Chapter 9
Refrigeration and Freezing Equipment

9.1 Introduction

Refrigeration is used in cooling/chilling and freezing of foods. The freezing temperature of foods (around 0 °C) is the borderline between the cooling and the freezing processes. In food processing, low temperatures are applied (a) for preservation and (b) in facilitating other non-preservation processes or manufacturing products that are directly or indirectly related to foods. In preservation, the main aim is to extend the shelf life of fresh or processed products through cooling, chilling, freezing, and subsequent storage, by reducing the activity of microorganisms, enzymes, and chemical and biological reactions. In preservation, two main categories are distinguished: (a) the application of refrigeration without any other additional method and (b) the application of refrigeration in connection with some other methods of preservation.

Examples of using refrigeration for preservation without any additional method are the use of low temperatures for preserving fresh products, such as vegetables, through precooling and chilling; the cold storage of fruits, vegetables, meat, and fish; and the freezing of meat and fish. Examples of application of low temperatures in connection to some other methods of preservation are the freezing of vegetables following blanching and the cooling of milk, immediately after heating, in pasteurization. Examples of using low temperatures for facilitating other non-preservation processes or manufacturing products used directly in foods are (a) influencing the texture (e.g., fatty meat is cut easier when frozen), (b) influencing chemical and biological reactions (e.g., influence on color and ripening, stopping of wine fermentation), and (c) producing ice and cryogenic liquids.

There are several methods for reducing temperature, the most important being the following: (1) mechanical compression methods, (2) physical–chemical methods, (3) use of ice, (4) direct evaporation, and (5) electrical methods.

In the mechanical compression and the physical–chemical methods, low temperatures are created by evaporation of refrigerants. A "pump" provides a

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permanent partial pressure reduction over a refrigerant (which is a liquid evaporating at relatively low temperature), so that temperature reduction, through the evaporation of the liquid, occurs. In the case of the mechanical methods, this "pump" is a compressor. In the physical–chemical (absorption) method, two heat exchangers (the absorber and the column generator) and a pump play the role of the compressor. In the absorber, which is a shell-and-tube-like heat exchanger, a weak aqueous ammonia solution, coming from another heat exchanger, the column generator, absorbs the ammonia vapors that come from the evaporator. The mixture creates a strong ammonia solution that is pumped back to the column generator, where it is heated indirectly with steam, resulting in evaporation of the ammonia. The ammonia vapors go to the condenser, while the liquid mixture, after the release of ammonia, becomes a weak ammonia solution.

In ice cooling, the temperature reduction is achieved by heat absorption during ice melting. In the vacuum method, the reduction of temperature is achieved by the evaporation of the excess water of the product (e.g., water on its surface), when placed in a vacuum chamber. In the electrical methods, the temperature reduction is achieved by the reverse of the Peltier effect (energy input for maintaining a temperature difference between the joints of two different metals, such as Cu–Constantan). Plank and Kuprianoff (1960) and Moersel (1967) describe several processes for producing low temperatures. However, in this book, only the compression refrigeration system will be discussed, since it is the most frequently used in relation to foods. Ice production, which is used in fish and sausage manufacturing and direct evaporation, will be discussed in connection with food refrigeration, since these three methods are mainly used in foods. The compression methods find the widest application, while ice is used in fish cooling and the direct evaporation in chilling of fresh vegetables.

The refrigeration equipment that is used for food may be classified as (a) refrigeration-producing equipment and (b) refrigeration-using equipment. Table 9.1 presents a classification of refrigeration equipment covering production and use of refrigeration. The refrigeration-using equipment may be further classified into equipment: (1) for applications above the freezing point of food (cooling), (2) for temperatures below the freezing point of food (freezing), (3) for distribution, (4) for retail, and (5) for ice production.

9.2 Refrigeration Equipment

9.2.1 Refrigeration Cycles

In compression methods, there are several variations of refrigeration cycles, but in all cases of simplified typical mechanical refrigeration cycles, four basic elements are distinguished (Fig. 9.1a): the evaporator (A), the compressor (B), the condenser (C), and the control or throttling valve (D). The evaporator is the element of the

Table 9.1 Classification of refrigeration equipment

Main category	Use of refrigeration		Equipment and use		Type of equipment
Production of refrigeration	Compression refrigeration		Compressors		Reciprocating
					Rotary
					Centrifugal
					Screw
			Evaporator		Indirect heat exchange
					Direct heat exchange
			Condenser		Water-cooled
					Air-cooled
					Evaporative type
			Other		Defrosting systems
					Pumps and fans
					Intercooler
					Oil separator
					Receiver
					Valves and control system
Use of refrigeration	Processing and preservation	Cooling	Solids	Continuous	Hydrocooling
					Surface cooling
					Tunnel
				Batch	Evaporative
					Tunnel
					Surface cooling
					Cold stores
			Liquids	Continuous	Surface cooling
				Batch	Surface cooling
					Mixing
		Freezing	Temperature down to –40 °C	Continuous	Tunnel
					Plate
					Belt
					Fluidized bed
				Batch	Tunnel
					Plate
			Temperature <–40 °C	Brine	Immersion, spraying
				Cryogenic liquids	CO_2
					N_2
	Distribution		Truck		Compression refrigeration
					Cryogenic liquids
			Container		Compression refrigeration
					Cryogenic liquids

(continued)

Table 9.1 (continued)

Main category	Use of refrigeration	Equipment and use	Type of equipment
	Retail	Closed type	Shelf cabinet
			Deep freezer
		Open type	Shelf cabinet
	Ice production	Production	Blocks
			Flakes
			Tubes
		Storage, transport	Screws, belts, bucket

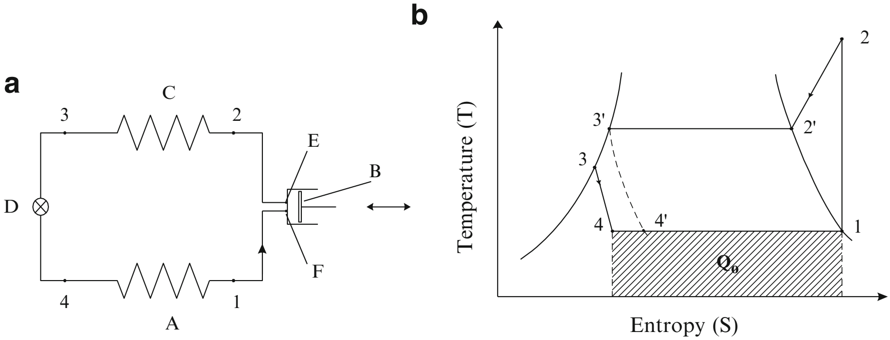


Fig. 9.1 Single-stage refrigeration cycle

refrigeration cycle coming directly or indirectly in contact with the food. In most cases, the heat exchange between product and evaporator is indirect, since a medium, such as air, glycol, etc., intervenes. Direct contact between product and evaporator exists in cases such as the freezing of food by plate freezing equipment. If a food processing equipment, using low temperatures, is small, the whole refrigeration-producing unit is part of this equipment. In larger equipment, only the evaporator is part of it. In large cold stores, only the evaporator is in the same area with the food.

The heat that flows from the product to the evaporator (Q_o) causes the partial evaporation of the fluid refrigerant. The compressor, subsequently, sucks the evaporated refrigerant, while the suction valve (E) opens, and the discharge valve (F) closes. The continuous removal of the vapor over the liquid refrigerant secures the steady heat absorption from the environment. The sucked refrigerant vapor is subsequently compressed, while the suction valve closes and the discharge valve opens. The high-pressure hot refrigerant comes to the condenser, where it is liquefied by a cooling medium, which is usually water or air. Subsequently, the liquefied refrigerant is eventually subcooled (e.g., by indirect contact with low-temperature refrigerant), and it is cooled as it passes through the control valve. Finally, the cool liquid refrigerant flows back into the evaporator. This cycle is repeated, and each time, a new amount of heat is taken away from the product, resulting in the reduction of its temperature.

As indicated in the *T–S* diagram (Fig. 9.1b), the amount of the heat removed (Q_o) is represented by the surface below the line 4–1. If no subcooling takes place, the amount of heat removed (area below 4′–1) is smaller. The lines, 2′–3′ and 4–1, are isotherms ($T =$ constant) as well as isobars ($p =$ constant). The line 3–4 (3′–4′), indicating the throttling process, is isenthalpic ($H =$ constant). The actual process differs from the process described above, since there are losses during heat exchange and compression.

The process described above is a single-stage process. If very low temperatures (usually $T < -20$ °C) have to be created, a two-stage process is required (Fig. 9.2). Such a process consists effectively of two single-stage processes, in which two compressors are used: one compressor (B_1) for the low-pressure stage (I) and a second (B_2) for the high-pressure stage (II). The condenser of the low-pressure stage is the evaporator of the high-pressure stage. The equipment combining the condenser of stage I and the evaporator of stage II is the intercooler. The intercoolers may be classified as open- and closed-type units (Dossat 1978). In the open type, the liquid refrigerants, which come from both cycles, are mixed in a vessel (Fig. 9.2a). In the closed type, the compressed gas of the low-pressure stage is cooled down indirectly, e.g., as it passes through a coil submerged in the refrigerant of the high-pressure stage. In this case, the refrigerants of the two stages may be different. A two-stage compression is used, if the pressure ratio between the condenser (p_c) and the evaporator (p_o) exceeds certain limits. For ammonia $p_c/p_o \propto 8$. The pressure in the intercooler (p_m, bar) is given by (9.1) (Moersel 1967).

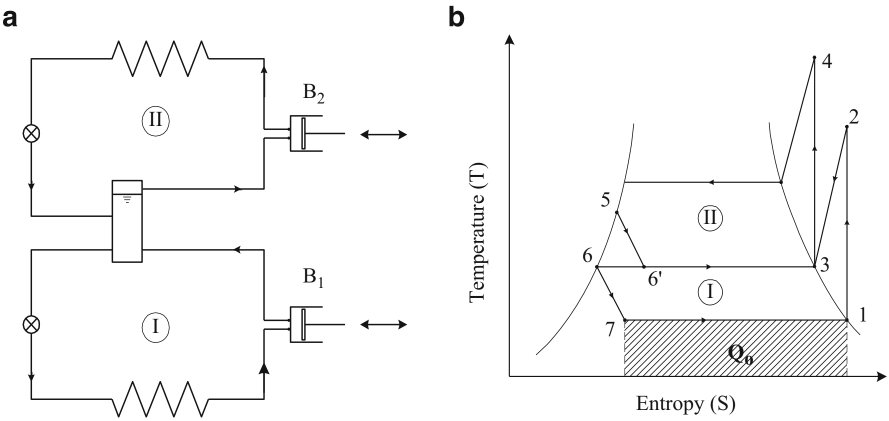


Fig. 9.2 Two-stage refrigeration cycle

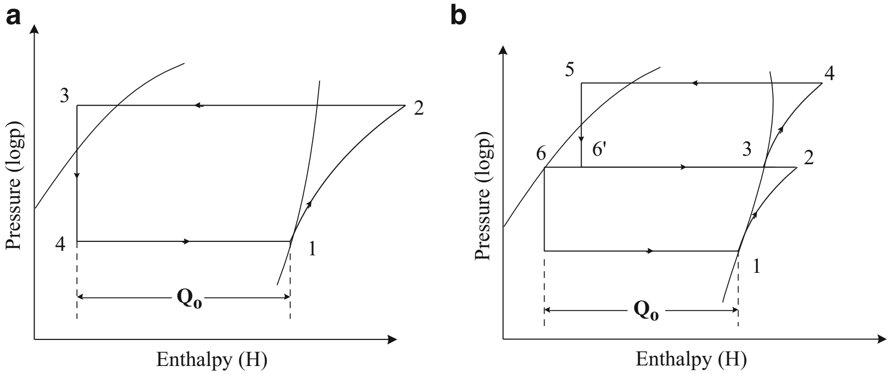


Fig. 9.3 Pressure–enthalpy diagrams for single- (**a**) and two-stage (**b**) compression

$$p_m = (p_c p_o)^{1/2} + 0.35 \tag{9.1}$$

Figures 9.1b and 9.2b indicate the one- and the two-stage refrigeration cycle in a *T–S* diagram. However, in calculations, the pressure–enthalpy [log(p)–H] diagram is preferred. In this diagram, the heat quantity Q_o absorbed is represented by a straight line (Fig. 9.3).

In comparing refrigeration equipment, the reference to certain condensation (T_c) and evaporation (T_o) temperatures is important. Often, in comparing one-stage compressor performances, the temperatures $T_c = 25$ or 30 °C and $T_o = -10$ or −15 °C are used. In a two-stage compression, the reference temperatures are $T_c = 25$ or −10 °C and $T_o = -25$ °C. However, a guarantee for an installation should not be given for these temperatures but for the temperatures that would be applied, because other factors could play a role in the final temperature development (Pohlmann et al. 1978).

In a refrigeration cycle, besides the abovementioned four basic elements, several other additional elements and controlling instruments are required. In cold stores, e.g., the condensed liquid refrigerant is collected in a container (receiver), before being further distributed to the evaporators. In the case that the distance between storage and machinery room (compressors and condensers) is large, pumps are also used for transporting the refrigerant to the evaporators. Furthermore, devices for defrosting the evaporators and for oil separation of refrigerants are required.

Evaporators differ in the way heat is transferred to them from the product. Since in almost all cases, a relative movement between products and the heat transfer medium exists, it is very important to maintain this relative motion (velocity) as high as possible. However, this should be done without significant negative effects on the product quality, such as product weight loss during cold storage or poor economic effectiveness of the process, due to energy increase caused by additional ventilation.

Advantages of refrigeration are (1) extension of the shelf life of the products with minor alternation of the original condition of food, (2) product application possible, even without packaging, (3) environmentally friendly, and (4) temperature scale-up possible.

The disadvantages of refrigeration are (1) expensive process; (2) weight loss of product, if process control is inadequate; (3) product freeze burn, if packaging and temperature not proper; (4) continuous supervision required; and (5) in freezing, the refrigeration chain must not be interrupted.

9.2.2 Compressors

The main types of compressors used in the production of low temperatures are the reciprocating compressor, the rotary compressor, the centrifugal compressor, and the screw compressor. These compressors are used as single units or combined with others for increasing the required capacity or the versatility of a refrigeration system. In combined application, they can work parallel with other compressors of the same type or, in some cases (e.g., in the production of very low temperatures), with compressors of other types. The reciprocating compressor is used in systems of small to medium refrigeration capacities. The rotary compressor is used in the production of very small to small capacities. The centrifugal compressor is engaged in the production of very large capacities. The screw compressor is used in the production of medium to large refrigeration capacities.

9.2.2.1 Reciprocating Compressors

General Characteristics

The reciprocating compressor is the most common type of compressor used in compression refrigeration. It is used in the production of a very wide spectrum of refrigeration loads (0.5–350 kW) for applications above or below the freezing temperature (T_f) of the food. A compressor basically consists of (a) cylinders with reciprocating pistons, (b) inlet and outlet valves, and (c) a lubrication system (Fig. 9.4a). A compressor usually has four or more cylinders. The larger number of cylinders, in addition to the capacity increase, results in a more smooth pumping of the refrigerant. Furthermore, pistons are often arranged in V, instead of line formation, for reducing vibration and the size of the machine.

The capacity of a reciprocating compressor depends on the following: (1) temperature of evaporation (T_e), (2) temperature of condensation (T_c), (3) number and size of the cylinders, (4) revolutions of the crankshaft (rpm), (5) losses in cylinder and valves, (6) condition and construction quality, (7) lubrication system, and (8) type of refrigerant.

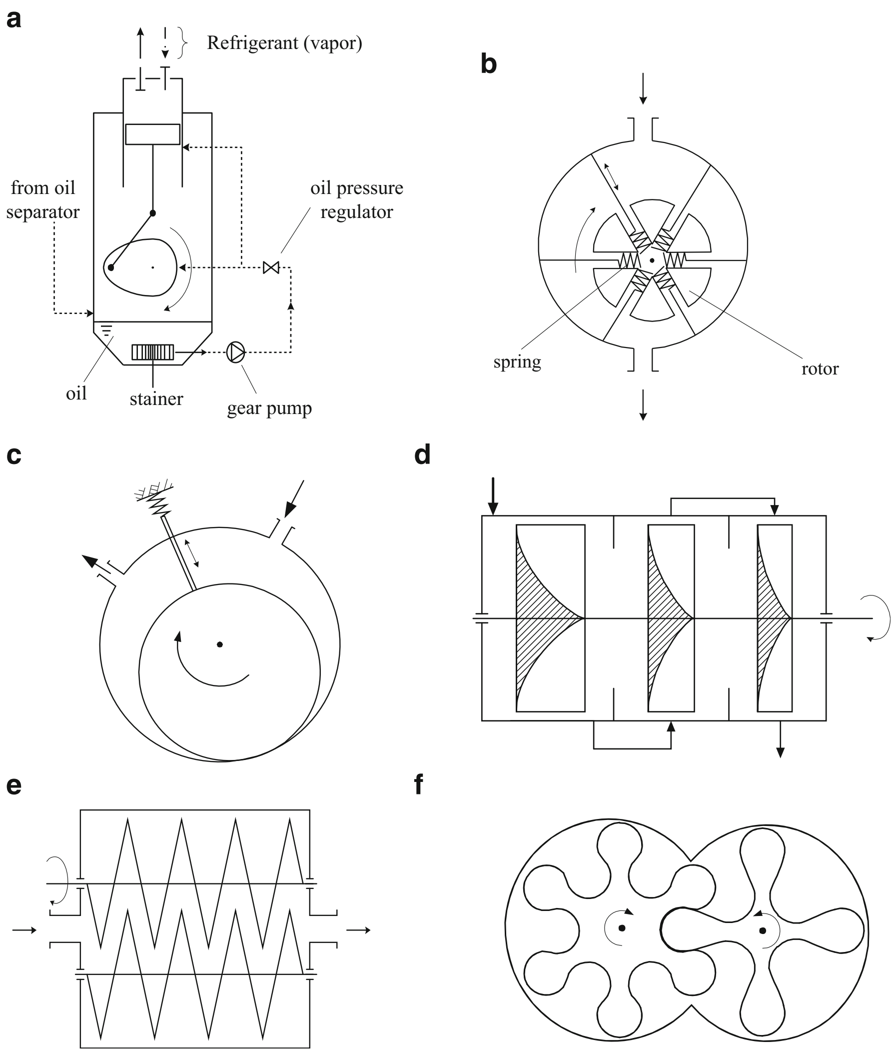


Fig. 9.4 Types of compressors (see text)

The capacity of a compressor depends on the displacement of its pistons. This depends on the volume of the cylinders and the piston speed. The displacement of the pistons ($V_{p,}$ m^3) and the piston velocity (u_z, m/s) are given by (9.2) and (9.3), respectively:

$$V_p = 60\,A\,h\,N\,z \tag{9.2}$$

where A, area of the base of cylinder (m^2); h, the length of the stroke (m); N, the revolutions of the crankshaft (rpm); and z, the number of cylinders.

$$u_z = hN/30 \tag{9.3}$$

Small reciprocating compressors are advantageous, when performing at high loads. However, since small compressors have small cylinder volume, for increasing their load, high crankshaft revolutions (N) are required. Nevertheless, a piston speed (u_z) should not exceed 2.5 m/s, since, otherwise, the vapor velocity through the valves would be too high (Plank and Kuprianoff 1960). As a result, according to (9.3), the length of stroke (h) decreases when the revolutions increase. Therefore, according to (9.2), for increasing the piston displacement and load, the only possibility is to increase the piston area (A). But a significant increase of A would increase the blow-by effect of the pistons, reducing the compression efficiency (Dossat 1978). Therefore, according to good design practice, certain limits must be put to the relation piston area (A) to length of stroke (h) or to the relation cylinder bore diameter (d) to stroke (h). Thus, according to Plank and Kuprianoff (1960) and Dossat (1978), the ratio (d/h) should be 1.0:1.2, and the crankshaft revolutions should not exceed 1200–1750 rpm. The larger the length of the stroke, the lower the piston velocity. For reducing the size of compressors, high crankshaft revolutions are preferred (usually 1200–1750 rpm). This results in a reduction of the ratio of machine weight to load capacity. However, for practical reasons, the piston speed cannot exceed 2.5 m/s (Plank and Kuprianoff 1960). A possibility of keeping the vapor velocity in acceptable limits would be the increase of the valve opening area. However, this would require very wide cylinders, but such construction is not possible, since a certain ratio of limits of cylinder diameter (d) to stroke (h) has to be considered. The volumetric efficiency (ε) of a cylinder is the ratio of the volume of suction vapor of refrigerant compressed per minute (V_a) to the actual volume of the cylinder (V_z). The volumetric efficiency decreases, when the compression ratio (high to low pressure) increases. The greater the pressure differences between evaporation and condensation, the lower the refrigeration load the machine performs. The volumetric efficiency of a reciprocating machine is reduced by losses in the cylinder and the valves. The loss in the cylinder is due to heat exchange with the environment and leakage, caused by wear. The loss in the valves is due to constructional restrictions and wear or inefficient adjustment. The indicator diagrams of compressors (pressure–volume diagrams) testify the reason of efficiency reduction, presenting the eventual sources of loss (Fig. 9.5). Such sources can be the delay in opening of valves, the leakage when these close, wet vapor, and "dead" space in the cylinder, due to construction restrictions. Pohlmann et al. (1978) and Dossat (1978) analyze several sources of reduction of the efficiency of reciprocating compressors on the basis of such indicator diagrams.

Water-cooling of the cylinders would move the compression toward the isothermal process, increasing the efficiency of the compressor. However, this is not practical, since it would also result in the discharge of saturated liquid from the compressor. Water-cooling is therefore restricted only to some systems using refrigerants of high discharge temperatures. Nevertheless, even then, this is not done for increasing the efficiency of the compressor but for reducing the rate of oil carbonization and the formation of acids, since both increase, when the discharge temperature increases (Dossat 1978).

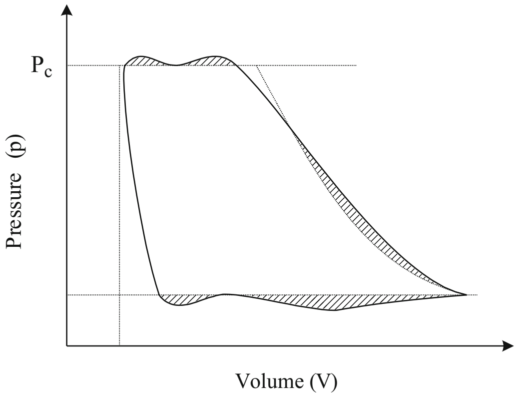


Fig. 9.5 Indicator diagram of reciprocating compressor

One reason of the wide use of reciprocating compressors is that several kinds of motors (electrical motors, diesel, gasoline engines, etc.) can drive them. This makes also their use possible in mobile units. The driving of reciprocating compressors can be direct or through v-belts.

Some indicative data of reciprocating compressors are volume of vapor sucked by piston displacement $V_p = 100–250\ m^3/h$, piston speed $u_z = 2–3$ m/s, and revolutions of crankshaft $N = 500–2000$ rpm. Reciprocating compressors in V arrangement, without motor, may have overall dimensions $1 \times 1 \times 1$ m. Two-stage compressors have more cylinders in the first stage. A compressor of four cylinders may have, e.g., three cylinders in the first stage. Since the evaporation in a two-stage reciprocating compressor takes place at lower pressures than in a single-stage compressor, when the temperature of condensation is the same, the total refrigeration load of the single-stage compression is lower. For example, for the same swept volume and number of cylinders, the refrigeration load of a two-stage ammonia compressor, at $T_c = 25$ °C and $T_o = –40$ °C, is only about 20 % of that of the single stage (T_c=25 °C, $T_o = –10$ °C). The weight of single- and two-stage compressors depends on the number of cylinders, varying between 1 and 3 tons, without the motor.

Selection of Reciprocating Compressors

In selecting recipocating compressors for a certain load, the brake power N_b (kW) can be estimated from (9.4).

$$N_b = Q_o / K_{th} \eta\, \eta_m \tag{9.4}$$

where Q_o is the refrigeration load (kW); K_{th}, the theoretical specific refrigeration load; $\eta_{I,}$ the indicated efficiency (fractional), and η_m, the mechanical efficiency (fractional).

K_{th} is estimated from data of the refrigeration cycle [e.g., log(p)–H, diagram], taking into account the operating time, e.g., 16 h/day. The indicative efficiency (η_I) takes into consideration the loss in the cylinders. For ammonia, it can be estimated by the method suggested by Linge (1950) and Moersel (1967), as a function of the ratio of evaporation to condensation pressures and the corresponding temperatures, the vapor volume displaced (V_d), and the cylinder dead space ($e = 2$–8 %). According to this method, for $T_c = 25$ °C, $T_o = -25$ °C, $V_d = 50$ m^3/h, and $e = 4$ %, the indicative efficiency is $\eta_I = 0.78$. The mechanical efficiency (η_m) is usually 0.85–0.93 (Moersel 1967).

