

# **FOOD REFRIGERATION SYSTEMS AND AIR CONDITIONING SYSTEMS**

**- Course support for Erasmus students -**

# 1. Thermodynamics

Refrigeration and air conditioning involves various processes such as compression, expansion, cooling, heating, humidification, de-humidification, air purification, air distribution etc. In all these processes, there is an exchange of mass, momentum and energy. All these exchanges are subject to certain fundamental laws. Hence, in order to understand and analyze refrigeration and air conditioning systems, a basic knowledge of the laws of thermodynamics, fluid mechanics and heat transfer that govern these processes is essential.

## 1.1. Definitions

*Thermodynamics* is the study of energy interactions between *systems* and the effect of these interactions on the system *properties*. Energy transfer between systems takes place in the form of *heat and/or work*. Thermodynamics deals with *systems in equilibrium*.

A *thermodynamic system* is defined as a quantity of matter of fixed mass and identity upon which attention is focused for study. In simple terms, a system is whatever we want to study. A system could be as simple as a gas in a cylinder or as complex as a nuclear power plant. Everything external to the system is the *surroundings*. The system is separated from the surroundings by the *system boundaries*, which may be real (the walls of a cylinder) or imaginary. Thermodynamic systems can be further classified into *closed systems, open systems and isolated systems*.

A system is specified and analyzed in terms of its *properties*. A property is any characteristic or attribute of matter, which can be evaluated quantitatively. The amount of energy transferred in a given process, work done, energy stored etc. are all evaluated in terms of the changes of the system properties. A *thermodynamic property* depends only on the state of the system and is independent of the path by which the system arrived at the given state. Hence all thermodynamic properties are point functions. Thermodynamic properties can be either intensive (independent of size/mass, e.g. temperature, pressure, density) or extensive (dependent on size/mass, e.g. mass, volume). The fundamental thermodynamic properties are *pressure, volume and temperature*.

Thermodynamic properties relevant to refrigeration and air conditioning systems are temperature, pressure, volume, density, specific heat, enthalpy, entropy etc.

A *process* is defined as the path of thermodynamic states which the system passes through as it goes from an initial state to a final state. In a *simple process* (transformation) one fundamental property remains constant; in a *complex transformation*, all the fundamental properties change.

A system is said to have undergone a *cycle* if, beginning with an initial state, it goes through different processes and finally arrives at the initial state.

An *equation of state (EOS)* is a fundamental equation, which expresses the relationship between pressure, specific volume and temperature. The simplest equation of state is that for an incompressible substance (e.g. solids and liquids), which states that the specific volume is constant.

## 1.2. Heat and work

*Heat* is a way of changing the energy between a system and its surroundings by virtue of a temperature difference only. Any other means for changing the energy of a system is called *work*. We can have push-pull work (e.g. in a piston-cylinder, lifting a weight), electric and magnetic work (e.g. an electric motor), chemical work, surface tension work, elastic work, etc.

Both heat and work are *process properties* and not properties of the system: the system is changing energy only when passing from one state to another (when it undergoes a process).

**Sign convention for work and heat transfer:** Most thermodynamics books consider the

work done by the system to be positive and the work done on the system to be negative. The heat transfer to the system is considered to be positive and heat rejected by the system is considered to be negative.

The *sensible heat* (which changes the temperature of the system) is given by the equation:

$$dQ = m \cdot c \cdot dT \quad [J],$$

where  $m$  is the mass of the system [kg],  $c$  is the specific heat [J/kg.K] and  $dt$  is the temperature change [K].

The *specific heat*  $c$  is an intensive property that represents the amount of heat that needs to be added to 1 kg of material to raise its temperature by 1°K. For gases it is important to know if the system undergoes a transformation at constant pressure or at constant volume; hence, we may define:

- $c_p$  – specific heat at constant pressure;
- $c_v$  – specific heat at constant volume.

The specific heat ratio  $c_p/c_v = k$  is also called the coefficient of adiabatic compression of the respective gas.

For an ideal gas, the specific heat depends on the nature of the gas and its temperature:

$$c = a + b \cdot T + c \cdot T^2 + \dots,$$

where  $a$ ,  $b$ ,  $c$  etc. are constants. For narrow temperature ranges the specific heat may be considered constant and we get the following relationship of the sensible heat:

$$Q_{12} = \int_1^2 m \cdot c \cdot dT = m \cdot c \cdot \Delta T.$$

*Pressure-volume work* (or *PV work*) occurs when the volume  $V$  of a system changes. For a reversible process in a closed system, PV work is represented by the following differential equation:

$$dL = p \cdot dV,$$

where  $p$  denotes the pressure inside the system and outside the system, against which the system expands; the two pressures are practically equal for a reversible process;  $dV$  denotes the infinitesimal increment of the volume of the system.

*Flow work* is taken into account when, in an open system fluid enters and leaves the system. It requires flow work for the fluid to enter the system against the system pressure and at the same time flow work is required to expel the fluid from the system. It can be shown that the specific flow work is given by the product of pressure,  $p$  and volume,  $V$ :

$$L_d = p \cdot V,$$

or:

$$dL_d = p \cdot dV + V \cdot dp.$$

### 1.3. Ideal (perfect) gases

A **perfect gas** is a theoretical gas that differs from real gases in a way that makes certain calculations easier to handle; the following assumptions apply to a perfect gas: (1) the gas consists of a large number of [molecules](#), which are in random motion and obey [Newton's laws of motion](#); (2) the volume of the molecules is negligibly small compared to the volume occupied by the gas; (3) no forces act on the molecules except during elastic collisions of negligible duration. Although no gas has these properties, the behaviour of real gases is described quite closely by the general gas law at sufficiently high temperatures and low pressures, when relatively large distances between molecules and their high speeds overcome any interaction.

#### 1.3.1. Simple transformations of a perfect gas

a) The **constant volume** (isochoric) process is defined by the following equation:

$$\frac{p}{T} = ct.$$

An example of this process is the heating or cooling of a gas stored in a rigid cylinder.

b) The **constant pressure (isobaric) process**

If the temperature of a gas is increased by the addition of heat while the gas is allowed to expand so that its pressure is kept constant, the volume of the gas will increase in accordance with Charles law:

$$\frac{V}{T} = ct.$$

c) The **constant temperature (isothermal) process**

According to Boyle's law, when a gas is compressed or expanded at constant temperature, the pressure will vary inversely with the volume:

$$p \cdot V = ct.$$

1.3.2. *Equation of state of the perfect gas*

The perfect or ideal gas equation of state (EOS) is given by:

$$p \cdot v = R \cdot T,$$

where  $p$  is the pressure [Pa],  $v$  is the specific volume ( $v = V/m$ ) [ $m^3/kg$ ],  $T$  is the absolute temperature [K] and  $R$  is the gas constant [J/kg·K].

The ideal gas equation is satisfactory for low molecular mass, real gases at relatively high temperatures and low pressures.

It can be proven that  $R = c_p - c_v$ .

For  $m$  kg of gas, the gaz, the EOS becomes:

$$p \cdot V = m \cdot R \cdot T,$$

and for 1 kmol of gas, with the molecular weight  $M$  we get:

$$M \cdot p \cdot v = M \cdot R \cdot T.$$

Considering that:

- $M \cdot v = v_M$  - molar volume ( $v_M=22,414 \text{ m}^3/\text{kmol}^1$ , la  $101325 \text{ Pa}$  și  $0 \text{ }^\circ\text{C}$ ),
- $M \cdot R = R_M$  - universal constant of the ideal gas ( $R_M = 8314,472 \text{ J/kmol}\cdot\text{K}$ ),

we get:

$$p \cdot v_M = R_M \cdot T.$$

Taking into account that  $m$  kg of gas contain  $\mu$  kmols ( $\mu = \frac{m}{M}$ ), the EOS may be written

as:

$$p \cdot V = \frac{m}{M} \cdot R_M \cdot T = \mu \cdot R_M \cdot T.$$

1.3.3. *Complex transformations of the real gas*

a) The adiabatic process

An adiabatic process is one in which no heat transfer takes place to or from the system during the process. For a fluid undergoing an adiabatic process, the pressure, temperature and volume satisfy the following relations:

$$p \cdot v^k = ct.,$$

$$T \cdot v^{k-1} = ct.,$$

$$T \cdot p^{\frac{1-k}{k}} = ct.$$

where  $k$  is the coefficient of adiabatic compression or expansion.

b) Polytropic process

When a gas undergoes a reversible process in which there is heat transfer, the process frequently takes place in such a way that a plot of  $\log P$  vs  $\log V$  is a straightline, implying that:

<sup>1</sup> 1 kmol  $\rightarrow$  M [kg]  $\rightarrow$   $v_M$  [ $m^3$ ]

$$p \cdot v^n = ct.,$$

$$T \cdot v^{n-1} = ct.,$$

$$T \cdot p^{\frac{1-n}{n}} = ct.,$$

The value of  $n$  can vary from  $-\infty$  to  $+\infty$ , depending upon the process (fig.1.1):

- For an isobaric process,  $n = 0$  and  $P = \text{constant}$ ;
- For an isothermal process,  $n = 1$  and  $T = \text{constant}$ ;
- For an isentropic process,  $n = k$  and  $s = \text{constant}$ ;
- For an isochoric process,  $n = -\infty$  and  $v = \text{constant}$ .

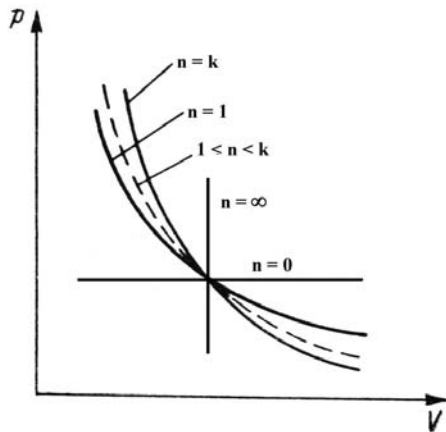


Fig. 1.1 – Simple and complex transformations of the perfect gas

#### 1.4. Fundamental laws of thermodynamics

**0.** The *Zeroth law of thermodynamics* states that when two systems are in thermal equilibrium with a third system, then they in turn are in thermal equilibrium with each other. This implies that some property must be same for the three systems. This property is temperature. Thus this law is the basis for temperature measurement. Equality of temperature is a necessary and sufficient condition for thermal equilibrium, *i.e.* no transfer of heat.

**1.** The *First law of thermodynamics* is a statement of law of conservation of energy, adapted for thermodynamic systems.

a) It was well known that heat and work both change the energy of a system. Joule conducted a series of experiments which showed the relationship between heat and work in a thermodynamic cycle for a system. He used a paddle to stir an insulated vessel filled with fluid (fig. 1.1). The amount of work done on the paddle was noted (the work was done by lowering a weight, so that work done =  $m \cdot g \cdot h$ ). Later, this vessel was placed in a bath and cooled. The energy involved in increasing the temperature of the bath was shown to be equal to that supplied by the lowered weight. Joule also performed experiments where electrical work was converted to heat using a coil and obtained the same result.

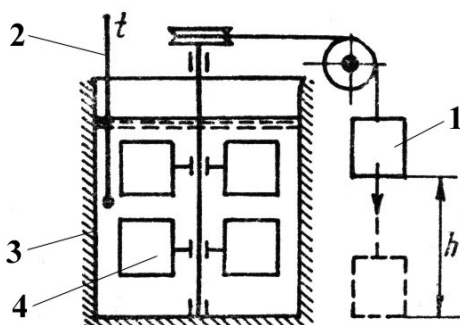


Fig. 1.1 – Joule's experimental setup

- 1-weight;
- 2-thermometer;
- 3-insulated vessel;
- 4-stirrer.

According to Joule, the amount of mechanical work generated by friction needed to raise the temperature of a pound of water by one degree Fahrenheit was 772.24 foot pound force. In today's units, using [J] for both heat and work, the formulation of the first law for closed systems and cyclic processes is:

$$Q = W.$$

b) For a closed system undergoing an *open transformation* (1-A-2, fig. 1.2), we introduce to imaginary paths (B and C) in order to obtain cyclic processes.

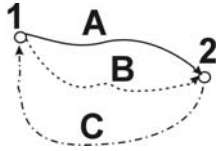


Fig. 1.2 – Applying the first principle for open processes

Using the previously mentioned formulation for cyclic processes, we get:

- for the cycle (1) → A → (2) → C → 1:  $(Q_{1A2} + Q_{2C1}) - (W_{1A2} + W_{2C1}) = 0$ ;
- for the cycle (1) → B → (2) → C → 1:  $(Q_{1B2} + Q_{2C1}) - (W_{1B2} + W_{2C1}) = 0$ .

Rearranging the equations we get:

$$Q_{1A2} - W_{1A2} = Q_{1B2} - W_{1B2} = W_{2C1} - Q_{2C1},$$

or :

$$Q_{12} - W_{12} = ct. = \Delta E_{tot},$$

wher  $\Delta E_{tot}$  is the overall variation of energy. The energy may be written as:

$$E_{tot} = U + E_c + E_p,$$

where U is the internal energy,  $E_c$  is the cinetic energy and  $E_p$  is the potential energy.

Neglecting the variations of cunetic and potential energy we get:

$$Q_{12} - W_{12} = \Delta U,$$

or:

$$Q - W = \Delta U.$$

Using specific quantities ( $q=Q/m$ ,  $w=W/m$ ) results in:

$$q - w = \Delta u,$$

and respectively:

$$dq = dw + du,$$

or:

$$dq = p \cdot dv + du,$$

The internal energy of a system represents a sum total of all forms of energy viz. thermal, molecular, lattice, nuclear, rotational, vibrational etc. It can be proven that:

$$du = c_v \cdot dT ,$$

which shows that internal energy is the energy exchanged by the system with the environment in an isochoric process.

c) For the *open system* shown in fig. 1.3,  $m_1$  and  $m_2$  are the mass flow rates at inlet and outlet,  $c_1$  and  $c_2$  are the inlet and outlet velocities and  $z_1$  and  $z_2$  are the heights at inlet and outlet with reference to a datum; q and w are the rate of heat and work transfer to the system.

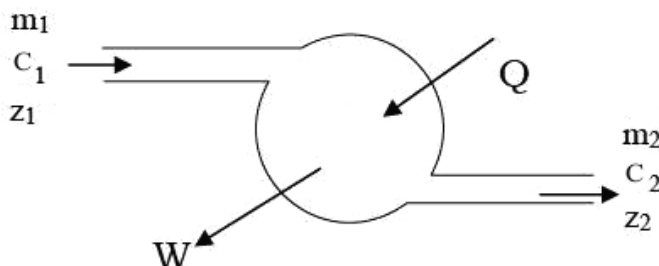


Fig. 1.3 - First law of thermodynamics for an open system

In steady state process, the time rate of change of all the quantities is zero, and mass is

also conserved ( $m_1 = m_2 = m$ ). As a result, the mass and total energy of the system do not change with time, hence, using the principle of energy conservation, we get:

$$q + u_1 + p_1 \cdot v_1 + \frac{c_1^2}{2} + g \cdot z_1 = w + u_2 + p_2 \cdot v_2 + \frac{c_2^2}{2} + g \cdot z_2,$$

where  $p \cdot v$  is the specific flow work.

Assuming that  $c_1 \approx c_2$  and  $z_1 \approx z_2$ , we get:

$$q - w = u_2 + p_2 \cdot v_2 - (u_1 + p_1 \cdot v_1),$$

or:

$$q = w + \Delta h \text{ or } dq = dw + dh,$$

where  $u + p \cdot v = h$  is the specific enthalpy. It can be proven that:

$$dh = c_p \cdot dT,$$

which shows that enthalpy is the energy exchanged by the system with the environment in an isobaric process.

## 2. The *Second law of thermodynamics*

The first law is symmetrical with respect to the initial and final states of an evolving system. But the second law asserts that a natural process runs only in one sense, and is not reversible. It is common sense that heat will not flow spontaneously from a body at lower temperature to a body at higher temperature. In order to transfer heat from lower temperature to higher temperature continuously (that is, to maintain the low temperature) a refrigeration system is needed which requires work input from external source. This is one of the principles of second law of thermodynamics, which is known as *Clausius statement of the second law*.

### **Clausius' statement of second law**

*It is impossible to transfer heat in a cyclic process from low temperature to high temperature without work from external source.*

It is also a fact that all the energy supplied to a system as work can be dissipated as heat transfer. On the other hand, all the energy supplied as heat transfer cannot be continuously converted into work giving a thermal efficiency of 100 percent. Only a part of heat transfer at high temperature in a cyclic process can be converted into work, the remaining part has to be rejected to surroundings at lower temperature. If it were possible to obtain work continuously by heat transfer with a single heat source, then automobile will run by deriving energy from atmosphere at no cost. A hypothetical machine that can achieve it is called Perpetual Motion Machine of second kind. This fact is embedded in *Kelvin-Planck Statement of the Second law*.

### **Kelvin-Planck statement of second law**

*It is impossible to construct a device (engine) operating in a cycle that will produce no effect other than extraction of heat from a single reservoir and convert all of it into work.*

## **Reversible and Irreversible Processes**

*A process is reversible with respect to the system and surroundings if the system and the surroundings can be restored to their respective initial states by reversing the direction of the process, that is, by reversing the heat transfer and work transfer. The process is irreversible if it cannot fulfill this criterion.*

If work is done in presence of friction, say by movement of piston in a cylinder then a part of the work is dissipated as heat and it cannot be fully recovered if the direction of process is reversed. Similarly, if heat is transferred through a temperature difference from higher temperature to a lower temperature, its direction cannot be reversed since heat transfer from lower temperature to higher temperature would require external work input. These are two

examples of irreversible processes. Reversible process is a hypothetical process in which work is done in absence of friction and heat transfer occurs isothermally. Irreversibility leads to loss in work output and loss in availability and useful work.

The historical origin of the second law of thermodynamics was in Carnot's principle. It refers to a cycle of a [Carnot engine](#), fictively operated in the limiting mode of extreme slowness known as quasi-static, so that the heat and work transfers are between subsystems that are always in their own internal states of thermodynamic equilibrium. The Carnot engine is an idealized device of special interest to engineers who are concerned with the efficiency of heat engines.

The Carnot cycle when acting as a heat engine consists of the following steps (fig. 1.4):

1. Reversible isothermal expansion of the gas at the "hot" temperature,  $T_1$  (isothermal heat addition or absorption). During this step (1 to 2) the gas is allowed to expand and it does work on the surroundings. The temperature of the gas does not change during the process, and thus the expansion is isothermal. The gas expansion is propelled by absorption of heat energy  $Q_1$  from the high temperature reservoir.
2. Isentropic (reversible adiabatic) expansion of the gas (isentropic work output). For this step (2 to 3) the mechanisms of the engine are assumed to be thermally insulated, thus they neither gain nor lose heat. The gas continues to expand, doing work on the surroundings, and losing an equivalent amount of internal energy. The gas expansion causes it to cool to the "cold" temperature,  $T_2$ .
3. Reversible isothermal compression of the gas at the "cold" temperature,  $T_2$  (isothermal heat rejection - 3 to 4). Now the surroundings do work on the gas, causing an amount of heat energy  $Q_2$  flow out of the gas to the low temperature reservoir.
4. Isentropic compression of the gas (isentropic work input). (4 to 1). Once again the mechanisms of the engine are assumed to be thermally insulated. During this step, the surroundings do work on the gas, increasing its internal energy and compressing it, causing the temperature to rise to  $T_1$ . At this point the gas is in the same state as at the start of step 1.

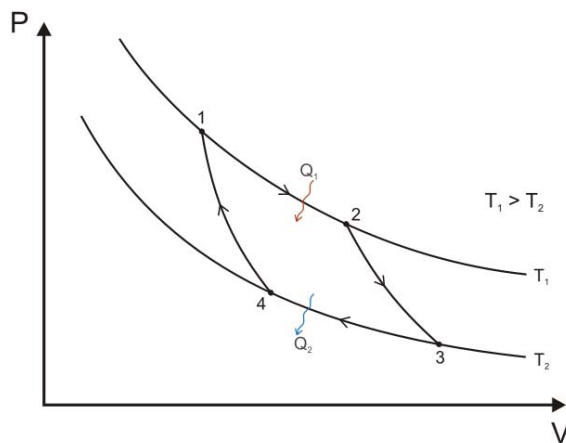


Fig. 1.4 – The Carnot cycle

The thermal efficiency of the direct Carnot cycle is:

$$\eta_{Tc} = \frac{Q_1 - Q_2}{Q_1} = 1 - \frac{T_2}{T_1}.$$

The ideal refrigeration systems work according to the reversed Carnot cycle (fig. 1.5) the coefficient of performance (COP) is given by the relation:

$$COP = \frac{\text{heat absorbed}}{\text{work supplied}} = \frac{q_c}{w_{tot}} = \frac{T_c}{T_h - T_c}.$$



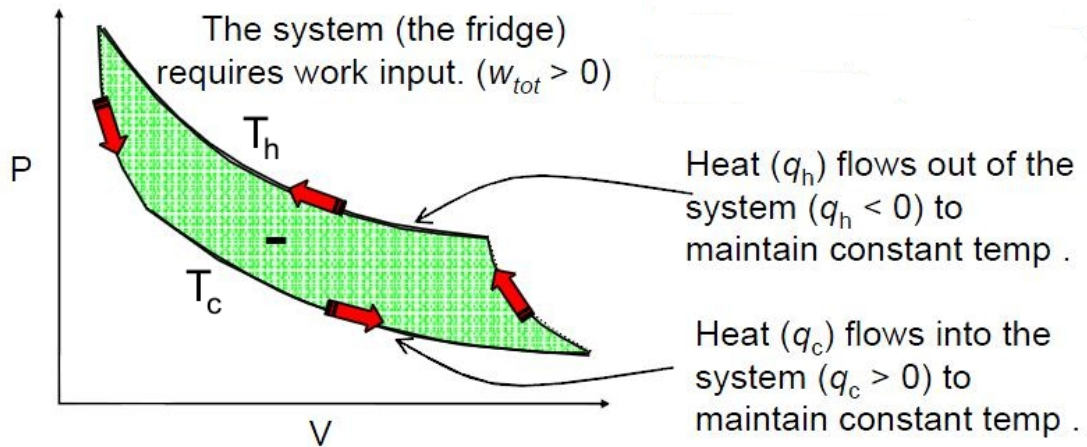


Fig. 1.5 – Reversed Carnot cycle

The mathematical formulation of the second law of thermodynamics for a *reversible cyclic process* assumes the decomposition of the cyclic process into elementary Carnot cycles (fig. 1.6).

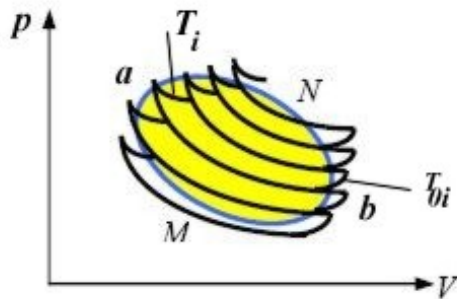


Fig. 1.6 – Decomposition of the cyclic process into elementary Carnot cycles

As a result we finally get:

$$\oint \frac{dQ}{T} = 0 .$$

$\oint \frac{dQ}{T}$  is called the Clausius integral and  $\frac{dQ}{T}$  is called entropy:

$$dS = \frac{dQ}{T} .$$

Entropy is a measure of amount of disorder in a system. It is also a measure of the extent to which the energy of a system is unavailable.

For *irreversible cycles* the second law becomes:

$$dS > \frac{dQ}{T} .$$

The formulations for the first and second laws of thermodynamics can be combined into the fundamental thermodynamic relation:

$$dU = T \cdot dS - p \cdot dV .$$

### 1.5. Real gases

As opposed to a perfect or ideal gas, the real gases exhibit properties that cannot be explained entirely using the ideal gas law. To understand the behaviour of real gases, the following must be taken into account:

- compressibility effects;
- variable specific heat capacity;
- van der Waals forces;
- non-equilibrium thermodynamic effects;

- issues with molecular dissociation and elementary reactions with variable composition.

Thomas Andrews, in 1869, experimented isothermal compression; he used carbon dioxide as the test gas. Using the apparatus shown in fig. 1.7, he plotted a series of isotherms (PV curves, fig. 1.8) to test the validity of Boyle's law over a wide range of pressures. The gases were compressed by tightening the screw and the pressure was estimated using the nitrogen assuming that under the conditions of the experiment it still obeyed Boyle's law.

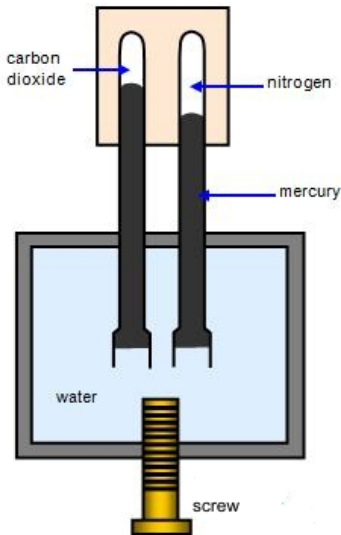


Fig. 1.7 – Andrews's apparatus

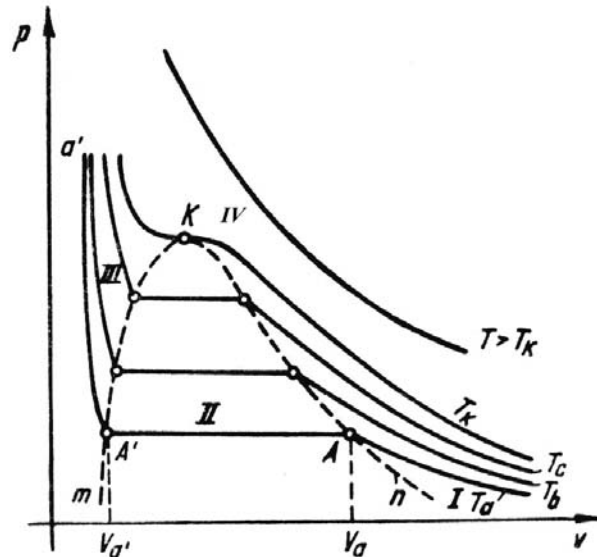


Fig. 1.8 – Isothermal compression of a real gas

At temperature  $T_a$ , decreasing the volume leads to the increase of the pressure; when point A is reached, the first droplets of liquid appear into the mass of gas. When the volume is further decreased (from A to A'), the pressure remains constant and more and more gas turns into liquid; when point A' is reached, the entire amount of gas was transformed into liquid. Because liquid is incompressible, the pressure rapidly increases when volume is decreased from A' to a'.

For the temperature  $T_b > T_a$  the behavior is the same, but the length of the constant pressure plateau diminishes.

For a certain temperature  $T_k$  the gas turns into liquid instantly; point K is called *critical point*. At the critical temperature  $T_k$  the densities of the liquid and gas become equal – the boundary disappears. The material will fill the container so it is like a gas, but may be much denser than a typical gas, and is called a 'supercritical fluid'. The isotherm at  $T_k$  has a horizontal inflection at the critical point K.

For temperatures higher than the critical temperature the isotherms start to become similar to the isotherms of the ideal gas (fig. 1.9). The substance can not be liquefied through isothermal compression when  $T > T_k$ .

The curve m-K-n is called saturation curve; the left side of the curve (m-k) is the condensation branch and the right side (n-K) is called the vaporization branch.

It should be noted that the phase change (the horizontal lines) takes place at constant temperature and pressure.

The saturation curve and the critical temperature isotherm are boundaries, separating regions where the substance is in different aggregation states:

- I – superheated vapors;
- II – moist saturated vapors (liquid+vapor);
- III – liquid;
- IV-supercritical fluid.

There are dry saturated vapors on the saturation line (n-K) and there is saturated liquid on the condensation line (m-K).

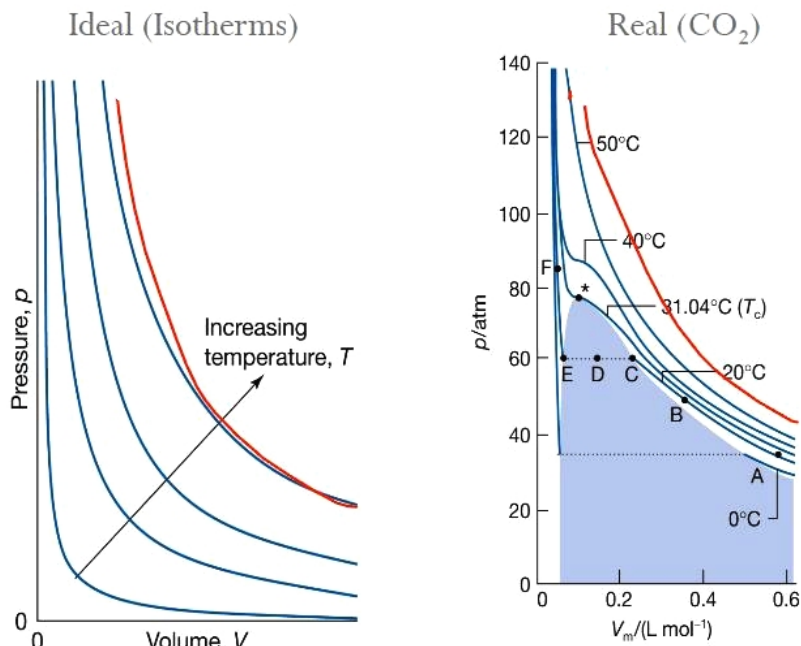


Fig. 1.9 – Isothermal compression for ideal and real gases

Several equations of state for the real gas were proposed over time; table 1.1 summarizes some of them.

Table 1.1.

Equations of state (EOS) of the real gases

<b>van der Waals</b>	$(P + a / v^2) * (v - b) = RT$
	$a = (27 / 64) * (R^2 T_c^2 / P_c)$
	$b = (1 / 8) * (R T_c / P_c)$
<b>Reidlich-Kwong</b>	$[P + a / T^{1/2} * v * (v + b)] * (v - b) = RT$
	$a = 0.42748 * (R^2 T_c^{2.5} / P_c)$
	$b = 0.08664 * (R T_c / P_c)$
<b>Reidlich-Kwong-Soave</b>	$P = (RT / v - b) - (a\alpha / v * (v + b))$
	$a = 0.42748 * (R^2 T_c^2 / P_c)$
	$b = 0.08664 * (R T_c / P_c)$
	$a = [1 + \kappa * (1 - T_r^{1/2})]^2$
	$\kappa = 0.480 + 1.574 * \omega - 0.176 * \omega^2$
	$\omega = \text{acentric factor}$
<b>Peng-Robinson</b>	$P = (RT / v - b) - (a\alpha / v * (v - b) + b * (v - b))$
	$a = 0.45724 * (R^2 T_c^2 / P_c)$
	$b = 0.07780 * (R T_c / P_c)$
	$a = [1 + \kappa * (1 - T_r^{1/2})]^2$
	$\kappa = 0.37464 + 1.54226 * \omega - 0.26992 * \omega^2$
	$\omega = \text{acentric factor}$
<b>Kammerlingh-Onnes</b>	$Pv = RT * (1 + B/V + C/V^2 + \dots)$
<b>Holborn</b>	$Pv = RT * (1 + B'P + C'P^2 + \dots)$

## 2. Basics of heat transfer

Heat transfer is defined as energy-in-transit due to temperature difference. Heat transfer takes place whenever there is a temperature gradient within a system or whenever two systems at different temperatures are brought into thermal contact. Heat, which is energy-in-transit cannot be measured or observed directly, but the effects produced by it can be observed and measured. Since heat transfer involves transfer and/or conversion of energy, all heat transfer processes must obey the first and second laws of thermodynamics. However unlike thermodynamics, heat transfer deals with systems not in thermal equilibrium and using the heat transfer laws it is possible to find the rate at which energy is transferred due to heat transfer. From the engineer's point of view, estimating the rate of heat transfer is a key requirement. Refrigeration and air conditioning involves heat transfer, hence a good understanding of the fundamentals of heat transfer is a must for a student of refrigeration and air conditioning.

Generally heat transfer takes place in three different modes: conduction, convection and radiation. In most of the engineering problems heat transfer takes place by more than one mode simultaneously.

### 2.1. Conduction heat transfer

Conduction heat transfer takes place whenever a temperature gradient exists in a stationary medium. Conduction is one of the basic modes of heat transfer. On a microscopic level, conduction heat transfer is due to the elastic impact of molecules in fluids, due to molecular vibration and rotation about their lattice positions and due to free electron migration in solids.

The fundamental law that governs conduction heat transfer is called Fourier's law of heat conduction, it is an empirical statement based on experimental observations and is given by:

$$Q_h = -\lambda \cdot A \cdot \frac{dt}{dx} \quad [W],$$

where  $Q_h$  is the rate of heat transfer,  $\lambda$  is the *thermal conductivity* of the substance [W/m·K],  $A$  is the surface [ $m^2$ ] and  $dt/dx$  is the temperature gradient in the  $x$  direction (heat transfer direction). The minus sign shows that the temperature decreases along the direction of heat transfer.

The thermal conductivity of materials varies significantly; generally it is very high for pure metals and low for non-metals. Thermal conductivity of solids is generally greater than that of fluids. Thermal conductivity of solids and liquids vary mainly with temperature, while thermal conductivity of gases depend on both temperature and pressure. For isotropic materials the value of thermal conductivity is same in all directions, while for anisotropic materials such as wood and graphite the value of thermal conductivity is different in different directions. In refrigeration and air conditioning high thermal conductivity materials are used in the construction of heat exchangers, while low thermal conductivity materials are required for insulating refrigerant pipelines, refrigerated cabinets, building walls etc.

