \end{aligned}$$

The rate of air infiltration must now be calculated at a velocity of 80 km/h. The velocity head of a flowing fluid, $\Delta P/\rho$, is $V^2/2$. ρ is the fluid density.

$$\Delta P = \frac{(80,000)^2 \text{ m}^2}{\text{h}^2} (0.5) \frac{\rho \text{ kg}}{\text{m}^3} \frac{1 \text{ h}^2}{(3600)^2 \text{ s}^2} = 247 \rho \text{ Pa}$$

The specified infiltration rate of 3.397 m³/h at a pressure of 124 Pa can be converted to the rate at the higher pressure by using the relationship that rate of flow is directly proportional to the square root of the pressure.

$$Q = 3.397 \left[\frac{247}{124} \right]^{\frac{1}{2}} = 4.79 \text{ m}^3/\text{h}$$

Using the ideal gas equation, $n_{ai} = PV/RT$:

$$n_{ai} = \frac{(4.79 \text{ m}^3/\text{h})(1000 \text{ L/m}^3)(1 \text{ atm})}{0.08206(\text{L})(\text{atm})/(\text{gmole K})(274.5 \text{ K})} = 212.7 \text{ g moles/h}$$

Substituting values for n_{ai} , n_{ro} and n_{rc} in the O_2 and CO_2 balance equations:

$$\begin{aligned}
 212(0.21) &= (212 + x + y)(0.02) + 4.176 \\
 x + y &= \frac{44.52 - 4.176 - 4.24}{0.02} = 1805 \\
 x &= 1805 - y \\
 y + 4.176 &= (212 + x + y)(0.05) \\
 0.95y &= 0.05x + 6.424 \\
 0.95y - 0.05(1805 - y) &= 6.424 \\
 1.0y &= 6.424 + 90.25 \\
 y &= 96.67 \text{ g moles } CO_2/h \\
 x &= 1708.3 \text{ g moles } N_2/h
 \end{aligned}$$

Note that because of air infiltration, it would not be possible for product respiration to maintain the necessary CA composition inside the trailer. Furthermore, the respiration rate for this particular product is so low that flushing of the trailer with N_2 and CO_2 would have to be done initially to bring the CA to the desired composition within a short period of time.

10.5 MODIFIED ATMOSPHERE PACKAGING

Modified atmosphere (MA) and CA are similar, and are sometimes used interchangeably. A feature of CA is some type of control that maintains constant conditions within the storage atmosphere. MA packaging involves changing the gaseous atmosphere surrounding the product at the time of packaging and using barrier packaging materials to prevent gaseous exchange with the environment surrounding the package. The use of gas permeable packaging materials that is tailored to promote gas exchange to maintain relatively constant conditions inside the package may be considered a form of CA packaging. Reduction of respiration rates of live organisms by lowering oxygen and elevating CO_2 has been discussed in the previous section. MA packaging, as is now practiced, is primarily designed to control the growth of microorganisms and in some cases, enzyme activity, in packaged uncooked or cooked food. The atmosphere used, if the product itself is non-respiring, is usually 75% CO_2 , 15% N_2 , and 10% O_2 . Bakery products, which are generally of lower moisture content than most foods, could be preserved longer in 100% CO_2 . Dissolution of CO_2 and O_2 in the aqueous phase of the product is responsible for microbial inhibition. High CO_2 is inhibitory to most microorganisms. Although CO_2 lowers the pH, the inhibitory effect is more than from the pH reduction alone. The presence of O_2 has slight inhibitory activity against anaerobes resulting in slower growth compared with a 100% CO_2 atmosphere.

Other gas combinations and the products on which they have been used are 80% CO_2 and 20% N_2 for luncheon meats; 100% N_2 for cheese; 30% CO_2 and 70% N_2 for fresh poultry and fish, and 30% O_2 , 30% CO_2 and 40% N_2 for red meats.

Typically, MA packaging involves drawing a vacuum from the product in a high barrier packaging material, displacing the vacuum with the modified atmosphere, and sealing the package.

Another approach that is suitable for engineering modeling involves the use of gas-permeable films as the primary packaging material and enclosing the packages in a larger high barrier bag filled with the desired gaseous mixture. The gas mixture composition and the package film permeability are

balanced to achieve the desired constant MA inside the package. This is a rapidly growing area in packaging research. So far there are no satisfactory models that can adequately predict shelf life of MA packaged products.

PROBLEMS

- 10.1. An ammonia refrigeration unit is used to cool milk from 30°C to 1°C (86°F to 33.8°F) by direct expansion of refrigerant in the jacket of a shell and tube heat exchanger. The heat exchanger has a total outside heat transfer surface area of 14.58 m^2 (157 ft^2). To prevent freezing, the temperature of the refrigerant in the heat exchanger jacket is maintained at -1°C (31.44°F).
- If the average overall heat transfer coefficient in the heat exchanger is $1136\text{ W/m}^2 \cong \text{K}$ ($200\text{ BTU/h} \cong \text{ft}^2 \cong ^{\circ}\text{F}$) based on the outside area, calculate the rate at which milk with a specific heat of 3893 J/kg K ($0.93\text{ BTU/lb} \cong ^{\circ}\text{F}$) can be processed in this unit.
 - Determine the tons of refrigeration required for the refrigeration system.
 - The high-pressure side of the refrigeration system is at 1.72 MPa (250 psia). Calculate the horsepower of the compressor required for the refrigeration system assuming a volumetric efficiency of 60%.
- 10.2. A single-stage compressor in a freon 12 refrigeration system has a volumetric efficiency of 90% at a high side pressure of 150 psia (1.03 MPa) and a low side pressure of 50 psia (0.34 MPa). Calculate the volumetric efficiency of this unit if it is operated at the same high side pressure but the low side pressure is dropped to 10 psia (68.9 kPa). Assume R12 is an ideal gas.
- 10.3. Calculate the tons of refrigeration for a unit that will be installed in a cooler maintained at 0°C (32°F) given the following information on its construction and operation:

The cooler is inside a building.