Advantages of reciprocating compressors are (1) wide range of application (processing, storage, retail); (2) wide range of refrigerants can be used; (3) for small capacities, possibility to construct them as compact units (including heat exchangers and condensers); (4) possibility to use them (two stage, booster) for producing very low temperatures, and (5) long manufacturing experience.

The disadvantages of reciprocating compressors are (1) vibration during operation, (2) being noisy, (3) capacity limitations, (4) being heavy and space occupying, and (5) many moving parts (i.e., many spare parts).

9.2.2.2 Rotary Compressors

In rotary compressors, a rotor is placed eccentrically in a stator/case. During rotation, the space between rotor and stator is reduced, resulting in compression of the vapor. There are several structures for achieving this progressive compression. In one case (Fig. 9.4b), the rotor is made up of blades that slide in ducts as the rotor moves. The contact between the ends of the blades and the case is steadily tight. Unequal compartments are formed as the blades rotate. The refrigerant vapor is sucked in the most spacious of the compartments and it is progressively compressed until its discharge. In another variation (Fig. 9.4c), a rotating cylinder is eccentrically positioned inside a larger cylinder, in which there are a suction and a discharge port. A diaphragm, which is in permanent contact with the internal cylinder with a spring device, is placed between the two ports. It provides steady separation of the space between the two cylinders during rotation of the internal cylinder. The refrigerant vapor, sucked in the equipment, is progressively compressed, as the interior cylinder rotates. A valve, placed outside the discharge port, can control the compression. For preventing freezing of the cylinder, caused by low suction temperatures, the external cylinder can be jacketed for heating. Advantages of the rotary equipment are the lack of valves and the small dimensions ($0.5 \times 0.30 \times 0.25$ m, without motor). Their main disadvantage is the relatively high wear of seal and stator wall and the difficulty to obtain high-pressure difference between suction and discharge ($\Delta p < 5$ bar). For reducing the wear, the rotation of the cylinder in larger equipment is up to 750 rpm and for smaller than $\Delta p < 5$ bar 1500 rpm, both putting limits to the capacity of the machine (swept volume, 30–40 m^3/h, load about 120,000 kJ/h or 33.3 kW). Therefore, a rotary compressor is suitable only for small equipment, acting as a booster, or for cases in which a reciprocating compressor, due to very low-pressure differences, would be less efficient.

9.2.2.3 Centrifugal Compressors

The centrifugal compressor is very similar to the centrifugal pump. It is mainly used in air conditioning of large installations, and it works competitively when large amounts of gas (>2000 m^3/h) have to be sucked and compressed at relatively low-pressure differences (1.4–2.0 bar per stage). Therefore, refrigerants of high molecular weight and more impeller wheels (usually 2–4) are used (Fig. 9.4d). The impeller wheels become progressively smaller in the direction of gas flow, for compensating the reduction of the gas volume due to compression in the previous stages. The compressors are usually assembled as compact units together with shell-and-tube heat exchangers. For reducing pressure losses, compact units are preferred, in which the heat exchangers are the largest part. The smallest vapor suction capacity, for profitability, is 2000 m^3/h. For achieving a propulsion of large quantities of gas, the rotors run at >3000 rpm (Pohlmann et al. 1978). These units are usually used in large buildings/factories and in ships. The main advantage of centrifugal compressors is their ability to produce large scale of refrigeration load with relative small units. Further advantages are the entrainment of very low quantities of oil in the refrigeration system and the simple construction. Disadvantages for a food plant are the relative small pressure difference (evaporation–condensation) achieved and the economic limits, restricting the use to large-scale installations only.

9.2.2.4 Screw Compressors

The screw compressor is the newest development among refrigeration compressors. It can be engaged in producing temperatures as low as –50 °C. A screw compressor is best suited for loads, supplementing the refrigeration loads of reciprocating compressors, e.g., for ammonia, $Q > 1{,}000{,}000$ kJ/h (278 kW) at $T_c = 25$ °C and $T_o = -10$ °C and $Q > 300{,}000$ kJ/h (83 kW) at $T_c = -10$ °C and $T_o = -40$ °C. The basic element of the compressor is the counterrotating screws (rotors). One rotor (the male) has four lobes and the other (female) six lobes. They are fitted in a stator consisting of two cylinders intermeshing longitudinally (Fig. 9.4e, f). Usually one screw is connected, through a gearbox or belts, to the power source, driving the other during its rotation. The refrigerant enters at the one end, and it is driven along the axis, compressed between the lobes. The pressure difference between vapor inlet and vapor outlet depends on the refrigerant and can vary between 7 and 20 bar. The volumetric efficiency (i.e., the relationship between the effective vapor propulsion and the geometric volume) of the equipment depends on the "internal leakage" (vapor set back), caused when a minimum critical clearance between stator and rotors is exceeded. For reducing the internal leakage, the screws must rotate at high speed ($N > 3000$ rpm). The swept volume of such a compressor depends on the refrigerant and the temperatures used, and it can be 700–10,000 m^3/h. For the same load, the swept volume of ammonia equipment is about 20 % larger than that of refrigerant R22. The required brake power depends on the refrigerant load. For smaller units, it may be about 20–150 kW, but for larger units exceeding 12 million kJ/h, it can be more than 700 kW. The dimensions of a screw compressor are not large. The length of a single screw may be 0.40–1.0 m and the diameter of the rotor 0.15–0.40 m. The maximum overall dimensions of a combined unit, including heat exchangers and motor, may be about $5.0 \times 2.0 \times 2.5$ m. The maximal weight of such a unit (without the motor) may be up to 6 tons.

Advantages of screw compressors are (1) high specific output (with economizer); (2) small dimensions, i.e., less construction materials, less weight, and less space occupied, e.g., about 30 % lighter and 40 % less space than comparable reciprocating units; (3) less vibration, e.g., only 1/100 of comparable reciprocating compressors; (4) lower refrigerant losses (because of less vibrations); (5) no valves (therefore higher rotation speeds possible); (6) compared to centrifugal compressors, screw compressors can use high-pressure refrigerants; and (7) wide-range capacity control (the refrigeration load can be easily reduced from 100 % down to 10 %).

Disadvantages of screw compressors are (1) screw wear, the high speed of rotation and the tight contact of the lobes increase friction; (2) for achieving high speeds, gearboxes or other transmissions are required, which increase (about 5 %) the wear and the energy consumption; (3) if the manufacturing accuracy or the materials are poor, or if the bearings and screws are worn, internal leakage occurs, reducing the efficiency of the equipment (bearings should be replaced every 20,000 h); and (4) both screws have to be replaced at the same time, even though their wear is uneven.

9.2.3 Evaporators

Evaporators are heat exchangers (see Chap. 6) that are part of a refrigeration cycle, used in absorbing heat from the products that have to be cooled or from the environment. There are several ways to classify evaporators used in reducing the temperature of food. They can be classified according to the: (a) final use of equipment, (b) medium contacting the evaporator externally, (c) way that the evaporator transfers heat from the products to the refrigerant, (d) form of the heat-exchanging surface, and (e) construction of the equipment.

With respect to the final use, they can be distinguished between evaporators that are part of a food processing unit, e.g., fluidized bed freezer or plate freezer, and evaporators that are part of an installation, e.g., an installation in a cold storage room or in a building. The medium coming directly in contact with an evaporator

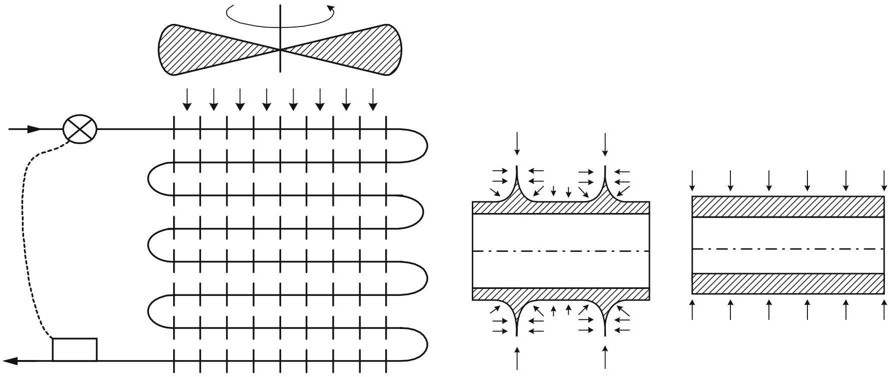


Fig. 9.6 Blower evaporator with fins

externally can be a gas (usually air), a liquid (e.g., brine), or a solid (e.g., food product). The evaporators contacting a liquid directly can be forced or free convection units, which use a fan in the case, e.g., of air or a pump in the case, e.g., of water or brine cooling.

The way heat is transferred from a product to the evaporator can be direct, if no medium exists between the surface of the heat exchanger and the product, or indirect, if some fluid (gas or liquid) intervenes. The heat-exchanging surface can be a tube (coil) or a flat surface. A tube can be bare or finned (Fig. 9.6). The distance of fins must be >8 mm for avoiding increased ice accumulation. Heat-exchanging flat surfaces of evaporators are double-wall structures. They can be plain or structured (Fig. 9.7). Fins on or in the tubes, or structured surfaces, increase the heat-exchanging surface area. The cooling medium in the evaporator can be cold water, brine, or a refrigerant liquid having a high boiling point. With respect to construction, the two main types are the flooded and the dry-expansion evaporators (Fig. 9.8).

Table 9.2 presents a classification of refrigeration evaporators, based on the external heated medium.

9.2.3.1 Forced Convection Air Coolers

General Characteristics

The air cooler is widely used for cooling air that is further used in food processing, food preservation, or air-conditioning applications. The air cooler may be, together with the other refrigeration-cycle elements, part of a food processing equipment, or it may be installed separately from the compressor and the condenser, in the room or equipment that needs low-temperature air. In most cases, the system used is the direct expansion, but in larger installations, the flooded system is also applied. The equipment basically consists of one or more fans; a bank of parallel tubes or plates,

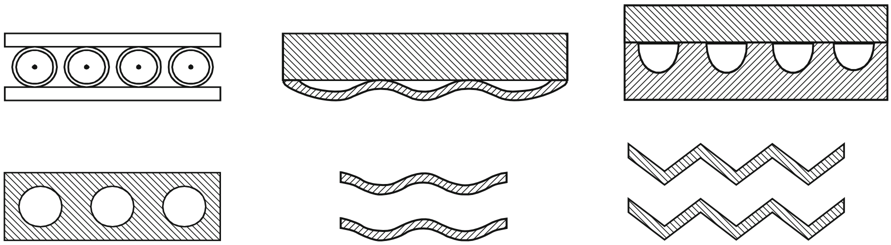


Fig. 9.7 Double-wall and structured plain surfaces

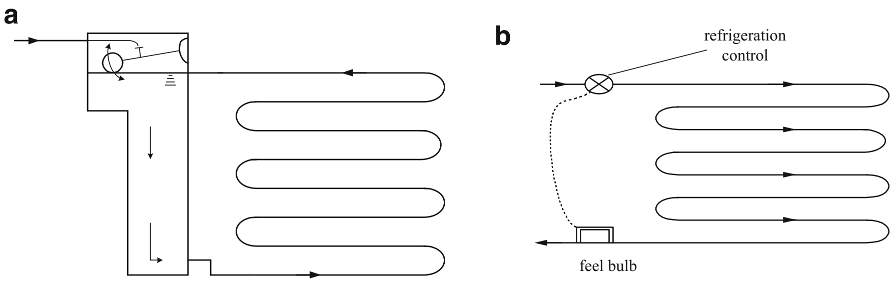


Fig. 9.8 Flooded type (**a**) and dry-expansion (**b**) evaporators

in which refrigerant circulates; and the defrosting installation. The tubes and the plates can be bare or finned. The most commonly used tube evaporator is described as follows:

In the tube air cooler, air is blown through the tubes by means of a fan. The tubes of the air coolers, using no ammonia refrigerants, usually are made of copper. This increases the heat transfer rate and reduces corrosion problems. The heat transfer of air coolers is also enhanced, when small-finned tubes (coils) are used. When air is blown vertically on tubes of internal and external diameters 24 and 28 mm, respectively, the surface area of the tubes with fins is seven times larger and the external heat transfer coefficient three times larger than that of the corresponding bare-tube evaporators. Each air cooler usually has 1–4 axial fans. This kind of fan is used for blowing large amounts of air at relatively low-pressure differences. When this evaporator is used in cold stores, the air velocity must not be very high, because this causes drying of unpacked food. Usually, the air velocity through the evaporator tubes is about 2–3 m/s and that above the products about 0.10–0.25 m/s. In cold stores of frozen products, the air velocity is 0.5–2.5 m/s. High refrigerant and air velocity increase the overall heat transfer coefficient of the evaporator. When the tubes have fins, the distance between the fins depends on the desired air temperature and the "air throw" (how far air is blown). For a higher velocity and air throw, this distance must be relatively large. For increasing the surface of a heat exchanger, fins can be only 3–5 mm apart. However, in evaporators used for freezing of food, the distance between the fins of tubes should be higher than 8 mm. This reduces the

Table 9.2 Classification of refrigeration evaporators

Heated medium	Type of evaporator	Type of heat exchanging surface	Examples
Gas	Forced convection	Bare	Cold storage, freezing tunnel, fluidized bed freezing equipment, multishelf produce sale case
		Finned	
		Plate	Shop refrigerators
	Free convection	Bare	Cold store rooms (coiled ceiling)
		Finned	
		Plate	Home freezer
Fluid	Bath and tube/plate	Bare	Cooling of liquids (brine), ice production
		Finned	
		Plate	
	Double pipe		Juice cooling, pasteurization (cooling stage)
	Shell and tube		Air conditioning, brine cooling
	Shell and coil		Quick cooling of liquids
	Double wall	Smooth	Freezing of juice
		Structured	Plate heat exchanger
		Jacketed	Scraped surface heat exchanger
	Baudelot		Juice cooling
Solids	Double wall	Smooth	Plate freezer
		Structured	Cooling of dried grain (tower)
		Jacketed	Meat cutter (preparation of meat mash)
	Rotating coil		Crystallization

pressure drop, caused by ice accumulated between the fins. Fins are used when the heat transfer coefficient between the refrigerant and the internal surface of the tube (h_i) is larger than between the tubes and the air blown on them ($h_i > h_a$). If $h_i < h_a$, as in the case of forced circulation of liquids outside tubes, the fins should be inside the tubes. Nevertheless, although such a measure increases the heat transfer between the tube and refrigerant, it also increases the pressure drop in the tubes (Fig. 9.9). Therefore, a thorough analysis of the whole system is required, whenever such measures should be applied. When the evaporator lies in vats with agitated brine, then, since $h_i = h_a$, no fins are required.

In a cold store, according to Dossat (1978), the temperature difference between air incoming and leaving the air cooler is approximately equal to the temperature difference between the mean temperature of the evaporator and cold storage room. Usually the temperature differences between inflowing and outflowing air is about 6–7 °C. In cold store rooms, the air coolers are placed preferably above the doors, at about 0.6 m away from the walls, the ceiling, and the products (e.g., stacked boxes).

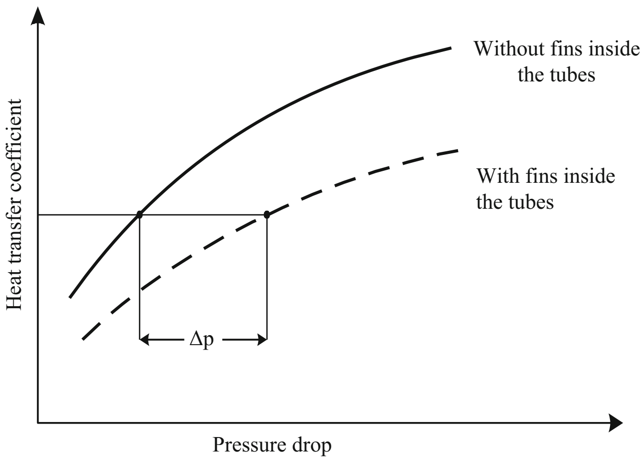


Fig. 9.9 Pressure drop in the finned coil

Indicative values for large dry-expansion finned air coolers (e.g., capacity 70–80 kW) are heat exchange surface, 500–600 m^2; refrigerant content, 60–70 kg; number of fans 3–4; energy consumption per fan, 0.5–0.8 kW ($N = 3000$–4000 rpm); air-blown volume, about 20 m^3/s; overall dimensions, $5.0 \times 1.0 \times 1.5$ m; and empty weight, about 1 ton. The noise level of fans must be below 80 dB.

Defrosting

An evaporator must be periodically defrosted for maintaining a high performance. The frequency and the length of each defrosting depend on the cooling system (temperature, air velocity, type of evaporator, etc.), the product cooled (kind of product, quantity, etc.), and the defrosting method applied. Frequent defrosting reduces the defrosting time. Usually, in air-cooling evaporators operating in rooms of high humidity, ice is accumulated quickly. Therefore, defrosting of such units may take place every one or every half an hour. Defrosting of small air coolers or other small evaporators can be done "naturally." In this case refrigeration stops, and the fan operates until the ice on the evaporator is melted. Large evaporators are defrosted artificially. Common methods are spraying of evaporators with hot water, electrical heating, and reversing of the refrigeration cycle. In hot water defrosting, water is sprayed on the tubes by means of a distribution pan, located above the heat exchanger, while a second pan, with a drain connection underneath, collects the water. In the electrical system, an electrical resistance induces heat, defrosting the tube and keeping the pan under the tube ice-free. The reversed-cycle defrosting is based on reversing the role between the evaporator and the condenser.

The evaporator is heated, as it condenses the hot vapor that comes from the compressor. This system requires an additional expansion valve, for reducing the temperature of the refrigerant vapor that is sent into the condenser.

9.2.3.2 Bath and Tube Evaporators

Bath and tube evaporators consist of a bank of straight tubes, placed vertically or parallel in a bath, which is filled up with the liquid that has to be cooled (Fig. 9.10a, b). The liquid is often brine, which is subsequently pumped for use as secondary refrigerant. The equipment is applied in cooling large quantities of liquid and meeting of frequent fluctuations in refrigeration demand. The bath and tube evaporator is not used in the direct cooling of liquid food, due to sanitary restrictions, caused by the open-type construction of the bath and the dense placement of the tubes. The open-type construction results in contamination of the liquid by several substances (microorganisms, dust, etc.) and odor. The dense placement of the tubes reduces the volume of the evaporator, by increasing the quantity of liquid in the bath, and makes cleaning difficult. A pump circulates the liquid of the bath at 0.3–0.7 m/s. This kind of equipment is usually a flooded evaporator. Since relatively large quantities of refrigerants are required in flooded evaporators, ammonia, which is a relatively cheap refrigerant, is preferred. In this case, the evaporator is made of carbon steel or wrought iron, since ammonia in the presence of moisture attacks copper and brass. Since these evaporators are almost fully filled up with refrigerant, they have a high efficiency. The overall heat transfer coefficient of equipment, in which brine circulates at a rate of about 1 ton/h, may be about 1400 W/m^2 K (Pohlmann et al. 1978). However, due to large dimensions and weight, this equipment is mainly used in large installations. A float control maintains the evaporator always filled up with refrigerant (Fig. 9.8). The upper tubes are less filled, since evaporated ammonia moves upward before leaving the equipment. Some indicative values of flooded evaporators of a capacity of 400,000 kJ/h (111.1 kW) are heat exchange surface area 50–52 m^2; overall dimensions, $5.0 \times 0.5 \times 1.5$ m; weight, 1.5 tons; and ammonia content, 650 L.

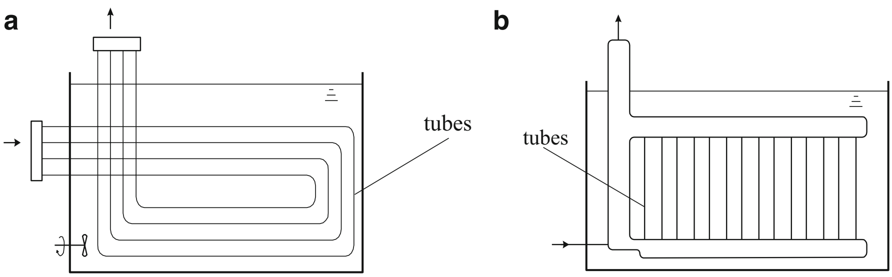


Fig. 9.10 Bath and tube evaporators

Advantages of bath and tube evaporators are (1) high capacity, (2) relatively high overall heat transfer coefficients, and (3) possibility to overcome fluctuations of refrigeration demand.

Disadvantages of bath and tube evaporators are (1) not meeting the sanitary requirements for food, (2) being bulky and heavy, and (3) not being easily cleaned.

9.2.3.3 Shell-and-Tube Evaporators

The shell-and-tube evaporator is one of the most common types of heat exchangers. This heat exchanger can operate as flooded or as direct expansion equipment. It consists of a cylindrical shell containing parallel straight tubes, supported at their ends by tube sheets (Fig. 9.11). Two heads cover both ends of the shell. Depending on the type of equipment and the liquid (product) that has to be cooled, either the product or the refrigerant flows in the tubes. In the case that the product flows in the tubes, this comes in one of the compartments formed between the heads and the tube sheets and continues to flow to the similar second compartment at the other end of the equipment. The refrigerant flows through the compartment formed between the external surface of the tubes and the shell. In some cases, vertical baffles

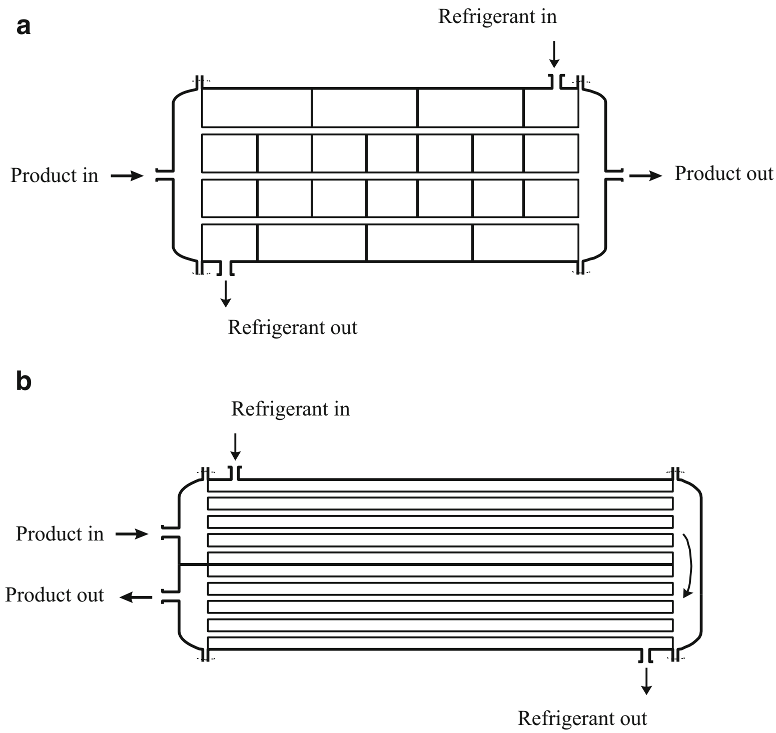


Fig. 9.11 Shell-and-tube evaporators. (**a**) Single pass and (**b**) double pass

increase the flow path of the refrigerant liquid in the shell. There are several variations of such equipment. The product, e.g., may pass only in one direction through the tubes (one-way equipment, Fig. 9.11a) or change direction every time it arrives at the tube sheets (two or more passes equipment, 9.11b). The shell-and-tube evaporator is built in several variations and sizes and can be part of a larger installation or even of a mobile compact refrigeration unit of a food processing machine. It can be installed horizontally or vertically. Special care is required for the gaskets of the heads and for the connections between tubes and sheets. Tubes are welded, and, in construction, stress forces, due to rapid temperature change or fluctuations, must be considered. In refrigeration, the diameter of such units can vary, e.g., from 0.30 to 1.00 m, the overall length from 1.50 to 6.00 m, the weight from 0.3 to 5.5 tons, and the heat exchange surface from 3 to 150 m^2. The overall heat transfer coefficient depends on the product, the refrigerant, and the applied conditions (temperatures, flow velocities), and it may be about 1000 W/m^2 K. The heat duty of large units, e.g., for brine or water-cooling, can be as large as 1000 kW ($T = -5$ °C/+35 °C), Huette (1960).

Advantages of shell-and-tube evaporators are (1) sanitary operation, (2) little floor space required, (3) easily adapted to other equipment, and (4) relatively good heat exchange.

The disadvantage of shell-and-tube evaporators are (1) thermal stresses; (2) probable freezing up of product if flowing in tubes, and (3) high cost.