For the case of one-dimensional, steady heat conduction through plane wall, with no heat generation (fig. 2.1) the rate of heat transfer is given by the relation:

$$Q_h = A \cdot \frac{\lambda_p}{\delta_p} \cdot (T_{p1} - T_{p2}).$$

For a plane wall containing several layers (fig. 2.) the rate of heat transfer through conduction is:

$$Q_h = A \cdot \frac{t_1 - t_4}{\sum_{i=1}^3 \frac{\delta_i}{\lambda_i}}.$$

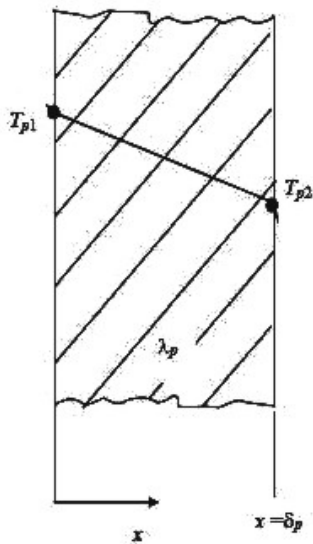


Fig. 2.1 – Schematics for the heat conduction through a plane wall

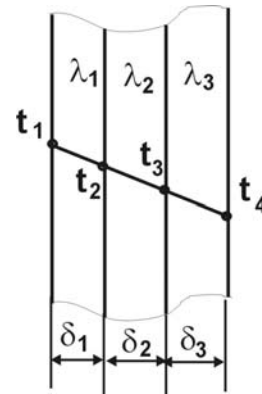


Fig. 2.2 – Heat conduction through a plane wall with multiple layers

## 2.2. Convection heat transfer

Convection heat transfer takes place between a surface and a moving fluid, when they are at different temperatures.

When fluid flows over a surface, its velocity and temperature adjacent to the surface are same as that of the surface due to the no-slip condition. The velocity and temperature far away from the surface may remain unaffected. As a result there is a sharp temperature gradient in this vicinity of the surface if the temperature of the surface of the plate is different from that of the flow stream.

Convection heat transfer takes place according to Newton's law:

$$Q_h = \alpha \cdot A \cdot (T_\infty - T_p),$$

where  $\alpha$  is the convective heat transfer coefficient [ $\text{W}/\text{m}^2 \cdot \text{K}$ ],  $T_\infty$  is the temperature of the fluid far away from the surface and  $T_p$  is the temperature of the surface.

Traditionally, from the manner in which the convection heat transfer rate is defined, evaluating the convective heat transfer coefficient has become the main objective of the problem. The convective heat transfer coefficient can vary widely depending upon the type of fluid and flow field and temperature difference. Evaluation of convective heat transfer coefficient is difficult as the physical phenomenon is quite complex. Analytically, it can be determined by solving the mass, momentum and energy equations. However, analytical solutions are available only for very simple situations, hence most of the convection heat transfer data is obtained through careful experiments, and the equations suggested for convective heat transfer coefficients are mostly empirical. Since the equations are of empirical nature, each equation is applicable to specific cases. Generalization has been made possible to some extent by using several non-dimensional numbers such as Reynolds number, Prandtl number, Nusselt number, Grashoff number, Rayleigh number etc.

## 2.3. Radiation heat transfer

Radiation is another fundamental mode of heat transfer. Unlike conduction and convection, radiation heat transfer does not require a medium for transmission as energy transfer occurs due to the propagation of electromagnetic waves. A body due to its temperature emits electromagnetic radiation.

For a real surface the radiation energy given by Stefan-Boltzmann's law is:

$$Q_h = \varepsilon \cdot \sigma \cdot A \cdot T^4,$$

where  $\varepsilon$  is the emissivity of the surface,  $\sigma=5,669 \cdot 10^{-8}$  [W/m<sup>2</sup>·K] is the Stefan-Boltzmann and T is the absolute temperature of the surface.

The emissivity is a property of the radiating surface and is defined as the emissive power (energy radiated by the body per unit area per unit time over all the wavelengths) of the surface to that of an ideal radiating surface. The ideal radiator is called as a “black body”, whose emissivity is 1.

The radiation heat exchange between any two surfaces 1 and 2 at different temperatures  $T_1$  and  $T_2$  is given by:

$$Q_{h1-2} = F_A \cdot F_\varepsilon \cdot \sigma \cdot A \cdot (T_1^4 - T_2^4),$$

where  $F_\varepsilon$  is the surface optical property factor,  $F_A$  is the geometric shape factor and  $T_1$  and  $T_2$  are the temperatures of the respective surfaces.

## 2.4. Multimodal heat transfer

In most of the practical heat transfer problems heat transfer occurs due to more than one mechanism. An example of this is transfer of heat from outside to the interiors of an air conditioned space (fig. 2.3). There is a convective transfer of heat from room 1 to the wall (which is made up of different layers, having different heat transfer properties), a conductive transfer through the wall and again a convective transfer to room 2. Using the notations from fig. 2.3, the rate of heat transfer from room 1 to room 2 is:

$$Q_h = A \cdot \frac{T_1 - T_2}{\frac{1}{\alpha_1} + \sum_{i=1}^3 \frac{\delta_i}{\lambda_i} + \frac{1}{\alpha_2}} \quad [W],$$

where  $\delta_i$  is the thickness of layer i and  $\lambda_i$  is its conductivity.

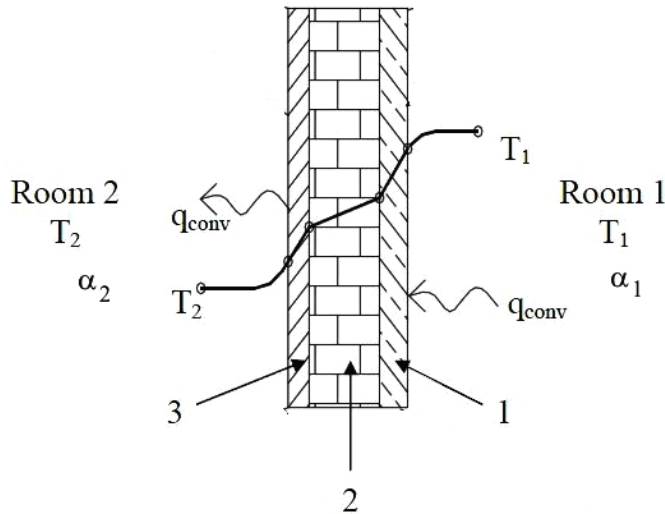


Fig. 2.3. – Multimodal (convection & conduction) heat transfer

1, 2, 3 – layers of the wall;  
 $\alpha_1, \alpha_2$  – convective coefficients.

Using the notation:

$$k = \frac{1}{\frac{1}{\alpha_1} + \sum_{i=1}^3 \frac{\delta_i}{\lambda_i} + \frac{1}{\alpha_2}} \quad \left[ \frac{W}{m^2 \cdot K} \right],$$

the previous relationship becomes:

$$Q_h = A \cdot k \cdot (T_1 - T_2) \quad [W],$$

where k is the overall heat transfer coefficient.

### 3. Compressors

A typical refrigeration system consists of several basic components such as compressors, condensers, expansion devices, evaporators, in addition to several accessories such as controls, filters, driers, oil separators etc.

A compressor is the most important and often the costliest component (typically 30 to 40 percent of total cost) of any vapor compression refrigeration system (VCRS). The function of a compressor in a VCRS is to continuously draw the refrigerant vapor from the evaporator, so that a low pressure and low temperature can be maintained in the evaporator at which the refrigerant can boil extracting heat from the refrigerated space. The compressor then has to raise the pressure of the refrigerant to a level at which it can condense by rejecting heat to the cooling medium in the condenser.

Compressors can be classified in several ways:

a) Based on the working principle:

- Positive displacement type;
- Roto-dynamic type.

In **positive displacement** type compressors, compression is achieved by trapping gas in an enclosed space and then reducing its volume; the pressure rises as the volume is reduced. Since the flow of refrigerant to the compressor is not steady, the positive displacement type compressor is a *pulsating flow device*. However, since the operating speeds are normally very high the flow appears to be almost steady on macroscopic time scale. Since the flow is pulsating on a microscopic time scale, positive displacement type compressors are prone to high wear, vibration and noise level. Depending upon the construction, positive displacement type compressors used in refrigeration and air conditioning can be classified into:

- Reciprocating type;
- Rotary type with sliding vanes (rolling piston type or multiple vane type);
- Rotary screw type (single screw or twin-screw type);
- Orbital compressors;
- Acoustic compressors.

In **roto-dynamic** compressors, the pressure rise is achieved by imparting kinetic energy to a steadily flowing stream of gas by a rotating mechanical element and then converting into pressure as the gas flows through a diverging passage. Unlike positive displacement type, the roto-dynamic type compressors are steady flow devices, hence are subjected to less wear and vibration. Depending upon the construction, roto-dynamic type compressors can be classified into:

- Radial flow type;
- Axial flow type.

Centrifugal compressors (also known as turbo-compressors) are radial flow type, roto-dynamic compressors. These compressors are widely used in large capacity refrigeration and air conditioning systems. Axial flow compressors are normally used in gas liquefaction applications.

b) Based on arrangement of compressor motor or external drive:

- Open type;
- Hermetic (or sealed) type;
- Semi-hermetic (or semi-sealed) type.

In open type compressors the rotating shaft of the compressor extends through a seal in the crankcase for an external drive. The external drive may be an electrical motor or an engine (e.g. diesel engine). The compressor may be belt driven or gear driven. Open type compressors are normally used in medium to large capacity refrigeration systems, for all refrigerants and for ammonia (due to its incompatibility with hermetic motor materials). Open type compressors are characterized by high efficiency, flexibility, better compressor cooling and serviceability. However, since the shaft has to extend through the seal, refrigerant leakage from the system cannot be eliminated completely. Hence refrigeration systems using open type compressors

require a refrigerant reservoir to take care of the refrigerant leakage for some time, and then regular maintenance for charging the system with refrigerant, changing of seals, gaskets etc.

In hermetic compressors, the motor and the compressor are enclosed in the same housing to prevent refrigerant leakage. The housing has welded connections for refrigerant inlet and outlet and for power input socket. As a result of this, there is virtually no possibility of refrigerant leakage from the compressor. All motors reject a part of the power supplied to it due to eddy currents and friction, that is, inefficiencies. Similarly the compressor also gets heated-up due to friction and also due to temperature rise of the vapor during compression. In Open type, both the compressor and the motor normally reject heat to the surrounding air for efficient operation. In hermetic compressors heat cannot be rejected to the surrounding air since both are enclosed in a shell. Hence, the cold suction gas is made to flow over the motor and the compressor before entering the compressor. This keeps the motor cool. The motor winding is in direct contact with the refrigerant, hence only those refrigerants which have high dielectric strength can be used in hermetic compressors. The cooling rate depends upon the flow rate of the refrigerant, its temperature and the thermal properties of the refrigerant. If flow rate is not sufficient and/or if the temperature is not low enough the insulation on the winding of the motor can burn out and short-circuiting may occur. Hence, hermetically sealed compressors give satisfactory and safe performance over a very narrow range of design temperature and should not be used for off-design conditions.

The COP of the hermetic compressor based systems is lower than that of the open compressor based systems since a part of the refrigeration effect is lost in cooling the motor and the compressor. However, hermetic compressors are almost universally used in small systems such as domestic refrigerators, water coolers, air conditioners etc, where efficiency is not as important as customer convenience (due to absence of continuous maintenance). In addition to this, the use of hermetic compressors is ideal in systems, which use capillary tubes as expansion devices and are critically charged systems. Hermetic compressors are normally not serviceable. They are not very flexible as it is difficult to vary their speed to control the cooling capacity.

In some (usually larger) hermetic units, the cylinder head is usually removable so that the valves and the piston can be serviced. This type of unit is called a semi-hermetic (or semi-sealed) compressor.

### ***3.1. Reciprocating compressors***

The construction and operating principle of the reciprocating compressor are shown in fig. 3.1.

The reciprocating movement of the piston (P) is achieved by the means of the crankshaft (CS) and the connecting rod (CR); this mechanism transforms the rotation motion of the crankshaft in reciprocating motion of the piston.

The piston moves between two extreme points: the BDC and the TDC; the volume described by the piston between these points is called displacement ( $V_s$ ).

The suction (Iv) and discharge (Ev) valves are placed in the cylinder head; the suction valve allows the flow of gas towards the cylinder (when the piston moves from TDC to BDC), while the discharge valve allows the out of the cylinder (when the piston moves from BDC to TDC).

The compression is achieved when the piston moves from BDC to TDC (process 1-2), both valves being closed. When the pressure is high enough, it opens the discharge valve and the gas flows out from the cylinder.

A small clearance ( $V_0$ ) is left between the cylinder head and piston at TDC to accommodate the valves and to take care of thermal expansion and machining tolerances. When the piston moves from TDC to BDC, the gas trapped in the clearance volume expands (process 3-4) and only afterwards the pressure inside the cylinder becomes low enough to allow the opening of the suction valve. As a result, the volume of gas that enters the cylinder is lower than



the displacement ( $V_a < V_s$ ).

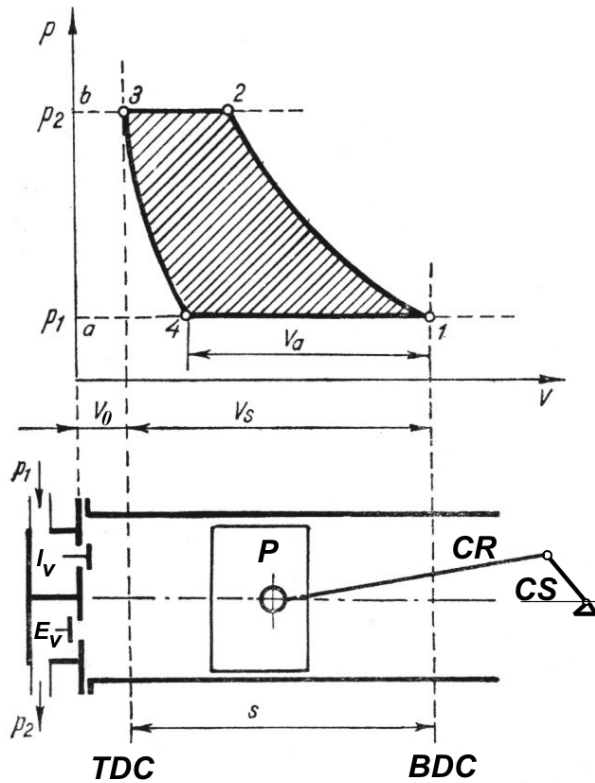


Fig. 3.3 – Construction and operating cycle of the reciprocating compressor

- $V_0$ -clearance volume;
- $V_a$ -swept volume;
- $V_s$ -displacement;
- TDC-top dead centre;
- BDC-bottom dead centre;
- $I_v$ -suction valve;
- $E_v$ -discharge valve;
- P-piston;
- CR-connecting rod;
- CS-crankshaft.

The main operating parameters of the reciprocating compressor are:

- the displacement:

$$V_s = \frac{\pi \cdot D^2}{4} \cdot S \quad [m^3],$$

where D is the piston bore and S is the cylinder displacement;

- theoretical flow rate:

$$Q_t = V_s \cdot i \cdot \frac{n}{60} \quad [m^3 / s],$$

where i is the number of cylinders and n is the rotation speed of the crankshaft [rot/min];

- volumetric efficiency:  $\eta_v = V_a / V_s$ ;
- clearance ratio:  $\sigma = V_0 / V_s$ ;
- pressure ratio:  $\varepsilon = p_2 / p_1$ ;
- real flow rate:  $Q_r = \eta_v \cdot Q_t$ .

The clearance ratio  $\sigma$  depends on the arrangement of the valves in the cylinder and the mean piston velocity. Normally  $\varepsilon$  is less than 5 percent for well designed compressors with moderate piston velocities ( $\approx 3$  m/s), however, it can be higher for higher piston speeds.

Fig. 3.4 shows the effect of the clearance volume over the operating cycle of the compressor, for different discharge pressures ( $p_2 < p'_2 < p_{max}$ ). It is obvious that the swept volume decreases when the discharge pressure increases ( $V_a > V'_a$ ) and this means that the real flow rate of the compressor decreases when the discharge pressure increases. In the most extreme case, when the discharge pressure is  $p_{max}$ , the swept volume is zero and there is no flow of gas from the compressor. For this reason there is a limit for the pressure ratio ( $\varepsilon = 6 \dots 8$ ) and the principle of two stage compression is used when higher pressure ratios are needed (in a refrigeration system higher pressure ratios usually mean lower suction pressures and hence lower vaporization temperatures).

Fig. 3.5. presents the principle of the two stage compression; we notice that there are two compressors placed in series, and the gas discharged from the first stage (I), at the intermediary

pressure  $p_x$ , is swept into the secondary stage (II).

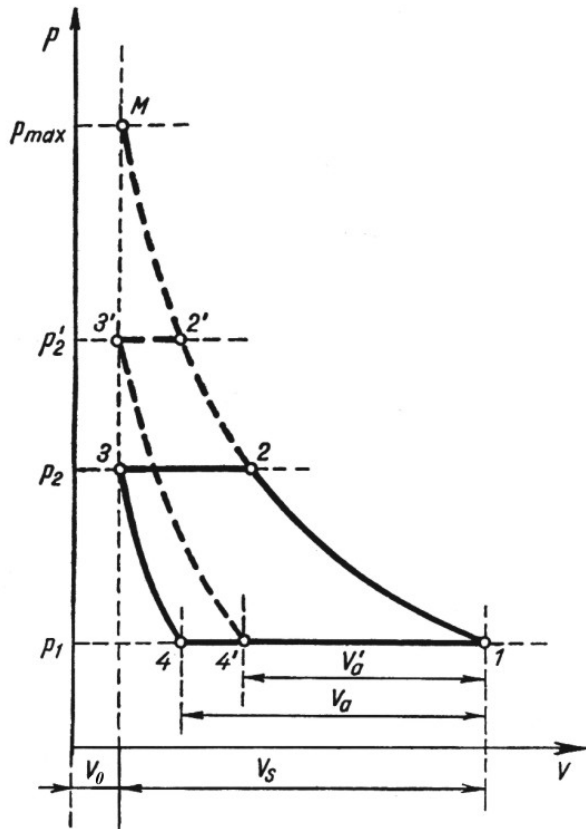
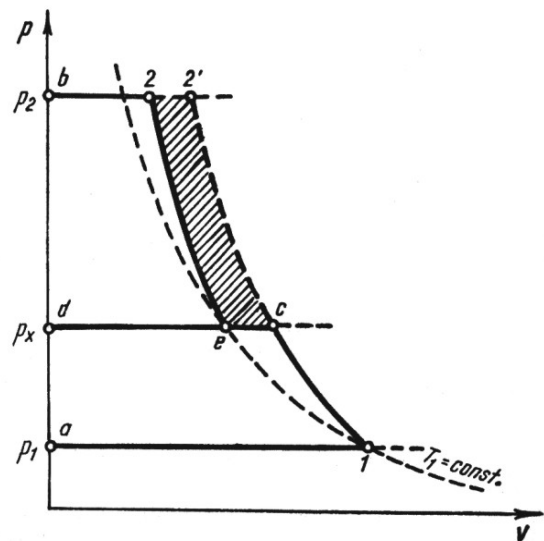
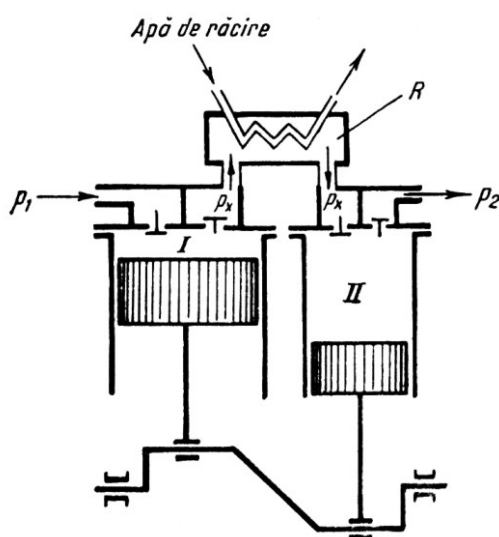


Fig. 3.4 – Effect of clearance volume and discharge pressure



a)

b)

Fig. 3.5 – Two stage compressor

I, II-compression stages; R-intermediary cooler;  $p_x$ -intermediary pressure

On the cycle diagram in fig. 3.5b, 1-c is the compression process in the first stage, c-e is the intermediary cooling at constant pressure and e-2 is the compression process for the second stage; 1-2' is the one stage compression, for the same pressure ratio  $p_2/p_1$ . The two stage compression requires less mechanical work than the single stage compression (the hashed surface is proportional with the decrease of the mechanical work); moreover, the discharge temperature in point 2 is lower than the discharge temperature in point 2' (one stage compression).

Assuming the same pressure ration for the both compression stages, the intermediary

pressure should be:

$$p_x = \sqrt{p_1 \cdot p_2} .$$

### 3.2. Other positive displacement refrigeration compressors

#### 3.2.1. Rolling piston (fixed vane) type compressors

Rolling piston or fixed vane type compressors are used in small refrigeration systems (up to 2 kW capacity) such as domestic refrigerators or air conditioners. These compressors belong to the class of positive displacement type as compression is achieved by reducing the volume of the refrigerant. In this type of compressors, the rotating shaft of the roller has its axis of rotation that matches with the centerline of the cylinder, however, it is eccentric with respect to the roller (fig. 3.6). This eccentricity of the shaft with respect to the roller creates suction and compression of the refrigerant as shown in fig. 3.6. A single vane or blade is positioned in the non-rotating cylindrical block. The rotating motion of the roller causes a reciprocating motion of the single vane.

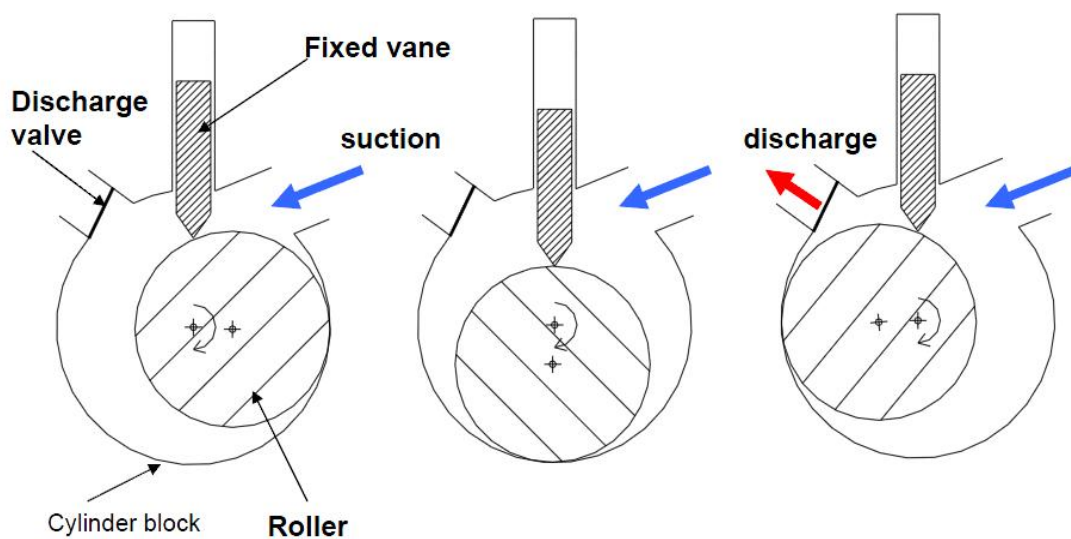


Fig. 3.6 - Working principle of a rolling piston type compressor

This type of compressor does not require a suction valve but requires a discharge valve. The sealing between the high and low pressure sides has to be provided:

- along the line of contact between roller and cylinder block;
- along the line of contact between vane and roller;
- between the roller and end-pates .

The leakage is controlled through hydrodynamic sealing and matching between the mating components. The effectiveness of the sealing depends on the clearance, compressor speed, surface finish and oil viscosity. Close tolerances and good surface finishing is required to minimize internal leakage.

Unlike in reciprocating compressors, the small clearance volume filled with high-pressure refrigerant does not expand, but simply mixes with the suction refrigerant in the suction space. As a result, the volumetric efficiency does not reduce drastically with increasing pressure ratio, indicating small re-expansion losses. The compressor runs smoothly and is relatively quiet as the refrigerant flow is continuous.

#### 3.2.2. Multiple vane compressors

As shown in fig. 3.7, in multiple vane type compressor, the axis of rotation coincides with the center of the roller (O), however, it is eccentric with respect to the center of the cylinder (O'). The rotor consists of a number of slots with sliding vanes. During the running of the compressor,

the sliding vanes, which are normally made of non-metallic materials, are held against the cylinder due to centrifugal forces. The number of compression strokes produced in one revolution of the rotor is equal to the number of sliding vanes, thus a 4-vane compressor produces 4 compression strokes in one rotation.

In these compressors, sealing is required between the vanes and cylinder, between the vanes and the slots on the rotor and between the rotor and the end plate. However, since pressure difference across each slot is only a fraction of the total pressure difference, the sealing is not as critical as in fixed vane type compressor.

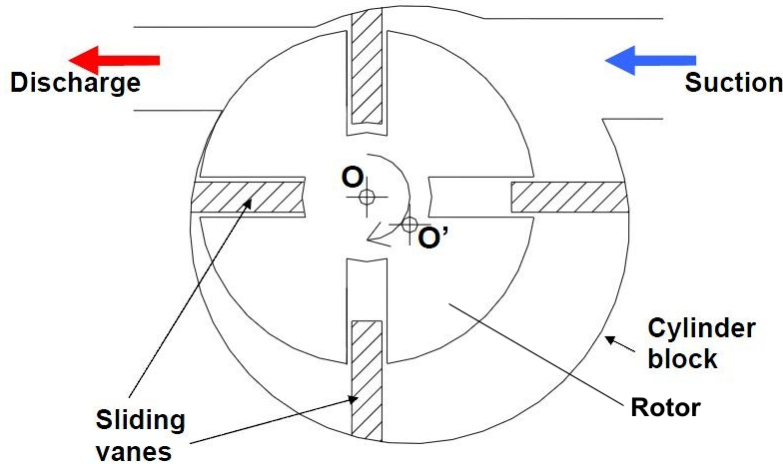


Fig. 3.7 – Working principle of a multiple vane, rotary compressor

### 3.2.3. Rotary, screw compressors

The rotary screw compressors can be either twin-screw type or single-screw type.

#### a) Twin-screw compressor

The twin-screw type compressor consists of two mating helically grooved rotors, one male and the other female. Generally the male rotor drives the female rotor. The male rotor has lobes, while the female rotor has flutes or gullies. The frequently used lobe-gully combinations are [4,6], [5,6] and [5,7]. Fig. 3.8 shows the [4,6] combination. For this [4,6] combination, when the male rotor rotates at 3600 RPM, the female rotor rotates at 2400 RPM.

As shown in fig. 3.9, the flow is mainly in the axial direction. Suction and compression take place as the rotors unmesh and mesh. When one lobe-gully combination begins to unmesh the opposite lobe-gully combination begins to mesh. With 4 male lobes rotating at 3600 RPM, 4 interlobe volumes are per revolution, thus giving  $4 \times 3600 = 14400$  discharges per minute.

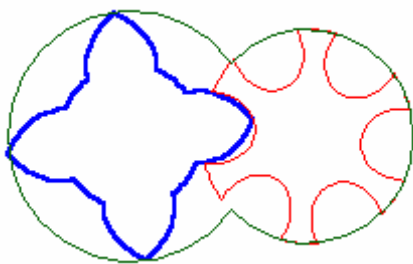


Fig. 3.8 - Twin-screw compressor with 4 male lobes and 6 female gullies

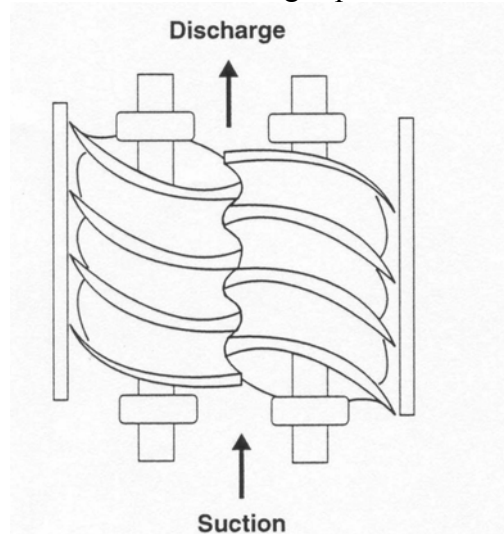


Fig. 3.9 – Direction of refrigerant flow in a twin-screw compressor

Discharge takes place at a point decided that depends entirely on the location of the

delivery port and geometry of the compressor. Since the built-in volume ratio is fixed by the geometry, a particular compressor is designed for a particular built-in pressure ratio. However, different built-in ratios can be obtained by changing the position of the discharge port.

Lubrication and sealing between the rotors is obtained by injecting lubricating oil between the rotors. The oil also helps in cooling the compressor, as a result very high pressure ratios (upto 20:1) are possible without overheating the compressor.

The capacity of the screw compressor is normally controlled with the help of a slide valve. As the slide valve is opened, some amount of suction refrigerant escapes to the suction side without being compressed. This yields a smooth capacity control from 100 percent down to 10 percent of full load. It is observed that the power input is approximately proportional to refrigeration capacity up to about 30 percent, however, the efficiency decreases rapidly, there after.

As the rotor normally rotates at high speeds, screw compressors can handle fairly large amounts of refrigerant flow rates compared to other positive displacement type compressors. Screw compressors are available in the capacity range of 70 to 4600 kW. They generally compete with high capacity reciprocating compressors and low capacity centrifugal compressors. They are available for a wide variety of refrigerants and applications. Compared to reciprocating compressors, screw compressors are balanced and hence do not suffer from vibration problems.

#### *b) Single-screw compressors*

As the name implies, single screw compressors consist of a single helical screw and two planet wheels or gate rotors. The helical screw is housed in a cylindrical casing with suction port at one end and discharge port at the other end as shown in fig. 3.10. Suction and compression are obtained as the screw and gate rotors un-mesh and mesh. The high and low pressure regions in the cylinder casing are separated by the gate rotors.

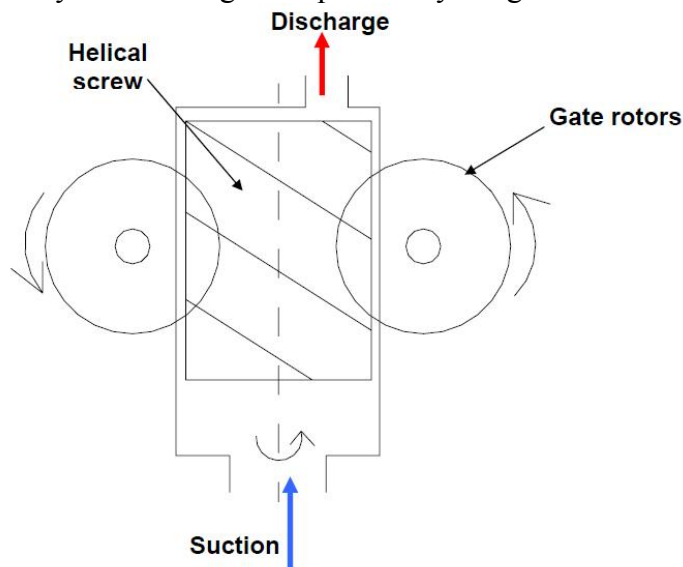


Fig. 3.10 – Single-screw compressor

The single screw is normally driven by an electric motor. The gate rotors are normally made of plastic materials. Very small power is required to rotate the gate rotors as the frictional losses between the metallic screw and the plastic gate rotors is very small. It is also possible to design the compressors with a single gate rotor. Similar to twin-screw, lubrication, sealing and compressor cooling is achieved by injecting lubricating oil into the compressor. An oil separator, oil cooler and pump are required to circulate the lubricating oil. It is also possible to achieve this by injecting liquid refrigerant, in which case there is no need for an oil separator.

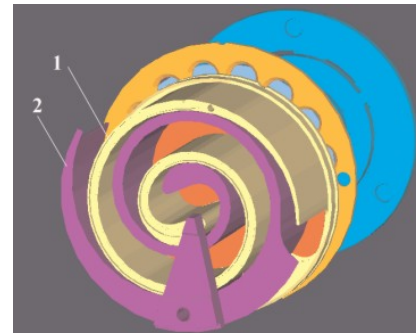
#### *3.2.4. Scroll compressors*

Scroll compressors are orbital motion, positive displacement type compressors, in which suction and compression is obtained by using two mating, spiral shaped, scroll members, one fixed and the other orbiting (fig. 3.11).

The refrigerant is forced into progressively smaller chambers towards the center (fig. 3.12). The compressed refrigerant is then discharged through the center of the fixed scroll. No inlet or exhaust valves are needed.



a)



c)



b)

Fig. 3.11 – Scroll compressor

a-fixed scroll;  
b-orbiting scroll;  
c-assembled compressor:  
1- fixed scroll;  
2- orbiting scroll.