Dimensions: $4 \times 4 \times 3.5\text{ m}$ ($13.1 \times 13.3 \times 11.5\text{ ft}$)

Wall and ceiling construction:

3.175 mm (1/8 in.) thick polyvinyl chloride sheet inside (k of PVC = $0.173\text{ W/m} \cong \text{K}$ or $0.1\text{ BTU/h} \cong \text{ft} \cong ^{\circ}\text{F}$)

15.24 cm (6 in.) fiberglass insulation

5.08 cm (2 in.) corkboard

3.17 mm (1/2 in.) PVC outside

Floor construction:

3.175 mm (1/8 in.) thick floor tile ($k = 0.36\text{ W/(m} \cong \text{k)}$ or $0.208\text{ BTU/(h} \cong \text{ft} \cong ^{\circ}\text{F}$)

10.16 cm (4 in.) concrete slab

20.32 cm (8 in.) air space

Concrete surface facing the ground at a constant temperature of 15°C (59°F)

Door:

1 m wide \times 2.43 m high ($3.28 \times 8\text{ ft}$)

Design for door openings that average four per hour at 1 minute per opening

Air infiltration rate:

$1\text{ m}^3/\text{h}$ ($35.3\text{ ft}^3/\text{h}$) at atmospheric pressure and ambient temperature

Ambient conditions:

32°C (89.6°F)

Product cooling load:

Design for a capability to cool 900 kg of product ($C_p = 0.76\text{ BTU/lb} \cong ^{\circ}\text{F}$ or $3181\text{ J/kg} \cong \text{K}$) from 32°C to 0°C (89.6°F to 32°F) in 5 hours. The freezing point of the product is -1.5°C (29.3°F).

- 10.4. The "stack effect" due to a difference in temperature between the inside and outside of a cooling room is often cited as the major reason for air infiltration. In this context, ΔP is positive at the lowest section of a cooler and is negative at the highest section, with a zone, called the neutral zone, at approximately the center of the room where the ΔP is zero. If the area of the openings at the lowest sections where ΔP is positive equals the area of the openings in the highest sections where ΔP is negative, air will enter at the top and escape at the openings in the bottom at the same volumetric rate of flow (assuming no pressure change inside the room). If the room allows air leakage at the rate of 2% of the room volume per minute at a ΔP of 0.5 in. wg (124 Pa), determine the rate of air infiltration that can be expected in a room that is 2 m (6.56 ft) high to the neutral zone if the interior of the room is at -20°C (-4°F) and ambient temperature is 30°C (86°F). The rate of gas flow through the cracks is proportional to the square root of ΔP . Assume air is an ideal gas. ΔP due to a column of air of height h at different temperatures $= g(\rho_1 - \rho_2)h$, where ρ_1 and ρ_2 are the densities of the columns of air.
- 10.5. For a 1-ton refrigeration unit (80% volumetric efficiency) using refrigerant 12 at a high side pressure of 150 psia (1.03 MPa) and a low side pressure of 45 psia (0.31 MPa), operating at an ambient temperature of 30°C (86°F), determine the effect of the following on refrigeration capacity and on $\text{HP}/(\text{ton})_r$. Assume the same compressor displacement in each case.
- Reducing the evaporator temperature to -30°C (-22°F). High side pressure remains at 150 psia (1.03 MPa).
 - Increasing ambient temperature to 35°C (95°F). (Low side pressure remains at 45 psia [0.31 MPa]). The high side pressure is to change such that ΔT between the hot refrigerant gas and ambient air remains the same as in the original set of conditions.
 - Air in the line such that the vapor phase of refrigerant always contain 10% air and 90% refrigerant by volume. Assume condensation temperature of hot refrigerant gas and temperature of cold refrigerant gas are the same as in the original set of conditions (partial pressure of refrigerant gas at the low and high side pressures are the same as in the original set of conditions, 45 and 150 psia or 0.31 and 1.03 MPa). Use $R = 1.987 \text{ BTU}/(\text{lbmole} \cong ^\circ\text{R})$ or $8318 \text{ J}/(\text{kg} \cong \text{K})$. The specific heat ratio C_p/C_v for air is 1.4.
 - Oil trapped in the vapor return line such that ΔP across the constriction is 10 psi (68.9 kPa). Assume evaporator temp. $= 0^\circ\text{C}$.
- 10.6. Chopped onions are frozen in a continuous belt freezer at using -50°C air at high velocity. When onions with a moisture content of 86% are loaded on the belt with a thickness of 2 cm., it took 20 minutes for the temperature to drop from 10°C to -20°C . Onion juice is added and mixed with the chopped onions in a ratio 0.10 parts juice to 0.90 parts of the chopped onions. The freezing point of the chopped onions is -0.5°C . The juice contains 1.5% solids (all soluble). The juice has a freezing point of -0.16°C . Assuming that the rate of heat transfer is the same, calculate the time required to freeze the onions with the added juice.

SUGGESTED READING

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