9.2.3.4 Shell-and-Coil Evaporators

This evaporator consists of a coil immersed in a tightly closed shell or vat (Fig. 9.12a). It can operate as a flooded (product in the coil) or as direct expansion heat exchanger (refrigerant in the coil). With respect to food, small units are often used as quick chillers of draft drinks. In this case, the refrigerant flows in the shell and the product (e.g., beer or juice) through the coil. Such units, besides quick

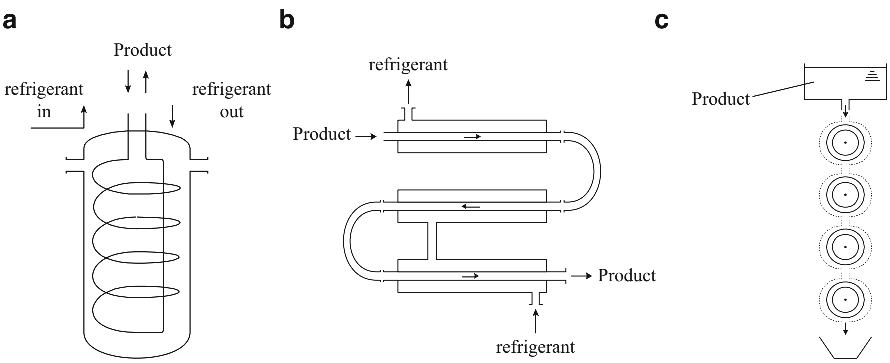


Fig. 9.12 Liquid cooling evaporators. (**a**) Immersed coil; (**b**) two concentric tube; (**c**) Baudelot

chilling, have also the advantage of holdup capacity and operation under sanitary conditions. In the case that the product flows through the coil, its temperature should not drop below the freezing point.

9.2.3.5 Double-Pipe Evaporator

This equipment consists of two concentric tubes (Fig. 9.12b). Usually, the product flows in the central tube. Heads at the end of the tubes, which can be taken off, facilitate cleaning. The refrigerant flows in the annulus countercurrently to the product. The heat transfer coefficient of such equipment is 280–830 W/m^2 K (Huette 1960). In food, this unit is used mainly for cooling fluids (wine and brewing industry) or viscous products (e.g., concentrated juice), or it is part of pasteurization equipment or aseptic processing (e.g., cooling section of concentrated tomato pasteurization). The advantage of this heat exchanger is the sanitary conditions of production, the simple construction, and the easy cleaning. This equipment is strong; however, it is fixed in a certain place and often requires significant headspace, as the tubes are placed one over the other.

9.2.3.6 Baudelot Evaporators

This heat exchanger consists of several tubes that are laid parallel, one over the other (Fig. 9.12c). The refrigerant circulates in the tubes and the product flows outside, forming a thin film around the tubes. External irrigation of the tubes starts above the highest tube. The chilled product is collected in a vat below and along the lowest tube. When the refrigerant is ammonia, the overall heat transfer coefficient of this equipment is 280–830 W/m^2 K (Huette 1960). This equipment is simple, is heat transfer efficient, and can be cleaned easily. However, since it operates open, it requires very strict sanitary processing conditions. Even in this case, it should be used covered, for finish cooling of consumer juices or products that will be further processed immediately (e.g., milk for cheese making).

9.2.3.7 Double-Wall Evaporators

Double-wall design is extensively used in several kinds of heat exchangers. It is used, e.g., as an element of jacketed vats, in which heated products are cooled down to certain processing temperatures, or it is the cooling element of plate pasteurization equipment. In this case, a structured double wall (Fig. 9.7) is formed between two plates and the intermediate gasket. Double wall is also used in direct freezing. Since, for achieving high heat transfer, the distance between the cooling plates is small, highly viscous fluids cannot be processed. Furthermore, for overcoming the wall thermal resistance, a significant temperature difference is required.

9.2.3.8 Cooling/Freezing Evaporators for Solids

Solid food can be directly cooled or frozen by several special structures (Fig. 9.13). Some examples are (a) pressing food between several double-wall plates; (b) falling of food (granules) down, through structured cooled double wall; (c) sweeping food over cooled double-wall surfaces; and (d) agitating food by a refrigerant-containing coil, in a double-wall refrigerated vat. Since all these cases are connected to the processing method applied, they are discussed together with these methods.

Improvement of Heat Transfer in Evaporators

Factors increasing the heat transfer of evaporators are (1) using the right refrigerant for the required conditions, (2) thorough filling of the tubes with refrigerant, (3) no oil in the evaporator, (4) high velocity of refrigerant inside and fluid outside the evaporator, (5) high thermal conductivity of the tube wall, (6) avoiding the accumulation of ice on the heat exchange surface, (7) not very dense arrangement of tubes, (8) using not very thin fins, (9) small diameter of air cooler tubes, and (10) adequate pressing of the product on the freezing surface (direct freezing of solid food).

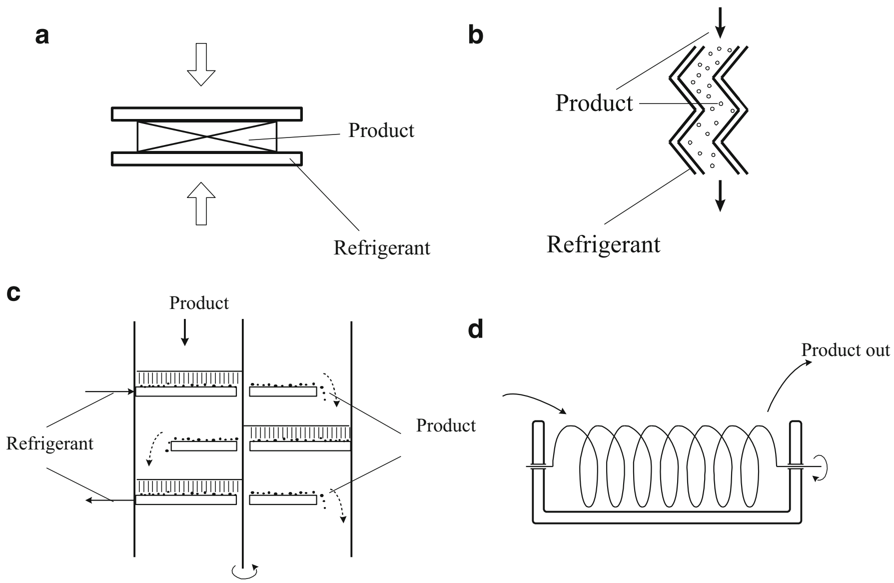


Fig. 9.13 Double-wall evaporators for direct cooling and freezing of solid food

9.2.4 Condensers

Condensers in refrigeration are heat exchangers used for the liquefaction and further cooling of the vapors, discharged from the compressor. The liquefaction of the refrigerants can be achieved by (a) water counterflow in tube heat exchangers (e.g., shell and tube), (b) evaporative condensation, (c) condensation in a cooling tower, and (d) air (Fig. 9.14). The liquid refrigerant, before being distributed to the evaporators, is collected in a container, the receiver. The receiver, besides continuous feeding of the evaporators with refrigerant, is also used for keeping it during repairs, i.e., when the refrigeration cycle is stopped and not hermetically closed.

9.2.4.1 Tube Condensers

The tube condensers are heat exchangers similar to the evaporators, described previously. Liquefaction of the refrigerant takes place, as cooling water flows in the tubes of, e.g., shell-and-tube heat exchangers or in the central tubes of double-pipe heat exchangers. The ammonia vapor, e.g., reaches to the condenser after compression, at about 80 °C. After being condensed, by means of water of about 15 °C, it leaves the heat exchanger at about 25 °C. The liquid ammonia is subsequently cooled to 17–18 °C, before continuing its way to the control valve and to the evaporator. In all cases, counterflow between refrigerant and cooling

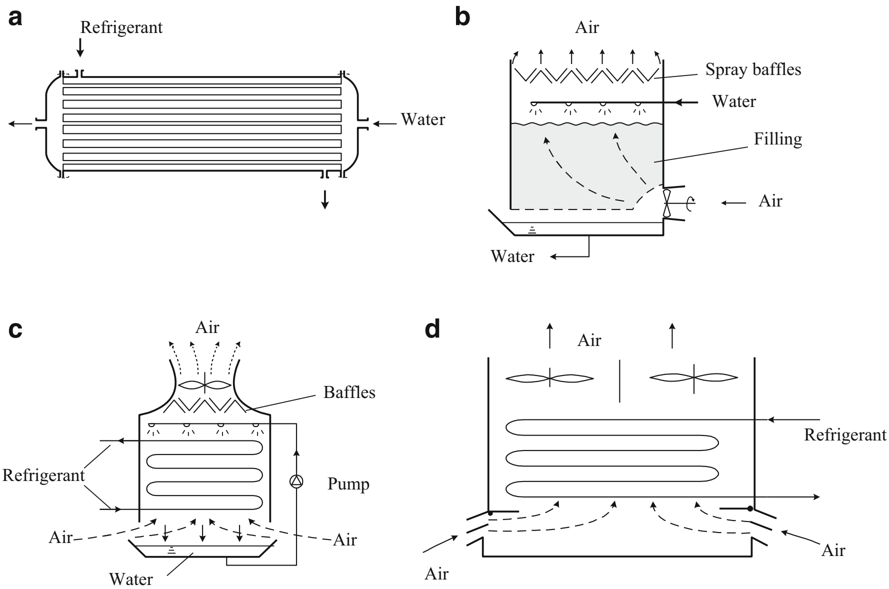


Fig. 9.14 Refrigerant condensers (see text)

water takes place. The velocity of water in the tubes is 1.0–1.5 m/s. The overall heat transfer coefficient of shell and tube, and double-pipe condensers, is about 800 W/ m^2 K (Huette 1960; Pohlmann et al. 1978). This type of condenser (especially the shell-and-tube equipment) is very widely used when plenty of cooling water is available. For water saving, the water leaving the heat exchanger is cooled, e.g., in a cooling tower and recycled. In the cooling tower, water is sprayed over packed material (often wood) filling the tower, while air is simultaneously blown through the space that is formed between the packing material. The cooled water is collected in a vat at the tower bottom. The quantity of water sprayed down is about 30 L/MJ h (Dossat 1978).

9.2.4.2 Evaporative Condensers

The evaporative condenser consists of a bank of parallel tubes, in which the refrigerant circulates. Water is sprayed from the top of the tubes downward, while air is blown from the bottom. The water that falls down is collected and pumped up, for spraying again (Fig. 9.14c). This condenser needs only about 5 % of the water that tube condensers consume, if no recycling takes place (1.3 L/MJ h) (Huette 1960). The overall heat transfer coefficient of this evaporator is 350–700 W/m^2 K. The air velocity is 1.6–2.5 m/s. The dimensions of an evaporative condenser of 250 kW may be about $3.5 \times 1.5 \times 3.5$ m and its weight 4.0–4.5 tons. The energy required by the fans can be 3–5 kW (Pohlmann et al. 1978).

9.2.4.3 Tower Condensers

The tower condenser is mainly used for high capacities. It consists of vertical long tubes in a shell. Water flows down spirally along the internal surface of tubes, from a vat on the top, by means of special nozzles, condensing the refrigerant in the shell. In a variation, this type of condenser also exists as flooded evaporator (refrigerant in the shell) for chilling water. The diameter of the tubes is about 60 mm. The overall heat transfer, due to the spiral motion of the water film, is high. For a tube diameter 60 mm and tube length 3 m, the overall heat transfer coefficient for ammonia condensation, depending on the temperature of the water and the velocity of the fluids, is 700–1650 W/m^2 K. The water consumption in such a condenser is high (double of a shell-and-tube condenser). Tower condensers require little floor space (tower diameter, 1–2 m). However, for capacities of 550–1500 kW and for temperature difference $\Delta T = 7$ °C, their height can exceed 8 m. The weight of such equipment may be 6–15 tons, Huette (1960) and Pohlmann et al. (1978).

9.2.4.4 Air Condensers

This equipment does not need water for cooling. The condensation is achieved by means of air blown through the refrigerant-containing tubes. Large units, e.g., 150–350 kW at $\Delta T = 7$ °C, have four or more air fans, blowing about 4.0–20.0 m^3/s. The fans and the heat exchangers can be installed vertically or horizontally. Usually, condensers have maintenance-free three-phase current axial fans, with IP54 motor protection. All fans have two or more ventilation speeds, depending on the voltage controllable delta-star changeover. Low ventilation speed reduces the noise of the fans (<80 dB). The noise level can be measured according to the BS 848 part II or DIN 45635. The output of each fan motor may be >700 W. The heat-exchanging surface of such a condenser is about 200–400 m^2. In the case that subcooling takes place, the refrigerant is collected in a receiver before being subcooled in an additional heat exchanger. The dimensions and weight of large air condensers (eight fans) may be $10.0 \times 2.5 \times 1.5$ m and 2.5 tons, respectively.

9.2.5 Capacity Control

The capacity of refrigeration may be controlled by three main methods: (1) refrigerant flow control in the evaporator, (2) control of the capacity by the compressor; and (3) flow control of water in the condenser.

In the first case, the control valve controls the capacity through the refrigerant flowing in the evaporator. All control valves regulate the refrigerant flow so that (a) the incoming refrigerant compensates the amount evaporated and (b) a constant pressure difference between evaporation and condensation of refrigerant is maintained. Five main valves and control devices are used: the hand expansion valve, the automatic or pressure valve (Fig. 9.15), the thermostatic valve, the pressure float, and the capillary tube. The pressure float is used mainly in connection with floated evaporators and the capillary tube in small units and

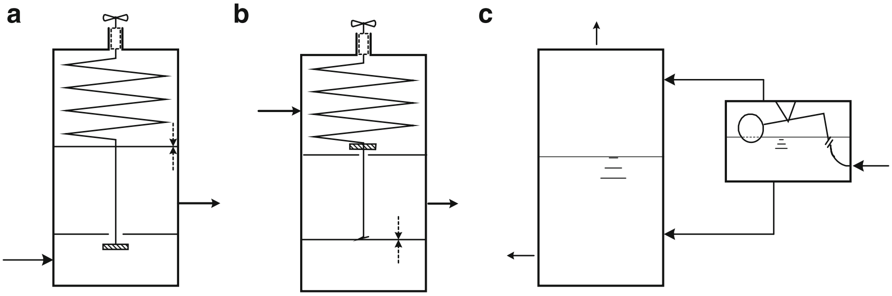


Fig. 9.15 Valves controlling evaporators in refrigeration

air-conditioning/heat pumps. In controlling the capacity through the compressor, the following possibilities exist: variation of the revolutions of the crank, intervention in the opening and closing of the valves, using cylinder bypass, and altering the cylinder volume. These methods are mainly applied to smaller compressors. In larger units, in which more compressors are used, control is done by the "on–off" operation of the compressors. Although this method is expensive, since more compressors are used for covering a certain load, it provides greater security with respect to load adequacy. Furthermore, it increases the versatility of the refrigeration system, and in the case that all compressors used are the same, the stock of spare parts is reduced. The control of water that flows in the condenser can also influence the refrigeration capacity, because it influences the high pressure of the system.

9.3 Refrigerants

9.3.1 Introduction

9.3.1.1 General Aspects

The liquid refrigerants, used in food refrigeration cycles, may be distinguished into natural and artificial. Furthermore, distinction may be based on their applications. They may be classified in liquids that are suitable for cooling or freezing, for smaller or larger equipment (e.g., domestic and commercial application or industrial application), used as stationary or in the transportation of refrigerated food.

Here, only compression refrigeration will be discussed, as it is the most common in food. A form of non-compression refrigeration seldom used for foods is the absorption method, which is mainly used in the chemical industry, in cases that there is excess of thermal heat (e.g., thermoelectric plans) and in small less noisy domestic refrigerators in hotel rooms. The absorption refrigeration gained some acceptance in the food industry since the ozone depletion problem. The most commonly used fluid combinations in such installations are water (as the refrigerant) and lithium bromide (as the absorbent) or ammonia as the refrigerant and water as the absorbent. Besides the already mentioned cases, absorption refrigeration may be used in food processing factories, when significant hot water quantities are necessary in food processing.

Since the invention of the compression refrigeration, several refrigerants have been tested. However since 1897 (when the effective use of NH_3 in vapor–compression systems was applied), this refrigerant dominated the refrigeration of food up to about 1930. After that, ammonia, although it has never ceased being used, was step-by-step replaced by the new refrigerants. These were chlorine-containing refrigerants such as chlorofluorocarbons (CFC) and later the hydrochlorofluorocarbons (HCFC). Finally, due to chlorine contribution in climatic concern about the ozone depletion potential (ODP) in the stratosphere of earth and partially in the increase of the global warming potential (GWP), it was decided (Montreal-Protocol 1987) to replace CFC and HCFC, as in particular the first caused the ODP and the second increased significantly the GWP of our planet.

Table 9.3a–d gives an overview of the refrigerants that have been used up to the Montreal Protocol and thereafter. The refrigerants are indicated by their characteristic ASHRAE number, their molecular formula, and their chemical name. The tables give also several properties, such as the flammability, toxicity, and temperature of inflammation and evaporation, and their applications.

9.3.1.2 Chlorine-Containing Refrigerants

Table 9.3a presents several CFC and HCFC refrigerants that had been used up to 1992. Typical CFC refrigerants were R11, R12, and R502. Typical HCFC refrigerants were the R22, R123, and R124. The commercial production chlorofluorocarbon (CFC) started in 1931 with the production of R12, and 5 years later, the hydrochlorofluorocarbon (HCFC) R22 was introduced. R12 was used in smaller refrigeration equipment for temperatures not much lower than –5 to –10 °C. R22 was used when lower temperatures (e.g., freezing) were required. CFC and HCFC refrigerants (also called “Freon” in the USA) dominated in food refrigeration for about 40 years, as they had good thermodynamic properties and were nontoxic and nonflammable. Furthermore, both of them, especially R12, were well miscible with the mineral oil lubricants that were used with natural refrigerants such as ammonia.

The intensive research that started to invent new refrigerants suitable to replace the chlorine-containing products started effectively after the Montreal Protocol and is still going on. However, it must be pointed out that the invention of a perfect refrigerant is a utopia. Up to now most of the new refrigerants that replaced R22 require 5–15 % more energy. It is pointed out that R12 and R22 are still references in operations at comparable conditions.

9.3.1.3 Requirements of Refrigerants

An ideal refrigerant should fulfill several requirements such as:
Thermodynamic and Energy Requirements

- Large latent heat of evaporation.
- Evaporation of refrigerant (change of liquid to gas phase) at low pressure.
- The required heat for liquefaction of the refrigerant should be preferably low.
- Both evaporation and condensation of the refrigerant achieved at reasonable pressure.
- The volume of the evaporated gas must be low.
- The heat of evaporation should be large.
- Easy to handle and of low cost.

Table 9.3a Refrigerants up to 1992

Category	Type	ASHRAE number	Molecular formula	Chemical name (IUPAC)	Application	Properties	Remarks
Chlorine-containing refrigerants (up to 1992)	Chlorofluorocarbons (CFC)	R11	CCl_3F	Trichlorofluoromethane	Ch	NF, *T*: 23.77 °C	For turbocompressors
		R12	CCl_2F_2	Dichlorofluoromethane	C, AC	NF, NTx, *T*: –29.9 °C	Widest used refrigerant, up to 1989
		R13	$CClF_3$	Chlorotrifluoromethane	F	FL, *T*: –81.5 °C	
		$R13b_1$	CF_3Br	Bromotrifluoromethane		*T*: –57.75 °C	
		R113	$C_2F_3Cl_3$	Trichlorotrifluoroethane		*T*: –81 °C	
		R114	$C_2F_4Cl_2$	Dichlorotetrafluoroethane	N, F	NF, *T*: 3.3 °C	
		R502	$CHClF_2 \cdot C_2F_5Cl$			*T*: –45.0 °C NFL	Azeotrope
	Hydrochlorofluorocarbons (HCFC)	R22	$CHClF_2$	Chlorodifluoromethane	F, C	NF, Tx, *T*: –40.7 °C	
		R123	$C_2HF_3Cl_2$	2,2-Dichloro-1,1, 1-trifloroethane	AC	NF, *T*: 23.8 °C	RP: R114
		R124	C_2HF_4Cl	2-Chloro-1,1,2, 2-tetrafluoroethane	C, AC	NF, LTx, *T*: –11.0 °C	RP: R114
		R141	$C_2H_3FCl_2$	1,1-Dichloro- 1-fluoroethane		FL, *T*: 32 °C	RP: R114
		R142b	$C_2H_3F_2Cl$	1-Chloro1, 1-difluoroethane	C, AC	FL, *T*: –10.0 °C	

TFL temperature of inflammation, *NFL* not flammable, *FL* flammable *HTx* high toxicity, *T* temperature of evaporation, *C* cooling (–10/+40 °C), *F* freezing (–35/+40 °C), *AC* air conditioning (+6/+55 °C), *Ch* chilling >0 °C, *RP* replacement of "X" (e.g., (panel a) X = R114, (panel b) X = R12, (panel c) X = R502)

Table 9.3b Natural refrigerants

Category	Type	ASHRAE number	Molecular formula	Chemical name (IUPAC)	Application	Properties	Remarks
Natural refrigerants (up to 1992)	Natural	R717	NH_3	Ammonia	F, C, AC	TFL: 651 °C, HTx, *T*: −33.3 °C, FL	
		R600	C_4H_{10}	Butane	C, AC	TFL: 430 °C, *T*: −12 °C FL	PR: R12, R134a
		R170	C_2H_6	Ethane	F	TFL: 530 °C, *T*: −88.6 °C	
		R744	CO_2	Carbon dioxide	F, AC	NFL, *T*: −78.5 °C	
		R290	C_3H_8	Propane	F	TFL: 510 °C, *T*: −42.6 °C, FL	RP: R22, R404a, R407a
		R764	SO_2	Sculpture dioxide		NFL, HTx, NFL	Not anymore used
		R600/R290	Blend		C		

TFL temperature of inflammation, *NFL* not flammable, *FL* flammable *HTx* high toxicity, *T* temperature of evaporation, *C* cooling (–10/+40 °C), *F* freezing (–35/+40 °C), *AC* air conditioning (+6/+55 °C), *Ch* chilling >0 °C, *RP* replacement of “X” (e.g., (panel a) X = R114, (panel b) X = R12, (panel c) X = R502)

Table 9.3c Refrigerants after 1992

Category	Type	ASHRAE number	Molecular formula	Chemical name (IUPAC)	Application	Properties	Remarks
No chlorine-containing refrigerants (after 1992)	"Single" hydrofluoro-carbons (HFC)	R23	CHF_3	Trifluoromethane			RP: R13, R503
		R32	CH_2F_2	Difluoromethane	C, AC	FL	RP: 502
		R125	CHF_5	Pentafluoromethane	F	*T*: –48.5 °C (1 bar), NFL	RP: 502
		R134a	$C_2H_2F_4$	1.1.2.2-Tetrafluoroethane	C, F, AC	NFL	RP: R12
		R143a	$C_2H_3F_3$	1.1.1-Trifluoroethane	F	*T*: –47.4 °C (1 bar)	RP: 502
		R227	C_3HF_7	Heptafluoropropane	C, AC	*T*: –16.5 °C (1 bar), NFL	
		R152a	$C_2H_4F_2$	1.1-Difluoroethane		*T*: –24.2 °C (1 bar), FL	

TFL temperature of inflammation, *NFL* not flammable, *FL* flammable *HTx* high toxicity, *T* temperature of evaporation, *C* cooling (–10/+40 °C), *F* freezing (–35/+40 °C), *AC* air conditioning (+6/+55 °C), *Ch* chilling >0 °C, *RP* replacement of "X" (e.g., (panel a) X = R114, (panel b) X = R12, (panel c) X = R502)

Table 9.3d Composition of Blends

Category	Type	ASHRAE number	Molecular formula	Chemical name (IUPAC)	Application	Properties	Remarks
	Blends			Refrigerants in Blend			
		R404a	$C_2HF_5{\cdot}C_2H_3F_3{\cdot}C_2H_2F_4$	R125, R143, R134a	F, C	*T*: –46.5 °C, NFL	FX-70, HP-62
		R407a	$CH_2F_2{\cdot}C_2HF_5{\cdot}C_2H_2F_4$	R32, R125, R134a	F, C	NFL, *T*: –45.6 °C	Klea 60
		R410	$CH_2F_2{\cdot}C_2HF_5$	R32, R125	F, C	*T*: –52.7 °C, NFL	AZ20 (near azeotropic) Puron, Suva9100, RP: R22
		R413a	$C_3F_8{\cdot}C_2H_2F_4{\cdot}C_4H_{10}$	R218, R134a, R600a	F, C		Isceon 49, RR: R12
		R417a	$C_2HF_5{\cdot}C_2H_2F_4{\cdot}C_4H_{10}$	R125, R134a, R600a	F, C		Isceon 59, RP: R22
		R421a	$C_2HF_5{\cdot}C_2H_2F_4$	R125, R134a			
		R422d	$C_2HF_5{\cdot}C_2H_2F_4{\cdot}C_4H_{10}$	R125, R134a, R600a			
		R507a	$C_2HF_5{\cdot}C_2H_3F_3$	R125, R143a	F, C	*T*: 46.7 °C NFL	AZ50, RP: R502
		R508a	$CHF_3{\cdot}C_2F_6$	R23, R116	F		RP: R13/R503
		R508b	$CHF_3{\cdot}C_2F_6$	R23, R116	F		
		R510	$C_2H_6O{\cdot}C_4H_{10}$	R170, R600	C		

TFL temperature of inflammation, *NFL* not flammable, *FL* flammable *HTx* high toxicity, *T* temperature of evaporation, *C* cooling (–10/+40 °C), *F* freezing (–35/+40 °C), *AC* air conditioning (+6/+55 °C), *Ch* chilling >0 °C, *RP* replacement of "X" (e.g., (panel a) X = R114, (panel b) X = R12, (panel c) X = R502)

9.3.1.4 Chemical Requirements

- Nontoxic
- No flammability
- Nonexplosive
- No equipment corrosiveness
- Chemical stability (no decay of refrigerant even at extreme situations)
- Easily detected by smell
- No problems in lubrication (e.g., compatible with lubricants)

9.3.2 Natural Refrigerants

Natural refrigerants besides air (R718) and water (R719) are:

Ammonia: NH_3 (R717)
Carbon dioxide: CO_2 (R744)
Propane: C_3H_8 (R290)
Isobutane: C_4H_{10} (R600a)

In some cases, water and air are also considered for precooling or short-time chilling purposes of fresh vegetables and fruits. This is especially done when climatic conditions, such as air temperature and relative humidity, allow it.