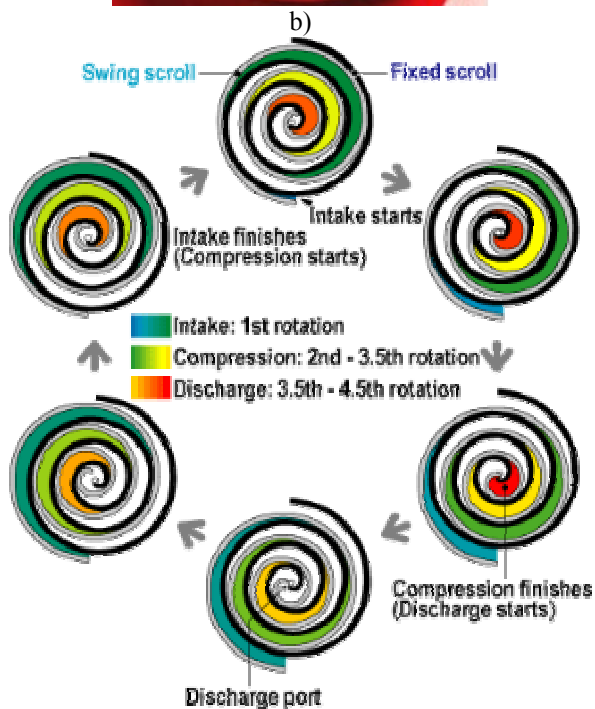


Fig. 3.12 – Operating principle of the scroll compressor

Currently, the scroll compressors are used in small capacity (3 to 50 kW) refrigeration, air conditioning and heat pump applications. They are normally of hermetic type. Scroll compressors offer several advantages such as:

- large suction and discharge ports reduce pressure losses during suction and discharge;
- physical separation of suction and compression reduce heat transfer to suction gas, leading to high volumetric efficiency;
- volumetric efficiency is also high due to very low re-expansion losses and continuous flow over a wide range of operating conditions;
- high compression efficiency, low noise and vibration compared to reciprocating compressors ;

- compact, with minimum number of moving parts.  
There are also some disadvantages to be taken into account:
- low capacity (flow rate);
- relatively expensive;
- the compressed refrigerant can get very hot, much hotter than compared to other types of compressors.

### 3.3. Centrifugal compressors

Centrifugal compressors, also known as turbo-compressors, belong to the roto-dynamic type of compressors. In these compressors the required pressure rise takes place due to the continuous conversion of angular momentum imparted to the refrigerant vapor by a high-speed impeller into static pressure. Unlike reciprocating compressors, centrifugal compressors are steady-flow devices hence they are subjected to less vibration and noise.

Figure 3.13 shows the working principle of a centrifugal compressor. As shown in the figure, low-pressure refrigerant enters the compressor through the eye of the impeller (1). The impeller (2) consists of a number of blades, which form flow passages (3) for refrigerant. From the eye, the refrigerant enters the flow passages formed by the impeller blades, which rotate at very high speed. As the refrigerant flows through the blade passages towards the tip of the impeller, it gains momentum and its static pressure also increases. From the tip of the impeller, the refrigerant flows into a stationary diffuser (4). In the diffuser, the refrigerant is decelerated and as a result the dynamic pressure drop is converted into static pressure rise, thus increasing the static pressure further. The vapor from the diffuser enters the volute casing (5) where further conversion of velocity into static pressure takes place due to the divergent shape of the volute. Finally, the pressurized refrigerant leaves the compressor from the volute casing (6).

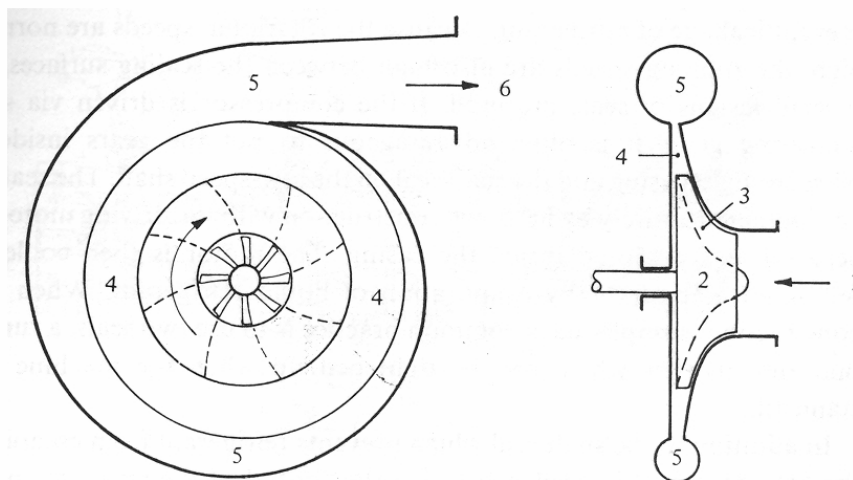


Fig. 3.13 – Operating principle of the centrifugal compressor

- 1-refrigerant inlet;
- 2-impeller;
- 3-refrigerant passages between the blades;
- 4- diffuser;
- 5-volute casing;
- 6-refrigerant discharge.

The gain in momentum is due to the transfer of momentum from the high-speed impeller blades to the refrigerant confined between the blade passages. The increase in static pressure is due to the self-compression caused by the centrifugal action.

Thus it can be seen that for a given refrigerant with a fixed density, the pressure rise depends only on the peripheral velocity or tip speed of the blade. The tip speed of the blade is proportional to the rotational speed (RPM) of the impeller and the impeller diameter. The maximum permissible tip speed is limited by the strength of the structural materials of the blade (usually made of high speed chrome-nickel steel) and the sonic velocity of the refrigerant. Under these limitations, the maximum achievable pressure rise (hence maximum achievable temperature lift) of single stage centrifugal compressor is limited for a given refrigerant. Hence,

multistage centrifugal compressors are used for large temperature lift applications. In multistage centrifugal compressors, the discharge of the lower stage compressor is fed to the inlet of the next stage compressor and so on. In multistage centrifugal compressors, the impeller diameter of all stages remains same, but the width of the impeller becomes progressively narrower in the direction of flow as refrigerant density increases progressively.

The refrigerant capacity of a centrifugal compressor depends primarily on the tip speed and width of the impeller.

Commercially both hermetic as well as open type centrifugal compressors are available. Open type compressors are driven by electric motors, internal combustion engines (using a wide variety of fuels) or even steam turbines.



## 4. Refrigeration systems

Refrigeration may be defined as the process of achieving and maintaining a temperature below that of the surroundings, the aim being to cool some product or space to the required temperature. One of the most important applications of refrigeration has been the preservation of perishable food products by storing them at low temperatures. Refrigeration systems are also used extensively for providing thermal comfort to human beings by means of air conditioning. Air Conditioning refers to the treatment of air so as to simultaneously control its temperature, moisture content, cleanliness, odour and circulation, as required by occupants, a process, or products in the space. The subject of refrigeration and air conditioning has evolved out of human need for food and comfort, and its history dates back to centuries.

Among other *natural cooling* methods (use of ice, use of certain salt solution) *evaporative cooling* is one of the oldest. As the name indicates, evaporative cooling is the process of reducing the temperature of a system by evaporation of water. Human beings perspire and dissipate their metabolic heat by evaporative cooling if the ambient temperature is more than skin temperature. Animals such as the hippopotamus and buffalo coat themselves with mud for evaporative cooling. Evaporative cooling has been used in India for centuries to obtain cold water in summer by storing the water in earthen pots. The water permeates through the pores of earthen vessel to its outer surface where it evaporates to the surrounding, absorbing its latent heat in part from the vessel, which cools the water. It is said that Patliputra University situated on the bank of river Ganges used to induce the evaporative-cooled air from the river. Suitably located chimneys in the rooms augmented the upward flow of warm air, which was replaced by cool air. Evaporative cooling by placing wet straw mats on the windows is also very common in India.

Refrigeration, as it is known these days, is produced by *artificial means*. Though it is very difficult to make a clear demarcation between natural and artificial refrigeration, it is generally agreed that the history of artificial refrigeration began in the year 1755, when the Scottish professor William Cullen made the first refrigerating machine, which could produce a small quantity of ice in the laboratory. Based on the working principle, refrigeration systems can be classified as *vapor compression systems*, *vapor absorption systems*, *gas cycle systems* etc.

The main goal of a refrigeration system which performs the reverse effect of a heat engine is to remove the heat from a low-level temperature medium (*heat source*) and to transfer this heat to a higher level temperature medium (*heat sink*). Figure 4.1 shows a thermodynamic system acting as a refrigeration machine. The absolute temperature of the source is  $T_L$  and the heat transferred from the source is the refrigeration effect (refrigeration load)  $Q_L$ . On the other side, the heat rejection to the sink at the temperature  $T_H$  is  $Q_H$ . Both effects are accomplished by the work input  $W$ . For continuous operation, the first law of thermodynamics is applied to the system.

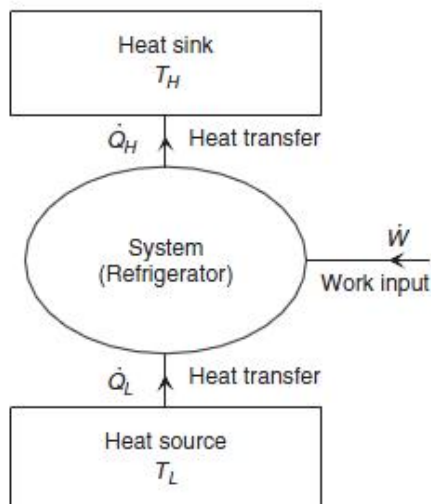


Fig. 4.1 - A thermodynamic system acting as a refrigerator

#### 4.1. Vapor compression refrigeration systems

The basis of modern refrigeration is the ability of liquids to absorb enormous quantities of heat as they boil and evaporate. Professor William Cullen of the University of Edinburgh demonstrated this in 1755 by placing some water in thermal contact with ether under a receiver of a vacuum pump. The evaporation rate of ether increased due to the vacuum pump and water could be frozen. This process involves two thermodynamic concepts, the vapor pressure and the latent heat. A liquid is in thermal equilibrium with its own vapor at a pressure called the saturation pressure, which depends on the temperature alone. If the pressure is increased for example in a pressure cooker, the water boils at higher temperature. The second concept is that the evaporation of liquid requires latent heat during evaporation. If latent heat is extracted from the liquid, the liquid gets cooled. The temperature of ether will remain constant as long as the vacuum pump maintains a pressure equal to saturation pressure at the desired temperature. This requires the removal of all the vapors formed due to vaporization. If a lower temperature is desired, then a lower saturation pressure will have to be maintained by the vacuum pump. The component of the modern day refrigeration system where cooling is produced by this method is called *evaporator*.

If this process of cooling is to be made continuous the vapors have to be recycled by condensation to the liquid state. The condensation process requires heat rejection to the surroundings. Vapors can be condensed at atmospheric temperature by increasing their pressure. The process of condensation was learned in the second half of eighteenth century. U.F. Clouet and G. Monge liquefied  $\text{SO}_2$  in 1780, while van Marum and Van Troostwijk liquefied  $\text{NH}_3$  in 1787. Hence, a *compressor* is required to maintain a high pressure so that the vapors can condense at a temperature greater than that of the surroundings. The condensation takes place in an *evaporator*.

In order to maintain the pressure difference between the condenser and the evaporator a throttling device is connected between them; in the *throttling (expansion) device* the pressure drops from the condensation pressure to the evaporation pressure.

Fig. 4.2 presents the basic diagram of a vapor compression refrigeration system; fig. 4.3 presents the operating cycle of the system. The refrigerant evaporates in the evaporator (V), at low pressure and low temperature (process 4-1, fig. 4.3), and absorbs the latent heat from the respective cooling space. The pressure of the vapors is raised by the compressor (process 1-2, fig. 4.3 – isentropic compression, in an ideal system). In the condenser the vapors cool from the discharge temperature to the condensation temperature (process 2-2’); then the condensation takes place (process 2’-3, at constant temperature and pressure) and saturated liquid (point 3) leaves the condenser. Inside the expansion valve the pressure drops from  $p_k$  (condensation pressure) to  $p_0$  (evaporation pressure).

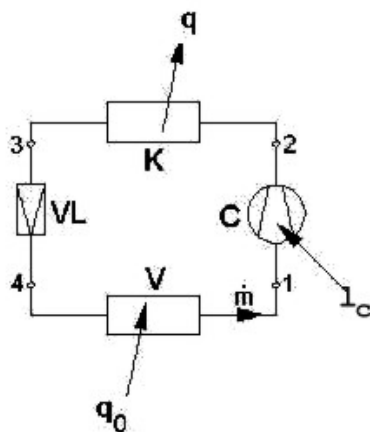


Fig. 4.2 – Standard vapor compression refrigeration system

V-evaporator;  
C-compressor;  
K-condenser;  
VL-expansion valve.

The efficiency of the vapor compression refrigeration system is:

$$COP = \frac{i_1 - i_4}{i_2 - i_1}$$

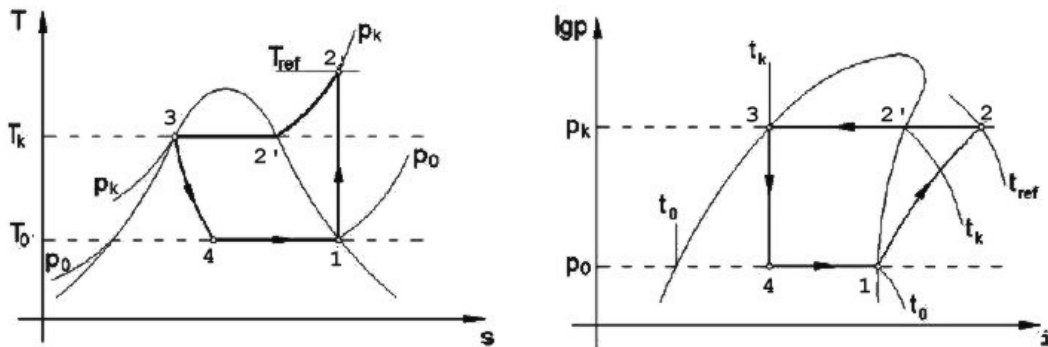


Fig. 4.3 – Operating cycle of the vapor compression refrigeration system

$T_0$ -evaporation temperature;  $T_k$ -condensing temperature  $T_{ref}$ -discharge temperature;  $p_0$ -evaporation pressure;  $p_k$ -condensing temperature.

## 4.2. Refrigerants

The requirements for the working fluid are as follows:

1. high latent heat of vaporization;
2. high density of suction gas;
3. non-corrosive, non-toxic and non-flammable; compatibility with materials of construction, with lubricating oils and with other materials present in the system; environmentally friendly;
4. critical temperature and triple point outside the working range;
6. convenient working pressures, i.e. not too high and preferably not below atmospheric pressure;
7. high dielectric strength (for hermetical compressors).

No single working fluid has all these properties and for this reason a great variety of chemicals substances were used over the years. The present situation is dominated by the need for fluids which are environmentally friendly.

The first designers of refrigeration machines, Jacob Perkins in 1834, and others later in the nineteenth century, used ethyl ether (R-610) as the first commercial refrigerant. Other and more appropriate refrigerants, for example, ammonia (R-717), carbon dioxide (R-744), ethyl chloride (R-160), isobutane (R-600a), methyl chloride (R-40), methylene chloride (R-30), and sulfur dioxide (R-764), were soon introduced, including air (R-729). Three of these refrigerants became very popular, that is, ammonia and sulfur dioxide for refrigerators and other small units and carbon dioxide preferably for ships' refrigeration. A large number of substances were tried over the following years, with varying success.

In the early 1930s, the introduction of chlorofluorocarbons (CFCs) was revolutionary as compared with the natural substances. In addition to their use as refrigerants in refrigeration and air-conditioning systems, CFCs were utilized as foam-blowing agents, aerosol propellants, and cleaning solvents since 1950.

Refrigerants be classified into the following categories:

- halocarbons;
- hydrocarbons (HCs);
- inorganic compounds;
- azeotropic mixtures;
- non-azeotropic mixtures.

### 4.2.1 Halocarbons

The halocarbons contain one or more of the three halogens – chlorine, fluorine, or bromine – and are widely used in refrigeration and air-conditioning systems as refrigerants.

These are more commonly known by their trade names, such as Freon, Arcton, Genetron, Isotron, and Uron. Numerical indication is preferable in practice.

In this group, the halocarbons, consisting of chlorine, fluorine, and carbon, were the most commonly used refrigerants (so-called *chlorofluorocarbons*, *CFCs*). CFCs were commonly used as refrigerants, solvents, and foam-blowing agents. The most common CFCs have been CFC-11 or R-11, CFC-12 or R-12, CFC-113 or R-113, CFC-114 or R-114, and CFC-115 or R-115. Although CFCs such as R-11, R-12, R-22, R-113, and R-114 were very common refrigerants in refrigeration and air-conditioning equipment, they were used in several industries as aerosols, foams, solvents, etc. Their use rapidly decreased, because of their environmental impact. In the past decade CFC phaseout in refrigeration became a primary political issue as well as, technically speaking, a more and more difficult problem. In addition to ozone layer depletion, the refrigeration and air-conditioning industry faces another problem – the increase in the greenhouse effect.

#### 4.2.2 Hydrocarbons

HCs are the compounds that mainly consist of carbon and hydrogen. HCs include methane, ethane, propane, cyclopropane, butane, and cyclopentane. Although HCs are highly flammable, they may offer advantages as alternative refrigerants because they are inexpensive to produce and have zero ozone depletion potential (ODP), very low global warming potential (GWP), and low toxicity.

There are several types of HC families such as the following:

- Hydrobromofluorocarbons (HBFCs) are the compounds that consist of hydrogen, bromine, fluorine, and carbon.
- HCFCs are the compounds that consist of hydrogen, chlorine, fluorine, and carbon. The HCFCs are one class of chemicals being used to replace the CFCs. They contain chlorine and thus deplete stratospheric ozone, but to a much lesser extent than CFCs. HCFCs have ODPs ranging from 0.01 to 0.1. Production of HCFCs with the highest ODPs will be phased out first, followed by other HCFCs.
- Hydrofluorocarbons (HFCs) are the compounds that consist of hydrogen, fluorine, and carbon. These are considered a class of replacements for CFCs, because of the fact that they do not contain chlorine or bromine and do not deplete the ozone layer. All HFCs have an ODP of 0. Some HFCs have high GWPs. HFCs are numbered according to a standard scheme.
- Methyl bromide ( $\text{CH}_3\text{Br}$ ) is a compound consisting of carbon, hydrogen, and bromine. It is an effective pesticide and is used to fumigate soil and many agricultural products. Because it contains bromine, it depletes stratospheric ozone and has an ODP of 0.6. Its production is banned in several countries, for example, in the United States since the end of December 2000.
- Methyl chloroform ( $\text{CH}_3\text{CCl}_3$ ) is a compound consisting of carbon, hydrogen, and chlorine. It is used as an industrial solvent. Its ODP is 0.11.

For refrigeration applications, a number of HCs such as methane (R-50), ethane (R-170), propane (R-290), *n*-butane (R-600), and isobutane (R-600a) that are suitable as refrigerants can be used.

#### 4.2.3 Inorganic Compounds

In spite of the early invention of many inorganic compounds, today they are still used in many refrigeration, air conditioning, and heat pump applications as refrigerants. Some examples are ammonia ( $\text{NH}_3$ ), water ( $\text{H}_2\text{O}$ ), air, carbon dioxide ( $\text{CO}_2$ ), and sulfur dioxide ( $\text{SO}_2$ ). Among these compounds, ammonia has received the greatest attention for practical applications and, even today, is of interest. Below, we will briefly focus on three compounds of this family – ammonia, carbon dioxide, and air.

##### a) Ammonia (R-717)

Ammonia is a colorless gas with a strong pungent odor which may be detected at low levels (e.g., 0.05 ppm). Liquid ammonia boils at atmospheric pressure at  $-33\text{ }^{\circ}\text{C}$ . The gas is lighter than air and very soluble in water. Despite its high thermal capability to provide cooling, it may cause several technical and health problems.

Despite its disadvantages, that ammonia is considered an excellent refrigerant and these possible disadvantages can be eliminated with proper design and control of the refrigeration system.

#### **b) Carbon Dioxide (R-744)**

Carbon dioxide is one of the oldest inorganic refrigerants. It is a colorless, odorless, nontoxic, nonflammable, and non-explosive refrigerant and can be used in cascade refrigeration systems and in dry-ice production, as well as in food freezing applications.

#### **c) Air (R-729)**

Air is generally used in aircraft air conditioning and refrigeration systems. Its coefficient of performance (COP) is low because of the light weight of the air system. In some refrigeration plants, it may be used in the quick freezing of food products.

### *4.2.4 Azeotropic Mixtures*

An azeotropic refrigerant mixture consists of two substances having different properties but behaving as a single substance. The two substances cannot be separated by distillation. The most common azeotropic refrigerant is R-502, which contains 48.8% R-22 and 51.2% R-115. Its COP is higher than that of R-22 and its lesser toxicity provides an opportunity to use this refrigerant in household refrigeration systems and the food refrigeration industry. Some other examples of azeotropic mixtures are R-500 (73.8% R-12 + 26.2% R-152a), R-503 (59.9% R-13 + 40.1% R-23), and R-504 (48.2% R-32 + 51.8% R-115).

### *4.2.5 Non-azeotropic Mixtures*

Non-azeotropic mixture is a fluid consisting of multiple components of different volatiles that, when used in refrigeration cycles, change composition during evaporation (boiling) or condensation.

Recently, non-azeotropic mixtures have been called *zeotropic mixtures* or *blends*. The application of non-azeotropic mixtures as refrigerants in refrigeration systems has been proposed since the beginning of the twentieth century. A great deal of research on these systems with non-azeotropic mixtures and on their thermo-physical properties has been done since that time. Great interest has been shown in non-azeotropic mixtures, especially for heat pumps, because their adaptable composition offers a new dimension in the layout and design of vapor-compression systems. Much work has been done since the first proposal to use these fluids in heat pumps.

Through the energy crises in the 1970s, non-azeotropic mixtures became more attractive in research and development on advanced vapor-compression heat pump systems. They offered the following advantages:

- energy improvement and saving,
- capacity control, and
- adaptation of hardware components regarding capacity and applications limits.

In the past, studies showed that widely used refrigerants such as R-11, R-12, R-22, and R-114 became most popular for the pure components of the non-azeotropic mixtures. Although many non-azeotropic mixtures (e.g., R-11 + R-12, R-12 + R-22, R-12 + R-114, R-13B1 + R-152a, R-22 + R-114, and R-114 + R-152a, etc.) were well known, a decade ago research and development mainly focused on three mixtures, R-12 + R-114, R-22 + R-114, and R-13B1 + R-152a.

### *4.2.6. Replacement of chlorine based refrigerants in commercial refrigeration applications<sup>2</sup>*

Fig. 4.4 presents the main refrigerants taken into account for commercial refrigeration

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<sup>2</sup> <http://www.emersonclimate.com/en-us/WhitePapers/ECT2005ECT162.pdf>

applications.

Snapped with HyperSnap-DX <a href="http://www.hyperionics.com">http://www.hyperionics.com</a>		GWP
R-12	CFC	10900
R-502	CFC	4700
R-22	HCFC	1810
R-134a	HFC	1430
R-404A	HFC	3922
R-407A	HFC	2107
R-407C	HFC	1774
R-410A	HFC	2088
R-422A	HFC	3143
R-422D	HFC	2729
R-32	HFC	675
R-1234yf	HFO	4
HC-290	Propane	20
R-717	Ammonia	0
R-744	Carbon Dioxide	1

Fig. 4.4. Refrigerants for commercial applications

gas release. Current technology has reduced greenhouse gas discharged by more than 80 percent since 1990. Projections show that by 2050, HFC emissions will make up less than two percent of potential future contributions for all greenhouse gases, as identified in the Kyoto Protocol.

The HFCs and equipment being produced for refrigeration appear to be satisfactory for these applications; however, there are several areas in which they differ from the refrigerants they are replacing:

- They require polyol ester (POE) oil rather than mineral oil.
- Most of the HFCs are mixtures, which can behave differently than pure compounds under some conditions.
- HFCs have higher vapor pressures than the refrigerants they are replacing, which can affect the settings of controls, valves and safety devices.

#### ***R-134a***

R-134a is the first non-ozone-depleting fluorocarbon refrigerant to be commercialized. Developed more than 25 years ago to have characteristics similar to R-12, it is used in medium- and high-temperature applications in which R-12 had been used. R-134a is used in automotive air conditioning because of its low hose permeability and high critical temperature. Domestic refrigerator producers also use R-134a.

R-134a has the benefit of being a single-component refrigerant and, therefore, does not have any glide. Also, the direct GWP of R-134a is low. The disadvantage of R-134a lies in its fairly low capacity compared to R-22. To use this refrigerant, all the tubing in the heat exchangers and between the components a system need to be significantly larger to minimize pressure drops and maintain an acceptable operating efficiency. Combined with the greater compressor displacements required, it results in a system that will be more costly than R-22 systems today. The heat-transfer coefficient of R-134a is also lower than that of R-22, and tests show that system performance degrades with its use. This may not be the case in larger commercial systems, in which large screw or centrifugal systems are used, and refrigerants like R-11 and R-12 were common. Here, R-134a may offer the best solution for a fairly low-investment, simple redesign to HFCs.

Emerson laboratory and field trials show that the refrigeration capacity and energy efficiency of R-134a are similar to R-12 for medium- and high-temperature applications. At evaporating temperatures below -23 degrees Celsius, R-134a loses its attractiveness for several

Technical and Economic Assessment Panel (TEAP) of the Montreal Protocol on Substances that Deplete the Ozone Layer reported in 1999 that HFCs are needed for the safe and cost-effective phase-out of CFCs and HCFCs and are essential substitutes for these products. Also, HFCs are needed to phase-out HCFCs in developing countries. As replacements for less energy-efficient, older equipment, HFC systems conserve energy and reduce global warming gases at electric power plants. These systems are being used responsibly, including recovery and reuse of HFCs to design of HFC-producing plants, with the goal of zero HFC emissions.

Substituting HFCs for CFCs has lowered the impact of greenhouse gas emissions, as HFCs cut total greenhouse

reasons:

- It experiences significant loss of capacity and efficiency compared to R-12.
- Pressure ratios become high, compromising compressor reliability.
- Low side pressures are sub-atmospheric (i.e., vacuum), resulting in system reliability concerns.

Except for ozone-depletion potential, R-134a possesses the same deficiencies as R-12 and represents a step backward for most commercial refrigeration and air conditioning applications. These deficiencies include larger-displacement compressors and larger-diameter tubing compared to that required for use with high-pressure refrigerants.

#### ***R-404A***

Equipment manufacturers use HFC refrigerant R-404A as the long-term replacement for R-502. R-404A is an excellent low and medium-temperature refrigerant, because of high energy efficiency and zero-ozone-depletion potential. R-404A is a near-azeotropic blend of HFC refrigerants R-125, R-143a and R-134a. It is commercially available from many sources and is becoming the most popular refrigerant of its class.

#### ***R-507***

This refrigerant is an azeotropic mixture of R-143a and R-125, with characteristics also similar to R-502. Emerson compressors developed for R-404A (except for a few hermetic reciprocating models) are also approved for R-507. R-404A and R-507 operate at slightly higher pressures and slightly lower discharge temperatures than R-502.

#### ***R-407C***

R-407C is a blend of R-32, R-125 and R-134a. Of the higher temperature HFC options, R-407C was designed to have operating characteristics similar to R-22. The major concerns surrounding R-407C are in its high glide (5.5 °C) and the efficiency degradation when compared to R-22; however, this refrigerant provides the simplest conversion of the HFC alternatives. In systems where glide is acceptable, R-407C has become a popular option for manufacturers who want to move quickly to an HFC alternative. In the long run, however, the lower-efficiency performance of this refrigerant may make it a less attractive alternative when compared to R-410A for medium-and high-temperature applications.

Care should be taken when applying R-407C in any applications in which glide can impact system performance by fractionation in flooded-evaporator or multi-evaporator designs.

Also, R-407C should not be viewed as a drop-in for R-22 systems or applications. Like all HFCs, R-407C requires POE lubricants, and other system design modifications may be required for R-407C to operate acceptably in R-22 systems.

#### ***R-407A***

R-407A is a blend of R-32, R-125, and R-134a in a 20/40/40 ratio by mass. R-407A was designed for low and medium temperature refrigeration applications. With an GWP Value of 2107, R-407A has 54% lower GWP than R-404A.

Manufacturers of this refrigerant claim a good capacity and efficiency match for R-22, making it well suited as a retrofit for R-22 in supermarket and food storage applications. Discharge temperatures of R-407A will be lower than R-22 but system pressures are higher. Compressor cooling should be taken into account for high ambient operation.

#### ***R-32***

R-32 is an HFC refrigerant that is slightly flammable, with an ASHRAE 34 flammability classification of A2L. It is best known as a component in R-410A. R-32 is not currently being marketed as a stand-alone refrigerant in North America, but is gaining interest in China.

#### ***R-410A***

R-410A has become the refrigerant of choice for use in residential air conditioning applications. Most major residential air conditioning manufacturers offer R-410A product lines.

R-410A delivers higher efficiency and better TEWI than other choices. The refrigerant also has many benefits that make it an ideal refrigerant for use in commercial refrigeration applications.

There are several distinct operational differences between R-22 and R-410A refrigerants.

R-410A operates at 50 percent higher pressure than R-22; however, the higher pressure allows the system to run at a lower temperature. Because of these differences, anyone handling these units should get training on the technical aspects of the new R-410A systems, where they can learn proper joint brazing and maintenance tips for this refrigerant.

R-410A is a near-azeotrope composition of 50 percent R-32 and 50 percent R-125. System testing has shown that R-410A delivers higher system efficiency than R-22. R-410A evaporates with a 35 percent higher heat-transfer coefficient and 28 percent lower pressure drop compared to R-22.

Other system performance enhancements have been gained by sizing for equal pressure drop and reducing the number of coil circuits needed to raise the mass flux. The higher density and pressure also permit smaller-diameter tubing, while maintaining reasonable pressure drops.

Because systems that use R-410A have been specially designed to use less tubing and fewer coils, R-410A has emerged as a cost-effective refrigerant. Fewer materials, along with lower refrigerant charge and better cyclic performance, also contribute to the affordability of R-410A.

R-410A is considered a high-pressure refrigerant. High-pressure refrigerants operate at pressures significantly higher than those normally seen with refrigerants such as R-22 and R-502. They cannot be used as retrofit refrigerants with existing equipment, but only in new equipment (including compressors) specifically designed for them. R-22 compressors cannot meet industry design standards with these higher pressures.

For refrigeration application, R-410A is potentially the most efficient refrigerant at medium-temperature conditions (-18...-1 °C). Other advantages include smaller line sizes and lower pressure drops; however, the system would require design for higher pressures. Potential changes in UL requirements may reduce the impact. Testing at lower temperatures has shown promising results. Research is ongoing at Emerson to understand the benefits of this refrigerant in commercial refrigeration.

#### ***R-417A***

R-417A was developed to be a “drop-in” refrigerant for new and service replacements of R-22, while using traditional HCFC lubricants, such as mineral oil and alkyl benzene (AB). This refrigerant is branded as ISCEON® 59 and Nu-22™ refrigerants and is a blend of R-125 (46.6 percent), R-134a (50 percent) and R-600 butane (3.4 percent). The hydrocarbon in the mixture was added to enhance oil return. ASHRAE designates the refrigerant as A1/A1 rated, meaning that it is nontoxic and nonflammable. The refrigerant manufacturer claims equivalent capacity and improved efficiency compared with R-22. It further claims that R-22 lubricants can be maintained, but recommends a consultation with the system and compressor manufacturer for current recommendations. The manufacturer’s claims about R-417A performance are not supported by independent test reports. Independent testing of R-417A has shown between nine and 10 percent lower system capacity when used as a drop-in refrigerant. This same testing shows efficiency losses of three to five percent. Independent testing of R-417A has also shown significant delays oil return.

Two more challenges are presented with R-417A. The refrigerant has a worse GWP rating than R-407C and R-22. As a blend, R-417A has the same fractionation and glide issues as R-407C. So a system leak may affect the composition significantly and, therefore, the properties of this refrigerant.

Emerson Climate Technologies does not expect R-417A to be a significant HFC alternative to R-22.

#### ***R-152a***

R-152a is chemically similar to R-134a, but it is different environmentally. R-152a has a much lower GWP (120 compared to 1,300) than R-134a but is considered ASHRAE A2 — flammable.

R-152a is being considered as an option to replace R-134a in automobile air conditioning; however, because of its flammability, R-152a is not a serious alternative for



commercial refrigeration systems.

#### ***R-422 refrigerants***

R-422A, R-422B and R-422D are other HFC refrigerants that were developed for replacement of R-22. All R-422 series refrigerants have the same blend of refrigerants. The last letter simply signifies slightly different percentages of the refrigerant mixes. R-422 refrigerants are branded as One Shot™ or ISCEON 9 series and are a blend of R-125 (85 percent), R-134a (11.6 percent) and R-600a (3.4 percent). The hydrocarbon in the mixture was added to enhance oil return.

The refrigerant manufacturers claim equivalent capacity and improved efficiency compared with R-22. They further claim that R-22 lubricants can be maintained, but recommend a consultation with the system and compressor manufacturer for current application considerations.

Independent testing of R-422A has shown that its capacity is 10 to 15 percent lower than R-22 and R-404A, especially in low-temperature conditions. Mass flow of R-422A is even higher than R-404A and about 55 percent higher than R-22. Pressures of R-422A are similar to those of R-404A and 20 percent higher than R-22.

Refrigerant changeover guidelines are available from Emerson that describe the retrofit procedures when transitioning from R-22 to R-422 series refrigerants.

#### ***Future low-GWP fluorocarbon refrigerants***

Several refrigerant manufacturers are developing refrigerants for automotive air conditioning applications that will meet European Union environmental standards for GWP substances. The most likely candidate to replace R-134a in automotive air conditioning applications is HFO-1234yf, a single-component refrigerant with a much lower GWP than R-134a. These future low-GWP refrigerants are being developed as non-carbon dioxide alternatives. Other possible fluorocarbon refrigerants under development are two-blend azeotropes with HFO and R-32 or R-134a.

#### ***Halogen-free refrigerants***

##### ***Ammonia***

Ammonia (NH<sub>3</sub>) is widely used as a refrigerant in large industrial refrigeration plants. As a halogen-free refrigerant, ammonia has the benefit of zero-ozone depletion potential and no direct GWP; however, its high toxicity limits its application to industrial refrigeration applications. In large ammonia systems, the efficiency is the same as similar systems with R-22 refrigerant.

Although ammonia is widely available and is a low-cost substance, there are significant challenges to applying ammonia as a refrigerant in commercial refrigeration systems. Ammonia systems have higher discharge pressures than R-22. Oil management becomes a major issue in ammonia systems, since the oils used are usually not soluble in ammonia. The low mass flow of ammonia compared to R-22 is an advantage for large ammonia plants, but becomes a challenge in smaller commercial systems. Also, ammonia is highly corrosive on copper materials, so refrigerant lines must be steel, and the copper in the compressor-motor windings must be insulated from the gas.

The major drawback of using ammonia in commercial refrigeration applications is its high toxicity and flammability level. This alone requires unique safety measures that are well beyond the scope of most commercial installations.

##### ***Carbon dioxide***

Environmental concerns about the potential direct emissions from HFC-based refrigeration systems have led to legislation and taxation schemes in parts of Europe that favor the usage of carbon dioxide (CO<sub>2</sub>) as a refrigerant. CO<sub>2</sub> is given the designation R-744. CO<sub>2</sub> is environmentally benign compared with other refrigerants, is nonflammable, has low toxicity, is widely available and is a low-first-cost substance. These are the reasons it was one of the original refrigerants, used nearly 100 years ago. Although thermodynamic performance of a simple CO<sub>2</sub>

cycle is poor — 30 to 50 percent worse than HFCs — “poor” refrigerants such as CO<sub>2</sub> tend to have good heat-transfer characteristics and respond well to cycle modifications.

Many CO<sub>2</sub> systems are designed for trans-critical operation. These systems tend to have lower energy efficiency, compared to conventional systems, and the system design is different from conventional systems. Trans-critical operation means that the CO<sub>2</sub> does not condense at the high pressure, and rather than using a traditional condenser, a gas cooler is used. The pressures created by CO<sub>2</sub> present significant challenges in its usage. This is a technical and cost challenge not only for the compressor, but also for the heat exchangers.

Typical cycle efficiency is 40 percent of the ideal refrigeration cycle Carnot<sup>15</sup>, where the Coefficient of Performance (COP) is 2.5, compared with 68 percent (COP 4.2) for an R-134a system at high-temperature conditions. Microchannel heat exchangers, gas/suction heat exchangers or CO<sub>2</sub> expanders improve system performance, with some added cost and complexity.

The cost impact of CO<sub>2</sub> in trans-critical systems is substantial. Because of the higher pressure, modifications are required on the compressor shell, valves, rings, terminal and seals, and the pressure relief valve and microchannel heat exchanger. Performance implications require CSHX, a discharge-pressure regulator valve and a low side accumulator to control excess charge. Another oil separator is required because of oil circulation and return problems. The bottom line is a 20 to 30 percent higher final cost for performance levels equal to those of an HFC.

The comparably high pressure level and thermodynamic properties of CO<sub>2</sub> as a refrigerant have driven system designers to consider sub-critical CO<sub>2</sub> systems. These systems operate much like conventional cascade refrigeration systems. In a sub-critical system, CO<sub>2</sub> is used as a direct expanding medium in the low-temperature stage, and different options exist for the medium-temperature stage. This way compressors in the low-temperature stage are only exposed to pressure levels similar to high-pressure air conditioning applications, such as with R-410A. Sub-critical operation might be the best application of CO<sub>2</sub> as a refrigerant for some commercial refrigeration applications.

In summary, CO<sub>2</sub> has many technical and cost challenges. The low efficiency and cycle complexity are the main limitations; however, CO<sub>2</sub> may become used in transport and low-temperature cascade systems, and in some heat-pump applications. Whether trans-critical or sub-critical CO<sub>2</sub> systems are considered, CO<sub>2</sub> technology cannot be seen as a drop-in replacement for any of the other refrigerants mentioned in this paper. Any application of CO<sub>2</sub> requires a thorough assessment of system efficiency, TEWI, life-cycle cost, technical feasibility, reliability and safety.

### ***Hydrocarbons***

The push for halogen-free refrigerants has led manufacturers to investigate hydrocarbons as a replacement for R-22. Propane (R-290) is considered as a replacement, because it is a halogen-free substance with no ozone-depletion potential and low direct GWP. Propane is widely available and is a low cost substance. The operating pressures of a refrigeration system with propane are similar to R-22. Propane has been applied in systems with low charge — less than 150 grams — and often outside the U.S. The disadvantage of propane and all hydrocarbons is that they are highly flammable. System costs are higher because of the required safety measures. Special considerations must be taken for excess pressures and electrical connections, and ventilation to prevent flammable gas mixtures. Commercial operators do not want to risk the safety-code issues and litigation risks associated with using propane in a refrigeration system.

The refrigeration industry is anticipating U.S. EPA SNAP approval of R-290 Propane refrigerant for Domestic refrigerators and small self-contained Commercial Refrigeration equipment soon. Also, several Foodservice operators and OEMs are exploring the performance and economics of transitioning refrigeration applications to HFC-free, or ‘natural’ refrigerant alternatives over the next several years, both in the U.S. and globally. The primary benefits of these transitions would be to:

- Reduce the Direct Global Warming Potential (GWP) of the refrigerant to near zero to

significantly mitigate the refrigerant's impact on global climate change.

- Potentially improve equipment energy efficiency to cut the Indirect GWP of CO<sub>2</sub> emissions generated during electricity production, while using less electricity.
- Employ 'renewable' refrigerant gases which are byproducts of various processes, thus generating a lower cost and lower carbon footprint refrigerant than today's manufactured alternatives

The most common natural refrigerant alternatives being considered for small commercial refrigeration applications are R-290 Propane and CO<sub>2</sub>, while new slightly flammable (A2L Class) chemical blends (i.e., HFO1234yf) are being considered as future low GWP options. China is also evaluating an A2L alternative known as R-32 (a component of R-410A).

#### **4.3. Secondary Refrigerants**

Secondary refrigerants play a role in carrying heat from an object or a space being cooled to the primary refrigerant or the evaporator of a refrigeration system. During this process, the secondary refrigerant has no phase change. In the past, the most common secondary refrigerants were brines, which are water-salt (e.g., sodium chloride and calcium chloride) solutions, and even today they are still used in spite of their corrosive effects. Also, the antifreezes, which are solutions of water and ethylene glycol, propylene glycol, or calcium chloride, are widely used as secondary refrigerants.

Of these fluids, *propylene glycol* has the unique feature of being safe when in contact with food products. A decade ago dichloromethane (CH<sub>2</sub>Cl<sub>2</sub>), trichloroethylene (C<sub>2</sub>HCl<sub>3</sub>), alcohol solutions, and acetone were also used in some special applications. The following features are considered as main criteria in the selection of a proper secondary refrigerant:

- satisfactory thermal and physical properties,
- stability,
- non-corrosiveness,
- non-toxicity,
- low cost, and
- usability.

#### **4.4. Subcooling**

The term subcooling refers to the cooling of the liquid refrigerant below the condensation temperature; subcooling aims to raise the cooling power of the system and to improve its COP. Subcooling can be performed in the final portion of the condenser or in a specially designed heat exchanger, placed after the condenser.

When a special heat exchanger is used for subcooling, the cooling medium is:

- cold water;
- cold refrigerant vapors, coming from the evaporator.

The system presented in fig. 4.5 uses a water cooled heat exchanger (SR), placed after the condenser (K); the liquid refrigerant passing through the subcooler is cooled below the condensing temperature  $T_k$  (process 3-3'); as a result, the cooling capacity of the system increases with  $\Delta q_0$ . Because the mechanical work required for the refrigerant compression (process 1-2) does not change, the efficiency (COP) of the system increases when this subcooling method is used.

The system presented in fig. 4.6 uses cold refrigerant vapors for subcooling, by the mean of a liquid-vapor regenerative heat exchanger (Sr). The cold vapors coming from the evaporator pass through the heat exchanger and cool the liquid refrigerant coming out from the condenser; thus, the liquid refrigerant is subcooled (process 3-3') and the vapors are superheated (process 1-1').

The cooling capacity increases when this method is used; vapor superheating could adversely effect the mechanical work for vapor compression. As a result the refrigeration

efficiency can increase or decrease, depending on the type of refrigerant. For this reason this subcooling method is not applied for refrigeration systems using ammonia as refrigerant.

Due to the different specific heats of vapor and liquid, the degree of subcooling ( $T_3 - T_{3'}$ ) is approx. half the degree of superheating ( $T_{1'} - T_1$ ).

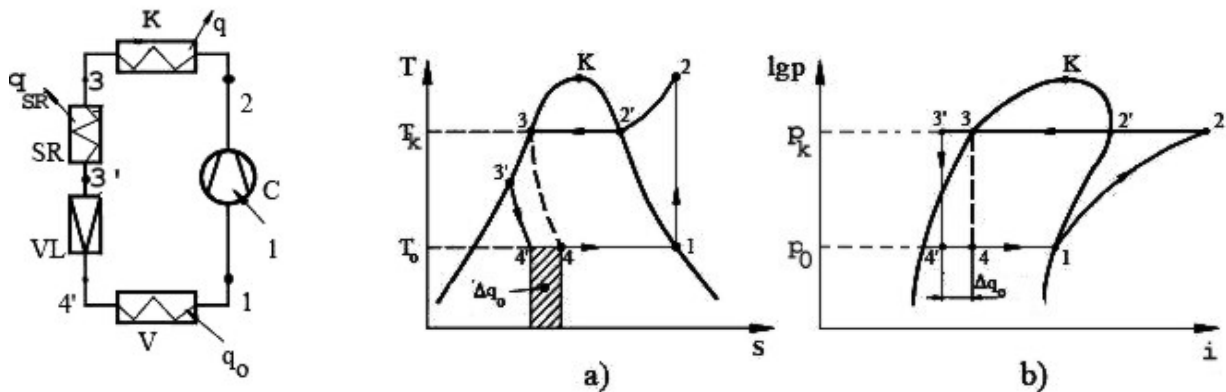


Fig. 4.5 – Water subcooling

SR-subcooler heat exchanger; 4'-1: evaporation process, with subcooling; 4-1: evaporation process, without subcooling.

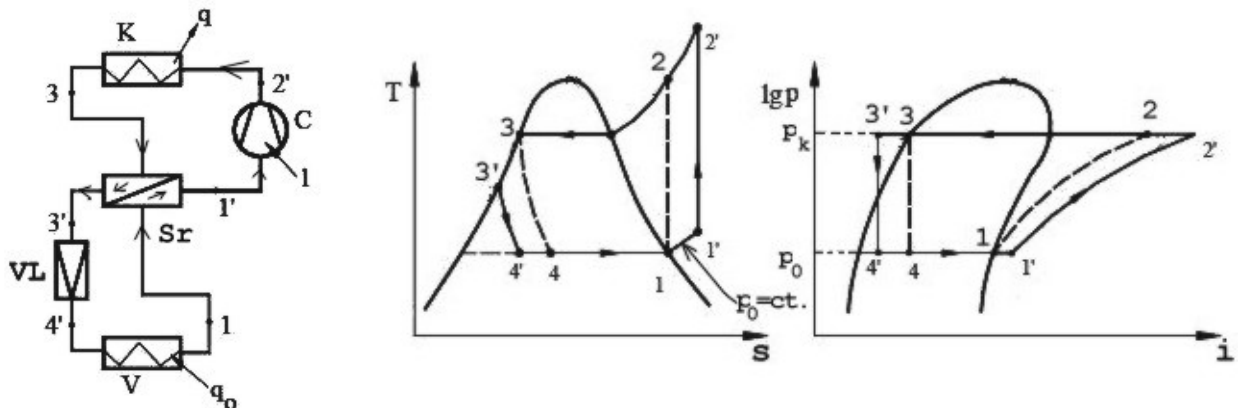


Fig. 4.6 – Subcooling with a liquid-vapor regenerative heat exchanger

Sr-heat exchanger; 4'-1: evaporation process, with subcooling; 4-1: evaporation process, without subcooling; 1-2: compression process, without superheating of vapor; 1'-2': compression process, with vapor superheating.

#### 4.5. Multi-stage refrigeration systems

A single stage vapor compression refrigeration system has one low side pressure (evaporator pressure) and one high side pressure (condenser pressure). The performance of single stage systems shows that these systems are adequate as long as the temperature difference between evaporator and condenser (*temperature lift*) is small. However, there are many applications where the temperature lift can be quite high. The temperature lift can become large either due to the requirement of very low evaporator temperatures and/or due to the requirement of very high condensing temperatures. For example, in frozen food industries the required evaporator can be as low as  $-40^{\circ}\text{C}$ , while in chemical industries temperatures as low as  $-150^{\circ}\text{C}$  may be required for liquefaction of gases. On the high temperature side the required condensing temperatures can be very high if the refrigeration system is used as a heat pump for heating applications such as process heating, drying etc. However, as the temperature lift increases the single stage systems become inefficient and impractical. Decreasing the evaporation temperature and/or increasing the condensing temperature results in the decrease of the cooling capacity of the system and in the increase of the compression work. Due to these drawbacks, single stage systems are not recommended when the evaporator temperature becomes very low and/or when the condenser temperature becomes high. In such cases multi-stage systems are used in practice.

Generally, for fluorocarbon and ammonia based refrigeration systems a single stage system is used up to an evaporator temperature of  $-30^{\circ}\text{C}$ . A two-stage system is used up to  $-60^{\circ}\text{C}$  and a three-stage system is used for temperatures below  $-60^{\circ}\text{C}$ .

One of the most common solution for multistage systems is the two stage compression, with intermediary cooling of the refrigerant vapor between the two stages.

Inter-cooling of the vapor may be achieved by using either a water-cooled heat exchanger or by the refrigerant in the flash tank.

The refrigeration system presented in fig. 4.7 uses a water cooled heat exchanger (Ri) for the intermediary cooling of the refrigerant. The two stages compression process is 1-2-2'-2'', where 1-2 represents the compression in the first stage, 2-2' is the intermediary cooling of the refrigerant at the constant pressure  $p_i$  and 2'-2'' represents the compression process for the second stage. The hashed surface represents the reduction of compression work ( $\Delta l$ ) for the two stage compression in comparison with the one stage compression process (1-II).

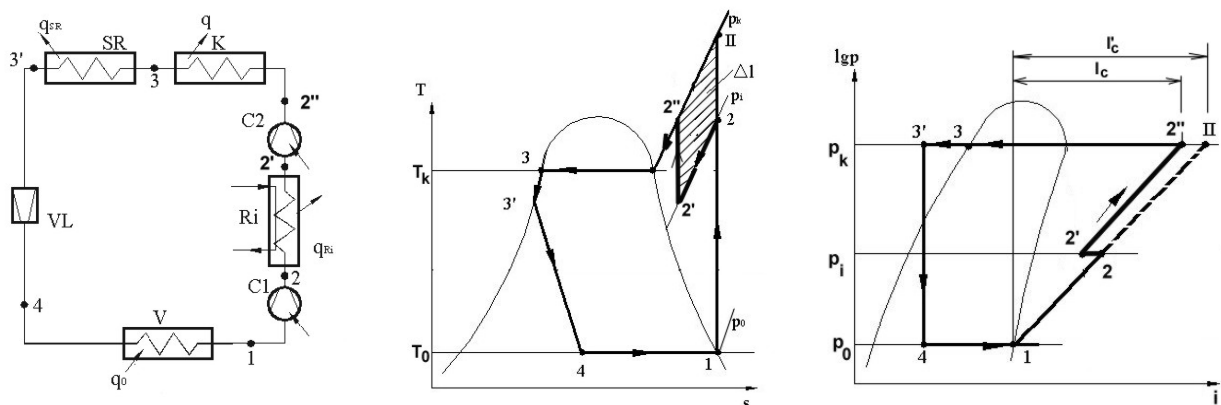


Fig. 4.7 – Refrigeration system with two stage compression and water intermediary cooling  
C1, C2-compression stages; Ri-water cooled heat exchanger; K-condenser; SR-subcooling heat exchanger; VL-expansion valve; V-evaporator.

Inte-cooling not only reduces the work input but also reduces the compressor discharge temperature leading to better lubrication and longer compressor life.

Inter-cooling may not be always possible using water-cooled heat exchangers as it depends on the availability of sufficiently cold water to which the refrigerant from low stage compressor can reject heat. Moreover, with water cooling the refrigerant at the inlet to the high stage compressor may not be saturated.

Inter-cooling using liquid refrigerant from condenser in the flash tank may or may not reduce the power input to the system, as it depends upon the nature of the refrigerant. This is due to the fact that the heat rejected by the refrigerant during intercooling generates additional vapor in the flash tank, which has to be compressed by the high stage compressor. Thus the mass flow rate of refrigerant through the high stage compressor will be more than that of the low stage compressor. Whether total power input to the system decreases or not depends on whether the increased power consumption due to higher mass flow rate is compensated by reduction in specific work of compression or not. For ammonia, the power input usually decreases with inter-cooling by liquid refrigerant however, for refrigerants such as R12, R22, the power input marginally increases. Thus inter-cooling using liquid refrigerant is not effective for R12 and R22. However, as mentioned, one benefit of inter-cooling is the reduction in compressor discharge temperature, which leads to better compressor lubrication and its longer life.

It is also possible to intercool the refrigerant vapor by a combination of water-cooled heat exchanger and the refrigerant liquid in the flash tank. As a result of using both water-cooling and flash-tank, the amount of refrigerant vapor handled by the high-stage compressor reduces leading to lower power consumption. However, the possibility of this again depends on the availability of cooling water at required temperature. Fig. 4.8 presents the diagram of a refrigeration system using a water-cooled heat exchanger and the refrigerant liquid in the flash tank in order to

achieve inter-cooling.

In the flash tank (BRI) the vapors coming from the first compression stage are blown through the cold liquid refrigerant from the tank (process 2'-3), after being cooled in the heat exchanger (process 2-2'). It is assumed that in this process (2'-3) the superheated refrigerant vapor gets completely de-superheated and emerges out as a saturated vapor at state 3.

In order to increase the refrigeration capacity it is possible to use the relatively low temperature of the liquid in the flash tank for subcooling, as shown in fig. 4.9. In this case the second subcooler (SR<sub>2</sub>) is a coil immersed into the cold liquid refrigerant from the flash tank (BRI); in the log p – i diagram the corresponding process is 6-8, the liquid refrigerant from the condenser being subcooled by exchanging heat with the liquid refrigerant in the flash tank. The flash tank itself is used for intercooling of the refrigerant vapor between the low and high-stage compressors, as in the previous system (fig. 4.8).

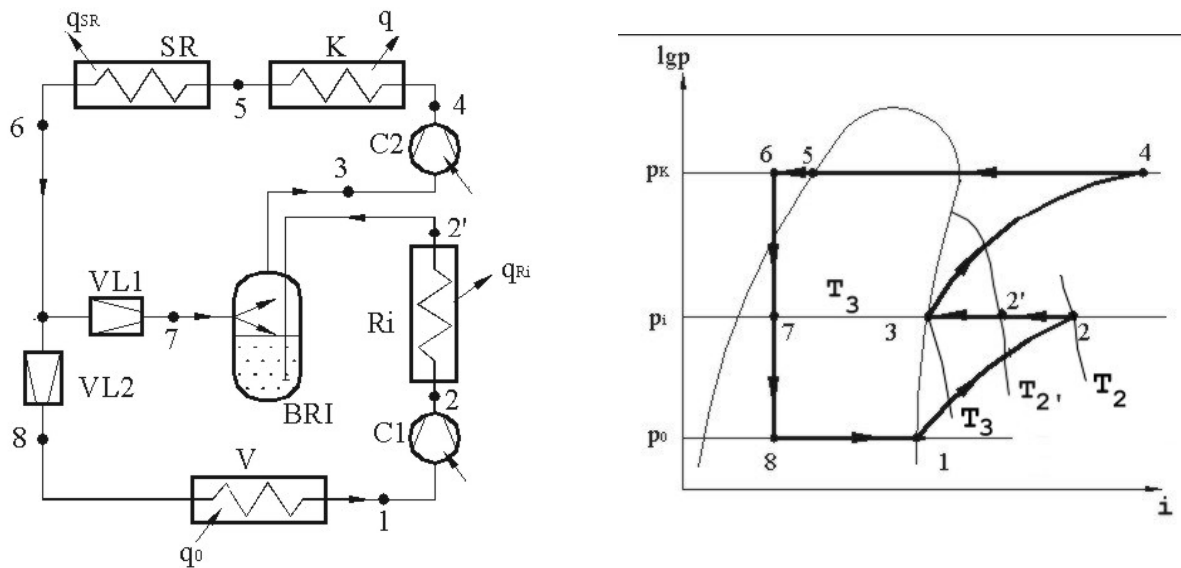


Fig. 4.8 – Two stage compression refrigeration system, with water cooled heat exchanger and flash tank intercooling

C1, C2-compression stages; Ri-water cooled heat exchanger; BRI-flash tank; K-condenser; SR-subcooling heat exchanger; VL<sub>1</sub>, VL<sub>2</sub>-expansion valves; V-evaporator.

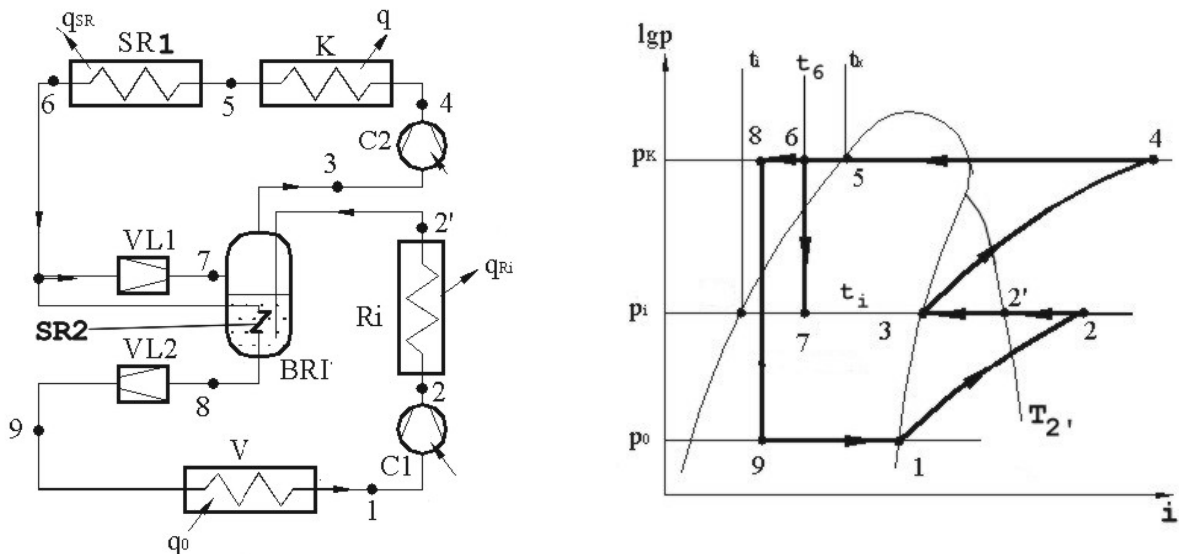


Fig. 4.9 – Subcooling in the flash tank

SR<sub>1</sub>-water cooled heat exchanger for subcooling; SR<sub>2</sub>-second subcooler; BRI-flash tank.

#### 4.6. Cascade refrigeration systems

Multi-stage systems have certain limitations:

- a) since only one refrigerant is used throughout the system, the refrigerant used should have high critical temperature and low freezing point;
- b) the operating pressures with a single refrigerant may become too high or too low. Generally only R12, R22 and NH<sub>3</sub> systems have been used in multi-stage systems as other conventional working fluids may operate in vacuum at very low evaporator temperatures; operation in vacuum leads to leakages into the system and large compressor displacement due to high specific volume.
- c) possibility of migration of lubricating oil from one compressor to other leading to compressor break-down.

The above limitations can be overcome by using cascade systems. In a cascade system a series of refrigerants with progressively lower boiling points are used in a series of single stage units. The refrigerants can be chosen with the appropriate properties, avoiding large dimensions for the system components. Refrigerants used in each stage may be different and are selected for optimum performance at the given evaporator and condenser temperatures.

In this type of systems, the condenser of lower stage system is coupled to the evaporator of the next higher stage system and so on. The component where heat of condensation of lower stage refrigerant is supplied for vaporization of next level refrigerant is called as *cascade condenser*.

Fig. 4.10 shows the schematic and operating cycle diagram of a two-stage cascade refrigeration system. As shown, this system employs two different refrigerants operating in two individual cycles. They are thermally coupled in the cascade condenser.

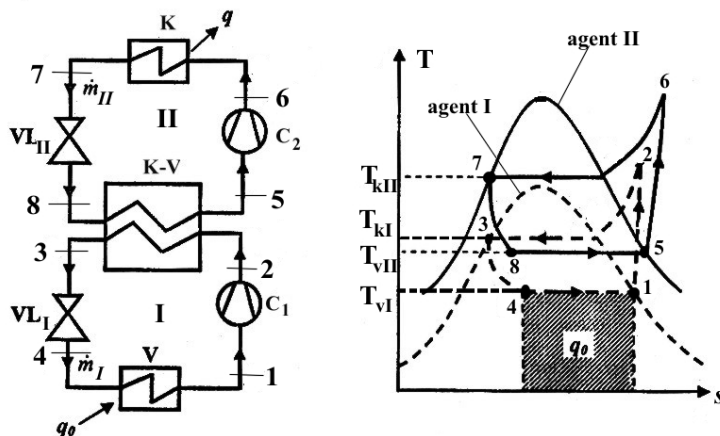


Fig. 4.10 - Two-stage cascade refrigeration system

K-V – cascade condenser;  
 I-low temperature refrigeration system;  
 II-high temperature refrigeration system.

The condenser of system II (high temperature stage) is usually fan cooled by the ambient air. In some cases a water supply may be used, but air cooling is much more common. The evaporator of system II is used to cool the condenser of system I (low temperature stage).

As previously mentioned, the refrigerants selected should have suitable pressure-temperature characteristics. An example of refrigerant combination is the use of carbon dioxide (NBP = -78.4°C, T<sub>k</sub> = 31.06°C) in low temperature cascade and ammonia (NBP = -33.33°C, T<sub>k</sub> = 132.25°C) in high temperature cascade. It is possible to use more than two cascade stages, and it is also possible to combine multi-stage systems with cascade systems.

#### 4.7. Vapor absorption refrigeration systems

Vapor Absorption Refrigeration Systems (VARS) belong to the class of vapor cycles similar to vapor compression refrigeration systems. However, unlike vapor compression refrigeration systems, the required input to absorption systems is in the form of heat. Hence these systems are also called as heat operated or thermal energy driven systems. Since these systems run on low-grade thermal energy, they are preferred when low-grade energy such as waste heat or solar energy is available. Since conventional absorption systems use natural refrigerants such as water or ammonia they are environment friendly.

Fig. 4.11 presents a comparison between vapor compression refrigeration systems (VCRS) and vapor absorption refrigeration systems (VARs). It can be seen that as far as the condenser, expansion valve and evaporators are concerned, both compression and absorption systems are identical. However, the difference lies in the way the refrigerant is compressed to condenser pressure. In vapor compression refrigeration systems the vapor is compressed mechanically using the compressor, whereas in absorption system the vapor is first converted into a liquid and then the liquid is pumped to condenser pressure using the solution pump. Since for the same pressure difference, work input required to pump a liquid (solution) is much less than the work required for compressing a vapor due to very small specific volume of liquid the mechanical energy required to operate vapor absorption refrigeration system is much less than that required to operate a compression system. However, the absorption system requires a relatively large amount of low-grade thermal energy at generator temperature to generate refrigerant vapor from the solution in generator. Thus while the energy input is in the form of mechanical energy in vapor compression refrigeration systems, it is mainly in the form of thermal energy in case of absorption systems. The solution pump work is often negligible compared to the generator heat input. The COP of vapor compression refrigeration system will be much higher than the COP of an absorption system.

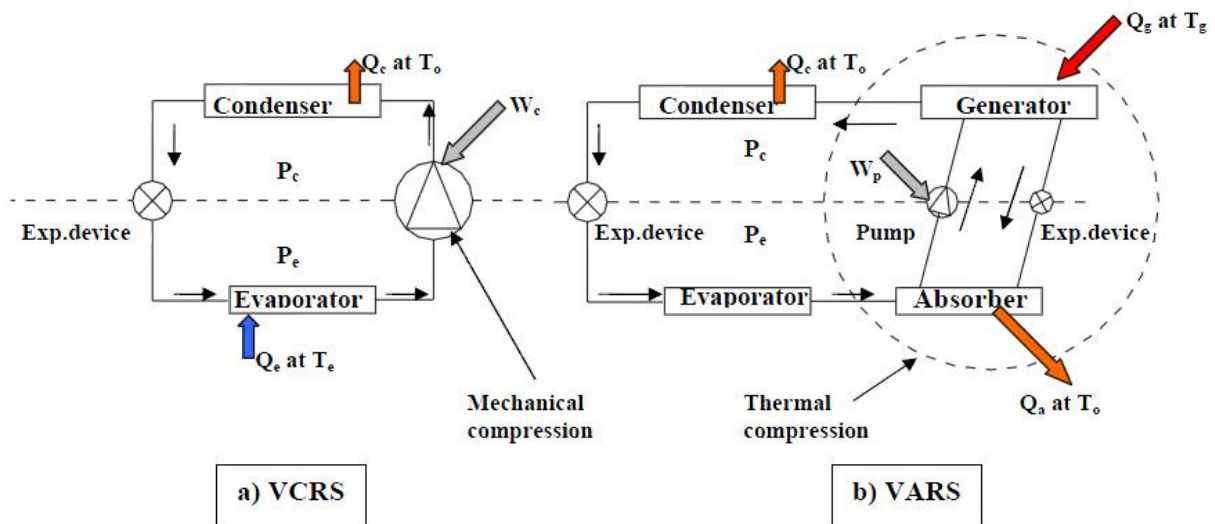


Fig. 4.11 – Comparison between vapor compression refrigeration systems and vapor absorption refrigeration systems

The solution used in absorption refrigeration systems may be considered as a homogeneous binary mixture of refrigerant and absorbent. The refrigerant-absorbent mixtures used in absorption refrigeration systems must be completely miscible under all conditions both in liquid and vapor phases and there should be large difference in the boiling points of refrigerant and absorbent (greater than  $200^{\circ}\text{C}$ ), so that only refrigerant is boiled-off in the generator. This ensures that only pure refrigerant circulates through refrigerant circuit (condenser-expansion valve-evaporator) leading to isothermal heat transfer in evaporator and condenser.

The most commonly used refrigerant-absorbent pairs in commercial systems are:

- Water-Lithium Bromide ( $\text{H}_2\text{O}-\text{LiBr}$ ) system for above  $0^{\circ}\text{C}$  applications such as air conditioning. Here water is the refrigerant and lithium bromide is the absorbent.
- Ammonia-Water ( $\text{NH}_3-\text{H}_2\text{O}$ ) system for refrigeration applications with ammonia as refrigerant and water as absorbent.

Figure 4.12 shows a basic vapor absorption refrigeration system with a solution heat exchanger on a pressure vs. temperature diagram. As shown in the figure, low temperature and low pressure refrigerant vapor from evaporator at state 1 enters the absorber and is absorbed by solution weak in refrigerant (state 8). The heat of absorption ( $Q_a$ ) is rejected to an external heat



sink at  $T_\infty$ . The solution, rich in refrigerant (state 2) is pumped to the generator pressure ( $p_g$ ) by the solution pump (state 3). The pressurized solution gets heated up sensibly as it flows through the solution heat exchanger by extracting heat from hot solution coming from generator (state 4). Heat is supplied to this solution from an external heat source in the generator ( $Q_g$  at  $T_g$ ), as a result refrigerant vapor is generated (absorbent may also boil to give off vapor in case of ammonia-water systems) at state 5. This high-pressure refrigerant vapor condenses in the condenser by rejecting heat of condensation to the external heat sink ( $Q_c$  at  $T_\infty$ ) and leaves the condenser as a high pressure liquid (state 9). This high pressure refrigerant liquid is throttled in the expansion device to evaporator pressure  $p_e$  (state 10) from where it enters the evaporator, extracts heat from low temperature heat source ( $Q_e$  at  $T_e$ ) and leaves the evaporator as vapor at state 1, completing a cycle. The hot solution that is weak in refrigerant (state 6) leaves the generator at high temperature and is cooled sensibly by rejecting heat to the solution going to the generator in the solution heat exchanger (state 7). Then it is throttled to the evaporator pressure in the throttle valve (state 8), from where it enters the absorber to complete the cycle.