Table 9.3b presents several natural refrigerants along with their applications and several basic properties of them. Natural refrigerants, such as R717 (NH_3), R600/R600a (butane/isobutane), propane, and CO_2, have been used before 1989, and some of them are still being used.

9.3.2.1 Ammonia (R717)

Ammonia R717 (NH_3) is a natural product with good thermodynamic properties in the most interesting temperature range for foods. It is the oldest refrigerant that is still quite extensively used, especially in larger units (e.g., cold stores). It is environmentally friendly and economical. Main negative characteristics of the refrigerant are its toxicity and relative corrosiveness. The gas of ammonia is very toxic. Air, containing >0.2 vol.% NH_3, results in mortality of 0.5–1.0 % in 60 min. However, due to its intensive odor, its presence, caused by leakages, is easily recognized and hazards can be avoided on time. It is corrosive to several metals (except iron and cast iron). Ammonia absorbs large quantities of water and this is the main reason of its corrosiveness. Ammonia is flammable, but the energy required for its inflammation is 50-fold of that required for natural gas. Food in cold stores that is soaked with ammonia becomes useless [Huette II, Taschenbuch Kaelteanlagen].

9.3.2.2 Propane (R290) and Isobutane (R600a)

Propane R290 (C_3H_8) and Isobutane R600a (C_4H_{10}) are hydrocarbons, which have good thermodynamic properties, but they are flammable. Therefore, they are not recommended for refrigeration of vehicles. High-grade propane can be used in low temperatures replacing R22. It has relatively good thermodynamic properties, but it is flammable. Therefore, it is not used in the refrigeration in vehicles. Hydrocarbons propane and isobutane are used in domestic refrigeration equipment. Here they are efficient and less noisy.

9.3.2.3 Carbon Dioxide

Carbon dioxide R744 (CO_2) has a similar long tradition in refrigeration technology as that of ammonia. It is not flammable, chemically stable, and effectively nontoxic. Breathing difficulty starts when the CO_2 content in air exceeds 2.5 vol.%. It can be used in the refrigeration in transport. Restrictions in applications are the high pressure in condensation and the relatively low pressure in evaporation. Due to good thermodynamic efficiency, the CO_2 compressors are not large, but the high pressure for condensation requires stronger construction, making the relevant equipment heavy. Carbon dioxide can be used in a cascade system. This overcomes the difficulties due to the very high-specific volume of ammonia vapor at temperatures below –35 °C. Such CO_2 systems are more efficient than the two-stage NH_3 systems for low temperatures of the system in the range of 40–55 °C.

9.3.3 Fluorocarbon and Blend Refrigerants

9.3.3.1 Fluorocarbons

The replacement of the chlorine-containing refrigerants was done for CFC (R12) gradually, starting with the restriction of its availability. This occurred in 1995 in Europe and 1 year later globally. The use of HCFC R22 in new equipment was banned in Europe up to 2000. It should be gradually totally removed up to about 2015 in Europe and up to 2020 in the USA.

The replacers of chlorine-containing refrigerants belonged to three categories:

1. The enforced usage of already existing natural refrigerants
2. The development of the hydrofluorocarbons (HFC)
3. The use of certain azeotropic fluids and blends of already permitted refrigerants

In all cases, the basic effort was to approximate the operational conditions of the removed CFC and HCFC refrigerants. Typical HFC refrigerants were R23, R32, and R125.

Table 9.3c includes refrigerants that are in use after 1989. Two subcategories of the table are (a) the "single" HFC refrigerants and (b) the blends. Typical *single refrigerants* are the R23, the R32, and the R134a. The last one was developed as a replacer of R12. However, as all the HFC refrigerants, R134a is not miscible with mineral oils. Furthermore, its efficiency is relatively low, as its energy consumption is high. It is mainly used in domestic and automotive refrigeration. But since the product is an inert gas, besides refrigeration, it is also used in other applications such as in plastic foam blowing, as propellant of materials in the pharmaceutical and cosmetic industry, in air-drying (moisture removing from compressed air), and as solvent in organic chemistry.

Moisture in refrigeration systems plays a part in corrosion processes and in case of hermetic compressor systems in the degradation of wire coatings. High water concentration enforces hydrolysis. Hydrolysis, on the other hand, increases with temperature and impurities, which may act as catalysts. Water that freezes out may block expansion devices and stick several valves. Finally, excess of water in the circulated refrigerant degrades its quality by thinning. Therefore, the moisture level in the refrigerant should be carefully controlled.

9.3.3.2 Blends

The blend*s* are a mixture of non-chlorine-containing refrigerants. The blends, R407 and especially R410a, are most common in domestic refrigeration, including air conditioning. R410a is an almost azeotropic product (AZ20). It is a high-pressure refrigerant with good per unit volume capacity. This results in the possibility to construct cheaper compressors. However, its rather high GWP is not satisfactory.

Table 9.4 indicates the basic combinations of blends. With the exception of R410a and R421, which are blends of two single compounds, all other are combinations of three compounds. Blends are usually mixture of R32, R125, and R134a. In some cases, R143a and R600/600a are part of mixtures as well. In a blend, R32 provides the heat, R134a reduces the pressure, and R125 reduces the flammability.

Table 9.4 Composition of blends

	Single hydrofluorocarbons (HFC)					Hydrocarbons (HC)		
Blends	R32	R125	R134a	R143	R218	R600	R600a	Number of single compounds pro-blend
R404a		X	X	X				3
R407a	X	X	X					3
R410 (AZ 20)	X	X						2
R413			X		X		X	3
R417		X	X			X		3
R421		X	X					2
R422		X	X				X	3

Table 9.5 Replacers of CFC and HCFC refrigerants and lubricants for replacers

CFC, HCFC refrigerant	Replacer	Lubricant	Application
R22	R407a	POE	Refrigeration
	R407c	POE	Refrigeration/AC
	R410a	MO or AB, POE	Refrigeration
	R417a	MO or AB, POE	Refrigeration
	R422d	MO or AB, POE	Refrigeration
	R438a	MO or AB, POE	Refrigeration/AC
R12	R423a	MO or AB, POE	Chillers
	R437a	MO or AB, POE	Refrigeration
	R123	POE	Chillers
	R134a	POE/PAG(*)	Refrigeration/chillers (*)AC—cars
R502	R404a	POE	Freezing
	R422a	MO or AB, POE	Freezing
R503	R508b	POE	Freezing
R13	R23	POE	Freezing

AB alkylbenzene, *POE* polyol ester, *MO* mineral oil, *AC* air conditioning
[Ref.: DuPoint Refrigerants. US General Replacement Guide]
*It indicates that R134a, may be also used in air conditioning (AC) of cars

Table 9.5 gives the replacers of the basic CFC and HCFC refrigerants that were used up to 1992. For R22, important replacers are R410a and R407 and 407a/c. For freezing, important replacers of R502 are R404a and R422d. For lower refrigeration capacities (chilling and automotive use), important replacers seem to be R134a and R423a.

9.4 Lubricants

9.4.1 Main Types of Lubricants

The introduction of new refrigerants in compression refrigeration resulted in changes in the application of lubricants. Two of the major categories of lubricants used in compression refrigeration are (a) mineral oils (MO) and (b) synthetics. The mineral oils were used with CFC and HCFC refrigerants, and they are still used in ammonia and HC. The synthetic lubricants were developed to match the requirements of the new refrigerants after banning chlorine in refrigeration liquids. Such lubricants were (a) alkylbenzenes (AB), (b) polyalkylene glycol (PAG), and (c) polyol esters (POE).

9.4.2 Function of Lubrication

Lubrication is very important in the compression refrigeration systems. Its application is quite versatile. The following are indicated contributions of lubricants in refrigeration systems:

1. Helping in removing heat excess
2. Sealing of unintentional gaps
3. Keeping the refrigeration system clean
4. Increasing the efficiency of the compressor
5. Reducing foams and noise
6. Reducing moisture in the system

High pressure in the refrigeration system and friction due to moving parts are two sources of heat generation. Lubricants come in contact with heated surfaces removing their heat excess and reducing friction, increasing the efficiency of the compressor system. At the same time, lubricants mainly of large molecular value seal fine gaps in leakages of the refrigeration system. The sealing effectiveness of a lubricant depends on factors such as the density and the viscosity of the lubricant and the pressure in the refrigeration system.

Lubricants are trapped in refrigerants in some extent. Although usually some kind of filtration of the refrigerant exists in refrigeration systems, the "mixed lubricants" (refrigerants with lubricants) assist in the transport of solid contaminants to the filter for their subsequent discharge. This is especially worthwhile when, e.g., some waxes in fluids or solids, such as fine residuals of the equipment manufacturing or the assembling process or dust, are present in the refrigerant–lubricant mixture. Furthermore, foaming must be managed, because besides its contribution to the reduction of the efficiency of the refrigeration system, it also increases noise during the operation of the compressor. Some lubricants may influence the moisture content of the refrigerant–lubricant mixture. This is the case when the lubricants are hygroscopic.

9.4.3 Requirements for Good Lubrication

The lubricants are mainly concentrated in the crankcase of compressors, or when being mixed with a refrigerant, they circulate all over the refrigeration system. The lubricant dissolved in any refrigerant depends on:

1. The pressure of the refrigerant vapor
2. The temperature of the lubricating oil
3. The length of time that the lubricant remains in contact with the refrigerant
4. The degree of lubricant miscibility of the refrigerant

The requirements of a good lubrication are:

1. Adequate miscibility of refrigerant and lubricant
2. Chemical stability
3. Physical stability
4. Small quantities in condensers and evaporators of the refrigeration system
5. Low wax content
6. High dielectric strength

The quantity of entrapped lubricants in the circulated refrigerant usually may be about 5 %. However, more lubricant may be present if not proper measures are taken. Such measures can be the installation of oil separators and care for enough low viscosity and increased miscibility.

The *miscibility* of lubricant oil with a refrigerant has some advantages if the right lubricant is used. In this case, the circulated oil returns to the compressor easily and lubricates even parts that cannot be otherwise easily reached. However, especially in the evaporator (cold side), the viscosity of lubricant in the refrigerant–lubricant mixture may increase, making difficult the return of the initial oil quantity back to the compressor. In such a case, besides the reduction of oil quantity in the crankcase, it influences negatively the heat transfer of the evaporator, as oil forms a film covering the inside part of the evaporator's surface.

The lubricant should neither react chemically with the refrigerant nor with the several parts of equipment or residuals in the refrigeration cycle. This is especially important in the high-pressure side of the system, in which the temperatures are relatively high, supporting chemical reactions. The lubricants should not be influenced by a probable presence of solid residuals in the solvent (refrigerant) which may act catalytically, enforcing high-temperature chemical reactions. The lubricant should have good flow ability in a broad range of temperatures. It should maintain the right viscosity at high as well as low temperatures. Therefore, generally, a low pour point of a lubricant is desirable.

Attention is also required to choose oil that is not degraded at relatively high compression temperatures. Such temperatures are usually above 100 °C, facilitating chemical reactions leading in the breakdown of oil in the refrigerant and eventually corrosion of several parts. The temperatures at which oil breakdown starts vary according to the type of the lubricant used. For *mineral oils* (MO) it is about 180 °C. For *alkylbenzenes* (AB) it is about 200 °C and for *polyol esters* (POE) it is about 250 °C. These data are valid for an environment without contaminants.

Even in relatively good miscibility of a lubricant with the refrigerant, small quantities of not mixed lubricant oils may be further present in the refrigeration cycle. This may cause problems in the heat exchangers (condenser, evaporator), the valves, and even some pipes after long operation. Therefore, it is desirable that the separated quantity of the selected lubricants remains low in the high as well as the low pressure of the refrigeration system.

In case of hermetic systems with rotary compressors, the dielectric properties of the refrigerant–lubricant mixture must ensure good insulating properties, as the mixture acts as insulation between the body of the compression unit and the motor.

9.4.4 Choice of Refrigerant Lubricants

For meeting the requirements of the new refrigerants, the selection of the proper lubricant is important. This is due to the fact that their quality may vary still for lubricants of the same type. Thus, since even minor alternations of the refrigerants may influence the efficiency of equipment, it is advisable to choose lubricants that follow the manufacturer's recommendations.

The mineral oil (MO) was used with CFC and HCFC refrigerants. It was and continues being used with natural refrigerants. The addition of hydrocarbons (HC) in the MO may help in thinning it, resulting to easier refrigerant–lubricant circulation. The MO cannot be used with HFC refrigerants as they are not mixable with them, forming a separate layer in the fluidized refrigerant. Nevertheless, in some cases, blends such as R410a, R422a, R422a, and R438 may alternatively use MO instead of POE lubricants. However, in a few cases of retrofits of older installations, POE lubricants that may be mixed with MO can be added in an already existing refrigeration system. But even in such a case, the modified lubricant does not seem to help its solubility in the already existing HFC refrigerant.

The synthetic polyalkylene glycol (PAG) lubricant was designed specially for meeting the requirements of R134a. It is a high-viscosity refrigeration lubricant that is well miscible with it, providing superior lubricity. Its pour point (the lowest temperature at which a fluid becomes semifluid losing its flow characteristics) is 44 °C. Generally, PAG is not compatible with mineral oils. It does not tolerate chlorine and is a nontoxic, fire resistant, and electrically insulating material. Its primary application is automotive air conditioning.

Probably the presently widest used lubricant along with HFC refrigerants is the polyol ester (POE). As indicated in Table 9.5, it is recommended in the lubrication of all systems containing refrigerants that replace CFC and HCFC refrigerants. POE has an excellent mixing with HFC, and it is also compatible with many lubricants in the market. POE has a very good viscosity in a broad range of temperatures. They can be used in quite low temperatures, because they keep their flowability, as they do not contain wax. However, its viscosity increases with temperature and it is, as PAG, very hydroscopic. This requires that the moisture content of the refrigerant-containing POE lubrication must be steadily controlled. Moisture may invade in the refrigerant by increased moisture in lubricants and improper handling of refrigerants or hydroscopic lubricants in assembling or retrofit operations or due to incorrect function of installed driers. High moisture can increase hydrolysis.

In this case that water reacts with esters, forming organic acids and alcohol. This process is influenced by temperature and acid value. Acids finally act, like impurities in the recalculated system, as catalysts of not desirable chemical reactions.

Finally, in the case of retrofits, it is indispensable that at least 95 % of the MO must be removed and the old installation must be well evacuated and special filter driers must be also used. In operation, the water in the refrigerants that contain POE lubricants is not influenced significantly if the moisture of the POE lubrication is below its saturation limits (i.e., $<$ 3000 ppm). In this case, there is no free water, and ice crystals are unlikely to be formed.

9.4.5 Additives

Besides lubrication, additives put into the refrigeration system act as a complementary antiwear protection, protection against foaming, protection against factors related with chemical reactions, etc. The use of additives must be applied after testing the results of their probable activity. These substances should be well miscible with refrigerants, they should not influence negatively the dryer filters, they should not cause deposits (e.g., in valves), they should not be influenced by temperature changes, and they should not form negative chemical reactions with existing refrigerants and lubricants. Additives with copper-based nanoparticles may boost the heat transfer properties of a refrigerant. The presence of chlorine in the old R12 refrigerant formed protective films of metal chlorides against corrosion. Additives based on sulfur and phosphorus chemistry may improve the anti-drag function of a lubricant, and some additives are active in reduction of foams.

9.5 Cooling of Foods

9.5.1 Chilling

The chilling (cooling) of foods at temperatures close to 0 °C is applied to extend the shelf life of "fresh" products, i.e., products immediately after harvesting or processing. Chilling covers all foods: fruits and vegetables, meat, fish, dairy products, cereals, etc., and complex food composed of all these, such as ready meals. In chilling, two main tasks are (a) the fast reduction of the initial temperature of the product down to the desired low temperature and (b) maintenance of the final temperature over a longer period. The fast reduction of the temperature is achieved by cooling equipment, in connection to some processing operation or in connection to storage. The analysis and design of refrigeration processes for foods is discussed by Cleland (1990).

The maintenance at a constant low temperature over a longer period is part of storage technology (see Sect. 9.5 of this chapter). In both cases, the refrigeration load, i.e., the sensible heat ($C_p\ \Delta T$) that must be removed, is important. In chilling, the specific heat of food above the freezing point is important. This is a function of the temperature of the product, which, with the exception of most fat, increases linearly with the temperature above 0 °C. Table 9.6 gives characteristic average values for the specific heat of foods at the temperature region above 0 °C. For most fat, the specific heat is almost constant at temperatures above 35–40 °C. Exceptions are sunflower oil, olive oil, and peanut oil, whose specific heat is almost constant at temperatures about 10–20 °C.

The specific heat C_p, for a temperature range (ΔT), can be also estimated, by using the equation $C_p = \Delta H/\Delta T$. The enthalpy change (ΔH) for the temperature range ΔT can be estimated by using the Riedel diagrams. There are three types of

Table 9.6 Indicative values of the specific heat (C_p) of food

Product	C_p (kJ/kg K)	Temperature (°C)
Water-containing food	3.5–3.9	$T > 0$
Water-containing food	1.8–1.9	$T < 0$
Dry food	1.3–2.1	$T > 0$ and $T < 0$
Fat	1.7–2.2	$T > 40$
Fat	1.5	$T < 0$

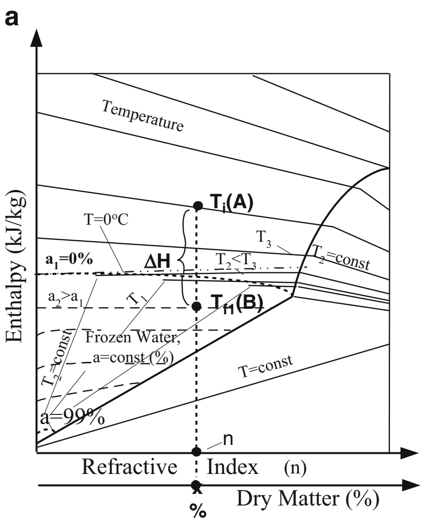


Fig. 9.16 Enthalpy–water content diagrams of Riedel

such diagrams. Two types (Fig. 9.16) give the enthalpy of several foods in relation to their temperature and water content. These diagrams also indicate the portion of food water that is crystallized, if the product is frozen. The third type of diagrams gives the enthalpy and the specific heat of fats and oils. They also indicate the portion of molten fats in relation to the temperature of the products. Figure 9.16a is used for fruits and vegetables (cases 1 and 2 in Table 9.7), while Fig. 9.16b is for all other cases in Table 9.7. As indicated in Fig. 9.16a, b, for cooling and freezing certain foods containing × % dry substance (by weight) or ξ_a kg water/kg product, from an initial temperature T_i (Point A) to a final temperature T_{f1} (Point B), the heat ΔH (kJ/kg) must be removed.

As indicated in Table 9.7 (extended application), these diagrams may be also used for other similar products. Such a case, e.g., is the use of the diagram that has been developed for juices, for estimating the specific heat and enthalpy of whole fruits and vegetables. In this case, if the dry matter of a fruit or vegetable is known, the specific heat is calculated by inserting the values of the Riedel diagram in the equation of the "mixing rule" (9.5), which relates the specific heat of a product (C_{Pr}) to the specific heats of its constituents (average values).

Table 9.7 Riedel diagrams for estimating the specific heat, the heat load of cooling/freezing of food, and the fraction of frozen water of food

	Product	Extended application	Reference
1	Fruit and vegetable juice	Fruits and vegetables	Riedel (1950a, b); DKV (8-02)
2	Sugar solutions		Riedel (1950a); DKV (8-06)
3	Fats and oils	Meat fat	Riedel (1955); DKV (8-10)
4	Lean beef	All other lean meats (fat less than 4 %)	Riedel (1957a, b, c); DKV (8-11)
5	Egg (albumen)		Riedel (1956a, b); DKV (8-12)
6	Egg (yolk)		Riedel (1957a); DKV (8-13)
7	Egg (whole)		Riedel (1957b); DKV (8-14)
8	Bread (white)	Wheat and rice starch Fat-free bakery products	Riedel (1959a, b); DKV (8-15)
9	Fish (lean)		Riedel (1956a); DKV (8-18)
10	Starch (potato)		Riedel (1959a); DKV (8-19)
11	Baking yeast		Riedel (1968); DKV (8-25)

$$C_{\mathrm{Pr}} = \xi_{\mathrm{w}} C_{\mathrm{w}} + \xi_{\mathrm{p}} C_{\mathrm{p}} + \xi_{\mathrm{c}} C_{\mathrm{c}} + \xi_{\mathrm{f}} C_{\mathrm{f}} + \xi_{\mathrm{s}} C_{\mathrm{s}} \tag{9.5}$$

where

C_{w} is specific heat of water (4.16 kJ/kg K)
C_{p} is specific heat of protein (1.55 kJ/kg K)
C_{c} is specific heat of carbohydrates (1.42 kJ/kg K)
C_{f} is specific heat of fat (1.70 kJ/kg K)
C_{s} is specific heat of salts (0.84 kJ/kg
ξ_{w}, ξ_{p}, ξ_{c}, ξ_{f}, ξ_{s} are content of water, proteins, carbohydrate, fat and salt, respectively (mass fractions).

For the calculation of ΔH for fruits and vegetables, (9.6) is used

$$\Delta H = (1 - \xi_{\mathrm{dm}})\Delta H_{\mathrm{J}} + C_{\mathrm{dm}} \cdot \xi_{\mathrm{dm}} \cdot \Delta T = \Delta H_{\mathrm{J}} - (\Delta H_{\mathrm{J}} - C_{\mathrm{dm}} \cdot \Delta T)\xi_{\mathrm{dm}} (\mathrm{kJ/kg}) \tag{9.6}$$

where

C_{dm} is specific heat of dry mater (1.21 kJ/kg °C)
ξ_{dm} is content of dry matter (mass fraction)
ΔH_{J} is enthalpy difference for temperature change ΔT of the juice (taken from the Riedel diagram).

In the same way, the specific heat of pork meat can be calculated if its fat content is known, by using the Riedel diagrams for fat (pork fat) and lean beef and the "mixing rule."

9.5.2 Cooling Equipment

Refrigeration equipment, used in processing and preservation of food, may be classified into two main categories, (a) the cooling and (b) the freezing equipment. In cooling of food, the low temperatures applied lie between about 13 °C and their freezing point (Schormueller 1966). The cooling methods can be direct evaporation and direct or indirect heat exchange. Furthermore, distinction is made among methods applied to solids or to liquids and batch and continuous operation. In freezing, two main categories are (a) application of temperatures between the freezing point and –40 °C and (b) application of temperatures below –40 °C.

9.5.2.1 Cooling of Solids

Cooling of solids can be carried out by hydrocooling, contacting cold surfaces, direct evaporation of surface water, and air. In some cases, the equipment used is quite similar to that applied in heating processes for solids (e.g., drying).

Hydrocooling

Hydrocooling is usually a continuous method applied to cooling of solids. It is used often in precooling of vegetables and some fruits (e.g., oranges and peaches). However, it is also applied in precooling of other products (e.g., meat). In hydrocooling, cold water is used at temperatures between 15 and 0 °C. For achieving temperatures around 0 °C, ice can be also mixed with water. A relative motion between product and cooling water is required for increasing the efficiency of cooling. Relative motion increases cooling two- to fourfold (de Fremery et al. 1977). The circulation of the water or the movement of the product can achieve this relative motion. Since precooling is a preliminary stage of further processing, especially in many cases of fruits and vegetables, it can be combined with washing or short-term storage in water basins (e.g., tomato), which are also preliminary operations. Often, the water contains a mild disinfectant, such as chlorine (5 ppm), or an approved phenolic compound (Fennema 1975).