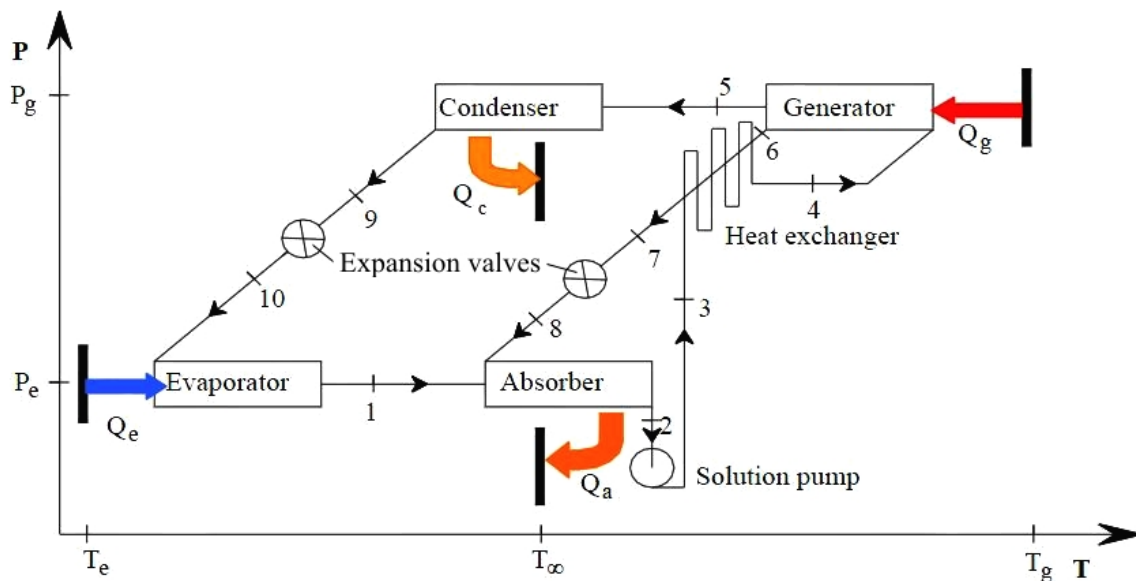


Fig. 4.12 – Schematics of a basic vapor absorption refrigeration system

It can be seen that though not an essential component, the solution heat exchanger is used in practical systems to improve the COP by reducing the heat input in the generator. The hot solution coming from the generator comes in thermal contact with the cold solution going to the generator. As a result of this heat exchange, less heat input is required in the generator and less heat is rejected in the absorber, thus improving the system performance significantly.

#### 4.8. Supermarket refrigeration

An important application of refrigeration is supermarket refrigeration. Nearly all supermarkets today use ozone-depleting HCFC refrigerant, usually R-22, or a blend consisting entirely or primarily of HFCs. HCFCs and the HFCs are also greenhouse gases. Most supermarkets use direct expansion refrigeration systems. Two of the most common advanced refrigeration technologies for supermarkets are distributed system and secondary loop system.

In a direct expansion system, the compressors are mounted together and share suction and discharge refrigeration lines that run throughout the store, feeding refrigerant to the cases and coolers (fig. 4.13). The compressors are located in a separate machine room, either at the back of the store or on its roof, to reduce noise and prevent customer access, while the condensers are usually air-cooled and hence are placed outside to reject heat. These multiple compressor racks operate at various suction pressures to support display cases operating at different temperatures.

The hot refrigerant gas from the compressors is cooled and condensed as it flows into the condenser. The liquid refrigerant is collected in the receiver and distributed to the cases and coolers by the liquid manifold. The refrigerant is expanded turning a fraction of liquid into vapor before flowing into the evaporator. After cycling through the cases, the refrigerant returns to the suction manifold and the compressors.

Supermarkets tend to have one direct expansion system for “low-temperature” refrigeration (e.g., ice cream, frozen foods, etc.) and one or two direct expansion systems for “medium-temperature” refrigeration (e.g., meat, prepared foods, dairy, refrigerated drinks, etc.).

Secondary loop systems have recently seen increased introduction into retail food equipment, and now make up about 4% of the market. These systems generally use R-404A or R-507A, although some earlier systems used R-22. Their average leak rate is between 2 and 15%.

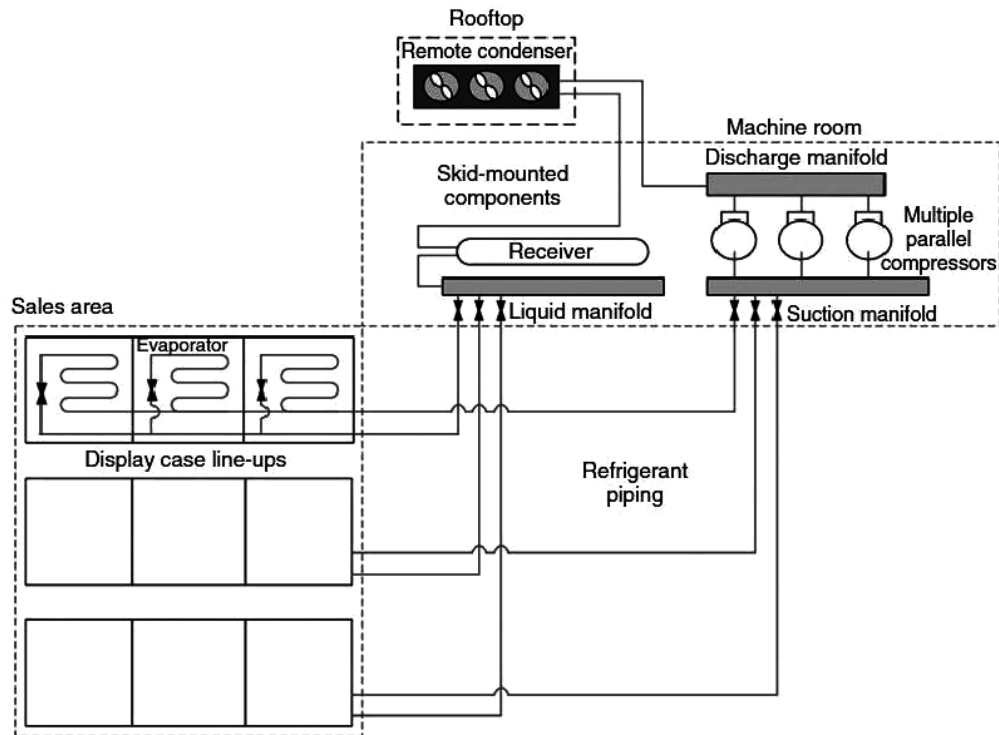


Fig. 4.13 - Schematic of direct expansion system

Secondary loop systems use a much smaller refrigerant charge than traditional direct expansion refrigeration systems, and hence have significantly decreased total refrigerant emissions. In secondary loop systems, two liquids are used. The first is a cold fluid, often a brine solution, which is pumped throughout the store to remove heat from the display equipment. The second is a refrigerant used to cool the cold fluid that travels around the equipment. Secondary loop systems can operate with two to four separate loops and chiller systems depending on the temperatures needed for the display cases.

As shown in fig. 4.14, the refrigerant is compressed in parallel compressors and the superheated refrigerant gas is cooled and condensed in a remote condenser. The liquid refrigerant is then collected in the receiver, expanded in a throttling device, and evaporated by absorbing heat from a cold fluid (i.e., brine). The cooled brine is distributed in the sales area (refrigerated area) absorbing heat from the products before returning to the evaporator to repeat the process.

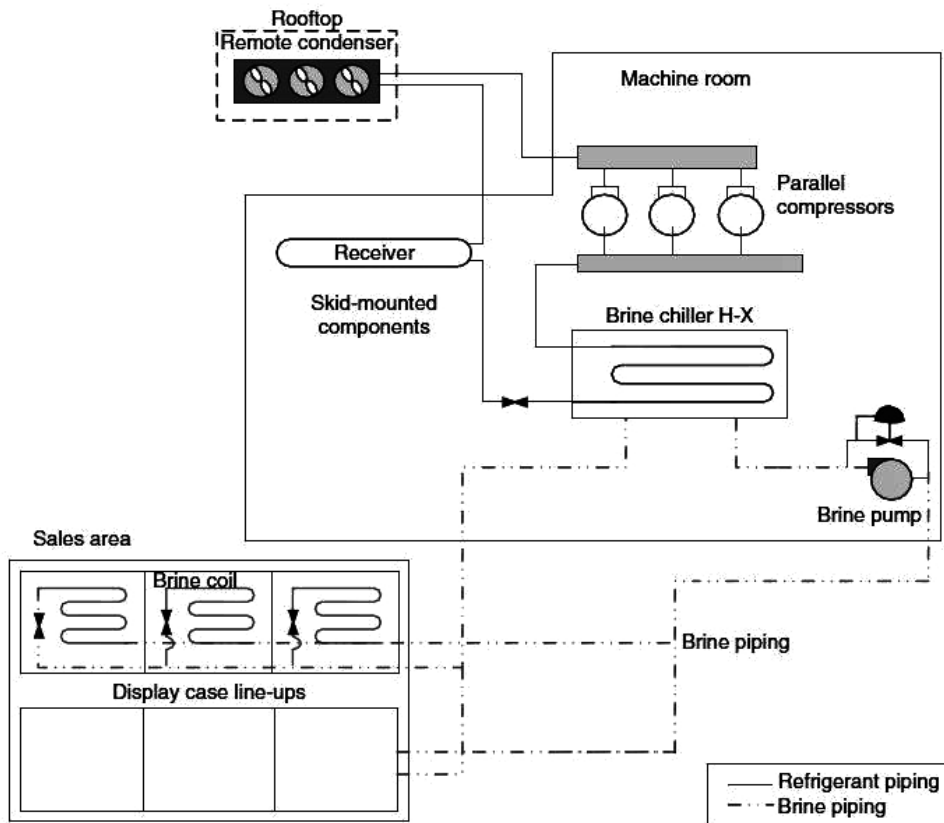


Fig. 4.14 – Schematic of the secondary loop system

## 5. Food chilling and freezing

Refrigeration slows down the chemical and biological processes in foods and the accompanying deterioration and the loss of quality. The *storage life* of fresh perishable foods such as meats, fish, fruits, and vegetables can be extended by several days by cooling, and by several weeks or months by freezing.

When the preservation period is longer than that acceptable for chilled meat, freezing must be used to minimize any physical, biochemical and microbiological changes affecting quality in storage. During freezing most of the water content of the meat, about 80%, solidifies into pure ice crystals, accompanied by a separation of dissolved solids.

### 5.1. Food chilling

Chilling is the unit operation in which the temperature of a food is reduced to between -1°C and 8°C. It is used to reduce the rate of biochemical and microbiological changes, and hence to extend the shelf life of fresh and processed foods. It causes minimal changes to sensory characteristics and nutritional properties of foods and, as a result, chilled foods are perceived by consumers as being convenient, easy to prepare, high quality and 'healthy', 'natural' and 'fresh'. Since the 1980s there has been substantial product development and strong growth in the chilled food market, particularly for sandwiches, desserts, ready meals, prepared salads, pizza and fresh pasta.

Chilling is often used in combination with other unit operations (for example fermentation or pasteurization) to extend the shelf life of mildly processed foods. There is a greater preservative effect when chilling is combined with control of the composition of the storage atmosphere than that found using either unit operation alone. However, not all foods can be chilled and tropical, subtropical and some temperate fruits, for example, suffer from chilling injury at 3–10°C above their freezing point.

Chilled foods are grouped into three categories according to their storage temperature range as follows:

- -1°C to +1°C (fresh fish, meats, sausages and ground meats, smoked meats and breaded fish).
- 0°C to +5°C (pasteurized canned meat, milk, cream, yoghurt, prepared salads, sandwiches, baked goods, fresh pasta, fresh soups and sauces, pizzas, pastries and unbaked dough).
- 0°C to +8°C (fully cooked meats and fish pies, cooked or uncooked cured meats, butter, margarine, hard cheese, cooked rice, fruit juices and soft fruits).

EU temperature legislation governs the chilling of beef, pork and lamb for the majority of abattoirs within the community. The only derogations are for very small abattoirs and for retail shops cutting meat for direct sale to the final consumer. The EC legislation does not define a chilling time, only a maximum final meat temperature of 7 °C before transport or cutting.

To chill fresh foods it is necessary to remove both sensible heat (also known as *field heat*) and heat generated by respiratory activity. The rate of respiration of fresh fruits is not necessarily constant at a constant storage temperature. Fruits which undergo 'climacteric' ripening show a short but abrupt increase in the rate of respiration which occurs near to the point of optimum ripeness. Climacteric fruits include apple, apricot, avocado, banana, mango, peach, pear, plum and tomato. Non-climacteric fruits include cherry, cucumber, fig, grape, grapefruit, lemon, pineapple and strawberry. Vegetables respire in a similar way to non-climacteric fruits. The optimum storage temperature and relative humidity, and expected storage times are shown in Table 5.1 for a variety of fresh fruits and vegetables.

In animal tissues, aerobic respiration rapidly declines when the supply of oxygenated blood is stopped at slaughter. Anaerobic respiration of glycogen to lactic acid then causes the pH of the meat to fall, and the onset of *rigor mortis*, in which the muscle tissue becomes firm and inextensible. Cooling during anaerobic respiration is necessary to produce the required texture

and color of meat and to reduce bacterial contamination. Undesirable changes, caused by cooling meat before rigor mortis has occurred, are termed *cold shortening*.

For processed foods, a reduction in temperature below the minimum necessary for microbial growth extends the generation time of micro-organisms and in effect prevents or retards reproduction. Chilling prevents the growth of thermophilic and many mesophilic micro-organisms. The main microbiological concerns with chilled foods are a number of pathogens that can grow during extended refrigerated storage below 5°C, or as a result of any increase in temperature (temperature abuse) and thus cause food poisoning. Previously it was considered that refrigeration temperatures would prevent the growth of pathogenic bacteria, but it is now known that some species can either grow to large numbers at these temperatures, or are sufficiently virulent to cause poisoning after ingestion of only a few cells.

Table 5.1

Optimum storage conditions for some fruits and vegetables			
Food	Temperature (°C)	Relative humidity (%)	Storage life (days)
Apricot	-0.5-0	90	7-14
Banana	11-15.5	85-95	7-10
Bean (snap)	7	90-95	7-10
Broccoli	0	95	10-14
Carrot	0	98-100	28-42
Celery	0	95	30-60
Cherry	-1	90-95	14-20
Cucumber	10-15	90-95	10-14
Eggplant	7-10	90-95	7-10
Lemon	10-14	85-90	30-180
Lime	9-10	85-90	40-140
Lettuce	0-1	95-100	14-20
Mushroom	0	90	3-4
Peach	-0.5-0	90	14-30
Plum	-1-0	90-95	14-30
Potato	3-10	90-95	150-240
Spinach	0	95	10-14
Strawberry	-0.5-0	90-95	5-7
Tomato	4-10	85-90	4-7
Watermelon	4-10	80-90	14-20

Chilling equipment is classified by the method used to remove heat, into:

- mechanical refrigerators
- cryogenic systems.

Batch or continuous operation is possible with both types of equipment, but all should lower the temperature of the product as quickly as possible through the critical warm zone (5–10°C) where maximum growth of micro-organisms occurs.

The chilling medium in mechanically cooled chillers may be air, water or metal surfaces. Air chillers (for example *blast chillers*) use forced convection to circulate air at around -4°C at high speed (4 m/s), and thus reduce the thickness of boundary films to increase the rate of heat transfer. Air-blast chillers are also used in refrigerated vehicles, but food should be adequately chilled when loaded onto the vehicle, as the refrigeration plant is only designed to hold food at the required temperature and cannot provide additional cooling of incompletely chilled food. *Eutectic plate systems* are another type of cooling that is used in refrigerated vehicles, especially for local distribution. Salt solutions (e.g. potassium chloride, sodium chloride or ammonium chloride) are frozen to their eutectic temperature<sup>3</sup> (from -3 to -21°C) and air is circulated across the plates, to absorb heat from the vehicle trailer. The plates are regenerated by re-freezing in an external freezer.

Retail chill cabinets use chilled air which circulates by natural convection. The cost of chill storage is high and to reduce costs, large stores may have a centralized plant to circulate

refrigerant to all cabinets. The heat generated by the condenser can also be used for in-store heating. Computer control of multiple cabinets detects excessive rises in temperature and warns of any requirement for emergency repairs or planned maintenance. Other energy-saving devices include night blinds or glass doors on the front of cabinets to trap cold air.

5.1.1. Chilling rate and chilling time

Chilling rate may be defined as:

$$w = \frac{dt}{d\tau} \left[ \frac{^{\circ}C}{s} \right],$$

where dt is the temperature variation during the period dτ.

In order to obtain the evolution of the product temperature during chilling and to calculate the chilling time some assumptions are considered:

- homogenous product;
- at a given time the temperature is the same in the entire mass of the product;
- the temperature of the cooling medium is constant;
- there is no mass transfer between the product and the cooling medium.

In these conditions we consider that the sensible heat removed from the product equals the convective heat transfer from the product to the cooling medium:

$$m \cdot c \cdot dt = -S \cdot \alpha \cdot (t - t_0) \cdot d\tau,$$

where m is the mass of the product, c is the specific heat, S is the surface of the product, α is the convective heat transfer coefficient and t<sub>0</sub> is the temperature of the cooling medium.

The above equation finally leads to:

$$t = t_0 + (t_i - t_0) \cdot e^{-\frac{\alpha \cdot S}{m \cdot c} \cdot \tau},$$

$$\tau_r = \frac{m \cdot c}{\alpha \cdot S} \cdot \ln \frac{t_i - t_0}{t_f - t_0},$$

where t<sub>i</sub> is the initial temperature of the product, t<sub>f</sub> is the final temperature of the product and τ<sub>r</sub> is the total chilling time.

5.1.2. Air chillers

Air chilling of foods is performed in chilling tunnels or chilling rooms; usually, the chilling tunnels operate continuously, while the batch chilling is used for chilling rooms.

In a **chilling tunnel** air circulates longitudinally, transversally or in a vertical direction (fig. 5.1, 5.2, 5.3).

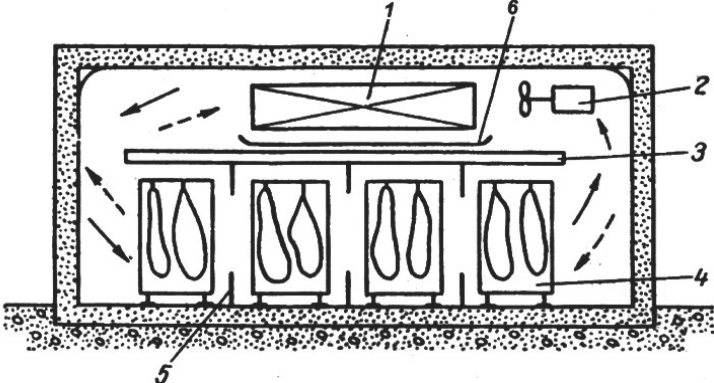


Fig. 5.1 – Air chilling tunnel, with longitudinal circulation of the air  
1-evaporator; 2-fan; 3-false ceiling; 4-product; 5-baffle.

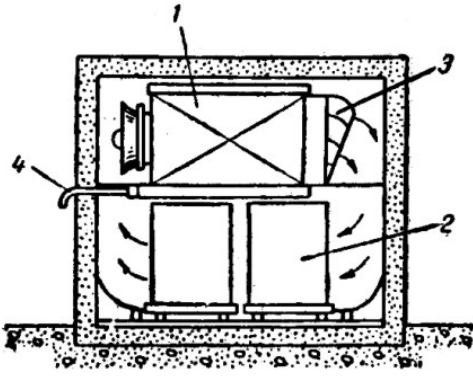


Fig. 5.2 - Air chilling tunnel, with transversal circulation of the air  
1-evaporator; 2-product; 3-baffle deflector; 4- water removal pipe.

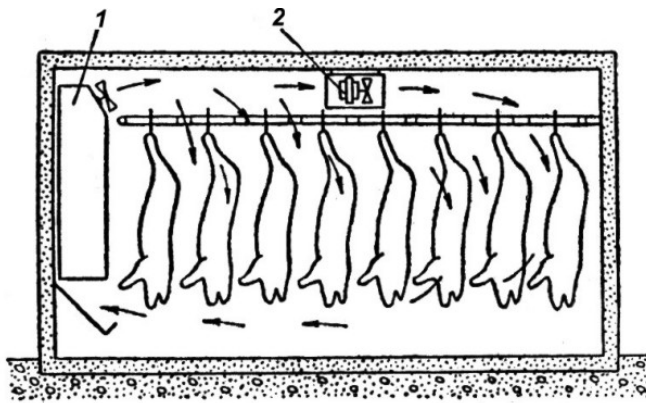


Fig. 5.2 - Air chilling tunnel, with vertical circulation of the air

- 1-evaporator;
- 2-auxiliary fan.

Two phase chilling can be used in order to speed up the process and to limit the evaporative weight loss; fig. 5.4 presents the schematic of two phase chilling tunnel.

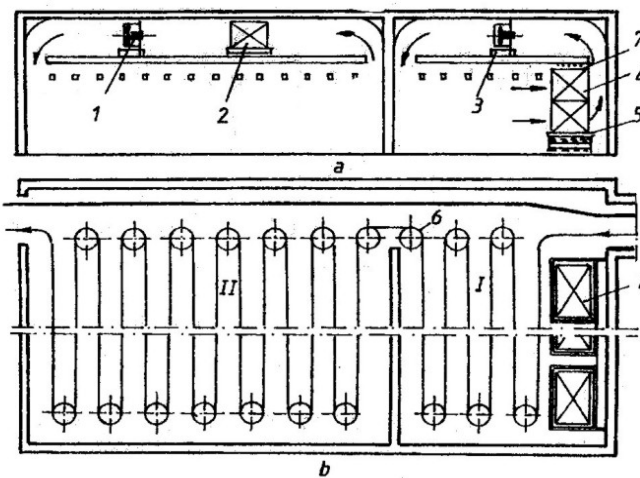


Fig. 5.4 – Two phase chilling tunnel

- 1, 3-fans;
- 2, 4-evaporators;
- 5-water collecting tray;
- 6-conveyer;
- 7-false ceiling;
- a-side view;
- b-upper view.

In the first chilling phase the air temperature is  $-10^{\circ}\text{C}$  and its speed is approx. 1 m/s; the products, hanged from the conveyer, travel trough section I of the tunnel in about 4-5 hours. The surface of the products is chilled rapidly in this first part. In the second part of the tunnel (II) air temperature is about  $0^{\circ}\text{C}$  and its speed is 0.3 m/s; the duration of the chilling process is 10-15 hours, until the product reaches a relatively uniform temperature in its entire mass. This system allows the diminishing of the evaporative weight loss compared with the one phase chilling system because the surface of the product is quickly cooled in the first phase and thus a smaller temperature difference between the product and the cooling medium is achieved in the second phase.

**Chilling rooms** have a lower capacity than chilling tunnels; due to the lower air speed (0.3 m/s), the duration of the chilling process increases. Fig. 5.5 presents some examples of chilling rooms.

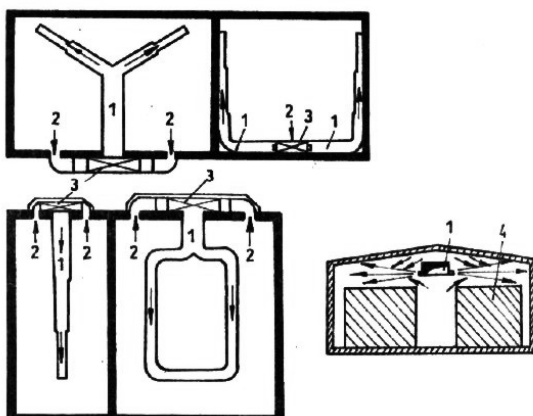


Fig. 5.5 – Chilling rooms

- 1-discharge duct;
- 2-intake port;
- 3-evaporator and fan;
- 4-products.

Both in chilling tunnels and chilling rooms air circulation over the coil of the evaporator is performed by the means of axial fans; sometimes centrifugal fans are used in chilling rooms (fig. 5.6).

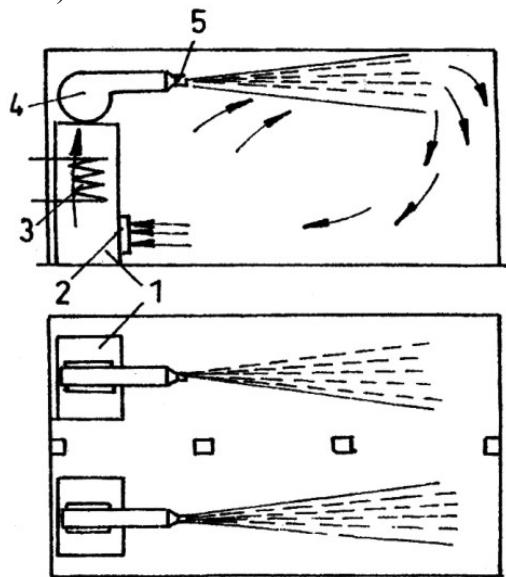


Fig. 5.6 – Chilling room with centrifugal fans for air circulation

- 1-evaporator case;
- 2-intake port;
- 3-evaporator coil;
- 4-centrifugal fan;
- 5-discharge nozzle.

#### 5.1.2. Chilling with liquid secondary agents

In this case a chilled secondary agent is used in order to refrigerate the product; the product is sprayed with cold agent or is immersed into the chilling agent. Due to the higher values of the convective coefficients, chilling with cold liquid agents requires less time than air chilling. The procedure is used for chilling poultry, fish and some vegetable products.

Depending on the final temperature of the product and on the type of product, chilling may be achieved with water, salt water or slurry ice; ice or vapor compression refrigeration systems are used in order to chill the secondary agent.

The device presented in fig. 5.7 uses ice in order to cool the water; the poultry carcasses are immersed in cold water and are moved from one to the other of the tank (5) by the blades (2) mounted on the conveyer (3). At the right end of the tank the elevator (4) extracts the chilled carcasses from the tank. Ice is produced by the ice generator (1).

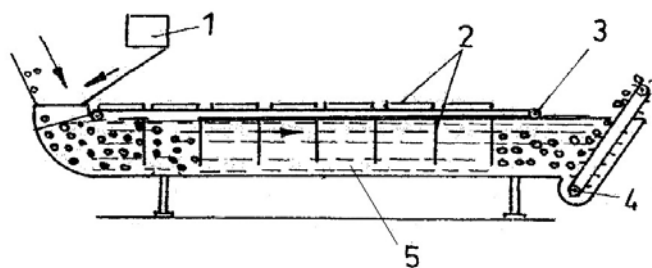


Fig. 5.7 – Poultry immersion chilling in ice water  
1-ice generator; 2-blades; 3-conveyer; 4-elevator; 5-tank.

Fig. 5.8 presents a tubular fish chiller, using sea water as a cooling medium ( $-2\dots-1^{\circ}\text{C}$ ). The seawater inside the tank is chilled by the evaporator of the vapor compression refrigeration system; the cold water is then pumped through the annulus towards the nozzles, creating a turbulent flow of water inside the fish chamber.

Slurry ice is a phase changing refrigerant made up of millions of ice “micro-crystals” (typically 0.1 to 1 mm in diameter) formed and suspended within a solution of water and a freezing point depressant; some compounds used in the field are salt (sodium chloride), ethylene glycol, propylene glycol, various alcohols (Isobutyl, ethanol) and sugar (sucrose, glucose).

This type of ice has many advantages in comparison with the traditional ice, such as flake



ice, shell ice, crush ice and so on: it can be used in direct contact with the object to be chilled; due to the large contacting area, it has the very good cooling performance; slurry ice can be pumped to the point of use, eliminating costly and maintenance intensive rakes, augers and ice conveying systems; operating at temperatures below the freezing point of water, ice slurry facilitates several efficiency improvements such as lowering the required temperature difference in heat exchangers due to the beneficial thermo-physical properties of ice slurry

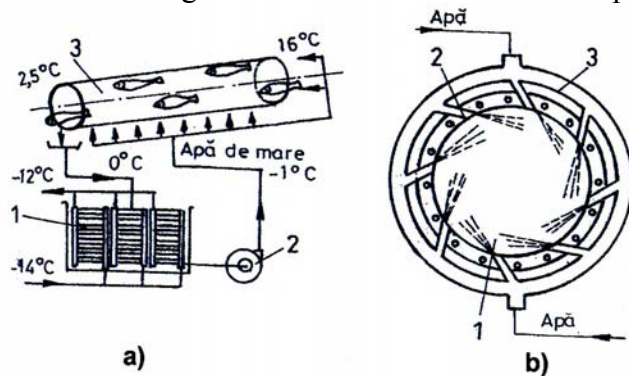


Fig. 5.8 – Tubular fish chiller

a-schematics:  
1-seawater tank;  
2-pump;  
3-tubular chiller.

b-view of the tubular chiller:  
1-fish chamber;  
2-nozzles;  
3-annulus.



Fig. 5.14 - Slurry ice with propylene glycol as depressant

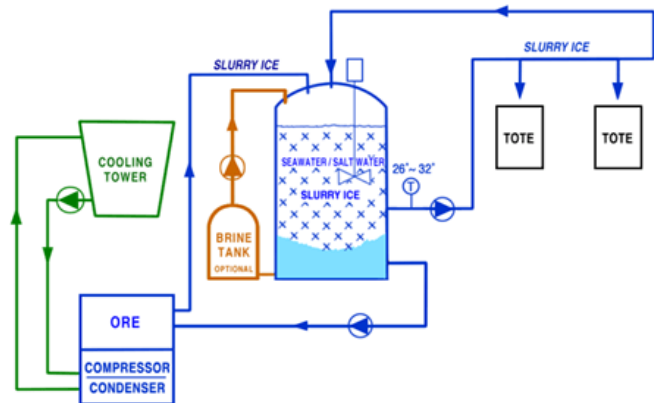


Fig. 5.15 – Schematics of a slurry ice installation  
ORE-ice crystals generator; TOTE-product packaging crate.

The slurry ice is produced in scrapped surface heat exchanger (SSHE); fig. 5.16 presents the working principle of SSHE for slurry ice.

The scrapped surface ice slurry generator consists of a circular shell-and-tube type heat exchanger, where outer shell side is cooled by an evaporating refrigerant and inside surface is scraped by spring loaded rotating blades to prevent any deposition of ice crystals on the cooled surface. This scraping action is required to prevent the formation of an ice layer on the inner walls of ice generator, which would otherwise introduce an additional thermal resistance and could seriously lower the heat transfer.

### 5.1.3. Ice chilling

This method is used for chilling fish and vegetables.

Ice is often made in the form of lumps or blocks, of various, weights from 10 to 200 kg. Alternatively, ice may be made in the form of smaller pieces (granular ice). Different types of granular ice exist such as, flake ice, tube ice, and plate ice.

In order to obtain ice blocks, the water to be frozen is filled into large metal moulds, which are left in a large tank of refrigerated brine for up to 24 hours (fig. 5.17). When the ice blocks are completely frozen, they are removed from the freezing tank and dipped into hot water. This melts the surface of the block so that it can be tipped out of its mould. The moulds can then be refilled with water, and then returned to the freezing tanks. The large ice blocks may then be broken down into smaller pieces in a crushing machine (fig. 5.18).

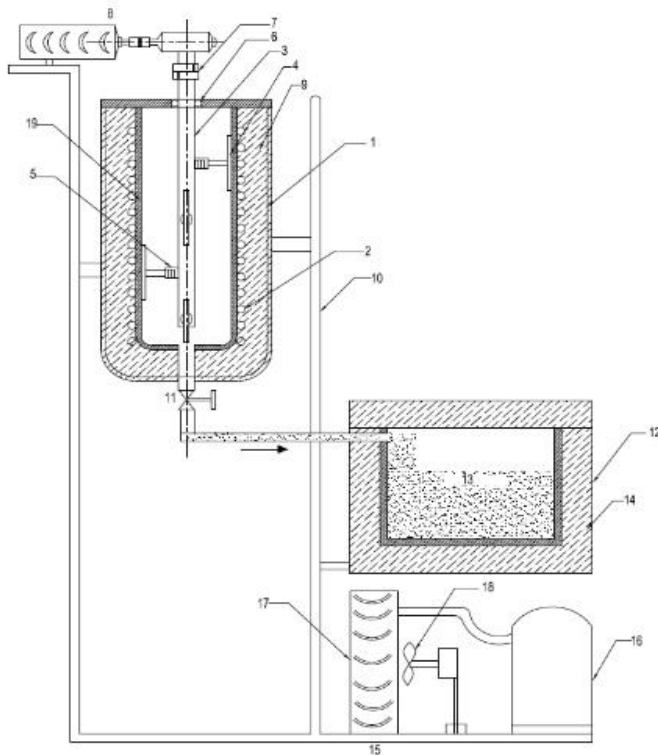


Fig. 5.16 - Schematic diagram of ice slurry generator<sup>3</sup>

- 1 - ice slurry generator;
- 2 - evaporator coil;
- 3 - scraper shaft;
- 4 - scraper blade;
- 5 - spring;
- 6 - bearing;
- 7 - coupling;
- 8 - motor;
- 9 - insulation;
- 10 - frame;
- 11- valve;
- 12 - ice slurry tank;
- 13 - ice slurry;
- 14 - insulation;
- 15 - condensing unit;
- 16 - compressor;
- 17 - condenser;
- 18 - condenser fan;
- 19 - inner shell.

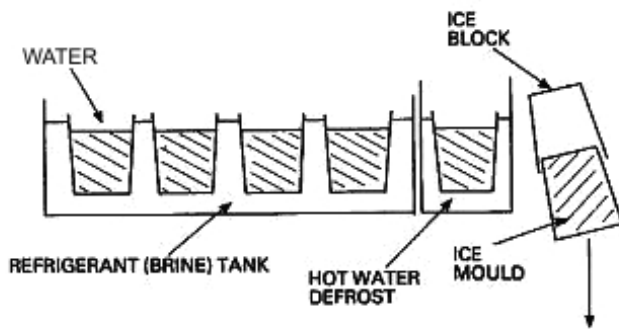


Fig. 5.17 - Production of lump ice

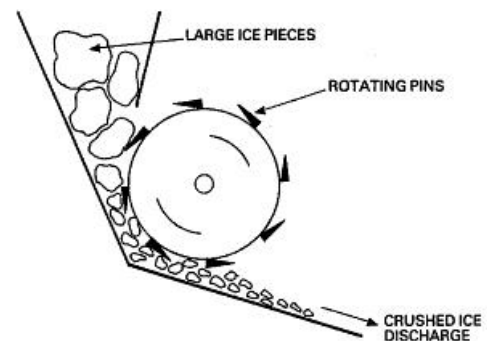


Fig. 5.18 - The crushing machine

The most common type of granular ice is flake ice. This is made by spraying water onto the surface of a rotating, refrigerated drum (fig. 5.19). The water freezes into a 2-3mm thick layer of ice. This is scraped off the drum as dry flakes of ice.

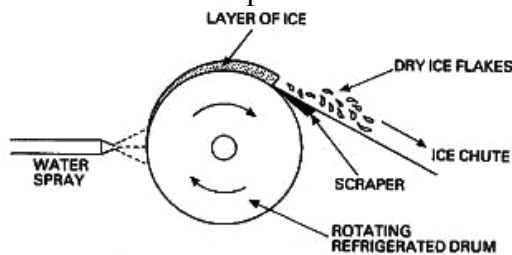


Fig. 5.19 - Production of flake ice

Fish stowed in plenty of ice are typically at a temperature of  $-0.5^{\circ}\text{C}$ . The ice has to do more than just chill the fish; it usually has to remove heat from the surrounding structure of the box or the storage room, it has to absorb the heat input through the structure during stowage from the warm air and sea outside, and it may also have to remove heat produced by the spoilage process in the fish themselves. It is therefore essential that plenty of ice is properly distributed throughout the catch to ensure efficient cooling. Ideally, each fish should be in contact only with ice and not even with other fish because fish touching one another do not cool as rapidly as those

<sup>3</sup> [http://www.ijesit.com/Volume%202/Issue%204/IJESIT201304\\_18.pdf](http://www.ijesit.com/Volume%202/Issue%204/IJESIT201304_18.pdf)

completely buried in ice. In practice, there are alternating layers of ice and fish (fig. 5.20).

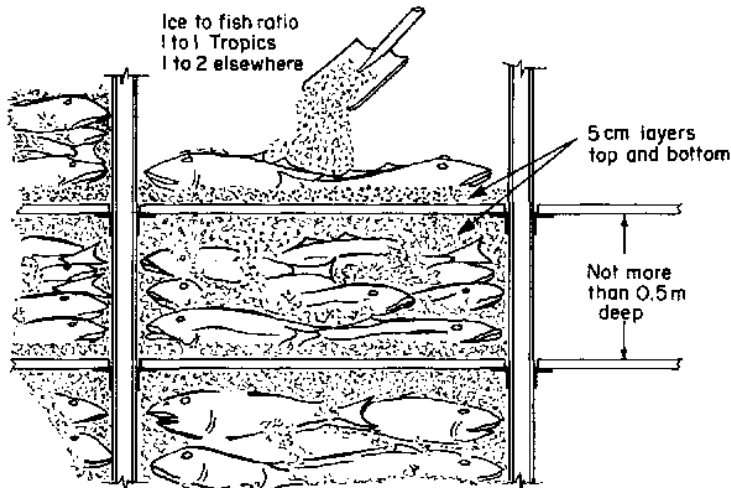


Fig. 5.20 – Fish chilling with ice

Vegetables are placed in wooden crates which are filled with ice. Fig. 5.21 presents the schematics of an ice filling machine; the ice flakes flow into the machine hopper (2) through the chute (1) and are poured into the crates through the hopper (4).

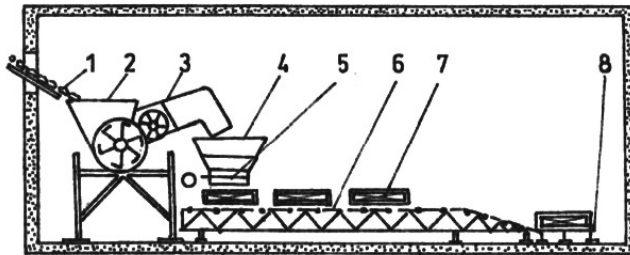


Fig. 5.21 – Ice filling machine

- 1-chute;
- 2, 4-hoppers;
- 3-ice feeder;
- 5-flap;
- 6, 8-conveyers;
- 7-crates.

#### 5.1.4. Chilling in heat exchangers

This chilling method is used for liquid food products (milk, cream, juices, beer, wine etc.). Refrigerants or secondary cooling agents are used on the cold side of the chilling device, in batch or continuous operation.

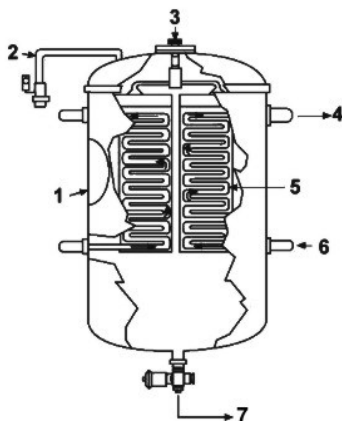


Fig. 5.22 – Chilling tank for batch operation  
1-tank; 2-CO<sub>2</sub> flange; 3-juice intake connection; 4, 6-refrigerant flanges; 5-coil; 7-juice out connection.

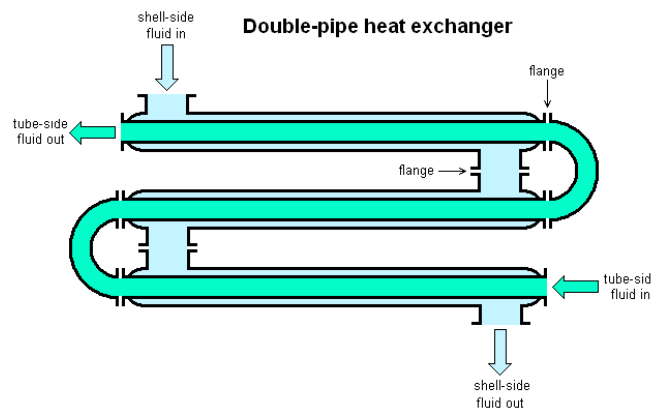


Fig. 5.23 – Double pipe heat exchanger

In batch operation mode, the product is fed into the tank (1, fig. 5.22) is cooled by the coil (5); when the final temperature is obtained, the product is purged from the tank through the discharge connection (7).

Fig. 5.23 presents a double pipe heat exchanger; the cooling medium circulates through the inner pipe, while the product circulates through the shell. The exchanger contains several

sections which are coupled through flanges; the number of sections depends on the temperature and flow rate of the fluids.

## 5.2. Food freezing

Freezing represents a preservation process for food where the product temperature is decreased to a temperature range resulting in the formation of ice crystals within the product structure.

Freezing is regarded as one of the best methods for long-term food preservation; frozen foods have an excellent safety record and freezing has never been reported to be the cause of food poisoning.

During freezing, water is converted to ice, thus chemical reactions and microbial growth are reduced at low temperatures; this apart, the formation of ice removes water from food systems, lowering the water activity.

The great advantage of freezing is that micro-organisms do not grow in foods when the temperature is  $-10^{\circ}\text{C}$  or colder. Foods preserved by means of other preservation methods (chilling, drying, curing, canning, etc.) have been more or less directly involved in food safety problems, because these foods are stored at temperatures that allow microbial growth. However, it should not be overlooked that although freezing kills some micro-organisms, it does not eliminate pathogenic micro-organisms nor microbial toxins present in the food product prior to freezing.

The final temperature of the frozen foods depends on the type of product; for many of them (meat, fish, ready-prepared meals) the final temperature is comprised between  $-18^{\circ}\text{C}$  and  $-25^{\circ}\text{C}$ .

### 5.2.1. The freezing process

Freezing of food starts when the food is placed in contact with a cold medium, which can be solid (for example, heat exchanger plates at  $-30$  to  $-40^{\circ}\text{C}$ , solid carbon dioxide (dry ice) at  $-78.5^{\circ}\text{C}$ ), liquid (immersion in a cooling mixture or cryogenic fluid such as liquid nitrogen at  $-196^{\circ}\text{C}$ ) or gas (a stream of air, gaseous nitrogen or  $\text{CO}_2$ ). The surface of the food cools faster than the centre of the food because the heat from the interior of the food has to reach the surface by conduction.

Figure 5.24 shows a typical temperature record during freezing, featuring the following phases: cooling from the initial temperature  $T_i$  to the super-cooling temperature  $T_s$  (phase I), freezing (phase II) and cooling until the final temperature  $T_F$  is reached (phase III). The temperature of the food may show super-cooling (point A) before increasing momentarily to approximately the initial freezing temperature  $T_F$ , and thereafter continuing along the B–C part as transfer of the latent heat of freezing from the food begins. The first ice crystals are formed between A and B and then further crystals are formed and grow between B and C.

Below its initial freezing point, a liquid is said to be super-cooled. This is a metastable state of the liquid; the liquid can continue to be in this state for a very long time, before nucleation of the first crystal takes place. Following this the crystals grow and spread throughout the volume rapidly. Pure water (free of impurities such as dust particles that would act as nucleation centers) can be super-cooled to around  $-40^{\circ}\text{C}$ . At lower temperatures water freezes due to homogeneous ice nucleation; in foods the degree of super-cooling is much smaller than in pure water because of heterogeneous nature of the solution and heterogeneous ice nucleation takes place. Super-cooling is important in nature since this is one of the mechanisms by which living plants and animals cope with sub-zero temperatures or minimize the damage of their tissue that ice formation can cause.

Ice formation undergoes two stages: nucleation and subsequent crystal growth.

Nucleation refers to the process by which a minimum crystal is formed with a critical radius which can then expand and grow. During nucleation the latent heat of solidification is

released (A-B, fig. 5.24); molecules aggregate into an ordered particle of a sufficient size to survive and serve as a site for further crystal growth.

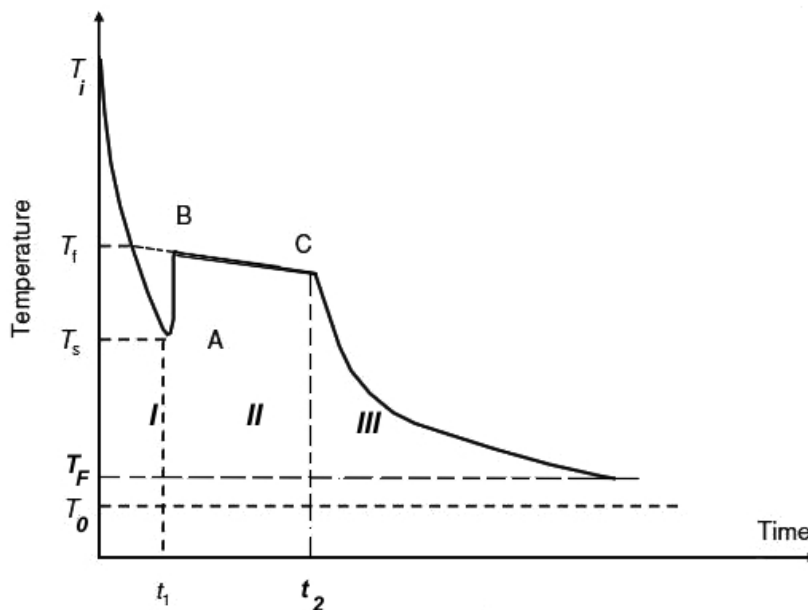


Fig. 5.24 – Freezing curve for foods

Nucleation (formation of ice seeds) takes place when a group of molecules gather in order to form an ice nuclei. Nucleation is necessary for freezing to initiate and the temperature can fall below  $0^{\circ}\text{C}$  without the formation of ice crystals. Point A (fig. 5.24) indicates the super-cooling of the liquid before crystallization begins.

Nucleation can be homogeneous or heterogeneous. Homogeneous nucleation occurs only in homogeneous particle-free liquids and happens due to random fluctuations of molecules (the random clusters of molecules momentarily assume the configuration of ice and act as seeds). In solid foods the nucleation is heterogeneous, with the cell surfaces acting as nucleation sites. The probability of nucleation at a site is enhanced if the molecular structure of the surface resembles that of ice, i.e. matches the lattice size of the ice crystal and acts as a template. This happens notably with ice nucleation active (INA) proteins found in some bacteria and plants.

As long as a stable ice nucleus is formed, further growth is possible by addition of molecules to the solid–liquid inter-phase. Growth is not instantaneous and is controlled by the rate of removal of the latent heat released during the phase change, as well as by the rate of mass transfer in the case of solutions (diffusion of water molecules from the surrounding solution to the surface of the ice crystals and counter-diffusion of solutes away from the growing crystal surface). The rate of crystalline growth is also a function of the super-cooling reached by the specimen.

Cooling from B to C (fig. 5.24) results in the growth of ice crystals and a substantial ice formation. A gradual increase in solute concentration is produced as water is separated in the form of relatively pure ice crystals and the declining freezing point (negative slope of B–C in fig. 5.24) reflects the change in concentration.

The freezing point of the solution is lower than the freezing point of pure water. The freezing point depression is determined by the number of dissolved solute molecules. Table 5.2 presents the freezing points of some food products.

### 5.2.2 Freezing rate

The rate of freezing determines the ice crystal size; the faster the rate, the more the nucleation, and a greater number of crystals of smaller size will result. The first simple definition of freezing rate is the rate of temperature change. In very small specimens, temperature gradients can be practically neglected and all the points in the sample will have similar freezing rates. However, in large-sized systems, temperature gradients along the sample are established; then freezing rate is position dependent and varies along the frozen sample. High freezing rates are

observed on the surface in contact with the refrigerant decreasing towards the thermal center [Handbook of frozen foods].

Table 5.2

Average Freezing Points of Some Food Categories

Food	X (%)	T <sub>f</sub> (°C)
Vegetables	78 to 92	-0.8 to -2.8
Fruits	87 to 95	-0.9 to -2.7
Apple juice	87.2	-1.44
Apple sauce	82.8	-1.67
Apple juice concentrate	49.8	-1.13
Meat	55 to 70	-1.7 to -2.2
Milk	87	-0.5
Egg	74	-0.5

Note: X represents the percent of frozen water.

For a given point in the sample, the freezing rate can be represented by the characteristic local freezing time ( $\tau_c$ ) that is the time needed to change the temperature from the initial freezing point to a temperature for which, for example, the 80% of the total water content is converted to ice. Each point along the sample will have a different  $\tau_c$  value [Handbook of frozen foods].

According to the International Institute of Refrigeration [Handbook of frozen foods], the freezing rate of a food is defined as the ratio between the minimum distance from the surface to the thermal center, and the time elapsed between the surface reaching 0°C and the thermal center becoming 10°C colder than the temperature of initial ice formation. In commercial practice, freezing rates vary between 0.2–100 cm/h; 0.2–0.5 cm/h correspond to slow freezing (bulk freezing in cold chambers), 0.5–3 cm/h to quick freezing (air blast and contact plate freezers), 5–10 cm/h to rapid freezing (individual quick freezing of small sized products in fluidized beds) and 10–100 cm/h to ultra rapid freezing by spraying or immersion in cryogenic fluids (liquid nitrogen, carbon dioxide).

The rates of freezing determine the type, size and distribution of ice formation. Slow freezing produces fewer larger ice crystals, fast freezing produces a greater number of smaller crystals. Whether large or small crystal size is preferable depends on the purpose of freezing. In ice cream, the ice crystals must be as small as possible so as to make the product as creamy and smooth as possible. However, to concentrate liquid food products, large crystals are easier to separate from the freeze concentrate. In freeze drying it is usually desirable to produce a small number of large crystals in order to accelerate the subsequent sublimation process. In foods that are frozen slowly, large ice crystals form and ice fills the extracellular space, causing dehydration of the cells. The ice crystals force the cells or tissue fibers apart. Although foods that are quick (flash) frozen produce small ice crystals, these ice crystals may grow larger over time through a process known as recrystallisation or Ostwald ripening (Smith and Schwartzberg, 1985). Recrystallisation occurs in frozen foods because larger crystals are thermodynamically more stable

### 5.2.3. Freezing time

Numerous equations and approaches to freezing time ( $\tau_c = t_2 - t_1$ , fig. 5.24) prediction have been proposed and utilized. One of the best known and most used of the prediction methods is based on Planck's equation (1913):

$$\tau_c = \frac{\rho_p \cdot l_{cp}}{t_p - t_0} \cdot \left( \frac{P \cdot h}{\alpha} + \frac{R \cdot h^2}{\lambda_p} \right),$$

where  $\rho_p$  is the product density,  $l_{cp}$  is the latent heat of freezing of food,  $t_0$  is the temperature of the cooling medium,  $t_p$  is the freezing temperature of the product,  $h$  is the thickness/dimension of object,  $\alpha$  is the convective heat transfer coefficient,  $\lambda_p$  is the thermal conductivity of the product and  $P$  and  $R$  are used to account for the influence of product shape.

#### 5.2.4. Types of food freezing systems

Industrial food-freezing systems can be divided into two broad groups — those using air as the cooling medium and those using other cooling mediums. Air-blast freezing systems can be further subdivided into still-air, blast-room or tunnel, belt, spiral-belt, and fluidized bed freezers. Other systems include plate, immersion, cryogenic freezers, and freezers for liquid foods.

##### a) Air-blast freezers

Air-blast freezers are the most common type of food freezer. Individual product items are placed in a recirculating air stream within a room or tunnel. The air is circulated by fans, which are often associated with the evaporator coils providing cooling. These freezers can be simple, operating in **batch** mode with manual loading and unloading of the product (fig. 5.25, 5.26), or more complex, with automated **continuous** operation (fig. 5.27, 5.28). Continuous freezers are best suited to processing large volumes of product. They have lower labor costs, and generally provide more uniform freezing conditions, but are less flexible.

Many different air and product flow configurations can be used. Horizontal air flow is probably most common (fig. 5.25), but there are a number of designs using vertical air flow, often to avoid air bypassing the product (fig. 5.25). In continuous systems, air and product flows can be co-current, countercurrent or cross-flow (fig. 5.27, 5.28). The latter two configurations are most common as air temperature rise is small, so the temperature driving force for cooling and thus the rate of heat transfer are maximized. Methods to present the product to the air depend on the size, shape and packaging of the product, and include trays, racks, trolleys, hooks, conveyors, and belts. A wide range of product types, sizes, shapes, and packaging types can be handled by such freezers.

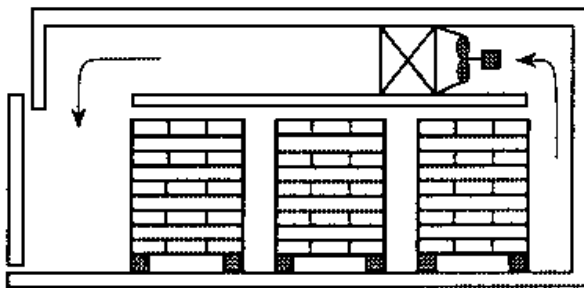


Fig. 5.25 - Schematic diagram of a batch air-blast tunnel freezer with racks of product and horizontal air flow.

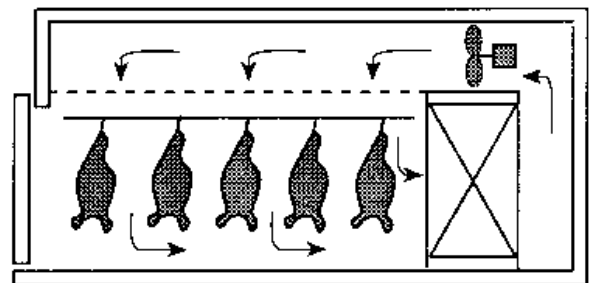


Fig. 5.26 - Schematic diagram of a batch air-blast carcass freezing room with vertical air flow through a distribution plenum.

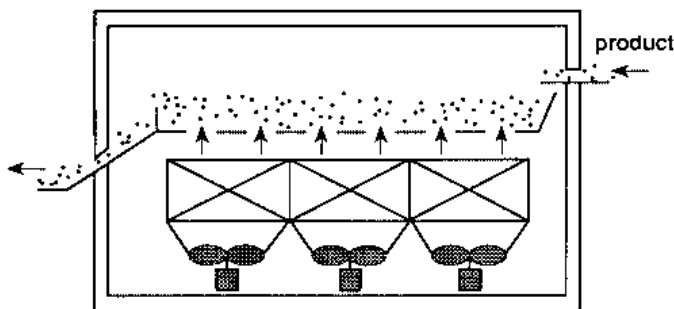


Fig. 5.27 - Schematic diagram of a continuous fluidized bed freezer. Belt freezers are similar but product is transported on a perforated conveyor belt.

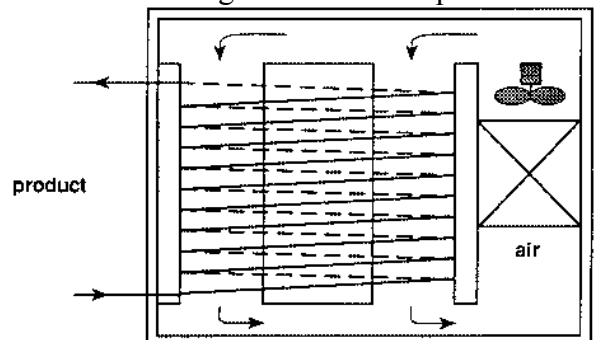


Fig. 5.28 - Schematic diagram of a continuous spiral belt freezer with vertical air flow

The major advantages of air-blast freezers are their simplicity and flexibility. The disadvantages are that using air limits the rate of heat transfer at the product surface, requires the use of substantial fan energy, and to achieve uniform air distribution can be difficult. Also, further disadvantages are that: evaporative weight loss can be significant from unwrapped product; bulging of packaged product can occur; defrosting evaporator coils or another means of frost prevention is required which can disrupt freezer operation; and the refrigeration system must operate with a low suction condition due to the air-to-refrigerant heat exchange.

*a.1) Still-Air Freezers*

The simplest type of freezer is one in which the product is placed in a refrigerated room that is usually used to store frozen product. The process is historically referred to as sharp freezing.

The shelves on which the product is placed within the room may be directly refrigerated, and the product may be bulk-stacked. Air flow over the product is minimal and the freezing rate is slow. Also, heat removal from the freezing product may cause undesirable temperature fluctuations in adjacent stored product.

*a.2) Air-Blast Room and Tunnel Freezers*

These freezers are commonly used for medium to large products where the rate of freezing is limited by the size of the product. The product does not need to be regular in shape. The product is placed on trays in racks or suspended so that air flow is possible around each individual product item. In continuous tunnel freezers, it is usual to have a mechanical system moving racks through the tunnel in a cyclic manner, automatic loading and unloading of the racks, and product arriving and leaving on a conveyor system (fig. 5.29, 5.30).

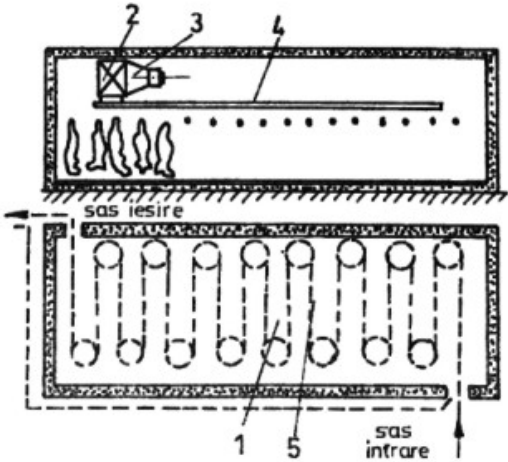


Fig. 5.29 – Continuous tunnel freezer

- 1-cooling space;
- 2-evaporator;
- 3-fan;
- 4-false ceiling;
- 5-conveyor.

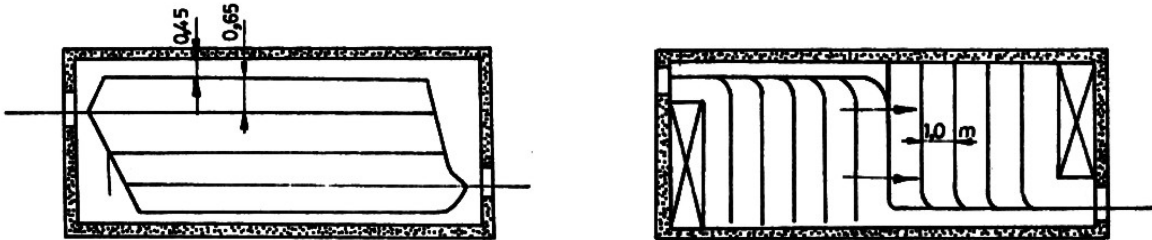


Fig. 5.29 – Conveyor systems in tunnel freezers

For batch freezers, the racks are manually loaded and positioned in the room or tunnel (fig. 5.25). In a tunnel system, the air is confined to flowing in the cross-section where the product is located. Also, the product is spaced evenly so that uniform air distribution and high air velocity is more easily achieved for a low total air flow rate and fan power. While most continuous tunnel freezers are restricted to one product size and shape in order to optimize the product loading configuration and air flow distribution, a range of products can be processed in



the same tunnel if a variety of rack sizes and tray spacing are used. In a blast room there is often less strict control of the air flow pathway and bypassing of the air around the product can more easily occur.

#### a.3) Belt Freezers

Belt freezers involve the product passing continuously through a tunnel freezer on a perforated belt (fig. 5.31). The air flow is directed vertically up through the belt and product layer.

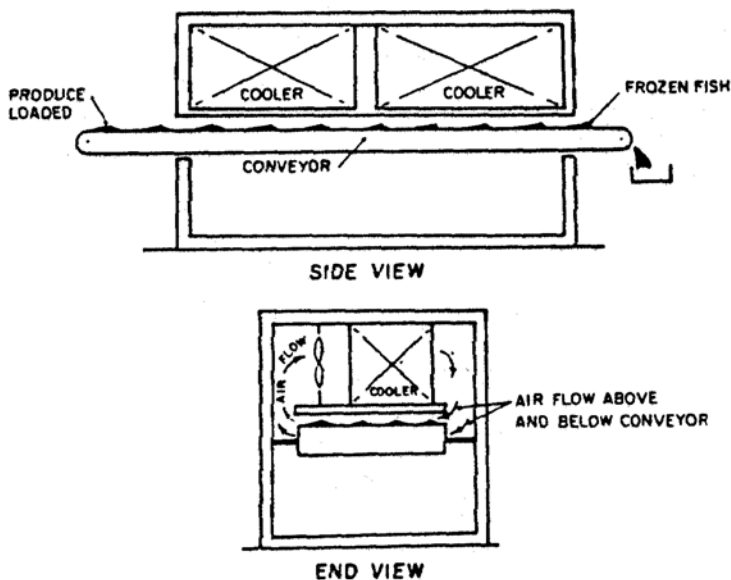


Fig. 5.31 – Belt freezer

There may be multiple belt passes (fig. 5.32); such freezers are commonly used for small unwrapped products with uniform shape in which a free-flow individually quick frozen (IQF) product is desired. The air velocities are typically in the range 1 to 6 m/s and the layer of product can be partially fluidized. This creates high rates of heat transfer between the air and product. Even distribution of the product across the belt is important to achieve uniform air distribution and freezing rate. Product transfers from one belt to another and/or mechanical devices are sometimes installed to reduce clumping and to redistribute the product. The belt speed can be varied to cope with changing production rates but care must be taken to maintain a uniform thickness of product on the belt.

#### a.4) Spiral Belt Freezers

Spiral freezers are a specialized type of belt freezer in which a continuous belt is stacked in a spiral arrangement (fig. 5.32); the belt (1) is wrapped around the rotating drum (3) allowing the products to be moved from the loading gate towards the discharge gate. Spiral freezers allow very long belts (long product residence times) in a compact area as long as sufficient overhead space is available. Therefore, they are suitable for processing products with longer freezing times compared with other belt freezers (e.g., larger products and packaged products for which the packaging impedes heat transfer). The size of the product is limited by the distance between each spiral tier and the total height of the stack. Air flow can be either horizontal across or vertical through the belts. Recent design improvements have included self-stacking belts to reduce mechanical wear and maintenance, and cleaning-in-place of the belt and freezer.

#### a.5) Fluidized Bed Freezers

Fluidized bed freezers are only suitable for small unwrapped IQF products of uniform size and shape, such as fruits and vegetables for which the energy requirements for fluidization are not excessive. In a manner similar to belt freezers, air is directed up through a perforated

plate and bed of the product but at a flow rate high enough to fluidize the product (fig. 5.33).

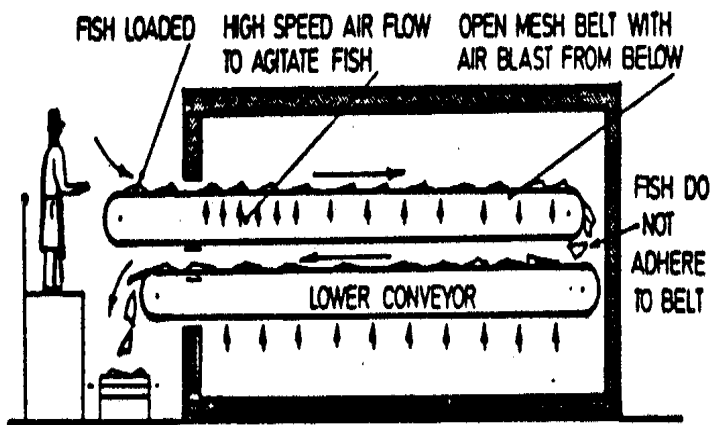


Fig. 5.31 - Semi-fluidized flow freezer with double belt

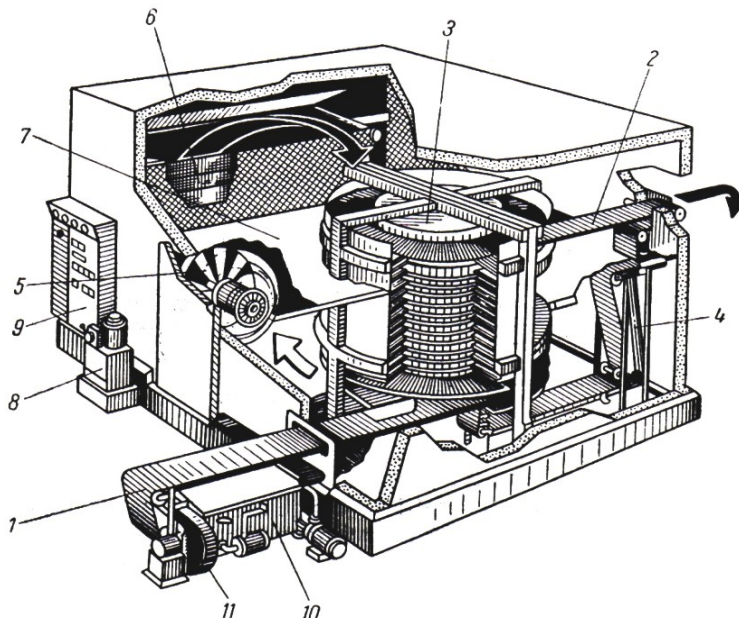


Fig. 5.32 – Spiral belt freezer

- 1-belt;
- 2-discharge gate;
- 3-rotating drum;
- 4-belt tensioning device;
- 5-fan;
- 6-air cooler;
- 7-fairing;
- 8-driving system;
- 9-electrical panel;
- 10-clean-in-place system;
- 11-belt drying fan.

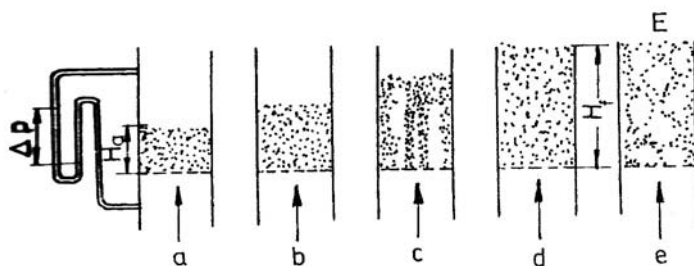


Fig. 5.33 – Formation of the fluidized bed

- a-fixed bed (initial phase; no air movement);
- b-loosening of the material layer (air speed too low);
- c-beginning of fluidization (air speed too low);
- d-fluidized bed (adequate air speed);
- e-entrainment of the material (air speed too high).

The product is fed in at one end and overflows out of the freezer at the other. Fluidization achieves good distribution of the product and prevents clumping, even with very wet incoming product, and the surface heat transfer is significantly enhanced. The product moves by flowing within the fluidized bed, but this can be aided by vibrating and/or sloping the air distribution plate. Individual items reside for different periods depending on the flow pattern in the bed. Fluidized bed freezers can be very compact because the small product size and high rates of convective heat transfer keep freezing times short.

Fig. 5.34 presents the construction of a belt fluidized bed freezer; the fluidized is obtained due to the combined action of the air current and oscillatory movement of the conveyor belts.

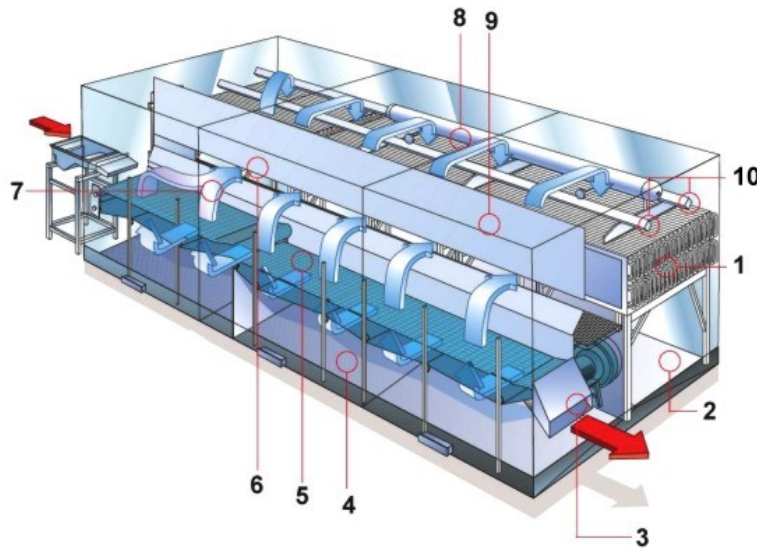


Fig. 5.34 – Belt fluidized bed freezer

- 1-evaporator;
- 2-fans discharge plenum;
- 3-discharge chute;
- 4-thermally insulated floor;
- 5-conveyor belts;
- 6, 9-air ducts;
- 7-air current;
- 8, 10-deicing systems.