Figure 9.17 shows several hydrocooling methods. These can be classified into methods in which the product is immersed in a bath, (a), (b), and (c), or the product is sprayed, (d), (e), and (f), and to combined methods, in which the product is immersed and sprayed, (g) and (h). The method in Fig. 9.17a is continuous and can be used for sensitive products. The products, which are put in hanging meshed

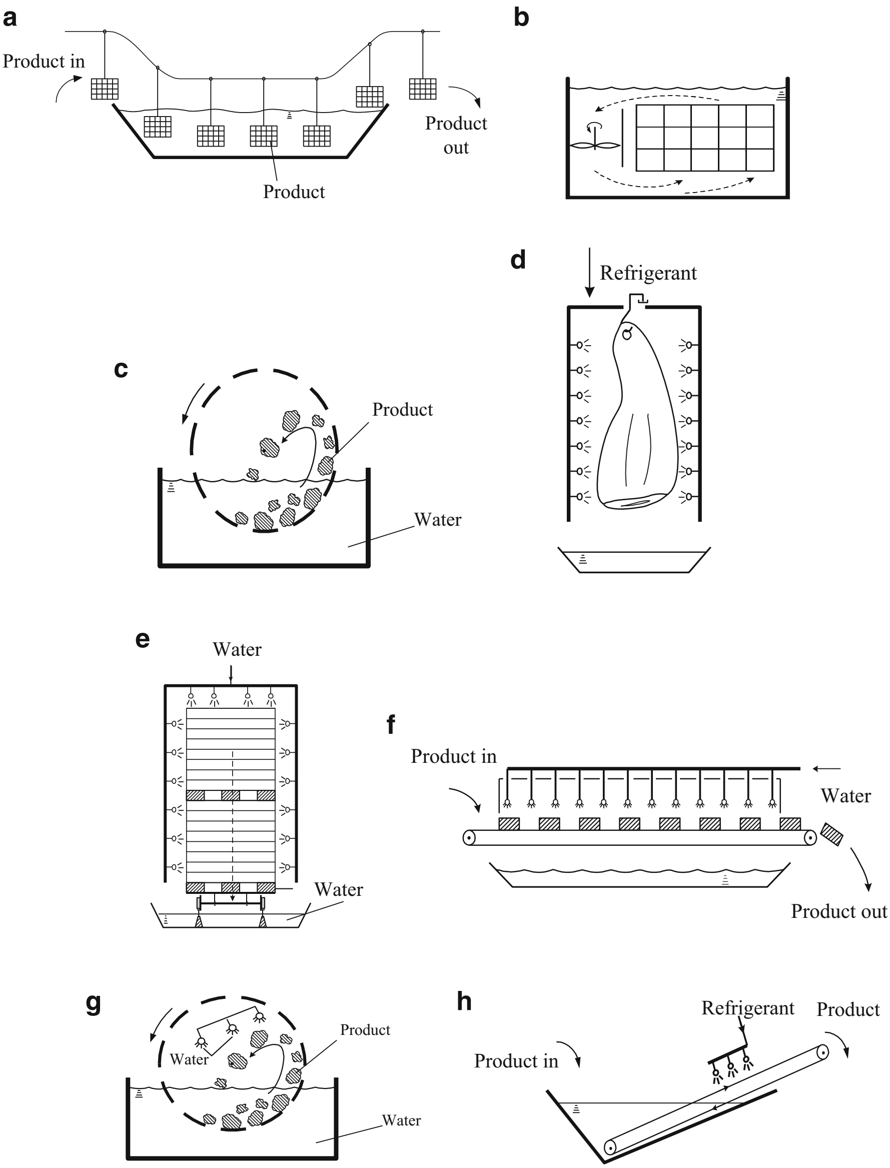


Fig. 9.17 Hydrocooling methods (see text)

boxes, are transported along a cool bath. In the equipment of the method in Fig. 9.17b, the product is immersed in a bath, while water circulates through the product. In the method in Fig. 9.17c, the relative motion between the product and the cool water is achieved by the rotation of a drum, which is partially immersed in cool water. Inside of the perforated drum, there are helical baffles, forwarding the product during the drum rotation. At the same time, the product is lifted by the rotating drum up a certain height, and it falls back to the water (tumbling). This type of equipment is often used for leafy vegetables (e.g., spinach). The method can be continuous or batch (if there are no screw baffles in the drum). This kind of equipment is also preferably used in chilling of poultry because, in comparison to air cooling, less weight is lost. For poultry chilling, 1–3 drums may be used (the two or three drums, in series), in which poultry rotates in counterflowing ice–water mixture, for 20–30 min. The consumption of ice is 0.6–0.9 kg/kg poultry. Such equipment requires a surface area of 18–20 m^2. The capacity of such drums is up to 6000 birds/h (de Fremery et al. 1977). Poultry, immersed in water for washing, precooling, and chilling, gains significant amount of water, especially in its skin and in its fatty tissue (about 8 %). USDA established tolerances, and an EU legislation (Regulation 2967/76) restrict the maximum water gain (James and Bailey 1990a, b).

The equipment in Fig. 9.17d, e is used for cooling meat. The method is called evaporative air chilling or spray chilling. The water activity of the surface of the product is maintained very high by wetting it with water sprayed through nozzles on both sides of the product. At the same time, cool air is blown on the food. In poultry, the carcasses can be sprayed with cold water (about 5 °C), 7–8 times in short burst intervals for about 1 h. This, together with cool air ($T < 8$ °C), reduces the temperature of the carcass from about 30 °C down to 4 °C. The quantity of water used is 0.5 L/carcass (Mulder and Veerkamp 1990). This method was also tried in chilling pork and beef carcasses. The chilling of beef lasted more than 8 h (James and Bailey 1990a). Water of 2–3 °C was sprayed repeatedly from 11 nozzles. Each time, 1 L/nozzle was sprayed for 30 s (James and Bailey 1990b). In the case of fruits (Fig. 9.16e), up to stacked three pallet boxes move on rails through a tunnel, in which nozzles spray cold water, containing a protective fungicide. The capacity of a unit of 12 × 2 × 4 m, cooling apples with water of about 7 °C, can be 200–220 pallet boxes/h. The required energy for the pumping system is 8–12 kW. The equipment in Fig. 9.16f can be used as part of immediate further processing of products that are sensitive to mechanical stresses. In comparison to the immersion method, advantages of spray chilling are the reduced water consumption and water product absorption. Disadvantages are the inducement of microbial growth, due to high water concentration on the surface of the carcasses and the relatively higher total energy requirement. The equipment in Fig. 9.17 h is similar to that of Fig. 9.17g. They differ in the additional spraying, occurring during the rotation. These machines are combination equipment, in which the product is immersed in cool water and sprayed. They are mainly used for fruits and vegetables. Water is cooled in a heat exchanger and recirculated.

Vacuum Cooling

Vacuum cooling can be preferably used when the ratio of product surface to its mass is large. This is, e.g., the case of several leafy vegetables. Furthermore, the method presumes that there is no great resistance to water removal from the interior of the product to its surface, which can happen, e.g., when a waxy layer covers the surface of the product. Vacuum cooling is a batch method. The product is put in pallets placed on wagons or in mobile trays on rails, and it is transported into the vacuum chamber, in which the pressure is reduced to 5.3–6.5 mbar (4.0–5.0 mmHg) (James and Bailey 1990a, b). In some vegetables, pre-spraying of the product with water contributes to a better cooling. The removal of 1 % water reduces the product temperature by 5 °C (Fennema 1975). In cooling of several vegetables, this kind of cooling is economically comparable with hydrocooling, and it gives better-quality results. However, as in all methods requiring vacuum, relative high capital investment is required. A chamber of $5.5 \times 1.8 \times 2.0$ m can cool 5 tons/h (five pallets, 1.20×1.00 m), using two vacuum pumps. The weight of such a chamber is about 12 tons, and the electrical power required, 40 kW.

Surface Contact Cooling

Basically, equipment that is used for heating can be modified for cooling. In the case of the double-wall surfaces, cool water or brine is circulated, instead of hot water or steam. Surface equipment can be used in cooling of solids in the form of granulates. The process can be batch or continuous. An example of using such a double-wall machine in a batch process is the vat with a rotating helical stirring device, used in crystallization (Fig. 9.13c). The cooling medium circulates in the double wall, but there are variations in which, for increasing the heat exchange, the helical stirring device is a tube in which the cooling medium also circulates. A modification of the equipment used in drying can be also used in the continuous cooling of granulates (Fig. 9.13b). In this case, several double-wall disks, lying one over the other, cool the granules as they are swept by stirring devices or brushes, falling from one disk to the next, after an almost full rotation.

Tunnel Cooling

Large food pieces can be chilled in heat-insulated tunnels, although, when storage follows, chilling is usually done in the rooms in which the products will be ultimately stored. Air cooling in tunnels does not require complicated installations and has also the advantage of cooling products of different size. The product comes into the tunnel in trolleys or racks (Fig. 9.18). Air, cooled in a finned-tube evaporator (Fig. 9.6), is recirculated after being blown to the product. The cooling time is reduced when low temperature and high air velocity are applied. Nevertheless, high air velocity increases the water loss of the product, and low temperatures, especially in combination with high air velocity, increase the possibility of surface or part-product freezing. Furthermore, increase of ventilation raises the required fan energy significantly. As James and Bailey (1990a) report, in cooling meat (140 kg beef), a fourfold increase of air velocity (from 0.5 to 2.0 m/s) results in a cooling time

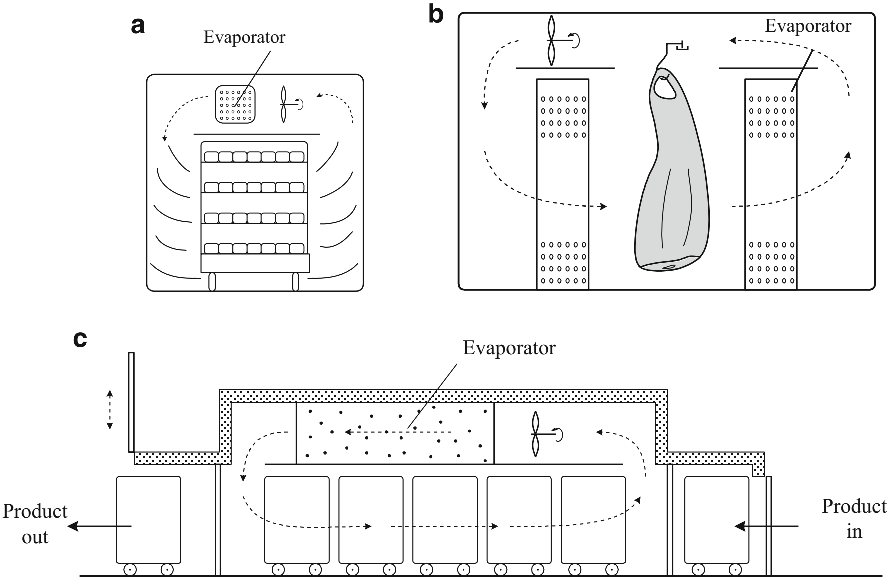


Fig. 9.18 Low-temperature tunnels (see text)

reduction of 18 %, but it also increases the energy consumption 64 times. In cooling of larger food pieces (e.g., carcasses), air temperature near the freezing point of food and air velocity around 1.0 m/s are used. In the batch system, the product does not move as long as it is in the tunnel, but in the continuous system, the trolleys enter and leave the tunnel from different doors (Fig. 9.18). In the continuous process, air is blown in counterflow to the movement of the trolley in the tunnel. The air cooler can be on the upper part or on the side of the tunnel.

9.5.2.2 Cooling of Liquids

Methods of liquid cooling may be batch or continuous. The equipment effectively consists of a heat exchanger, like those described in Chaps. 6 (heat transfer equipment) and 7 (evaporators) and in this chapter, and a pumping system. Basic equipment for liquid cooling is the plate heat exchanger, the scraped surface equipment, the double-wall vessel, the shell-and-coil equipment, the vacuum cooling equipment, and the equipment combining cooling with mixing.

Plate Heat Exchangers

The plate heat exchangers are double-wall constructions (Fig. 9.7) and are described in Chap. 6 (Figs. 6.4 and 6.5). This kind of equipment is, e.g., used in cooling of milk or juice continuously before storing in isothermal tanks. The same equipment is also used for heating and subsequent cooling of liquids (e.g., pasteurization). It is very popular, because it has good heat transfer properties, it is flexible (its surface can be increased, if required), and it can be cleaned easily. Since the distance between the plates is small (only a few millimeters), the density and the viscosity of the liquids must be low. Often, the cooling medium is cold (iced) water, which moves in counterflow to the product. The maximum capacity of such equipment depends on the number of plates, the pump system used, the physical properties of the product (thermal conductivity, viscosity), and the final temperature of the product.

Scraped Surface Exchangers

This type of equipment is used in cooling highly viscous liquids continuously. In vertical position, they are applied to manufacturing of ice (flakes). Horizontal equipment is used in manufacturing of ice cream. This heat exchanger is described in Chap. 6 (Fig. 6.9).

Jacketed Vessels

The vessels and the agitated kettles are double-wall equipment described in Chaps. 3 and 6. The cooling medium flows between the surfaces, while several types of stirring devices agitate the product. At constant speed and constant initial temperature of the cooling medium, the heat transfer of the vessel/kettle, besides the kind of the product, depends on the agitation.

Shell-and-Coil Equipment

The basic element of this equipment is the shell-and-coil heat exchanger. This heat exchanger is described in this chapter (Sect. 9.2.3, Fig. 9.12a). The continuously operating equipment is mainly used for cooling water and drinks quickly.

Vacuum Cooling Equipment

The cooling time of liquids in agitated jacketed vessels is reduced, if vacuum is applied due to additional evaporative cooling. The method consists of a closed jacketed vessel that can be vacuumed (pressure 6 mbar). When vacuum is applied, the liquid of the vessel can be cooled from 90 °C to ambient temperature ten times faster.

Combined Cooling with Mixing

Recirculation is one of the main methods of mixing. If this method is combined with cooling, it can be applied to flowable products, such as liquids and grains. The product comes out of the tank or silo gradually, and it flows back to them, after cooling in an efficient heat exchanger. This method can also be applied to food suspensions made up of liquid and solid components. The cooling of food is difficult when it is made up of a liquid (e.g., soup), in which fatty meat pieces are included. In this case, the much lower thermal conductivity of the solid pieces determines the cooling rate of the whole food. A possibility in this case could be the cooling of the two main components separately, before they are mixed in the vessel (James and Bailey 1990a).

9.6 Freezing of Food

9.6.1 Freezing

Freezing of food is a preservation method, combining long shelf life with good product quality. The products must be frozen as soon as possible after processing and maintained frozen up to thawing/consumption. Several authors have analyzed the freezing process and its effects on the final quality of food. Freezing damage of food is due to four reasons (Reid 1996): the chill damage, the solute-concentration damage, the dehydration occurring due to osmotic forces, and the mechanical damage from ice crystals. The mechanism of ice crystal formation and changes occurring during thawing were described by Fennema (1975), Fellows (1992), Heiss and Eichner (1995), and Schormueller (1966). The role of freezing rate on the quality of frozen food was analyzed by IIF (1975), Mardsen and Henrickson (1996), and Woolrich and Novak (1977). The effect of food properties on the freezing process was discussed by Heldmann (1992).

The quality of frozen foods depends on the: (1) good initial product quality; (2) correct application of additional processing methods, when required; (3) right freezing speed; (4) hygienic processing conditions; and (5) proper thawing.

Since freezing is a relatively expensive process and it neither kills microorganisms nor inactivates enzymes, it is very important to select products of very good initial quality. Furthermore, the hygienic conditions of food preparation and freezing must be very good to avoid any infection or contamination before freezing. Mechanical or thermal damage during the preparation for freezing must be avoided, because they result in the loss of valuable food components, increase the danger of post-contamination, and cause degradation of the quality of the final products. If, e.g., additional thermal processing, like blanching in the case of vegetables, is required, this should be effective but as minimum as possible, avoiding damage to

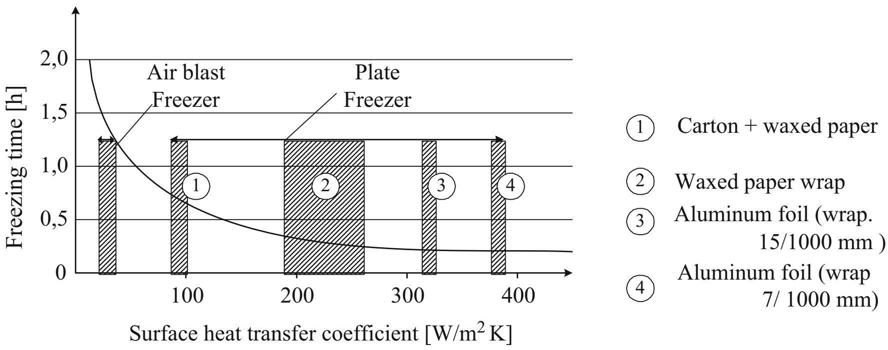
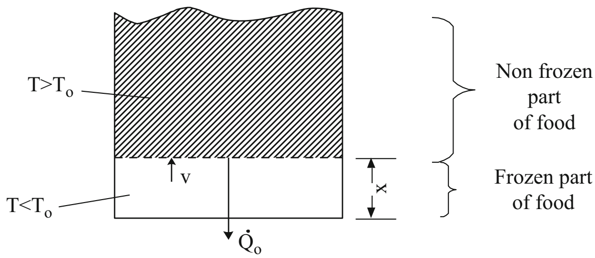


Fig. 9.19 Influence of packaging on the surface heat transfer coefficient

Fig. 9.20 Speed of freezing



the initial high quality of the product (see p. 467). Packaging protects food from any post-contamination and reduces the weight loss during freezing or storage. However, although packaging of food before freezing contributes to the good final quality of the product, it has the disadvantage of retarding the freezing speed. As indicated in Fig. 9.19, the surface heat transfer coefficient of fine aluminum wrapping is four times higher than that of carton wrapping covered with wax (IIF 1972).

In comparing freezing methods and freezing of foods, the freezing speed (Fig. 9.20) is helpful. It gives the time that is required for the "cold front," i.e., the borderline between frozen and unfrozen part of food, to move toward the side of the nonfrozen food (9.7). According to the recommendation of the International Institute of Refrigeration, the freezing speed (u) is distinguished in four categories: the slow freezing, $u = 0.1$–0.2 cm/h (bulk freezing in cold store rooms); the quick freezing, $u = 0.5$–3 cm/h (blast freezing or plate freezing); the rapid freezing, $u = 5$–10 cm/h (individual freezing of small pieces of food in fluidized freezers); and the ultra rapid freezing, $u = 10$–100 cm/h (spraying with liquids or cryogenic freezing). For retail packages, freezing speeds $u > 0.5$ cm/h and, for individual freezing, freezing speeds $u > 5$ cm/h are considered satisfactory (IIF 1972):

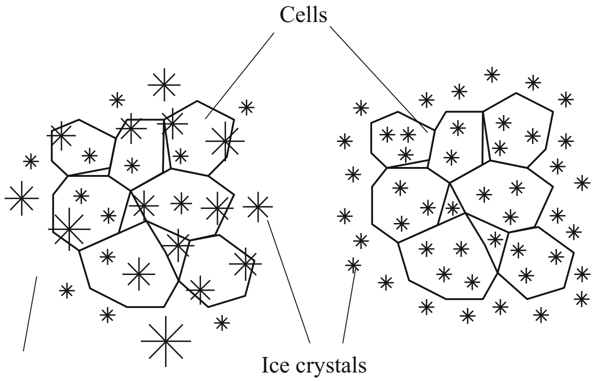


Fig. 9.21 Ice crystals in freezing of food cells

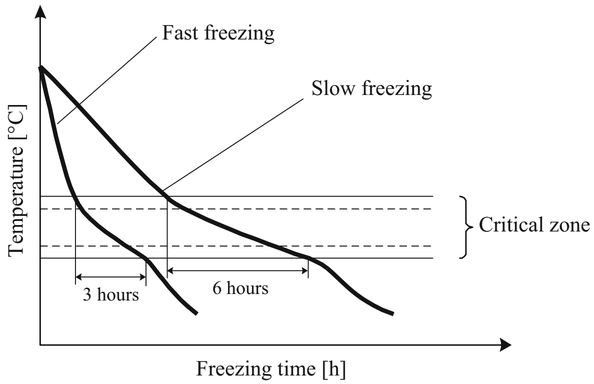


Fig. 9.22 Zone of crystallization in food freezing

$$u = x/t \tag{9.7}$$

where u is freezing speed (cm/h); x, distance from the surface contacting the freezing medium (cm); and t, freezing time (h).

Generally, freezing must be done quickly, resulting in the production of small ice crystals, which do not damage the cells of the products very much (Fig. 9.21). The large ice crystals damage the cell walls and valuable substances are lost during thawing (Heiss and Eichner 1995). For reducing the number of large ice crystals in the food, the product must not remain for a long time in the critical temperature zone (Fig. 9.22), in which crystallization takes place (Heiss and Eichner 1995; Fellows 1990). Thus, the number and size of crystals formed depend on several factors, influencing the heat transfer. Such factors are the mass of the product that must be frozen, the method of freezing, the freezing conditions, and the packaging. However, the quality of the food does not always depend on the high speed of freezing. In the case of beef, there is no remarkable difference between meat frozen at freezing speeds 0.03 and 200 cm/h (Spiess and Kostaropoulos 1977).

Thawing plays also an important role in the final quality of food. When using conventional methods (hot air, warm water, or steam), thawing of a certain piece of

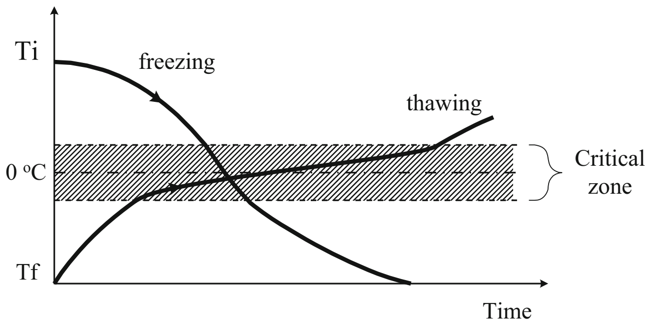


Fig. 9.23 Freezing and thawing of food. T_i initial, T_f temperature

food lasts longer than freezing (Fig. 9.23). This happens because the thermal conductivity of ice is almost fourfold of that of water (ice, 2.2 W/m K; water, 0.6 W/m K). As thawing proceeds, the water on the external part of the food causes a relative delay in heat transfer toward the still frozen product core (Fennema 1975; Heiss and Eichner 1995). Of course this does not hold, when electromagnetic thawing methods, such as dielectric and MW heating, are applied.

Thawing with MW is faster, but the product must be homogeneous. MW thawing, e.g., is not suitable for meat containing much fat in layers or many bones, since these components have different dielectric properties than meat flesh. In any case, in thawing, the product temperature should not exceed 10 °C, since this could increase the danger of microbial spoilage (Spiess and Kostaropoulos 1977).

In calculations related to the freezing of food, the freezing point, the fraction of frozen water, and the freezing time of the products are important. The freezing point is important in thawing estimations and in avoiding damage of products that should be chilled near their freezing point, without crossing it (e.g., cold storage of fruits). Table 9.8 gives the freezing points of several foods (Spiess and Kostaropoulos 1977).

The fraction of frozen water is important in estimating the most economic and proper freezing temperature. Freezing is completed, when no significant increase in the freezing of water of a product takes place. This fraction varies from product to product. At the end of freezing of white bread, its frozen water is only about 62 %, while in strawberries, it is about 95 %. Figure 9.24 gives the fraction of frozen water of several foods, for temperatures below 0 °C (–15 to –30 °C). Usually, freezing is completed at –18 to –20 °C.

The fraction of frozen water can be estimated through the Riedel diagrams. As indicated in Fig. 9.16, diagrams (a) and (b) give also the fraction of water that is frozen, when the temperature of a product is reduced below its freezing point. In the case, e.g., of lean meat (Fig. 9.16a) and fruit and vegetable juice (Fig. 9.16b), when the temperature is reduced from point A to point C, whose temperature is below 0 °C, the curves a_x that pass through point C indicate the fraction of frozen water.

Table 9.8 Freezing points of foods

Product	Initial freezing point (°C)
Meat	–0.6 to –1.2
Fish	–0.6 to –2.0
Milk	–0.5
Egg (white)	–0.45
Egg (yolk)	–0.65
Green salad	–0.40
Tomato	–0.9
Cauliflower	–1.1
Onion, peas, strawberries	–1.2
Peach	–1.4
Apple, pear	–2.0
Plum	–2.4
Cherry	–4.5
Nuts, chestnut	–6.7

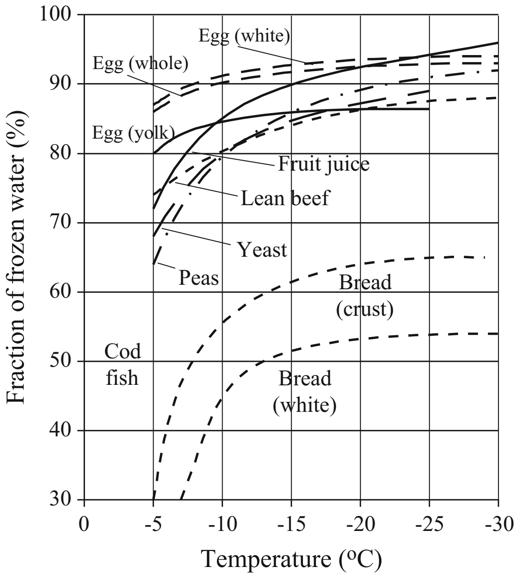


Fig. 9.24 Fraction of frozen water in foods

The freezing time of food is important in economic and technical analyses and estimations. It is important in estimating the output and the capacity of a unit, in selecting the right freezing equipment, in adjusting equipment for getting the best possible freezing result, and in manufacturing of proper equipment. In freezing time calculations, the Plank equation is used (9.8):

$$t_f = [\rho H_L/(T_f - T_A)]\left(Pa/h_c + Ra^2/\lambda\right) \tag{9.8}$$

where:

t_f, freezing time (s)
ρ, density of food (kg/m^2)
H_L, latent heat of crystallization (J/kg)
T_f, initial temperature of food (°C)
T_A, temperature of the freezing medium (°C), characteristic dimension (e.g., thickness of product parallel to direction of prevailing heat transfer) (m)
λ, thermal conductivity of the product (W/m K)
h_c, surface heat transfer coefficient (W/m^2 K)
R and P, constants for accounting the influence of the shape of the product. For sphere, $P = 1/16$, $R = 1/24$; for infinite plate, $P = 1/2$, $R = 1/8$; for infinite cylinder, $P = 1/4$, $R = 1/16$

Variations of this equation are presented by Schormueller (1966), IIR (1972), Brennan et al. (1990), Fellows (1990), Heldmann (1992), Singh(1995), Heiss and Eichner (1995), and Cleland and Valentas (1997). The Plank equation gives a rough estimate of the freezing time for basic geometric shapes, such as infinite plate, infinite cylinder, and sphere. According to this equation, the interrelation of freezing time of these three geometric shapes is t_{plate}:$t_{cylinder}$:$t_{cylinder} = 1.0$:0.5:0.33 (Heiss and Eichner 1995). However, since there are several limitations, restricting an accurate calculation of any food, several attempts have been undertaken to modify this equation or develop a new one. Nevertheless, although these attempts enable a more realistic approximation of the freezing time of real foods, application of the suggested solutions is still limited (Heldmann 1992; Singh 1995).

In estimating the required total heat load (i.e., including the latent heat) for freezing several foods, from initial temperatures above 0 °C to lower temperatures, the Riedel diagrams can be used. Otherwise, the calculation includes the heat load Q (J) for reducing the temperature of a product from an initial temperature T_i to the freezing temperature, the load for removing the latent heat, and the load for reducing the temperature of the product from its freezing point to the final temperature T_f ((9.8), (9.10), and (9.11)).

$$Q = mC_p(T_i - T_o) \tag{9.9}$$

$$Q = H_L m \tag{9.10}$$

$$Q = mC_{pf}(T_o - T_f) \tag{9.11}$$

where m is the mass of the product (kg); H_L, latent heat (J/kg); T_i, T_o, T_f are initial, freezing, and final temperature of the product (°C); and C_p, C_{pf} are specific heat (J/kg) of food above and below freezing, respectively.

The method of freezing has the following advantages: (1) good quality of final product, (2) extended shelf life of high-quality "raw products," and (3) versatility in the field of catering.

The disadvantages of food freezing include (1) relatively expensive process; (2) the freezing chain must not be interrupted up to the final consumption; and (3) dependence on high-quality raw materials.

9.6.2 Freezing Equipment

The freezing equipment can be classified according to the temperature applied (above or below –40 °C), the processed product (solid, liquid), the freezing medium (air, cold surface, liquid), and the way of processing (continuous, batch). Freezing methods are described by Venetucci (1995), Heiss and Eichner (1995), Persson and Loedahl (1996), James and Bailey (1990a), Woolrich and Novak (1977), IIR (1972), Schormueller (1966), and Cleland and Valentas (1997).

9.6.2.1 Air Freezing Equipment

Air is used in freezing food in tunnels, conveyor belts, and fluidized bed equipment. In all cases, air is blown countercurrently to the product, and depending on the freezing method, it is blown horizontally or vertically to the product. In tunnel freezing, the horizontal blow method prevails. In fluidized beds, air is blown vertically upward, and in belt freezing, both blowing methods are used. Since the specific heat of air is low, large air quantities are required for freezing.

Tunnel Freezers

Freezing in a tunnel is very similar to the tunnel cooling process. It is used in freezing of a wide range of products, extending from fine cut or minced products up to whole poultry or even half beef carcasses. It consists of an insulated room with one door or with two doors for a continuous operation (Fig. 9.24). The difference between the two types of tunnels is that in freezing, the air temperature is –30 to –40 °C, the insulation of the chamber is thicker, and the air velocity is higher (3–6 m/s). This requires larger heat exchangers and more powerful fans. Larger air velocity (e.g., 10 m/s) would reduce the freezing time, but the benefit of such reduction is not so significant, if the increased energy consumption is taken into account, because the energy increases with the third power of the air velocity (Schormueller 1966; Heiss and Eichner,1995). As in the case of cooling, the three main loading systems of a tunnel are (a) the push-through system, in which for each new trolley coming in the tunnel, a trolley with frozen product gets out; (b) the rack system, which is applied to freezing of carcasses; and (c) the chain drive system, in which the trolleys are pulled by a chain, in and out of the tunnel. When using trolleys, the product is first put on trays. If no large pieces, like poultry, are frozen, a trolley can be loaded with 40 trays and carry about 250–300 kg of product.

The freezing time depends on the size and thermal conductivity of the product. For product on trays, it lasts usually 1.5–6 h. There are single-row or double-row tunnels. Usually, the fans are in a channel above the trolleys. The heat exchangers are on both sides, and in the case of a double row, they are also between the trolleys. The capacity of a tunnel with 8 loaded trolleys is 1.5–4.0 tons/h. This corresponds to a specific capacity of about 25 kg/m^2 h of tray area. It is important to put the trolleys and trays in such a way that no free spaces between them are left.

The advantages of tunnel freezing are (1) flexibility, tunnels are suitable for a great variety of small product quantities; (2) easy cleaning; and (3) simplicity.

The disadvantages of tunnels include: (1) they require relatively large space; (2) more labor is needed than in belt or fluidized bed freezing; and (3) there is significant weight loss of product (2–3 %).

Fluidized Bed

The fluidized bed method is an individual quick-freezing (IQF) method, used in freezing small whole or cut pieces or food (diameter up to about 3 cm and length up to about 12 cm), such as, peas, French fries, sliced or cut carrots, beans, mushrooms, etc. The food pieces are frozen individually, as they hover in the air that freezes them quickly. The equipment consists of an inclined screen, fans (usually radial) blowing air upward through the perforated bottom, and heat exchangers cooling the air to –40 °C (Fig. 9.25). The air, streaming upward, freezes the product, which at the same time is transported by the air cushion formed. The product is frozen quickly, because (a) it is surrounded by cold air and (b) the heat transfer between air and product increases, as there is a relative motion between the product and the transporting air. Examples of freezing time are peas, 3–4 min, and French fries and strawberries, 9–13 min. The product layer over the screen depends on the product, e.g., 3–25 cm (usually about 12 cm). In proper design, the weight loss of the product is less than 1.5–2 %. A wet product surface and a high freezing rate reduce weight loss. This is achieved when the air velocity increases, causing better heat transfer, as the air moves faster along the product, and the rotation or tumbling of the product during its transport also increases. However, since high air velocity and low temperature tend to dry the product, a small relative reduction of air temperature

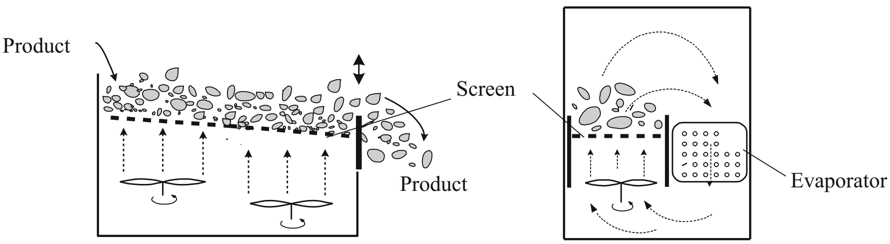


Fig. 9.25 Fluidized bed freezing

is beneficial. The capacity of fluidized bed freezing equipment varies between about 1 and 12 tons/h. The specific capacity for fruits and vegetables is about 160 kg/m^2 h. The dimensions of a fluidized bed freezer are length, 2.0–11 m; width, 2–9 m; and height: 3–6 m.

The advantages of fluidized bed equipment are (1) large specific capacity, (2) reduced product weight loss, (3) small dimensions, and (4) not many moving parts.

Disadvantages of fluidized bed equipment include (1) relatively high energy requirement, (2) not for universal use (only for small pieces), and (3) requiring homogeneity of the pieces.

9.6.2.2 Belt Freezers

This equipment consists of belts moving through a cold air steam (Figs. 9.26 and 9.27). The belts are either straight or curved, made of steel or plastic material, allowing air to pass through. In all cases, a special automatic mechanism maintains the tension of the belts constant. This kind of equipment is suited for freezing sensitive and relatively large or heavy pieces of food. Some examples of products frozen this way are apple slices, cauliflower, strawberries, artichokes, etc. Belt freezing equipment is also used in hardening of prefrozen food.

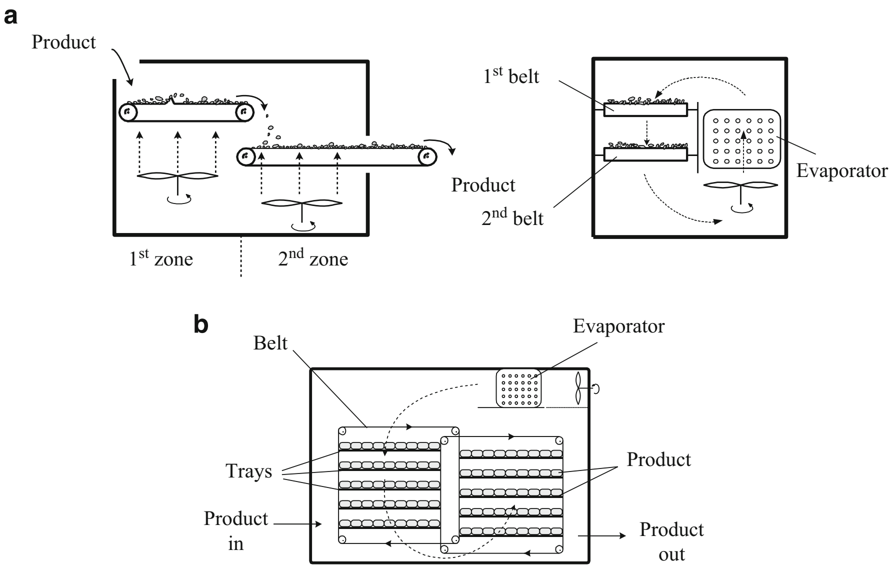


Fig. 9.26 Belt freezing equipment. (**a**) Straight belt and (**b**) elevator belts

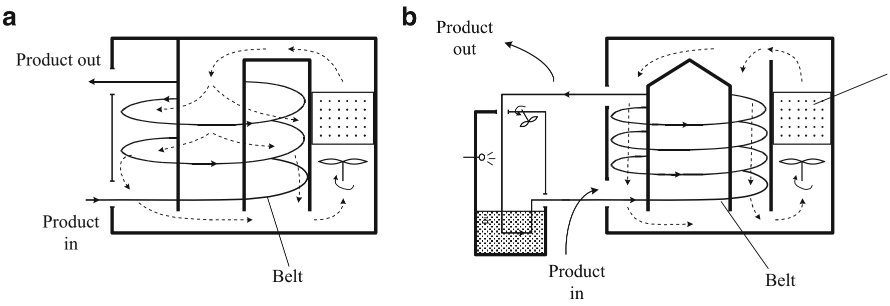


Fig. 9.27 Curved (*spiral*) belt freezing equipment (see text)

- Straight belts
  In some cases, the straight belts are separated into zones (Fig. 9.26a, b). In the first zone, the air recirculates vigorously, causing a surface freezing of the product ("crust freezing"). The freezing of the product is completed in the second zone. In some structures, buckles of the belt cause turning over of the product, contributing to more even freezing. Strawberries can be frozen in about 12 min and fish fillets in about 20 min. The capacity of belt freezing equipment is 0.2–6 tons/h. The overall length of such equipment is 5–13 m. For reducing the length, two or more belts may be placed one over the other. The overall width is 4–5 m. For increasing the versatility of the equipment, two or more belts may move parallel to each other, at different speeds. A single belt is usually 0.5–0.8 m wide. The heat exchanger lies in a separate part of the equipment, on the side of the belt. The overall height of the freezer is about 5 m. For freezing the same quantity of food, they require more floor space than the fluidized bed equipment, but about 30 % less than the spiral belt freezing equipment.
- Elevator system
  In Fig. 9.26b, the freezing equipment consists of parallel belts carrying large loaded shelves moving up, and after reaching the highest position in the room, they move again down. It is effectively an elevator system, in which freezing can be controlled by the speed of the belts, and it takes place during the up-and-down movement of the shelves. This method is often used in hardening of products like packed ice cream. The capacity of hardening equipment of this kind depends on the type of ice cream and the desired texture of the product. They can harden, e.g., 20,000 L/h. The method is very flexible. Besides freezing control, through the speed of the belt, it is also possible to load and empty the shelves at different positions, enabling the parallel freezing of different products or packages of different size. The method saves room but it requires more energy in comparison to flat belt structures.
- Curved belts
  Curved (spiral) belts are used for saving space. Two main types are the spiral and the semispiral freezing equipment, consisting of a combination of curved and straight belts. The spiral type is quite often used in freezing of hamburgers, fish

sticks, and ready meals. It is also used in hardening of frozen products. The combined type is mainly used in hardening. In the spiral construction, the length of the belt can exceed 300 m. The width is usually 4–7 m. Air is blown horizontally (Fig. 9.27a) or vertically through the product (Fig. 9.27b), which moves around a cylindrical core. In the first case the cylindrical core contributes to the air circulation. In both cases, this cylinder may contribute to the spiral movement of the belt, which winds around the core. Air is cooled in finned heat exchangers, placed in a separate room. The flexible belt is washed automatically after each full round. This may consist of hot water spraying, dipping in a detergent-containing vat, rinsing with cold water, and air-drying, before being reloaded. Large equipment may freeze continuously more than 5 tons/h. The specific capacity (about 40 kg/m^2 h) is not very high, due to the space occupied by the core. However, its main advantage is the continuous, gentle product transport and the flexibility in enabling the parallel freezing of products differing in size or in packaging. The floor space and height of an insulated room, containing a spiral freezing belt, may be 60–70 m^2 and 5–6 m, respectively.

Advantages of belt freezing equipment are (1) freezing of a wide range of delicate products, (2) freezing of wet and sticky products, (3) possibility to freeze also larger pieces, and (4) freezing of packaged or non-packaged food.

The disadvantages of belt freezing equipment include (1) relatively many moving parts (fans and belts), (2) relatively high energy consumption, (3) high initial capital, and (4) homogeneity of product distribution on the belt required.

9.6.2.3 Cold Surface Freezing

Food can be frozen quickly in plate freezing equipment (Fig. 9.28). It consists of several double-wall plates, in which a refrigerant circulates. Food is placed between the plates, which press the food by means of a hydraulic system lightly (0.06–0.1 bar), for reducing air pockets between cooling surface and packaging

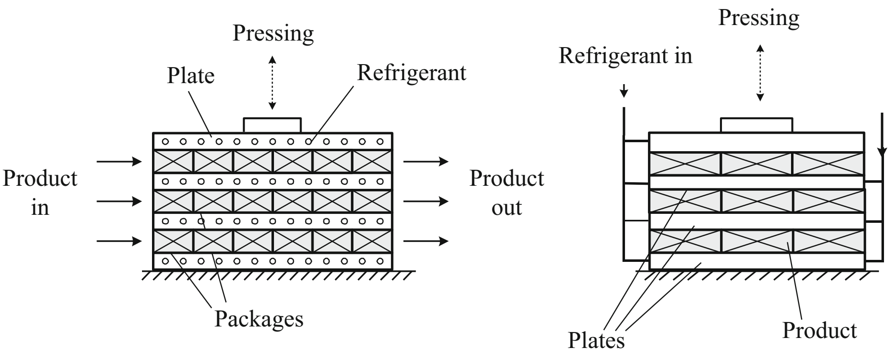


Fig. 9.28 Horizontal plate freezing equipment

(Guthschmidt 1973). When freezing is finished, the plates separate and the product is removed for reloading. The double-wall plates (Fig. 9.7) are made of extruded aluminum alloy of food quality. If the equipment is used for freezing fish in ships, the aluminum alloy used must be also seawater resistant. The plates can be parallel or vertical. Vertical plates are used in freezing fish in ships, because they require less free headroom. The number of parallel plates can be 5–20. Their spacing (distance) is up to 7 cm, and their surface is 1.5–2.0 m^2 (e.g., 1.5×0.8 m or 2.0×1.1 m). Vertical plate equipment usually has 12–16 plates, lying 5–9.5 cm apart. The surface of the plates can be, e.g., 1.2×0.6 m. Plate equipment is used in freezing of whole fish, fish fillets, pieces of meat (e.g., chops) product packed in rectangular packages, and liquid slurries. The last product is frozen in plastic bags, hanging between vertical plates. The capacity of plate equipment is 6–13 tons/24 h. The refrigeration capacity of large units is about 75 kW. The freezing of a 5-cm fish block can last about 1.15 h. The specific capacity of a plate freezing equipment is about 160 kg/m^2 h. The product, before entering the plate freezing equipment, is placed on metal trays. This is done for avoiding the icing of the plates, due to water loss of the product. The parallel plates are placed in an insulated cabinet. In the continuous system, two doors are used, one for feeding and one (on the opposite side) for emptying. The automatic feeding is based on the push-through system. Each time that the freezing of products of a tray is completed, a new tray enters, pushing that with the frozen products out. In the vertical plate equipment, the product (mainly fish) is frozen unpacked. It is taken out of the plates during defrosting (heating of the plates). The great advantage of plate freezers is their good specific capacity. This is about four times as high as that of freezing tunnels (85 kg/m^2 h). The overall dimensions of a cabinet containing 20 parallel plates may be $3 \times 2 \times 2$ m and its weight 1.8–2.0 tons.

9.6.2.4 Liquid Freezing

Liquid Freezers

In all cases, the equipment used is not complicated. In liquid freezing, cryogenic liquids such as liquefied nitrogen (N_2) and carbon dioxide (CO_2), brines, and nontoxic mixtures of water and solutes (e.g., sugar–alcohol solution in water) or other liquids (e.g., propylene glycol/water mixture) are used (Persson and Loedahl 1996). The food to be frozen is either immersed in the liquid or sprayed. Freezing by liquids is very fast, due to low temperature and direct contact with the whole product surface. The rate of freezing, e.g., by spraying a product with liquid nitrogen is 2.5 times faster than in fluidized bed freezing and 25 times faster by liquid immersion (Spiess and Kostaropoulos 1977). In the case of liquid N_2 or CO_2 and in some cases of brine (e.g., freezing of fish or meat that will be used in sausage manufacture), the freezing medium may contact directly the food. In all other cases, only packaged food is immersed in liquids or sprayed. Poultry, e.g., is packed in plastic bags that are vacuumed or shrunk by short immersion in hot water before freezing (Heiss and Eichner 1995).

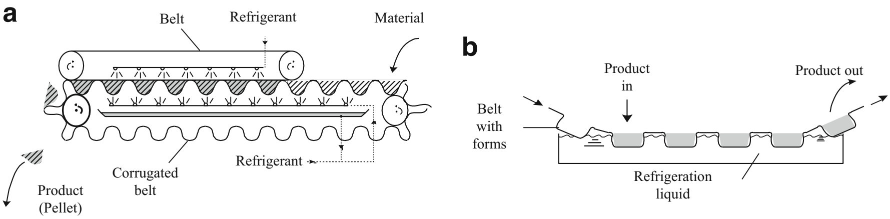


Fig. 9.29 Pellet freezing equipment (see text)

Frozen Pellets

Liquids are also used in the production of frozen pellets. In this case, foods, such as dairy products, liquid egg, fruit pulps, sauces, and vegetable purees, are frozen between two parallel moving metallic belts (Fig. 9.29). The corrugated lower belt gives the shape of the pellets, while the upper belt is flat. A liquid, e.g., a propylene glycol/water mixture, is sprayed on the external sides of both belts (Fig. 9.29a). In a variation of this equipment, only the lower belt is used, which is immersed in the liquid as it moves forward (Fig. 9.29b). The capacity of corrugated belt liquid freezers is 0.2–1.5 tons/h. Packaging of pellets can be done in bags or cartons or, in the case of further processing in the same fabric, in pallet boxes (0.5 tons/box). These pallet boxes can be stacked five high in cold store rooms.

Cryogenic Liquids

In the case of cryogenic N_2, a straight-belt-type freezing equipment is quite often used. However, if there is not enough space, spiral belts running around one or two cylindrical cores are used, or the product is directly immersed in liquid N_2. For small quantities, a batch process chamber can be used. A freezing unit of two spirals (one following the other) may have overall dimensions of $12 \times 6 \times 5$ m. The distance between the belts can be up to 15 cm and their width can be 0.5–2.5 m. The freezing time of spiral belt freezing units depends on the kind and size of the products. Usually it varies between 20 and 90 min. The total surface of 1–2 belt spirals is 20–200 m^2. Spiral freezing equipment using cryogenic N_2 may require, for the motion of the belts and the ventilation of evaporated N_2 gas, a power of 10–30 kW.

Figure 9.30 gives two examples of straight-belt equipment for freezing of food with cryogenic liquids. In freezing with N_2 (Fig. 9.30a), the product is put on a metallic perforated belt, which brings it in an insulated cabinet. The belt is up to 12 m long and 1.5–2.5 m wide. Three zones are distinguished, i.e., the precooling, the freezing, and the equilibrium zone. In a 12-m-long equipment, in the first zone, which is about 5–6 m, the product is precooled down to about –70 °C. Precooling is done, as the product passes through gas of just evaporated liquid N_2. In the second zone, which is 2–3 m long, liquid N_2 (–196 °C) evaporates, while it is sprayed

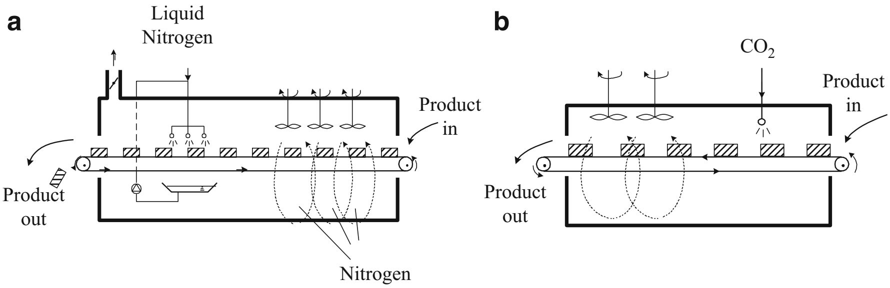


Fig. 9.30 Cryogenic liquid freezing equipment. (**a**) Liquid nitrogen and (**b**) carbon dioxide

directly onto the food. Due to the high heat transfer coefficient of the liquid N_2 during its evaporation, which is about 2300 W/m^2 K (Dinglinger 1977), the product is frozen very fast, reaching a surface temperature of about –190 °C. The N_2 gas is sucked and ventilated over the product entering the cabinet (first zone). In the third zone, the temperature of the product is equilibrated by air blown on its surface. The method can be used for packed and unpacked food. The capacity of N_2 freezing equipment can be up to about 1.5 tons/h. If immersion in liquid N_2 is used, freezing may last 15–100 s. For freezing 1 ton of small food particulates, a bath only 1.5 m long is required (Fellows 1990). Such a unit requires less than 1 kW power. However, since not all foods resist the freezing shock, caused by the sudden immersion in so low temperature, the method is restricted to products like berries, shrimp, and diced meat and fruits. In direct contact, 1 kg of liquid N_2, which corresponds to 1.245 L, may remove 330 kJ from a freezing product (Henrici and Haaf 1973). The liquid N_2 consumption is 1–1.5 kg/kg product (Heiss and Eichner 1995). In some cases, for reducing the N_2 consumption, the products processed by cryogenic N_2 are frozen only on their surface. The final freezing is carried out in other equipment (e.g., belt freezers) or in cold stores at –20 °C (IIF 1972). This can be done only if the subsequent slower freezing speed does not influence the quality of the food.