Table 5.3

Freezing time in fluidized bed freezers

<u>Product</u>	<u>Time to reduce temperature from 21°C to -18 °C, min</u>
Peas	3-4
Lima beans	4-5
Diced carrots	6
Cut green beans	5-12
Strawberries	9-13
French fried potatoes	8-12
Fish fillets	30

*b) Contact freezers*

In contact freezers the products are in direct contact with cold metal surfaces; this method is used for freezing regular shaped (packs of fish).

Fig. 5.35 and 5.36 present some types of metal plates used for food freezing; the refrigerant circulates through pipes (fig. 5.35) or through internal channels created between the plates (fig. 5.36).

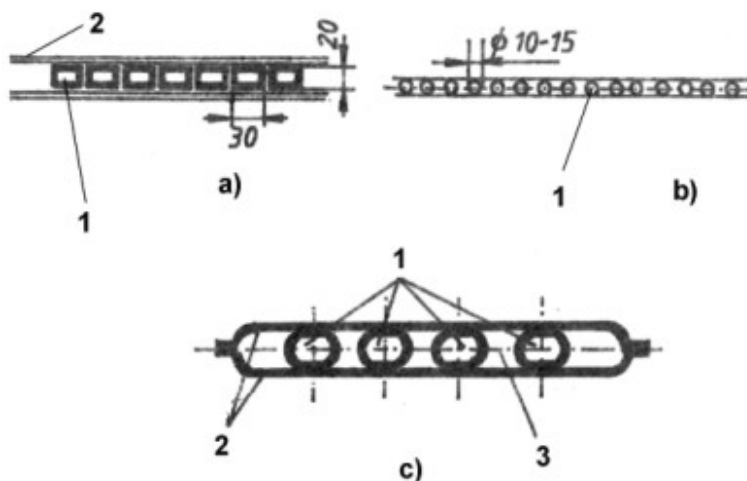


Fig. 5.35 – Metal plates for contact freezing

- a-square section pipes;
- b-round section pipes;
- c-elliptical section pipes;
- 1-pipes;
- 2-metal plates;
- 3-vacuumed space.

The plates are mounted parallel to each other and may be either horizontal or vertical. The spaces between the plates are variable, the plates being opened out for loading and, prior to the freezing operation, closed so that the surface of the plates is in intimate contact with the

packaged or unpackaged food.

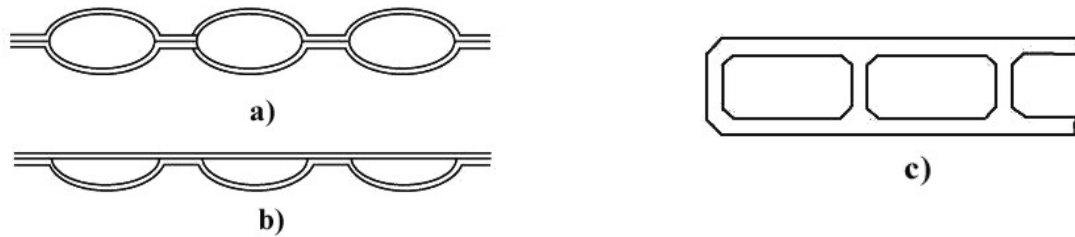


Fig. 5.36 – Metal plates for contact freezing  
a, b-welded metal plates; c-extruded.

Double contact plate freezers (fig. 5.37) are commonly used for freezing foods in retail packages. This equipment, which may be batch, semi automatic, or automatic, consists of a stack of horizontal cold plates with intervening spaces to accommodate single layers of packaged product. The filled unit appears much like a multilayered sandwich containing cold plates and product in alternating layers. When closed, the plates make firm contact with the two major surfaces of the packages, thereby facilitating heat transfer and assuring that the major surfaces of the packages do not bulge during freezing.

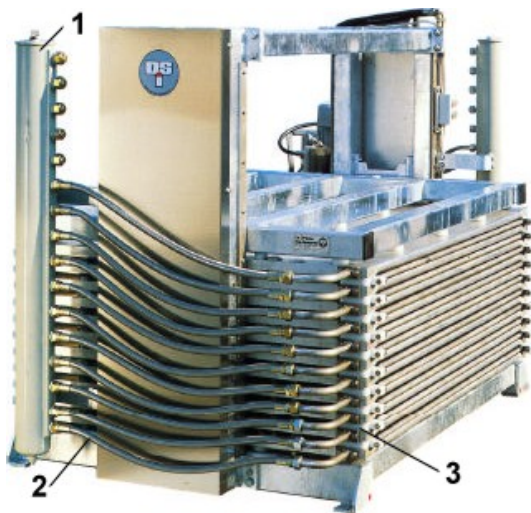


Fig. 5.37 – Horizontal plates freezer

1-refrigerant pipe;  
2-hose;  
3-plates.

Contact plate freezing is an economical method that minimizes problems of product dehydration, defrosting of equipment and package bulging. Two notable disadvantages do however exist:

- packages must be uniform of thickness;
- freezing occurs at a moderately slow rate as compared to other modern methods. For example, a compact product in a well fitted package 2.5- 4 cm thick, when cooled by plates at  $-33^{\circ}\text{C}$ , requires about 1-1.5 hr to freeze. Freezing times are extended considerably when the package contains a significant volume of void space.

Cold metal belt freezers (fig. 5.38) use a secondary agent, cooled at  $-35\dots-40^{\circ}\text{C}$ , which is sprayed over the inferior surface of metal belt conveyor (4). The products freeze in contact with the cold surface of the belt; cold air is used to freeze the upper part of the products.

### c) Liquid Immersion Freezing

Liquid immersion freezing (usually referred to as direct immersion freezing) is accomplished when a food product, either packaged or unpackaged, is frozen by immersion in or by spraying with a secondary agent that remains liquid throughout the process. Aqueous solutions of the following substances have been used as secondary agents: propylene glycol, glycerol, sodium chloride, and mixtures of salt and sugar. This technique, although not common,

is used commercially for canned citrus juice concentrate (cans of juice are passed continuously through a chamber containing the cold liquid); for poultry especially during the initial stages of freezing ( to impart a uniform, white color to the surface); occasionally for fish and shrimp.

The major advantages of liquid immersion freezing are that it results in rapid freezing (especially for foods which are unpackaged or are packaged in skin-tight films) and it is easily adapted to continuous operations.

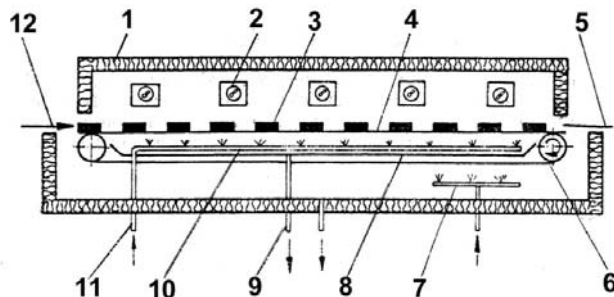


Fig. 5.38 – Cold metal belt freezer

1-casing; 2-air coolers; 3-product; 4-metal belt conveyor; 5-product discharge shute; 6-head pulley; 7-cleaning system; 8-secondary agent collecting tray; 9, 11-secondary agent pipes; 10-sparing nozzles; 12-loading shute.

#### d) Cryogenic freezing

Cryogenic freezing refers to very rapid freezing achieved by exposing food items, unpackaged or thinly packaged, to an extremely cold cooling agent, undergoing a phase change. The fact that heat removal is accomplished during a phase change is used to distinguish cryogenic freezing from liquid-immersion freezing. The most common food grade cryogenic agents are liquid nitrogen (LN) and liquid carbon dioxide (LCO<sub>2</sub>).

The rate of freezing obtained with cryogenic methods is much greater than that obtained with air-blast or plate freezing, but is only moderately greater than that obtained with fluidized bed or liquid immersion freezing. During cryogenic freezing, the ice crystals remain small, avoiding damage to the cell walls (fig. 5.39). As a result, the defrosted product is fresh and appetizing.

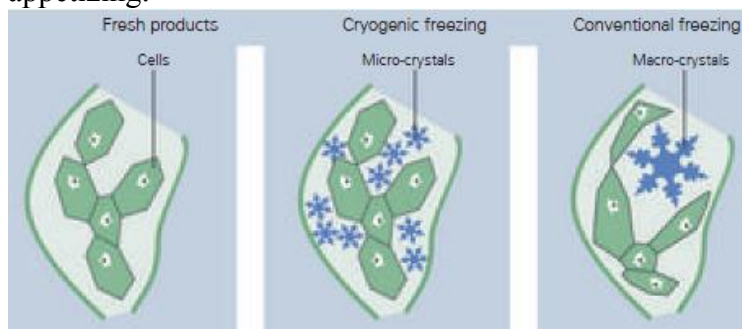


Fig. 5.39 – Rapid freezing results in small ice crystals

Nitrogen (N<sub>2</sub>) is an inert, reaction-inhibiting gas which is produced in its liquid form through cryogenic air separation; N<sub>2</sub> is the main component of air with 78 volume percent. Cryogenic liquid nitrogen (LIN) boils at -196 °C under atmospheric pressure. At 3 bar in a storage tank, liquid nitrogen has a boiling temperature of -185 °C. When applied to the product surface, LN evaporates and absorbs around 183 kJ/kg of heat (enthalpy of evaporation). The heat absorption is doubled if the cold nitrogen gas is used to pre-cool the products, as shown in fig. 5.40. Here the product is placed on a conveyor belt and moved into the insulated chamber, where it is cooled with moderately cold gaseous nitrogen moving counter current to the product. LN is sprayed or dribbled on the product. Following the desired exposure time, the product pass to the place where it is allowed to equilibrate to the desired final temperature (-18° to -30°C) before it is discharged. The final temperature of the product is usually no different than the one obtained

through conventional methods of freezing.

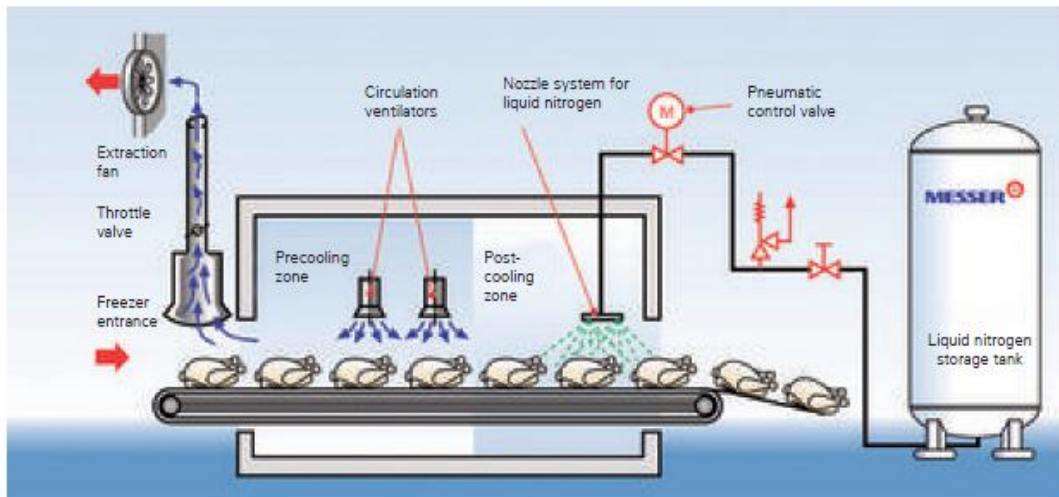


Fig. 5.40 - Process flow of a continuously operating nitrogen tunnel freezer

The immersion freezing principle may be also used with LN (fig. 5.41); a conveyor belt transports the food through the level-controlled nitrogen bath. The immersion freezer is mainly used in combination with other freezers in order to boost their effectiveness.

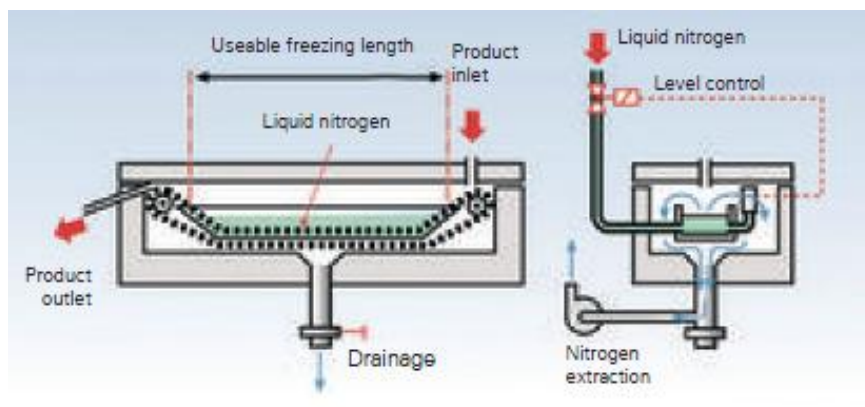


Fig. 5.41 - Operational principle of a LN immersion freezer

Carbon dioxide ( $\text{CO}_2$ ), also known as carbonic acid in the trade, is an inert, odorless, tasteless, colorless and non-flammable gas under normal conditions ( $0^\circ\text{C}$ , 1 bar). It is obtained from natural sources or produced industrially.  $\text{CO}_2$  can only be liquefied at a pressure of 5.18 bar.

Liquid carbon dioxide is stored in thermally insulated tanks. When 1 kg of liquid carbon dioxide, which was stored at 17 bar and  $-25^\circ\text{C}$ , expands to atmospheric pressure, this produces around 0.5 kg of dry ice snow and 0.5 kg of gas with a cold content of around 285 kJ. Both  $\text{CO}_2$  phases reach a temperature of  $-78.8^\circ\text{C}$  as a result of the expansion. When brought into contact with the comparatively warm product surface, the  $\text{CO}_2$  snow sublimates.

In a carbon dioxide freezer, it is the sublimation of the solid particles that provides the majority of the refrigerating effect. To maximize the contact time between the solid carbon dioxide particles and the product, carbon dioxide is usually sprayed throughout the entire length of the freezer tunnel.

Fig. 5.42 presents the schematics of cryogenic-mechanical freezer, using liquid  $\text{CO}_2$  and cold air. The liquid  $\text{CO}_2$  is sprayed over the entire length of the conveyor belt in the cryogenic part of the device; cold air is used in the mechanical part of the system, in order to allow the the products to equilibrate to the desired final temperature. Spiral belt  $\text{CO}_2$  freezers, with a construction similar to the one of the air freezers, are also widely used.

Table 5.4 presents a comparative analysis of LN and  $\text{LCO}_2$  freezing.

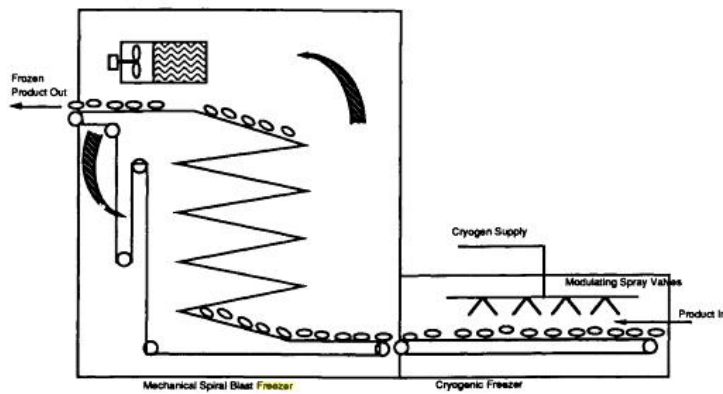


Fig. 5.42 – Cryogenic-mechanical freezer

Table 5.4

Comparative analysis of LN and LCO<sub>2</sub> freezing

	Liquid Nitrogen	Carbon Dioxide
Safety and Environmental Considerations	LN <sub>2</sub> is chemically inert. An increase in N <sub>2</sub> levels of 2% is not dangerous to humans. LN <sub>2</sub> is much easier to exhaust than CO <sub>2</sub> .	CO <sub>2</sub> is faintly reactive and toxic. OSHA's carbon dioxide exposure limit in the breathing atmosphere is just 0.5 percent. If incorrectly sized or operated, dry ice snow may collect on the process room floor.
Recommended Safety Equipment	Matheson Tri-Gas recommends an oxygen deficiency monitoring system installed in the processing room for both Liquid Nitrogen and Carbon Dioxide freezing equipment.	
Food Quality	Rapid freezing locks in moisture and preserves texture.	Slower freezing can negatively affect food quality.
Investment Costs	Lower freezing temperatures mean quicker freezing so equipment needed to produce same amount of product is smaller. Smaller size equipment means lower cost.	Higher freezing temperatures mean the equipment has to be proportionally larger to achieve the same throughput.
Operating Cost	LN <sub>2</sub> has about an 18% higher refrigeration value (see chart on back) when compared to CO <sub>2</sub> *. Therefore, on a pound per pound basis, the freezing process would use about 18% less LN <sub>2</sub> than CO <sub>2</sub> .	Even if LN <sub>2</sub> is more expensive than CO <sub>2</sub> , the price per pound of frozen product can be less using LN <sub>2</sub> since the freezing process would use less cryogen.
Maintenance Cost	Simple equipment with low maintenance requirements.	Complicated equipment requires more maintenance.
Production Rate	Lower operating temperatures and shorter dwell time result in almost twice the throughput in the same footprint. For example, a 20 foot LN <sub>2</sub> tunnel can deliver the same output as a 30 foot CO <sub>2</sub> tunnel	Higher operating temperatures and longer dwell time require 50% larger freezer to produce the same amount of product.
Operational Flexibility	Smaller freezing equipment leaves more room for expansion. Simple to operate and user-friendly. Precise sizing not as critical, can operate efficiently outside of the original design parameters.	Larger freezing equipment takes up more plant space. Less user-friendly, requires more vigilant operator. Freezer must be sized precisely for a specific product and throughput.

\*The actual value depends on the storage pressure of the gas, the type of freezer, the freezer's operating temperature, and other factors.

Table 5.5

## Commonly used freezing methods for various foods

Product Type	Commonly Used Freezing Methods
Meat, carcasses	Air blast
Meat, cartons	Air blast, plate
Meat, large individually wrapped cuts	Liquid immersion, air blast
Meat, small or diced pieces	Cryogenic, liquid immersion
Meat, cured or processed products	Air blast, liquid immersion, cryogenic
Meat, hamburger patties	Plate, impingement, air blast, cryogenic
Poultry, whole bird or pieces	Air blast, often preceded by liquid immersion or cryogenic, plate (in packages)
Poultry, processed or breaded products	Cryogenic, impingement, air blast, plate (in packages)
Fish, whole or eviscerated	Air blast, cryogenic, plate, liquid immersion (particularly aboard fishing vessels)
Fish, fillets or small diced pieces	Plate, air blast, cryogenic, impingement
Fish, minced blocks	Plate
Fish, processed or breaded products	Cryogenic, impingement, air blast, plate (in packages)
Shellfish	Air blast, cryogenic, plate (in packages)
Prawns and shrimp	Cryogenic, impingement, air blast
Fruits, small size (whole)	Air blast, cryogenic, plate (in packages), fluidized bed
Fruits, large size (sliced)	Air blast, plate (in packages)
Fruits, purée or pulp	Air blast, plate
Vegetables, small size (e.g., peas)	Fluidized bed, air blast, cryogenic, plate (in packages)
Vegetables, medium size (e.g., corn cobs)	Air blast, plate (in packages)
Vegetables, leafy (e.g., spinach)	Air blast, plate (in packages)
Cheese and butter	Plate, air blast
Dough, bread, and baked products	Air blast, plate
Pre-cooked ready meals	Air blast, plate (in packages)



## 6. Air conditioning

### 6.1. Air psychrometry

The term psychrometry (or psychrometrics) derives from the Greek *psuchron* (ψυχρόν) meaning "cold" and *metron* (μέτρον) meaning "means of measurement". Psychrometrics is the science of studying the thermodynamic properties of moist air.

Atmospheric air (moist air) is a mixture of dry air and water vapor. The amount of moisture vapor in the air depends upon atmospheric pressure and temperature. When the air is hot, it can contain a large amount of moisture vapor, sometimes as much as 5% by volume. When it is cold, its capacity to hold the moisture is reduced. When the temperature of warm air begins to fall, the vapor also cools and, if cooling continues, it will condense into tiny moisture droplets. In the atmosphere this results in the formation of clouds and eventually rain.

When air contains the maximum amount of water vapors it is said to be **saturated**; the maximum amount of water vapors increases with temperature and pressure.

Dry air and water vapors may be considered as perfect gases; therefore Dalton's law may be applied:

$$p_B = p_a + p_v,$$

where  $p_B$  is the barometric pressure,  $p_a$  is the partial pressure of dry air and  $p_v$  is the partial pressure of water vapors.

#### 6.1.1. Parameters of moist air

**Humidity ratio** is defined as:

$$x = \frac{m_v}{m_a} \left[ \frac{\text{kg water}}{\text{kg dry air}} \right],$$

where  $m_v$  is the mass of water vapors and  $m_a$  is the mass of dry air.

Using the EOS of perfect gases we get:

$$x = 0,622 \cdot \frac{p_v}{p_a},$$

or:

$$x = 0,622 \cdot \frac{p_v}{p_B - p_v}.$$

For saturated air:

$$x_s = 0,622 \cdot \frac{p_{vs}}{p_B - p_{vs}}.$$

The **relative humidity** of air represents the ratio between the amount of water vapors and the amount of water vapors when for saturated air:

$$\varphi = \frac{m_v}{m_{vs}} = \frac{p_v}{p_{vs}},$$

or:

$$\varphi_{\%} = \frac{p_v}{p_{vs}} \cdot 100 \quad [\%].$$

Using the relative humidity and saturation pressure the humidity ratio becomes:

$$x = 0,622 \cdot \frac{\varphi \cdot p_{vs}}{p_B - \varphi \cdot p_{vs}},$$

where  $p_{vs}$  is the partial pressure of water vapors at saturation, which can be calculated or taken from tables.

The **specific enthalpy** of moist air, for 1 kg of dry air, is:

$$I = t + x \cdot (2501 + 1,93 \cdot t) \left[ \frac{kJ}{kg} \right],$$

where  $t$  is the air temperature.

The **density** of moist air is:

$$\rho = \frac{1 + x}{R_a + x \cdot R_v} \cdot \frac{p_B}{T} \left[ \frac{kg}{m^3} \right].$$

### 6.1.2. Psychrometric charts

The psychrometric charts are graphical representations of the parameters of moist air; each point in a psychrometric chart represents a certain state of the air (in terms of humidity ratio, temperature, enthalpy, relative humidity).

There are two types of psychrometric charts in use today: Mollier and Carrier; the Mollier chart is used mainly in Europe, while the Carrier chart is used in USA.

Fig. 6.1 presents the main curves depicted in a Mollier chart; unlike the usual x-y charts, there is a  $120^\circ$  angle between the axes (humidity ratio,  $X$ , and enthalpy,  $i$ ) of a Mollier chart (fig. 6.1a). As a result the unsaturated area of the chart is enlarged and the constant enthalpy lines (4) are parallel and inclined; the constant temperature lines (2) are divergent, with the  $0^\circ\text{C}$  isotherm being horizontal (fig. 6.1b). In the saturated area of the diagram (beneath the saturation line  $\phi = 100\%$ ), the isotherms (6) may be considered parallel (fig. 6.1c).

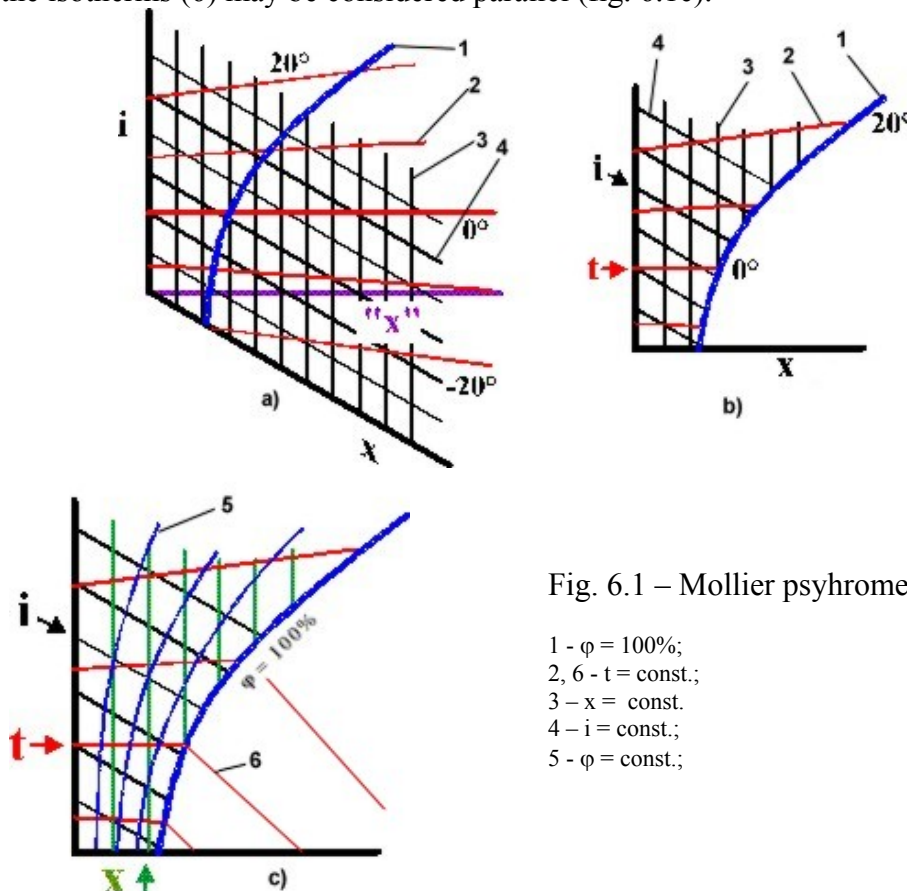


Fig. 6.1 – Mollier psychrometric chart

- 1 -  $\phi = 100\%$ ;
- 2, 6 -  $t = \text{const.}$ ;
- 3 -  $x = \text{const.}$
- 4 -  $i = \text{const.}$ ;
- 5 -  $\phi = \text{const.}$ ;

Using the Mollier chart the dew point temperature and wet bulb temperature may be evaluated.

The **Dew Point** is the temperature at which water vapor starts to condense out of the air (the temperature at which air becomes completely saturated). Above this temperature the moisture will stay in the air.

The **Wet Bulb** temperature  $T_u$  is the temperature of adiabatic saturation. This is the temperature indicated by a moistened thermometer bulb exposed to the air flow.

Wet Bulb temperature can be measured by using a thermometer with the bulb wrapped in wet muslin (fig. 6.2). The adiabatic evaporation of water from the thermometer and the cooling

effect is indicated by a "wet bulb temperature" lower than the "dry bulb temperature" in the air.

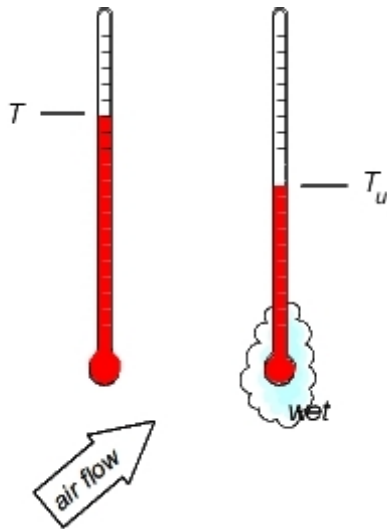


Fig. 6.2 – Dry bulb and wet bulb temperature

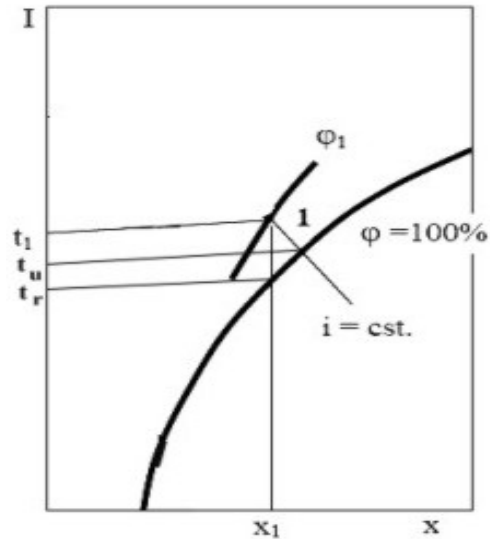


Fig. 6.3 – Evaluation of the dew point temperature and wet bulb temperature

The **Dry Bulb** temperature  $T$  refers basically to the ambient air temperature. It is called "Dry Bulb" because the air temperature is indicated by a thermometer not affected by the moisture of the air. The wet bulb temperature is always lower than the dry bulb temperature but will be identical with 100% relative humidity (the air is at the saturation line).

In the Mollier chart, the dew point temperature  $t_r$  is obtained where the vertical line from the point corresponding to the actual state of the air (1, fig. 6.3) intersects the saturation line  $\phi = 100\%$ . The wet bulb temperature  $t_u$  is obtained at the intersection between the constant enthalpy line ( $i = \text{cst.}$ ) and the saturation line.

The dew point temperature may also be calculated with the formula:

$$t_r = \frac{4030 \cdot (t + 235)}{4030 - (t + 235) \cdot \ln \phi} \quad [^{\circ}\text{C}],$$

where  $t$  is the dry bulb temperature and  $\phi$  is the relative humidity (in fraction).

Regarding the wet bulb temperature, the rate of evaporation from the wet bandage on the bulb and the temperature difference between the dry bulb and wet bulb depend on the humidity of the air. The evaporation is reduced when the air contains more water vapor. This basic principle is used for measuring the relative humidity with the Assmann psychrometer (fig. 6.4).

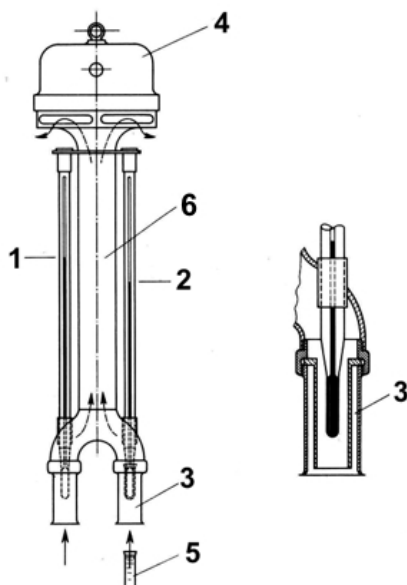


Fig. 6.4 – The Assmann psychrometer

- 1-dry bulb thermometer;
- 2-wet bulb thermometer;
- 3-tube of the wet bulb thermometer;
- 4-fan;
- 5-wet muslin;
- 6-intake pipe.

The fan (4) creates a current of air that flows over both the dry bulb and the wet bulb thermometer. The relative humidity may be calculated using a specific equation (when the parameters of the psychrometer are known) or may be obtained using the Mollier chart (fig. 6.5) or the Carrier chart (fig. 6.6).

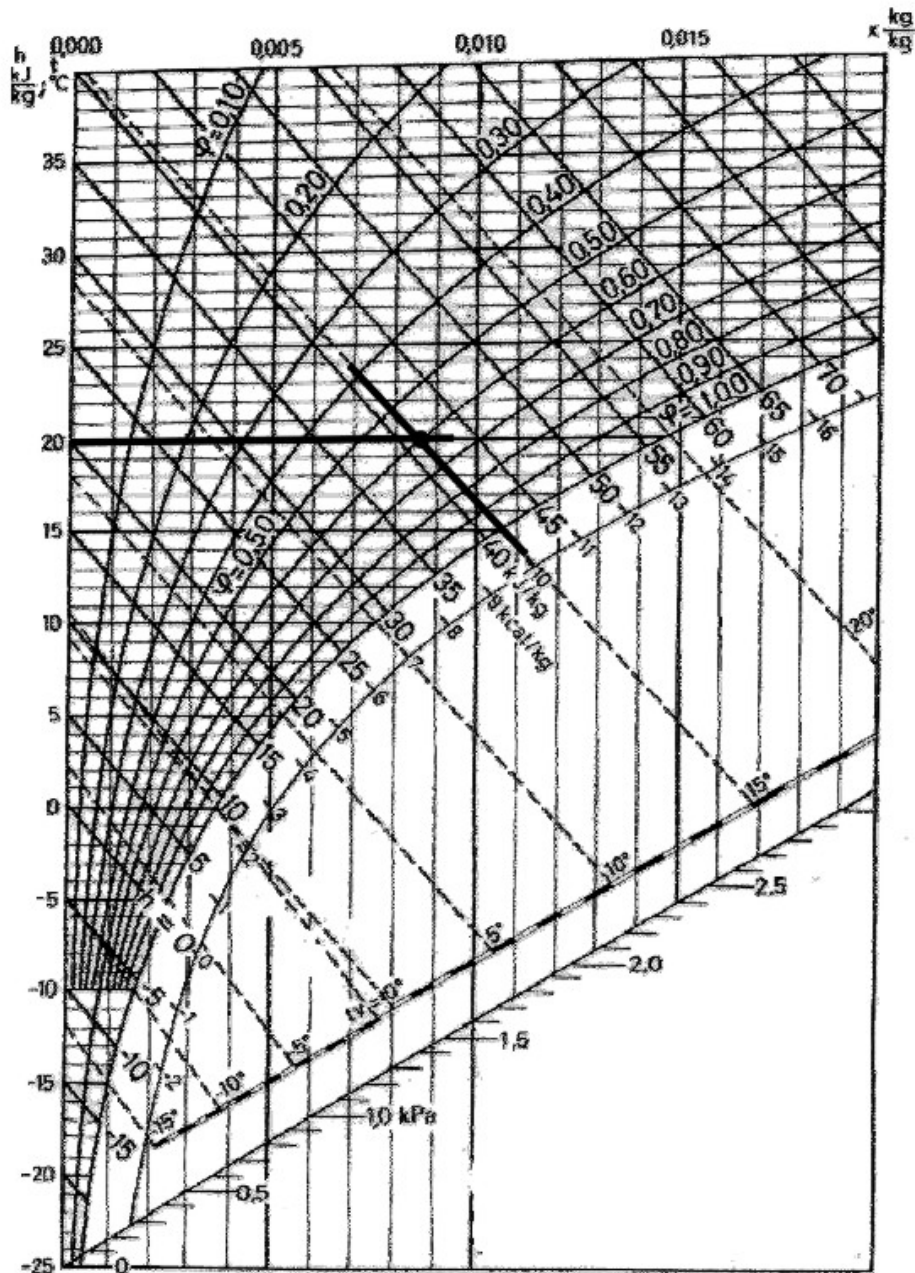


Fig. 6.5 – Evaluation of the relative humidity using the Mollier chart (20°C dry bulb temperature, 15°C wet bulb temperature,  $\phi=0.6$ ).