The triple point of carbon dioxide (CO_2) is –56.6 °C and 5.28 bar. Since it is expensive and not practical to work at so high pressure, the operation of the relevant equipment is based on cryogenic CO_2 ("dry ice"). When liquid CO_2 is released to the atmosphere, half of it becomes dry ice and the other half vapor. Both have the equilibrium temperature of –78.5 °C. Therefore, although CO_2 spraying equipment used in freezing of food is very similar to that of liquid N_2, it differs in the location of spraying. Since the CO_2 dry ice, produced during spraying, needs some time to sublime, spraying of the product with CO_2 is done close to its entrance in the cabinet (Fig. 9.30b). According to Woolrich and Novak (1977), freezing can be done in equipment consisting of a rotating cylinder in which dry ice is mixed with the product in proper analogy. The dry ice can be stored and ground before it is mixed with food. A 25-m-long cylinder, of about 30 cm diameter, could freeze 0.5 tons/h. If the product is subsequently packaged, no dry ice should be enclosed, since expansion of it later could damage the package. The overall cost of CO_2 freezing is comparable to that of N_2 freezing (Fellows 1990).

Advantages of cryogenic liquid freezing are (1) high freezing speed (high capacity, better quality), (2) low product weight loss during freezing (0.1–1.0 %), (3) low initial capital (1/3 of that of mechanical systems), (4) low floor space (specific capacity, 125 kg/m^2 h), (5) low maintenance cost (simple construction), and (6) easy handling.

The disadvantages of cryogenic liquid freezing include (1) cryogenic liquid expensive, (2) relatively high cryogenic liquid consumption, (3) dependence on relatively few cryogenic liquid suppliers, and (4) sophisticated storage installations. The plate equipment freezing and the cryogenic liquid freezing have better heat transfer coefficients than air blast freezing. Nevertheless, the overall fixed and operational costs of the plate freezing equipment are almost double as that of a fluidized bed (air blast) freezer (Woolrich and Novak 1977).

9.6.3 Thawing Equipment

The industrial process of thawing has become important due to the expansion of food freezing in connection to catering. The thawing processes may be subdivided into convective, vacuum, contact heating, and electrical methods. The convective methods are further distinguished into methods using air, water, or steam. In all cases, the equipment used is very similar to that of other heat exchange processes (e.g., cooling, heating, drying).

9.6.3.1 Convective Methods of Thawing

In all convective thawing methods, the heat has to be supplied through the external surface of the product. Therefore, the surface heat transfer coefficient, in transferring heat from a fluid medium to the product, is important. Furthermore, the speed of thawing depends also on the thermal conductivity of the thawed product, since heat has to pass through the melted zone of the product to the thermal center of the frozen food. Air blast thawing is slower than water or steam thawing. However, this method is the most often used, because it requires less capital and can be used in all products. The thawing tunnel is very similar to those used in freezing, cooling, or drying. However, since thawing is slower than freezing, in most cases trolleys, instead of belts, are used. In water thawing, the product is put in containers or bags, which are immersed in warm water. The water thawing equipment, except for the temperature of water used, is similar to that described in hydrocooling.

The advantages and disadvantages of air and water thawing are as follows (Hallstrom et al. 1988):

Advantages of air thawing are (1) low capital cost for batch operation, (2) versatility, (3) batch and continuous operation possible, and (4) low mechanical maintenance cost.

Disadvantages of air thawing are (1) large flow rate and turbulence necessary, (2) oxidation risk, (3) risk of drying of thawed products, (4) possibility of bacterial hazards, (5) cleaning difficult, (6) odor problems, (7) difficulty to heat product uniformly, and (8) continuous flow operation expensive.

Advantages of water thawing are (1) large heat transfer coefficients, (2) uniform heating of the product surface; (3) batch and continuous operation possible, and (4) low labor cost.

Disadvantages of water thawing are (1) leaching of flavor components; (2) possibility of bacterial contamination; (3) reusing of water often necessary, but costly; (4) corrosion of equipment; (5) continuous operation expensive; and (6) cleaning difficult.

Steam can be also used in thawing of frozen food. However, although this method is fast, it has the disadvantage that condensed steam is added to the thawed product.

9.6.3.2 Vacuum Thawing

Vacuum thawing has the advantage of a rapid rate, which is caused by the high mass transfer rate, according to the equation (Hallstrom et al. 1988):

$$dm/dt = D_{\mathrm{w}} \Delta p \tag{9.12}$$

where (m) is the mass of the product (kg), (t) is the time (s), D_{w} is the diffusivity of water vapor in the vacuum chamber (m^2/s), and (Δp) is the pressure drop at the surface of thawing (Pa). $\Delta p = (p_{\mathrm{s}} - p_{\mathrm{a}})$, where ($p_{\mathrm{s}}$) is the vapor pressure of water at the thawing surface and (p_{a}) is the vapor pressure of water in the vacuum chamber, which is practically equal to the air pressure in the chamber. In vacuum operation, both (D_{w}) and (Δp) will increase and, therefore, the thawing rate will increase sharply, according to (9.7). (D_{w}) will increase due to the inverse relationship to the gas pressure, while (Δp) will increase due to the sharp decrease of (p_{a}) in the vacuum chamber. Vacuum thawing is a batch process and the thawing equipment has a relatively small capacity (less than 2 tons). However, since the thawing rate is increased, a greater number of operating cycles is possible per day. The thawing equipment consists basically of a vacuum vessel, loaded with trays of layers of frozen product, which are stacked on trolleys.

9.6.3.3 Contact Thawing

As in the case of cooling, it is possible to use double-wall surfaces for transferring heat to the product that has to be thawed. This method is applied when the frozen food is available in small pieces. In this case, equipment similar to that of Fig. 9.13d is used. Semicylindrical jacketed vessels, equipped with a screw propeller in the center for agitating the frozen food, can thaw about 2.5 tons/h. The overall dimensions of such equipment are about 5.0×0.6 m.

9.6.3.4 Electrical Thawing

The two main methods of electrical thawing are dielectric thawing and microwave (MW) thawing. In both methods, the electrical energy is transformed to heat, as polarization or movement of molecules in an electric field takes place (see Chap. 6). The heat generated depends on the frequency and the electrical properties of food such as the dielectric constant (ε'), indicating the efficiency of a product to accumulate electrical energy, and the loss factor (ε''), indicating the electrical energy that can be transformed into heat, when the product is placed in an electromagnetic field (Table 9.9).

The main difference between the two methods lies in the frequency of the electromagnetic energy. In dielectric heating the frequencies are about 10 MHz, while in MW heating, certain radar frequencies for not interfering in other radar applications are used. These frequencies, according international agreement, are 915 and 2450 MHz.

In the dielectric heating, the frozen product is placed between plates or electrodes, which are connected to a source, supplying alternating high-frequency voltage. The frozen food forms the dielectric medium in an electrical capacitor. As electricity flows through the product, alternating from one plate to the other, heat is generated, thawing the food. In the case that larger blocks of frozen food (e.g., fish) with voids are to be thawed, this can be done after putting the iced blocks in trays filled up with water. The equipment consists of a rubber or plastic conveyor belt (about 1.5 m wide), which carries the food between electrodes or plates

Table 9.9 Dielectric properties of food and packaging materials (2.45 GHz, 25 °C)

Product	Dielectric constant (ε')	Loss factor (ε'')	Penetration (mm)
Beef	50	15	9
Beef (cooked)	35	12	10
Pork	58	16	
Pork (ham, cooked)	45	25	5
Potato	64	14	
Potato (puree)	65	21	7
Carrot	72	15	
Water (distilled)	77	11	16
Ice (–2 °C)	3	0.03	10 m
Polyethylene	2	0.001	28 m
Glass	6	0.005	10 m

Source: Data from Fellows (1990) and Schubert et al. (1991)

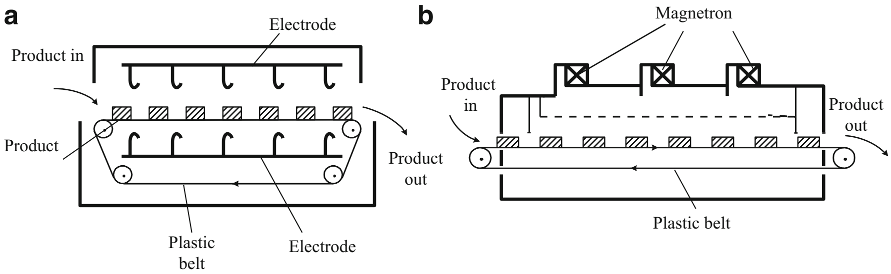


Fig. 9.31 Electrical thawing equipment. (**a**) Dielectric and (**b**) microwave (MW)

connected to electrical current (Fig. 9.31a). Each plate system can be an independent source of electrical/heating energy. The number and the power of the dielectric units depend on the capacity of the equipment and the kind of the frozen product (size, dielectric properties). The electric capacity of each unit may be about 10–100 kW. A unit of 20 kW can thaw about 350-kg fish, 450-kg meat, or 600-kg cakes per hour. A large unit can be 25 m long and 1.5 m wide and requires 120–150 kW. Such a unit can thaw up to 2.5 tons/h. The thawing time depends on the size and the kind of product and lasts less than 1 h (e.g., for frozen fish blocks, 10–60 min).

The industrial MW equipment consists of a belt transporting the food to a chamber in which magnetrons supply the electromagnetic energy. Of course, no metallic trays or packaging material should come in the chamber. For a more even distribution of the electromagnetic energy, fans are placed in the position the MW radiation enters the chamber or in a pre-chamber before meeting the food on the belt (Fig. 9.31b). The main advantages of both electrical methods is the rate of thawing and the reduced weight loss during thawing, which is about 2–9 % in comparison to the conventional heating. Due to shorter time, *minutes than hours*, there is less danger of contamination during thawing. The disadvantage of dielectric heating is that it can be applied only to products of restricted size. The main disadvantage of MW is nonuniform heating.

9.7 Cold Storage

9.7.1 General Aspects

There are three main categories of cold stores: the cold stores for temperatures up to about 0 °C; the cold stores for frozen products, in which the temperature is usually about –20 °C; and the controlled atmosphere (CA) cold stores, in which the temperature is as in the first category, but the atmosphere basically consists of reduced O_2 and increased CO_2. The basic equipment of these three cold store categories is the same. All have compressors, condensers, and evaporators. They differ in the type of compressor (e.g., two-stage reciprocating compressor for cold storage of frozen products), the type and dimensions of the evaporator, and the additional equipment required (e.g., humidifiers, scrubbers, etc.).

The efficient operation of a cold store depends on the following factors: (1) buildings (correct design and construction), (2) equipment (refrigeration load, reliability, flexibility), and (3) management (organization, in- and outflow policy).

These factors are interdependent. The correct stacking, e.g., depends on well-designed corridors, doors, and storage room. Successful cooling requires sufficient refrigeration load, good stacking, and efficient management. Management is important since the refrigeration efficiency depends also on the good maintenance of equipment and installations and the loading/emptying policy.

The shelf life of the products in all cold food depends on: (1) the kind of the products stored (in fruits, e.g., if they are climacteric), (2) the initial quality (e.g., ripeness, injuries), (3) the initial microbial count, (4) the temperature, and (5) the relative humidity of storage.

In all cases, the product exiting the cold storage should not have off-odors, and its texture should be as near to the initial as possible. For keeping away off-odors, mixed storage of products in the same room should be avoided, or in the case that this is not possible, only tightly packed foods should be stored together. Furthermore, the air in the cold storeroom should be renewed several times per day. With the exception of the CA storage, depending on the stored product, this renewal could take place up to six times per day. Preservation of the firmness (texture) of the products is especially important for fresh fruits and vegetables. This depends on holding the water, which means minimization of the products' weight losses.

In selecting the evaporators, compressors, and condensers of a cold store, the maximum refrigeration (heat) load is required. This is estimated for the case that the cold store room is full. With respect to the estimation of the refrigeration load in cold storage in temperatures above 0 °C, it is distinguished between chilling and storage cooling. The main aim of chilling is to reduce the temperature of the product from the ambient temperature to the storage temperature, as soon as possible. The refrigeration load in storage is needed for maintaining the temperature of the product constant during storage. Usually, chilling takes place in the cold storage rooms, when batches of products enter the room. In this case, the maximum refrigeration load is estimated for the refrigeration required for chilling the last batch of product, plus the refrigeration required for maintaining the temperature of the rest of the filled-up room constant. Examples of calculating the refrigeration load for storage are given by Pohlmann et al. (1978), Fellows (1990), Henze (1972), and van Beek and Meffert (1981).

The refrigeration load of a cold storage room (Q) includes calculations for the removal of the following heat loads:

(1) sensible heat of the product and its packaging (Q_s); (2) latent heat/water crystallization (Q_L) (storage of frozen products); (3) heat of respiration Q_R (when fruits and vegetables are stored); (4) heat produced by the air blowers (Q_V); (5) heat due to renewal of air (Q_A); (6) heat due to leakage through walls, ceiling, and floor

(Q_B); and (7) heat produced by personnel, light, and forklifts entering the cold storage room (Q_P).

The total refrigeration load for chilling (Q_{Ch}) or freezing (Q_F) is estimated on the basis of 1 h. The refrigeration load for storage (Q_{St}) is estimated for 24 h. For the final refrigeration load that must be supplied by the compressors (Q_o), the actual time of operation (t) must be also taken into consideration. The total load for each case is as follows:

1. Chilling:

$$Q_{Ch} = Q_s + Q_R + Q_V + Q_B \tag{9.13}$$

$$Q_{OCh} = Q_V / t_{Ch} \tag{9.14}$$

2. Freezing:

$$Q_F = Q_S + Q_L + Q_V + Q_B \tag{9.15}$$

$$Q_{OF} = Q_F / t_F \tag{9.16}$$

3. Storage:

$$Q_{St} = Q_L + Q_V + Q_A + Q_B + Q_P \tag{9.17}$$

$$Q_{Ost} = Q_{St} / t_{St} \tag{9.18}$$

The storage requirements of perishable food products and commodities are listed in tables published by the International Institute of Refrigeration (1967), the Institute of Food Science and Technology (1982), the ASHRAE (1989), Henze (1972), Dossat (1978), and Rao (1992). The tables contain data for several foods, recommendations about storage temperature, relative humidity, rate of respiration, expected storage life, etc. The tables of ASHRAE/Rao and Dossat use English units (BTU, lbs, etc.).

For estimating the heat transfer from a fluid medium to the products and the further transfer of refrigeration to its thermal center, the heat transfer coefficients and the physical properties of the products, thermal conductivity, thermal diffusivity, and specific heat, are necessary. Such properties can be found in Lewis (1990), Rao (1992), Saravacos and Maroulis (2001), Rha (1975), Rao and Rizvi (1995), and Kostaropoulos (1971). Some useful properties for design of food processing equipment are given in the Appendix B of this book.

The estimation of the sensible heat (reduction from the ambient temperature to the storage temperature, Q_s) has been described previously, in Chilling (Sect. 9.3). The latent heat Q_L (heat for freezing the water in the product or the water lost by the product during storage) can be calculated as mentioned in Sect. 9.4. The refrigeration load due to the respiration of fruits and vegetables Q_R can be estimated by multiplying the quantity of products $m(t)$ stored by the rate of respiration H_R (W/ton).

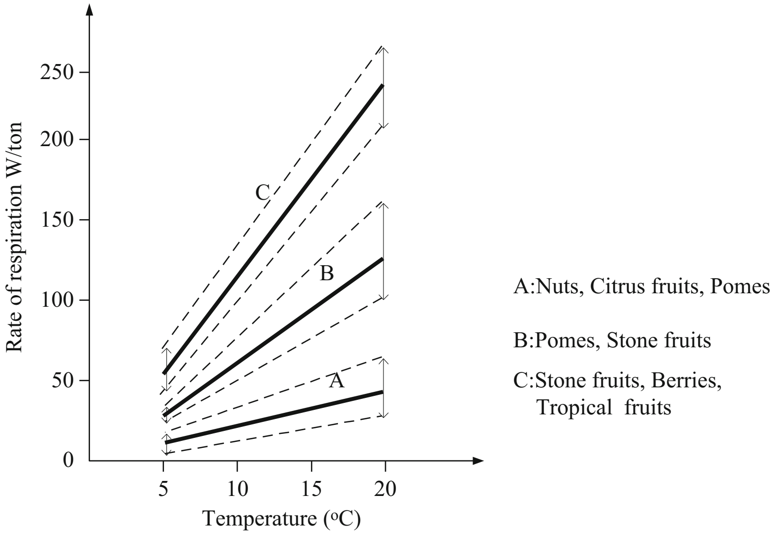


Fig. 9.32 Rate of respiration of fruits, in relation to their temperature

The respiration rate of fruits and vegetables is product specific and increases with the temperature. Fruits and vegetables are classified, according to their rate of respiration, into four categories (Fennema 1975): (a) rapid, $H_R > 150$ W/ton (e.g., asparagus, green peas, green beans); (b) moderately rapid, $H_R = 72$–150 W/ton (e.g., Brussels sprouts, spinach, strawberries); (c) moderately slow, 30–80 W/ton (e.g., apples, carrots, celery); and (d) slow, <30 W/ton (e.g., cabbage, grapes, lemons, oranges). Fruits could be classified as follows: (a) tropical fruits, berries, and some stone fruits, (b) pomes and some stone fruits; and (c) nuts, citrus fruits, and some pomes. Figure 9.32 gives the rate of respiration in relation of the temperature for products of these three categories. The respiration rates for a large number of fruits and vegetables are given by Hansen (1967a, b).

The motors of air blowers produce heat, which depends on the power (kW), the number, the position (e.g., inside the rooms or in channels), and the length of operation of the blowers. For estimating the heat for cooling the new air coming in the cold store, the quantity of the new air and the difference between the air enthalpy before and after entering the cold store are needed (9.19). The enthalpies can be obtained from the psychrometric chart (enthalpy–humidity), discussed in Chap. 8:

$$Q_R = n\rho V_A (H_{Aa} - H_{Ac}) \tag{9.19}$$

where n is the number of air changes per 24 h; ρ, density of air (1.29 kg/m^3); V_A, volume of each air inflow (m^3); and H_{Aa}, H_{Ac}, enthalpies of ambient and cold storage air (J/kg).

The estimation of the heat leakage is based on (9.20). The estimation is made for every wall, for the ceiling, and for the floor of each cold storage room. In the case that no values for the heat transfer coefficient (U) exist, either rough values (e.g., $U = 0.2$–0.9 W/m^2 K) are taken as recommended by Pohlmann et al. (1978) or they are calculated analytically, as already discussed in Chap. 6. The analytical calculation U takes into consideration the internal surface heat transfer coefficient h_i (W/m^2 K), the external heat transfer coefficient h_e (outside the cold store room), and the thermal conductivities of the materials composing the walls, the ceiling, and the floor of the cold storage room. For the external surfaces of the cold storage rooms (if no wind blows), the approximate (natural convection) value $h_e = 8$ W/m^2 K may be used. The surface heat exchange coefficient for the inside surface is higher:

$$Q_B = \sum (UA\Delta T) \tag{9.20}$$

where U is the overall heat transfer coefficient for the walls, ceiling, and floor, respectively (W/m^2 K); A, area of walls, ceiling, and floor, respectively (m^2); and ΔT, difference between the temperature on the external side of the walls, the ceiling, and the floor and the temperature of the cold storage room.

For estimating the heat produced by personnel, light, and opening of the doors during cold storage, an extra value $Q_o = (0.06$–$0.12) \times Q_{oSt}$ is usually added.

9.7.2 Reduction of Weight Loss

As indicated in Fig. 9.33, the weight loss of food ($\Delta W/W$) depends on the relative humidity (RH), the storage temperature (T), and the velocity of the circulated air (u_a), in the cold storage room. The weight loss is reduced, if in a storage room, at constant temperature, the relative humidity increases and the storage time decreases. Furthermore, at constant RH and temperature, the weight loss of stored non-packaged food increases, when the velocity of the circulated air is increased.

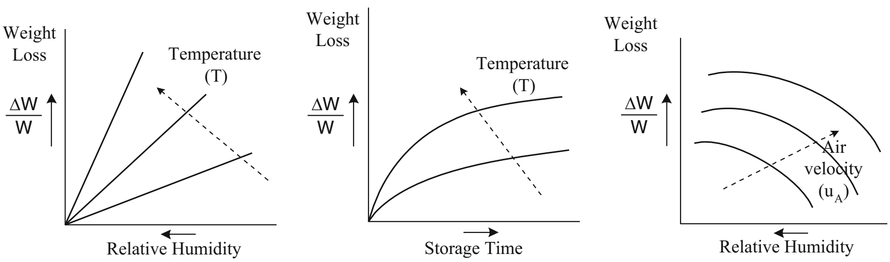


Fig. 9.33 Factors influencing the weight loss of food

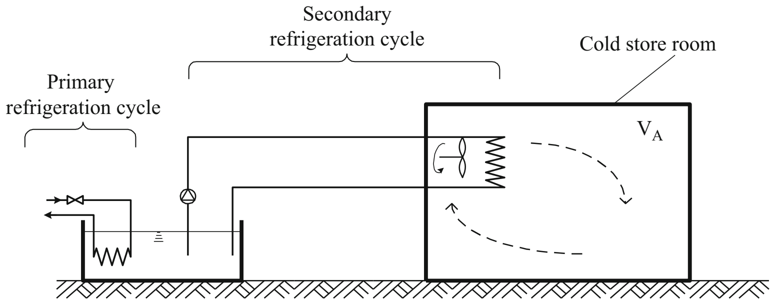


Fig. 9.34 Double refrigeration cycle

The measures that can be taken to reduce the weight loss of non-packaged, cold-stored food are:

1. Reducing the temperature difference between inblowing and outblowing air of an evaporator
2. Reducing the air velocity of the air blower
3. Use of humidifiers
4. Use of CA storage.

9.7.2.1 Temperature Adjustment

The temperature difference (ΔT) between in- and outblowing air of an evaporator can be reduced from about 7 to 5 °C, if oversize evaporator surfaces are used. A further reduction is possible, if indirect refrigeration is applied. This is, e.g., the case when cooled ethylene glycol solution is pumped in the evaporator of the air blaster, because the pump can regulate the inflow of refrigerant more accurately than several types of valves (Fig. 9.34).

Low air velocity reduces the weight loss of food. However, there are limits in the reduction of the air velocity, since air must reach the entire product stacked in the cold store room. In a cold storeroom, air blown by the air coolers circulates passing through the stacked product. The air coolers are placed in the room, as described in the section of air-cooling evaporators, preferably high above the door, at a distance from the ceiling and wall, allowing unhindered air in- and outflow. A uniform air distribution in the cold store is very important. Therefore, the stacked product should be placed in such a way that no possibility of "easy air escape" between the stacks exists. The air velocity near the product is usually about 0.1 m/s. In the storage of frozen products, the velocity is 0.5–2.5 m/s. The air circulation is 10–15 times per hour in cold storage and 25–30 times per hour in freeze storage. In the case of chilling, the circulation of air can be more than 70 times per hour (Pohlmann et al. 1978). The height of the stack is related to the volume of the room. The rooms of multi-floor cold stores, controlled atmosphere cold stores, and meat cold stores with racks are not high. The ground cold storage rooms for fruits and vegetables are preferably high, since they offer maximization of volume in relation to the external surface of the cold store. The height of a storeroom of a capacity of 300–350 tons may be 6–7 m. Cold storerooms for frozen products can be even higher. In multi-floor buildings, the rooms are usually 3–5 m high.

9.7.2.2 Humidification

Humidifiers are mainly used in relatively small cold storage rooms. In order to reduce the danger of mold growth, the water droplets, dispersed in the air of the cold storage room, must be as small as possible. This way the water condensation on the surface of the products is almost avoided. To achieve this kind of distribution, nebulizers are used. They consist, e.g., of piezoelectric crystals or other devices, vibrating in the water or on its surface, at an extremely high frequency. Such a device may nebulize (disperse droplets from liquid water) 1–4 kg water/h and consume about 0.5 kW. Their dimensions are about $0.5 \times 0.5 \times 1.0$ m.

9.7.2.3 Controlled Atmosphere

The controlled atmosphere (CA) storage can be applied to a wide range of foods (fruits and vegetables, ready meals, fresh meat, fish, and baked products). It reduces the weight losses during storage, as it makes possible the increase of the relative humidity in the cold storage room, up to 95 %. Significant research work has been done in determining the best CA conditions for storage of food. Henze (1972) and Gorini et al. (1990) describe the use of CA in fruits and vegetables, Finne (1982) discussed the use of CA in muscle foods, Wolf et al. (1975) describe the use in ready meals, and Knorr and Tolins (1985) discuss the effect of carbon dioxide-modified atmosphere in the compressibility of stored baked foods. In CA storage, the products are enclosed in air-tight rooms. Oxygen and carbon dioxide are reduced to 4–5 %. When starting CA storage of, e.g., fruits (apples), the initial oxygen (21 %) is reduced by catalytic combustion, after mixing it with propane or methane, to about 5 %. Such equipment, e.g., for reducing oxygen of 70 m^3/h inflowing air consumes 5 kg/h propane and 2 kW electrical energy. Their overall dimensions are $1.5 \times 1.0 \times 1.5$ m, and their weight is about 0.5 ton. Finally, the oxygen in the cold storage atmosphere is stabilized at 3 %, since the products consume also an amount of oxygen during respiration. Further reduction of oxygen is possible (e.g., in ultralow oxygen "ULO" storage), but in any case, oxygen should not be less than 1 %, because anaerobic conditions would prevail, with the consequence of formation of ethyl alcohol and undesirable physiological changes (Gorini et al. 1990).

During storage, it is important to maintain the oxygen and carbon dioxide concentrations constant, since in several products, increase of carbon dioxide and/or decrease of oxygen, above or below certain limits, can damage the products (Fennema 1975; Fellows 1990). There are many combinations of oxygen–carbon

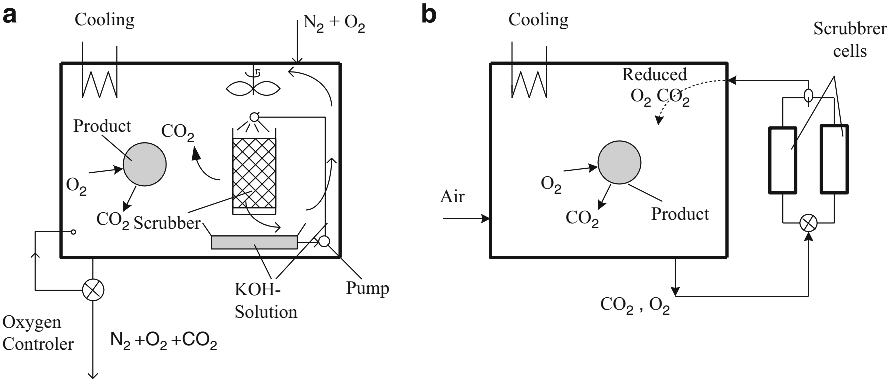


Fig. 9.35 Scrubbers for controlled atmosphere storage

dioxide. Henze (1972), e.g., gives only, for apple and pear varieties, 16 and 20 such combinations, respectively. The carbon dioxide content in the storage atmosphere depends on the temperature in the room and the product stored. In vegetables, it can vary according to Gorini et al. (1990) between 0 % (e.g., cucumber, 12 °C) and 14 % (asparagus, 1 to 4 °C).

The carbon dioxide is maintained constant through scrubbers. These are devices located inside or outside of the CA cold storage rooms, removing excess carbon dioxide. There are two methods for removing excess carbon dioxide, the chemical and the physical method (Meffert 1983). In the chemical method (Fig. 9.35a), substances such as dry lime, suspensions of lime in water, lye solutions, etc., can be used for absorbing carbon dioxide. In the physical method (Fig. 9.35b), carbon dioxide is absorbed in water or adsorbed in activated carbon. The activated carbon is one of the most often used substances in scrubbers. The construction of activated carbon scrubbers is simple. The unit consists of two cells, and each cell must be regenerated every 20 min with fresh air at ambient temperature. The adsorption capacity of a relatively large, e.g., for two-cell activated carbon units, is 50–450 kg CO_2/24 h. The advantages of such equipment are simple construction, low water absorption, and low electrical consumption. The overall dimensions of such equipment are 1.5 × 2.0.0 × 2.0 m, the weight is about 1.5 tons, and the required power is 4–5 kW. The disadvantages of such units are the capacity reduction at carbon dioxide concentrations below 2 % and the possibility of aroma absorption, when it is used in the storage of certain fruits. However, in several cases this capability is advantageous, since intensive aroma is developed, when the products are ripe, and significant ethylene is also produced, which are undesirable.

Activated carbon reduces also ethylene in the storeroom, retarding ripening. In the case that the products produce significant amounts of ethylene, as in the case of kiwi fruit, special scrubbers (ethylene converters) can be used, which consist of special catalyst bed fillings. Such units can keep the ethylene concentration in cold storerooms of fruits and vegetables below 0.1 ppm. They may circulate up to

500 m^3/h and require 7-kg ethylene-reducing substance of 10 tons per month and about 6 kW power (60 % for pumping the gas through the converter).

9.8 Ice Manufacturing

Ice was for a long time the most important source of cooling or cold storage. Until a few years ago, it was very important in cooling of railway wagons, for the transport of fresh products, as peaches or grapes. Nowadays, with respect to food, it is still used in two direct cooling applications, (a) chilling of processed food, such as minced meat in the cutter, or chilling poultry after slaughtering and (b) cooling of fresh food, such as fish at sea and ashore, or vegetables.

Ice is very important especially in the cooling of fish, since this kind of cooling has following advantages:

1. It is harmless in direct contact with food.
2. As it comes directly in contact with fish, it reduces its temperature rapidly.
3. The cooling temperature is just above the freezing point of fish.
4. It keeps fish moist (other methods dry it) and glossy.
5. Ice washes away slime, reducing bacterial growth.
6. It is a natural thermostat.

The largest ice-producing installations are harbors of fishing ships. Such installations consist of ice manufacturing and ice storage units and installations for automatic loading of ice to ships or transport vehicles. The ice-producing capacity of large units exceeds 150 tons/day, and the capacity of large ice storage units may exceed 300 tons.

Three types of ice can be manufactured: the "dull ice," the "clear ice," and the distilled water ice. In the first type, tap water is used and the natural salts of water are evenly distributed in the ice. In the second type, air is blown in the water during the ice production, and the natural salts are concentrated in the core if the ice formed. In the third category, distilled water is used, and this way the ice is free of any contamination or salts. The equipment of ice manufacturing can be classified in block and small ice equipment. In the past, the block ice was more important, while now the small ice manufacturing prevails, as this type is more flexible and requires less space and the whole process, from production to consumption, can be automated.

In the ice-block production (Fig. 9.36), water is filled in identical cans. The cans have a square cross section, and they are larger in the upper than in the lower part. The cans are either dipped in a cold brine bath (e.g., calcium chloride) or they have double walls in which refrigerant circulates (jacketed cans). In the first case, the cans stay in the brine bath until the contained water is frozen. There are different sizes of cans. A 25-kg ice block is, e.g., produced by a can whose upper side is about 0.2 m, its lower about 0.16 m, and its height about 1.0–1.1 m. The cans are grouped in 5–6 pieces, and all cans of a group are filled with water. If the temperature of the

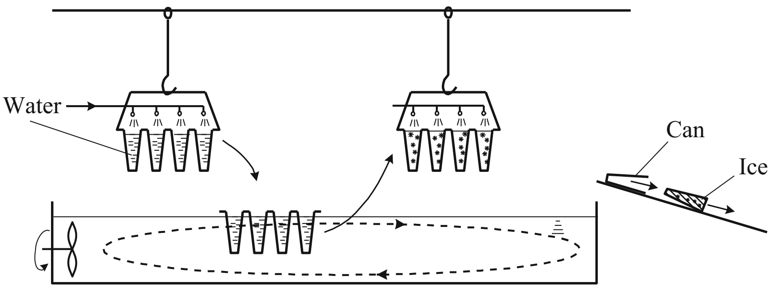


Fig. 9.36 Ice-block manufacturing

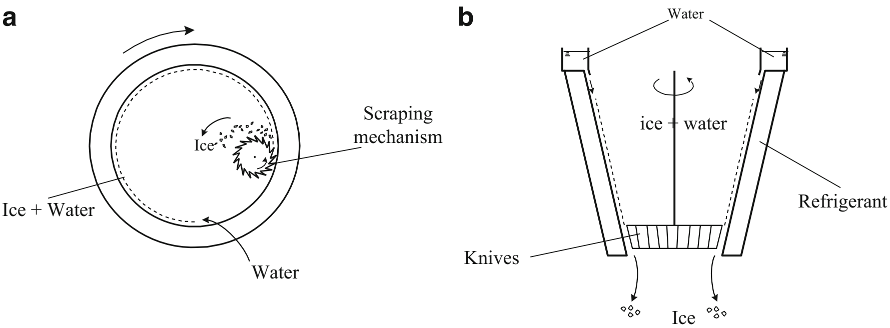


Fig. 9.37 Flake-ice manufacturing

filling water is not low (e.g., below 10–15 °C), the water is chilled. When ice blocks are created, the cans are taken out of the brine by a hoist/crane system; they are dipped in water of ambient temperature, until the ice contacting the can surface is molten; and finally they are dumped and emptied automatically. Agitators circulate the brine, for increasing the heat transfer. The capacity of a unit of 90 cans is 3 tons of ice block per 24 h. A 25-kg ice block is frozen in about 18 h. The main dimensions of a bath of about 7-m^3 brine are $4.5 \times 2 \times 1.5$ m. An ice-block unit requires 460–670 kJ/kg refrigeration, including heat loses. In some cases, air is blown in the lower part of jacketed cans.

Small ice is produced as "snow" flakes and in different simple geometrical forms (e.g., cylinders, cubes, etc.). Besides direct production methods, small ice can be also produced by grinding ice blocks. Flakes are manufactured by scraping off iced surfaces. Ice is formed on the internal or external frozen surface of jacketed cylinders, as water trickles down. Knives or other similar devices scrape the ice formed on the jacketed surface (Fig. 9.37). Such equipment can produce about 25 tons of ice in 24 h. When "snow," or ice with relative much nonfrozen water, is produced, ice pellets can be formed by pressing. In creating small ice pieces of simple geometrical formation, cylinder ice and tube ice are two common structures. They are produced by freezing water in tubes.

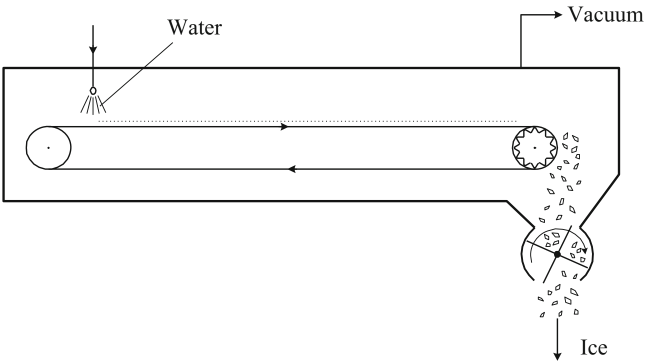


Fig. 9.38 Vacuum belt ice-producing equipment

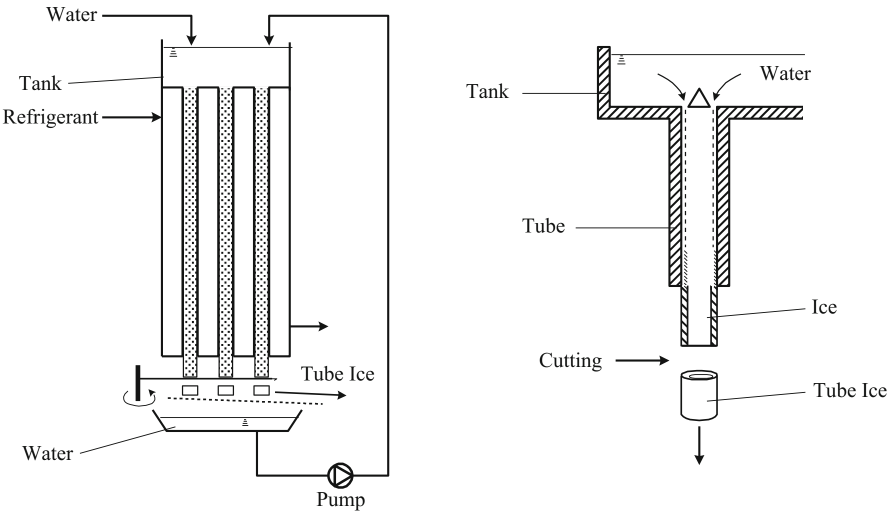


Fig. 9.39 Tube-ice manufacturing

In a variation of the above described flake-ice unit, the flakes are formed on a metallic belt (Fig. 9.38). The belt is in a vacuum chamber (pressure 6.1 mbar or 4.58 mmHg), in which the temperature is –3 °C. Water is spread on one end of the belt, and at the other end, the ice layer formed on the surface of the belt breaks in flakes, as the belt turns around.

Tube ice is manufactured as water, flowing inside tubes, is frozen (Fig. 9.39). The tube ice machine consists of a vertical tube and shell heat exchanger and a water tank on the upper part. A special water distributor on the top of each tube controls the water flow so that a steady film is formed on the inner surface of the tubes. The water freezes quickly due to the refrigerant that contacts the tubes externally. The excess of water is collected below the tubes and is sent back to

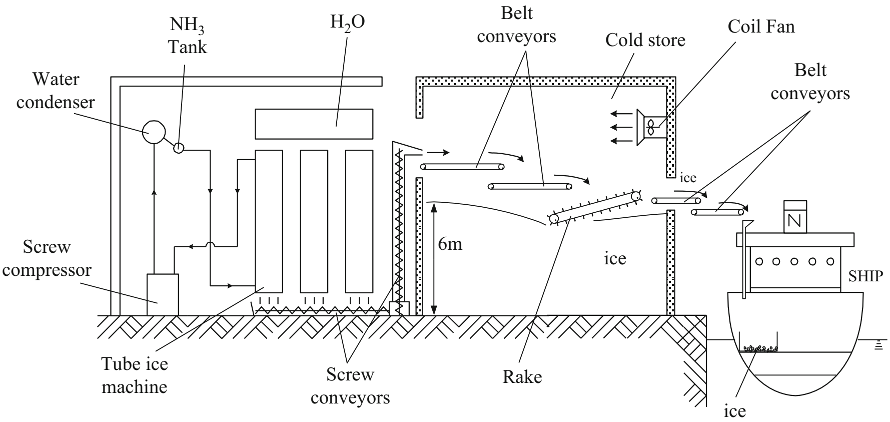


Fig. 9.40 Ice manufacturing, storage, and loading installation

the water tank. A rotating device cuts ice in the form of tubes, as it glides down and exits the low end of the tube. Harvesting is done during the nonfreezing cycles of the equipment. The size of the ice tubes depends on the diameter of the heat exchanger tubes and the cutting. Production can start properly after 10–15 min. The capacity depends on the initial temperature of water, the evaporating temperature, and the length of the freezing cycle. A unit of 25 m^2 can produce 18 tons s/h. The floor space of such equipment is only 15 % of the space required by equivalent ice-block installations. The power consumption requirements of a medium or large ice production plant are about 37 kWh/ton.

Small ice is stored bulk, up to 6 m high (Fig. 9.40). Air coolers provide auxiliary refrigeration compensating heat leakage from the environment of the cold store room. Loading is done automatically, by using screw conveyors or belts, operating high above the bulk stored ice and providing a uniform, loose filling of the cold store room. Emptying is done also automatically. Scoop or rake systems move the ice toward screw conveyers. Ice is then transported, by a system of screw elevators and belts, to the final destination (e.g., a ship).

Example 9.1 In a cold storage room, 312 tons of apples must be precooled and stored for 5 months. The well-insulated storage room is rectangular and its dimensions are 20 × 16.5 × 6.0 m. The external wall of the narrow side (16.5 m) is located northward, and the opposite wall adjoins a corridor. One of the long side walls is external and is located westward. The opposite wall adjoins a cold storage room for frozen products. The ceiling is flat. The temperatures in the apple cold storage room and its surroundings are:

Cold storage room, 2 °C
Corridor, 10 °C
Ambient (maximal external temperature during the storage period), 35 °C
Old storage room for frozen products, –18 °C
Ground temperature, 15 °C

The apples are delivered at 25 °C at the rate 6 tons/h, for 8 h/day, and are precooled in 20 h. They are packed and stored in wooden boxes (60 × 40 × 32 cm; tare, 2 kg), containing 20-kg fruits each, which are stacked on wooden Euro pallets (120 × 100 × 10 cm; tare, 5 kg).

The specific heat of apples and wood are 3.77 kJ/kg K and 1.89 kJ/kg K, respectively. The overall heat transfer coefficients (U) of the insulated walls, floor, and ceiling are assumed as follows:

External walls (north and west), $U_N = U_W = 0.16$ W/m^2 K; wall between apple storage and frozen products, $U_E = 0.17$ W/m^2 K; wall between apple storage and corridor, $U_C = 0.24$ W/m^2 K; ceiling, $U_D = 0.12$ W/m^2 K; and floor, $U_B = 0.22$ W/m^2 K.

Estimate (a) the refrigeration load required for precooling and cooling the apples and (b) the power of an ammonia reciprocating compressor for supplying this load, if the refrigerant condensation temperature is 25 °C and the subcooling is 15 °C.

Assumptions

It is assumed that 120 tons of apples will be precooled and 192 tons will be cold stored. The delivery of apples lasts 6.5 days, and precooling is completed after 3 days (72 h). The maximal refrigeration load required is when the last delivery of 6 tons of apples is precooled, and the precooling of the previous 19 deliveries (19 × 6 = 114 tons) has to be accomplished.

Precooling (120 tons, 20-h Basis)

1. Sensible heat of apples

$$Q_s = mc_a(T_{ia} - T_{fa}) = 120,000 \times 3.77 \times (25 - 2) = 10,405\,\text{MJ}$$

2. Sensible heat of wooden boxes and pallets

   The cold storage room has a capacity of 15 × 14 = 210 pallet stacks, each stack consisting of three pallets. Each pallet contains 5 × 5 = 25 boxes, containing 45 × 20 = 500 kg of apples.

   The height of a loaded pallet is (5 × 32) + 10 = 170 cm. The height of a stack is 3 × 170 = 510 cm and its weight 1500 kg. There is a 90-cm free pace above the stacks, for air circulation. For precooling of the 120-ton apples, the number of boxes will be 120,000/20 = 6000 and the number of pallets 6000/25 = 240.

   The sensible heat of boxes will be $Q_{box} = 6000 \times 2 \times 1.89 \times (25 - 2) =$ 521.6 MJ.

   The sensible heat of pallets will be $Q_{pal} = 240 \times 5 \times 1.89 \times (25 - 2) = 52.2$ MJ (about 10 % of the boxes).

   The sensible heat of both boxes and pallets will be $Q_{box} + Q_{pal} = 573.8$ MJ.

3. Heat of respiration of apples

   Assume heat of respiration of apples at 2 °C, $r = 14.6$ W/ton, $Q_r = 120 \times 14.6 \times 20 \times 3600 = 126.1$ MJ.

4. Heat due to the ventilation fans

   Assume six fans of 0.6 kW each, $Q_v = 6 \times 0.6 \times 20 = 72$ kWh = 260 MJ.

   Total cooling requirement, $Q_T = 10{,}405 + 573.8 + 126.1 + 260 = 11{,}364.9$ MJ/20 h = 157.8 kW.

5. Heat loss due to thermal leakage, $Q = UA\Delta T$

   Heat transfer areas: north and south walls, $A_N = A_S = 94$ m^2, west and east walls.

   $A_W = A_E = 114$ m^2. Area of ceiling and floor, $A_D = A_B = 330$ m^2. Heat losses: $Q_N = 0.16 \times 94 \times (35-2) = 0.5$ kW; $Q_W = 0.16 \times 114 \times (35-2) = 0.602$ kW + 5 % solar radiation = 0.632 kW; $Q_E = 0.17 \times 114 \times (2-(-18)) = 0.36$ kW; $Q_C = 0.24 \times 94 \times (10-2) = 0.18$ kW; $Q_D = 0.12 \times 330 \times (35-2) = 1.31$ kW + 25 % solar radiation = 1.64 kW; $Q_B = 0.22 \times 330 \times (15-2) = 0.94$ kW.

   Total heat leakage $Q_L = Q_N + Q_W + Q_C + Q_D + Q_B - Q_E = 0.50 + 0.632 + 0.18 + 1.64 + 0.94 - 0.36$, or $Q_L = 3.53$ kW (about 2 % of the total precooling requirement).

   Total refrigeration load for precooling the 120 tons of apples, $Q_{pcl} = 157.8 + 3.53 = 161.33$ kW.

Cold Storage (192 tons, 24 h Basis)

1. Heat absorption due to weight loss: it is assumed that the total water loss during the entire storage period (5 months) is 10 % of the weight of the apples (31.2 tons), corresponding to a water loss of 208 kg/24 h. Since the heat of evaporation of water at 2 °C is 2.5 MJ/kg, the heat absorption will be $Q_{\Delta W} = -208 \times 2500 = -520{,}000$ kJ = – 144 kWh.
2. Respiration rate: $Q_r = 192 \times 14.6 \times 24/1000 = 67.3$ kWh.
3. Thermal leakage: $Q_L = 3.53 \times 24 = 84.7$ kWh.
4. Heat loss due to the fans: $Q_v = 6 \times 0.6 \times 24 = 86.4$ kWh.
5. Air renewal: volume of the cold storage room $V = 1881$ m^3. From the psychrometric chart of the temperature range 0–50 °C (Zogzas 2001), the enthalpy change of water from ambient temperature 25 °C to cold storage temperature 2 °C will be $\Delta H = 51 - 13 = 38$ kJ/kg, and the air density $\rho = 1.22$ kg/m^3. Assuming three renewals of room volume per 24 h, the heat load will be $Q_A = 3 \times 1881 \times 1.22 \times 38 = 261.6$ kWh (for 24 h).

Total cold storage requirements for the 192 tons of apples, $Q_T = 67.3 + 84.7 + 86.4 + 261.6 - 144 = 356$ kWh. If the compressor, supplying this cooling load, is operated 14 h per 24 h, its capacity will be 356/14 = 25.4 kW.

The refrigeration load for precooling and cold storage will be 161.3 + 25.4 = 186.7 kW. Assuming an addition of 5 % refrigeration for meeting extra requirements, such as personnel entering the cold storage room, electric lights, and fluctuations of the temperature in the adjoining rooms, the total refrigeration load will be 186.7 + 9.3 = 196 kW.

Compressor

Assume that a reciprocating ammonia compressor will be used, operating at the standard conditions of evaporator $T_e = -15$ °C, $p_e = 2.3$ bar and condenser $T_c = -15$ °C, $p_c = 8.3$ bar. The heats of evaporation of ammonia at the evaporator and condenser temperatures are 1310 and 1170 kJ/kg, respectively, with the corresponding densities $\rho_e = 2$ kg/m^3 and $\rho_c = 8$ kg/m^3 (Perry and Green 1997).

The amount of ammonia required for the refrigeration load 196 kW will be 196/1310 = 0.15 kg/s or 0.15 × 3600 = 540 kg/h, corresponding to 540/2 = 270 m^3/h.

The mechanical power of the compressor P is given by the equation P = refrigeration load/COP, where COP is the coefficient of performance, i.e., the ratio of the refrigeration produced to the work supplied. The theoretical COP, i.e., the efficiency of the reversed Carnot cycle, is given by the equation COP $= T_e/(T_c - T_e)$. In this example, COP = 258/(298–258) = 6.45, and therefore, P = 196/6.45 = 30.4 kW. If the mechanical efficiency of the compressor is 75 %, the required power of the compressor will be P = 30.4/0.75 = 40.5 kW.

Notes

1. The COP of the compressor increases significantly, if the temperature difference between the condenser and the evaporator is reduced. Thus, for a refrigeration system operated at $T_c = 25$ °C and $T_e = -5$ °C, we have COP = 268/(298–268) = 8.93, and P = 196/8.93 = 21.9 kW. The mechanical power of the compressor of this example will be 21.7/0.75 = 29.2 kW.
2. The actual coefficient of performance COP can be calculated as the ratio COP $= \Delta H_{ref}/W_{comp}$, where ΔH_{ref} is the refrigeration load (heat removed) and W_{comp} is the mechanical power supplied by the compressor. Both these quantities can be obtained from the pressure–enthalpy diagram of the refrigerant (ammonia) (Perry and Green 1997). The actual COP is usually similar to the theoretical COP, calculated from the equation COP $= T_e/(T_c - T_e)$.
3. Most of the refrigeration load (90 %) is needed for precooling the apples, and only about 10 % is used for maintaining the cold storage temperature.
4. The water evaporated from the apples during storage (208 kg/24 h) is assumed to be removed from the storage room with the renewal air. Cooling and heating balances of frosting and defrosting of the cooling coils in the cold storage room are not considered in this example.

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