### 6.1.3. Representations of moist air processes in the Mollier diagram

Fig. 6.7 and 6.8 present some simple psychrometric processes, represented on the Mollier chart.

Sensible heating of moist air is achieved when air passes over a heating coil; the humidity ratio of air remains constant and its temperature increases from  $t_1$  to  $t_2$  as it flows over the heating coil (fig. 6.7).

When the air is cooled while flowing over a cold surface two cases are possible (fig. 6.8):

- if the final air temperature  $t_3$  is higher than the dew point temperature  $t_2$ , the humidity ratio remains constant;
- if the final temperature  $t_3$  is lower than the dew point temperature  $t_2$  the humidity ratio

decreases from  $X_1$  to  $X_3$ . If the air is then heated in order to reach the initial temperature  $t_1$ , its humidity ratio will be lower than the initial one (point 1'); thus the air is dehumidified.

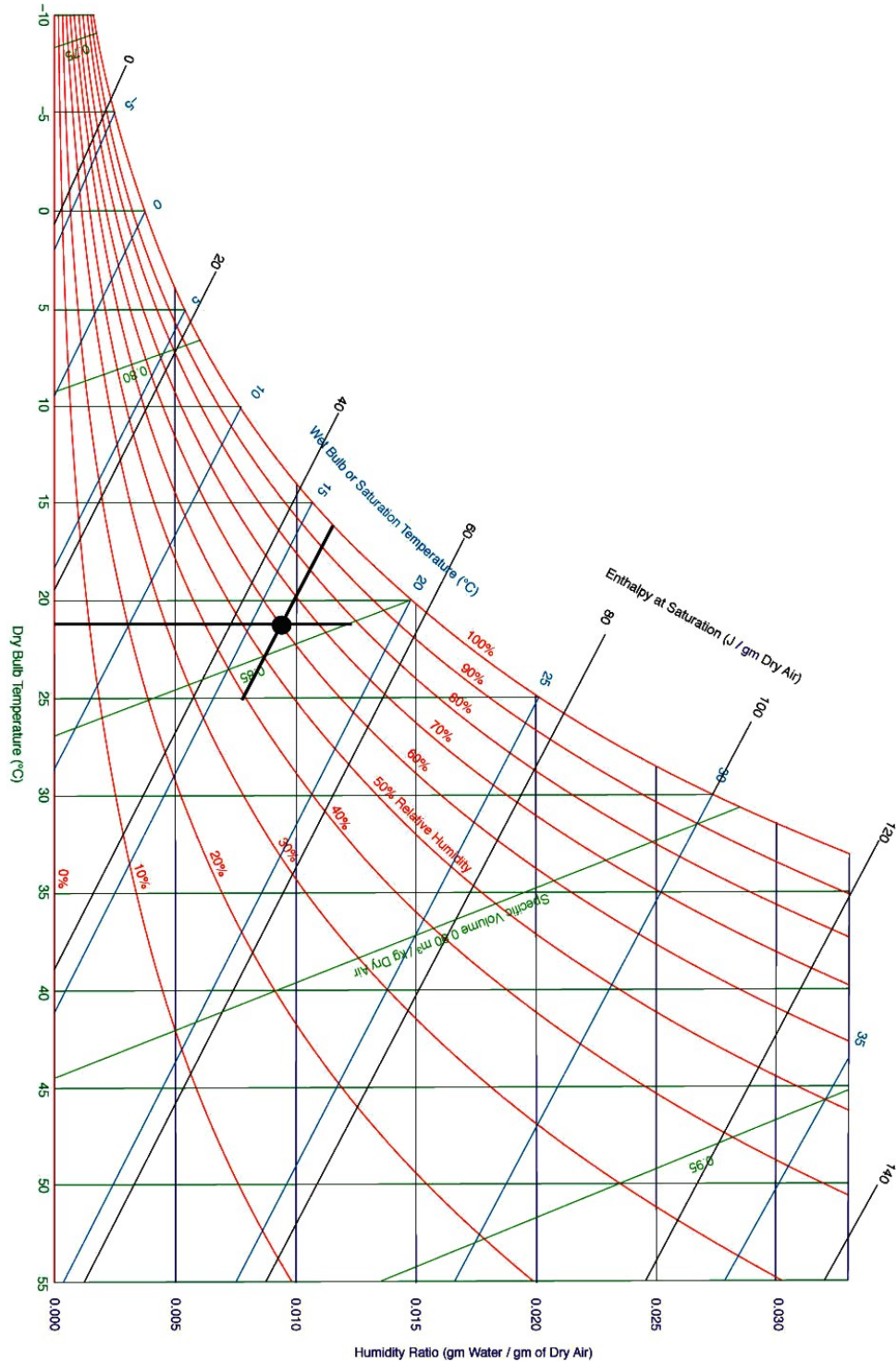


Fig. 6.6 – Evaluation of the relative humidity using the carrier chart (21°C dry bulb temperature, 16°C wet bulb temperature,  $\phi=60\%$ ).

Fig. 6.9 presents the process of mixing two moist air streams, with different parameters. Mixing of air streams with different states is commonly encountered in many processes, including in air conditioning.

The point (M) on the psychrometric chart representing the resulting mixture lies on a straight line connecting the two inlet states; from the mass and energy balance we get the parameters of the air mixture:

$$i_M = \frac{\dot{m}_1 \cdot i_1 + \dot{m}_2 \cdot i_2}{\dot{m}_1 + \dot{m}_2}, \quad x_M = \frac{\dot{m}_1 \cdot x_1 + \dot{m}_2 \cdot x_2}{\dot{m}_1 + \dot{m}_2}$$

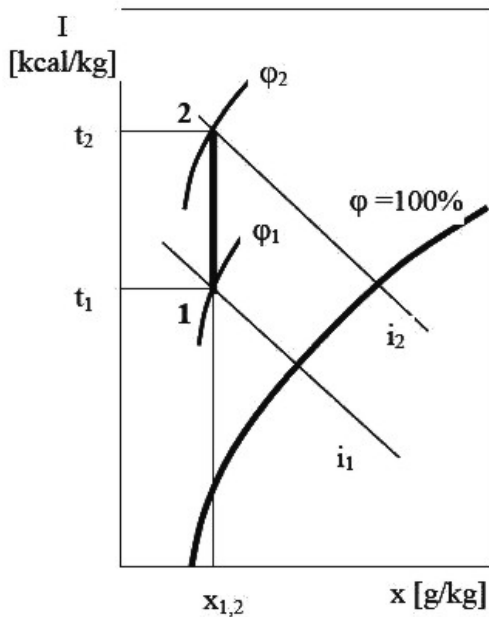


Fig. 6.7 – Heating of moist air

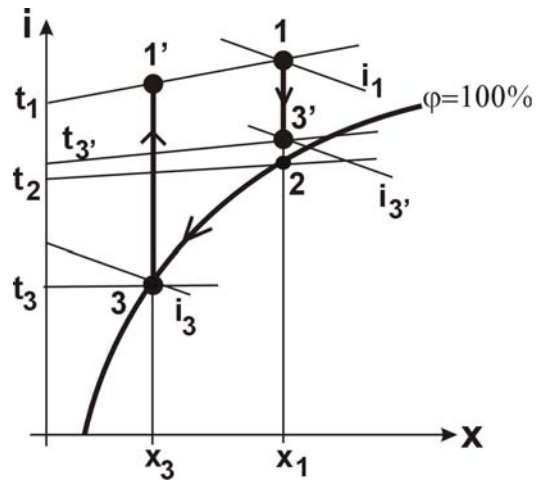


Fig. 6.8 – Cooling of moist air

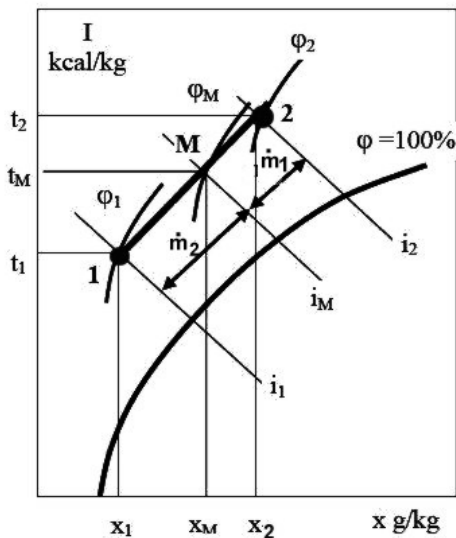


Fig. 6.9 – Mixing of two air streams

When very cold and dry air (point 2, fig. 6.10) mixes with warm air at high relative humidity (point 1), the resulting mixture condition (point M) may lie in the two-phase region (beneath the saturation line  $\phi=100\%$ ); as a result there will be condensation of water vapor. In order to avoid the formation of liquid water the cold air is preheated (process 2-3) and then it is mixed with the warm air.

## 6.2. Air conditioning systems

### 6.2.1. Types of air conditioning systems

**Air conditioning** (A/C) is the process of altering the properties of air (primarily temperature and humidity) to more favorable conditions; the conditioned air is then distributed to an occupied space in order to improve human comfort or to ensure the conditions imposed by the industrial process taking place in the respective space.

There are mainly two types of air conditioning system: centralized and independent. In centralized systems air is processed in the A/C plant and this processed air is then conveyed to

the conditioned space through insulated ducts using blowers and fans. This air extracts (or supplies in case of winter) the required amount of sensible and latent heat from the conditioned space. The return air from the conditioned space is conveyed back to the plant, where it again undergoes the required processing thus completing the cycle. No additional processing of air is required in the conditioned space

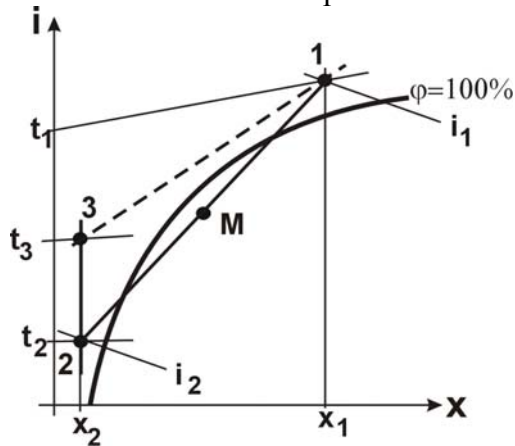


Fig. 6.10 – Formation of liquid water when mixing two air streams

The independent A/C systems are mounted in each conditioned space and are used where stringent control of conditioned space temperature and humidity is not required. These systems can be used for air conditioning individual rooms to large office buildings, classrooms, hotels, shopping centers, nursing homes etc. and are especially suited for existing building.

Fig. 6.11 presents a centralized type air conditioning system, with the air conditioning plant mounted on the roof of the building. The treated air is supplied to each space by the means of the supply ducts (9) and (10) and the return air leaves the conditioned spaces through the return air ducts (11). A part of the return air is recirculated through the duct (3), being mixed with fresh air, while the other part is vented to the atmosphere through the air exhaust duct (12) and the exhaust gate (13). In winter conditions the fresh air is preheated by the coil (2); the mixture of fresh and returned air is then treated (de-humidified or humidified, cooled or heated etc.) and then supplied to the conditioned spaces.

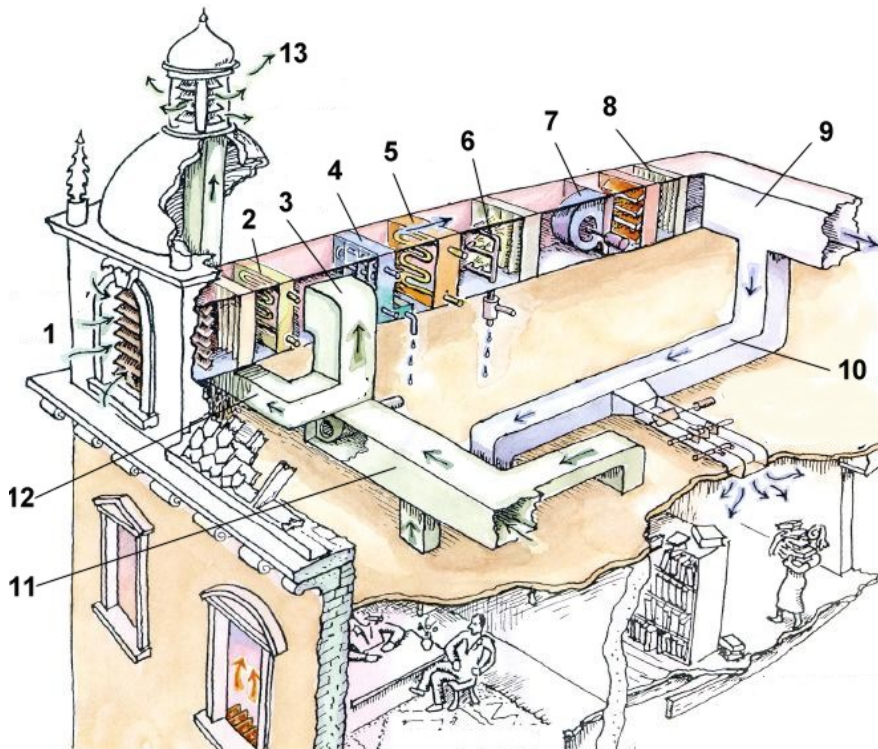


Fig. 6.11 – Centralized air conditioning system

1-fresh air intake; 2-preheating coil; 3-return air duct; 4-dehumidifying coil; 5-heating coil; 6-humidification unit; 7-fan; 8-filter; 9, 10-supply air ducts; 11-air return ducts; 12-air exhaust duct; 13-air exhaust gate.

Fig. 6.12 presents a split type independent air conditioning unit; the interior unit contains a fan and the evaporator (1, fig. 6.13) of the refrigeration system, while the exterior unit contains the condenser (4), the compressor (2) and the throttling device (6) of the refrigeration system. During summer the evaporator (V, fig. 6.14a) of the interior unit cools the air inside the conditioned space; during the cold season the roles of the condenser and evaporator are switched, the coil of the interior unit becoming the condenser of the refrigeration system (K, fig. 6.14b). For this purpose the unit is provided with a distributor (D, fig. 6.14) that changes the direction of the refrigerant flow through the system.

Split air conditioners include their own integral controls and are therefore relatively simply installed wherever their cooling function is required.

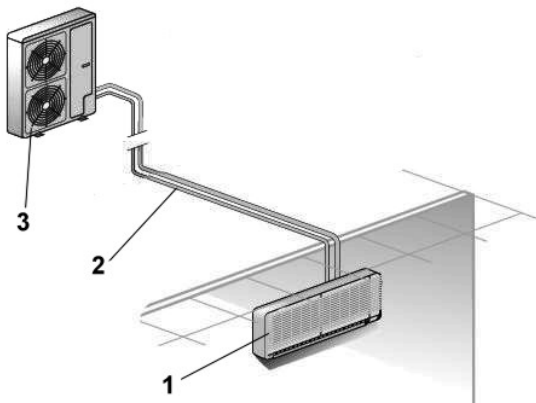


Fig. 6.12 – Split type independent air conditioning unit  
1-interior unit; 2-refrigerant pipes; 3-exterior unit.

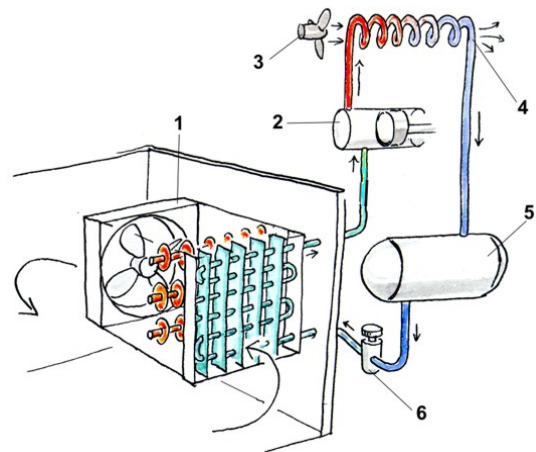


Fig. 6.13 – Schematics of the split type conditioning unit  
1-evaporator; 2-compressor; 3-condensing unit fan; 4-condenser; 5-refrigerant tank; 6-throttling device.

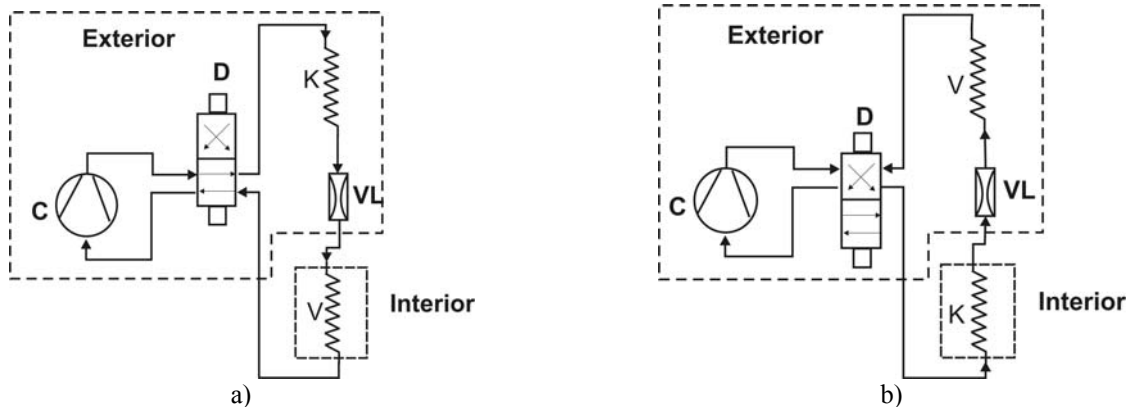


Fig. 6.14 – Schematics of a split type air conditioning unit, used for both air cooling and heating  
C-compressor; D-distributor; VL-capillary tube (throttling device); V-evaporator; K-condenser.

Multi-split is a general name applied to packaged systems where two or more remote coolers are run from one condensing unit. Liquid from the condenser passes directly through an expansion device, and the resulting mixture of cold liquid and flash gas is distributed to each of the fan-coil units on the circuit. On-off control of the cold liquid to each room is effected by a solenoid valve within each indoor unit, controlled by a room thermostat. Return refrigerant gas passes through a suction trap and/or a suction/liquid heat exchanger in order to ensure that unevaporated refrigerant is vaporized before entering the compressor. Fig. 6.15 presents the schematics of a multi-split type air conditioning unit.

### 6.2.2. Thermal comfort

As mentioned before air conditioning is required either for providing suitable comfort



conditions for the occupants (comfort air conditioning), or for providing suitable conditions for storage of perishable products (e.g. in cold storages) or conditions for a process to take place or for products to be manufactured (industrial air conditioning). The required inside conditions for cold storage and industrial air conditioning applications vary widely depending on the specific requirement. However, the required inside conditions for comfort air conditioning systems remain practically same irrespective of the size, type, location, use of the air conditioning building etc., as this is related to the thermal comfort of the human beings.

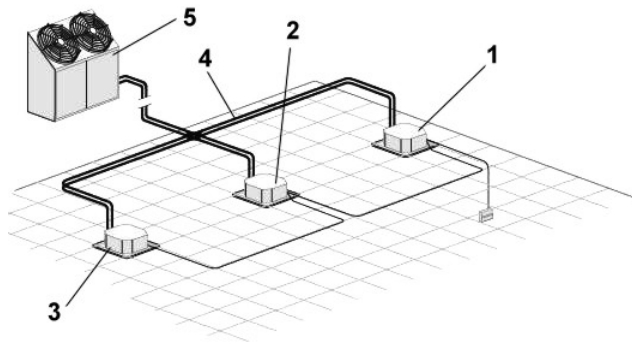


Fig. 6.15 – Multi-split type independent air conditioning unit

1, 2, 3-interior (cooling) units; 4-refrigerant pipes; 5- exterior unit.

Thermal comfort is defined as “that condition of mind which expresses satisfaction with the thermal environment”. This condition is also some times called as “neutral condition”, though in a strict sense, they are not necessarily same. Thermal comfort is affected by several factors:

1. Physiological factors such as age, activity, sex and health. These factors influence the metabolic rate. It is observed that of these factors, the most important is activity. Other factors are found to have negligible effect on thermal comfort.

2. Insulating factor due to clothing. The type of clothing has strong influence on the rate of heat transfer from the human body. The unit for measuring the resistance offered by clothes is called as “**clo**”. 1 clo is equal to a resistance of about  $0.155 \text{ m}^2 \cdot \text{K}/\text{W}$ . Typical clo values for different types of clothing have been estimated and are available in the form of tables. For example, a typical business suit has a clo value of 1.0, while a pair of shorts has a clo value of about 0.05.

3. Environmental factors. Important factors are the dry bulb temperature, relative humidity, air motion and surrounding surface temperature. Of these the dry bulb temperature affects heat transfer by convection and evaporation, the relative humidity affects heat loss by evaporation, air velocity influences both convective and evaporative heat transfer and the surrounding surface temperature affects the radiative heat transfer.

Apart from the above, other factors such as drafts, asymmetrical cooling or heating, cold or hot floors etc. also affect the thermal comfort. The objective of a comfort air conditioning system is to control the environmental factors so that comfort conditions prevail in the occupied space. It has no control on the physiological and insulating factors. However, wearing suitable clothing may help in reducing the cost of the air conditioning system.

A living human body may be likened to a heat engine in which the chemical energy contained in the food it consumes is continuously converted into work and heat. The process of conversion of chemical energy contained in food into heat and work is called as “metabolism”. The rate at which the chemical energy is converted into heat and work is called as “metabolic rate”. Knowledge of metabolic rate of the occupants is required as this forms a part of the cooling load of the air conditioned building. Similar to a heat engine, one can define thermal efficiency of a human being as the ratio of useful work output to the energy input. The thermal efficiency of a human being can vary from 0% to as high as 15-20% for a short duration. By the manner in which the work is defined, for most of the light activities the useful work output of human beings is zero, indicating a thermal efficiency of 0%. Irrespective of the work output, a human body continuously generates heat at a rate varying from about 100 W (e.g. for a sedentary person) to as high as 2000 W (e.g. a person doing strenuous exercise). Continuous heat

generation is essential, as the temperature of the human body has to be maintained within a narrow range of temperature, irrespective of the external surroundings.

The temperature of human body depends upon the energy balance between itself and the surrounding thermal environment (fig. 6.16). Taking the human body as the control volume, one can write the thermal energy (heat) balance equation for the human body as:

$$Q_{gen} = Q_{sk} + Q_{res} + Q_{st},$$

where  $Q_{gen}$  is the rate of heat generation inside the body,  $Q_{sk}$  is the rate of heat transferred from the skin,  $Q_{res}$  is the heat transfer rate due to respiration and  $Q_{st}$  is the rate at which heat is stored inside the body. For comfort the rate at which heat is stored inside the body should be zero and the thermoregulatory system of humans is responsible for maintaining a constant body temperature.

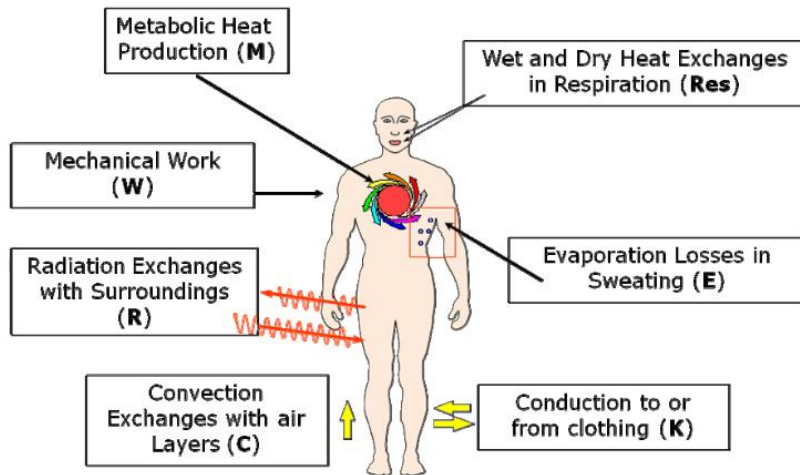


Fig. 6.16 - Human body thermal balance

The heat generation rate  $Q_{gen}$  is given by:

$$Q_{gen} = M - W,$$

where  $M$  is the metabolic rate and  $W$  is the work output. The metabolic rate and work output depend on the activity and are usually measured in “met”; a met is defined as the metabolic rate per unit area of a sedentary person and is found to be equal to about  $58.2 \text{ W/m}^2$ .

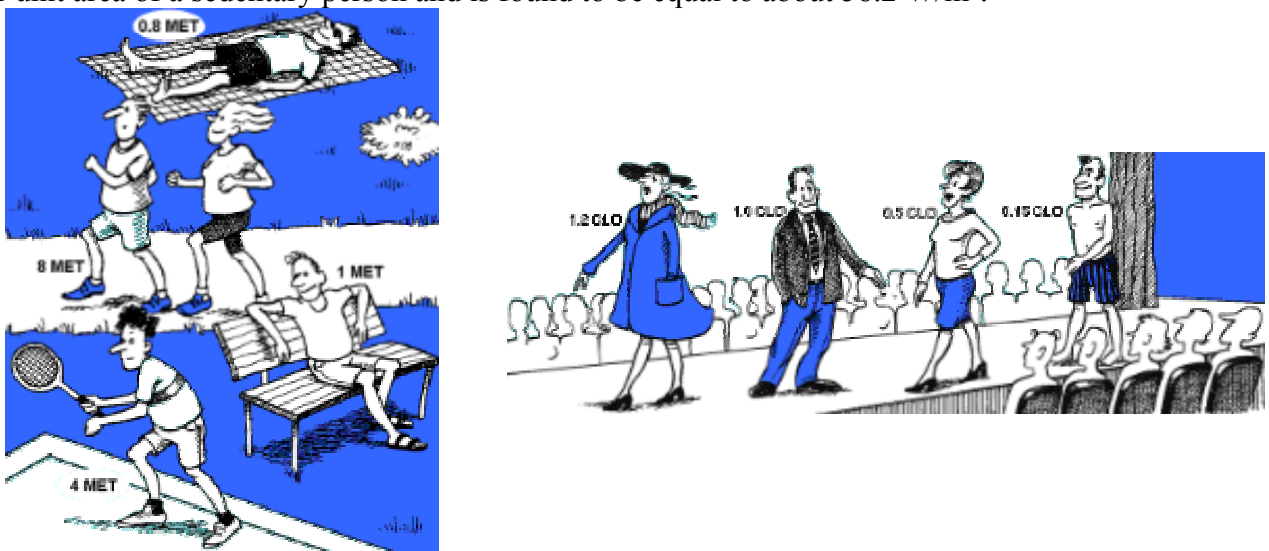


Fig. 6.17 – Metabolic rate and thermal resistance of clothes<sup>4</sup>

Thermal comfort is largely a state of mind, separate from equations for heat and mass transfer and energy balances. However, the perception of comfort is expected to be influenced by the variables that affect the heat and mass transfer in our energy balance model. The most common approach to characterizing thermal comfort for the purposes of prediction and building

<sup>4</sup> <http://www.blowtex-educair.it/DOWNLOADS/Thermal%20Comfort.htm>

design has been to correlate the results of psychological experiments to thermal analysis variables. A large number of the experiments have been performed at universities with college students as the human subjects: human subjects with various clothing levels and performing different activities were placed in environments with different air temperatures and surface temperatures, different humidity conditions and different airflow velocities and patterns, the subjects being then asked to express their level of comfort. The level of comfort is often characterized using the ASHRAE thermal sensation scale with values from +3 (hot) to -3 (cold) with 0 being the neutral condition. Using the response of a large number of subjects and combining this information with the thermal energy balance equations Fanger developed a set of correlations giving the PMV (Predicted Mean Vote) index as a function of six variables: air temperature, mean radiant temperature, air velocity, air humidity, clothing resistance, and activity level. PMV predicts the mean response of a large number of occupants based on the thermal sensation scale. Fanger related the PMV to Percent of People Dissatisfied (PPD) by the following equation:

$$PPD = 100 - 95 \cdot e^{-(0,03353 \cdot PMV^4 + 0,2179 \cdot PMV^2)}$$

where dissatisfied refers to anybody not voting for -1, 0 or +1. It can be seen from the above equation that even when the PMV is zero (i.e., no thermal load on body) 5 % of the people will be dissatisfied (fig. 6.18). When PMV is within  $\pm 0.5$ , then PPD is less than 10 %.

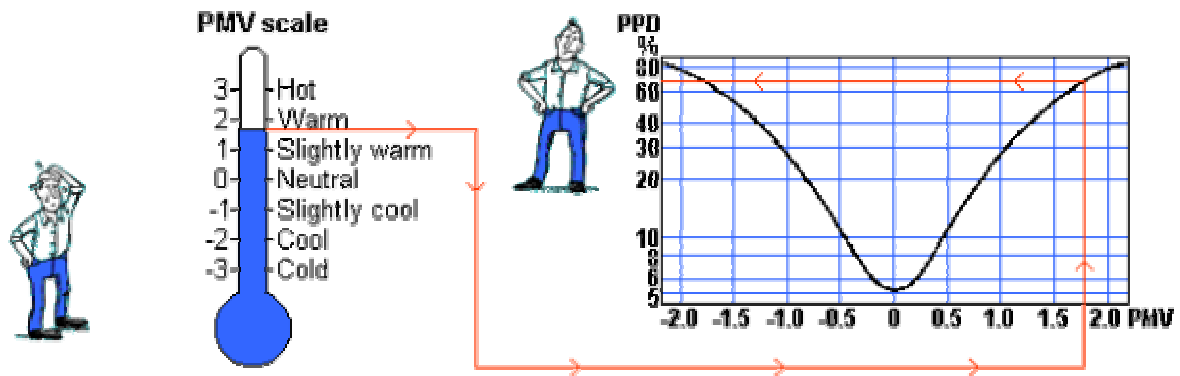


Fig. 6.18 – Correlation between PMV and PPD

Even though a person has a sensation of thermal neutrality, parts of the body may be exposed to conditions that result in thermal discomfort. This local thermal discomfort can not be removed by raising or lowering the temperature of the enclosure. It is necessary to remove the cause of the localised over-heating or cooling. Generally, local thermal discomfort can be grouped under one of the following four headings (fig. 6.19):

1. Local convective cooling of the body caused by draught
2. Cooling or heating of parts of the body by radiation. This is known as a radiation asymmetry problem.
3. Cold feet and a warm head at the same time, caused by large vertical air temperature differences.
4. Hot or cold feet, caused by uncomfortable floor temperature

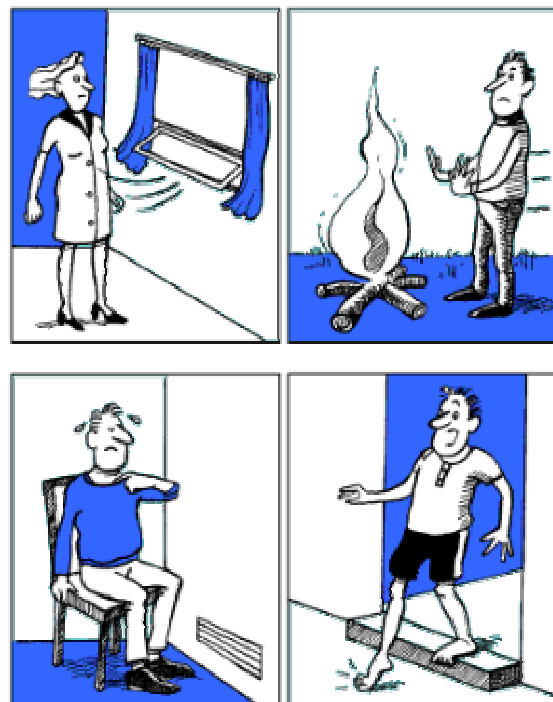


Fig. 6.19 – Local thermal discomfort

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# EVALUATION OF THE COOLING LOAD AND DESIGN OF THE COOLING EQUIPMENT

## THEME:

The aim of the project is to evaluate the operating parameters of the cooling equipment needed in order to store refrigerated pork meat in the unit DR1 (fig.16.1), using the air blast cooling method; the meat is indirectly cooled<sup>1</sup>. The height of the unit is 3.5 m.

A vapor compression cooling equipment is used in this application; the condenser of the equipment is placed outside the unit and is air cooled. The temperature of the outside air is 30°C.

### 16.1. Evaluation of the cooling load

The cooling load for 24 hours is given by the relation:

$$Q = Q_1 + Q_2 + Q_3 + Q_4, \text{ [kJ/24 h]}$$

where:

- $Q_1$  – heat transferred from the outside, through the walls of the unit;
- $Q_2$  – cooling load due to the product being chilled;
- $Q_3$  – cooling load in order to compensate the heat due to outside air ingress;
- $Q_4$  – supplementary cooling load.

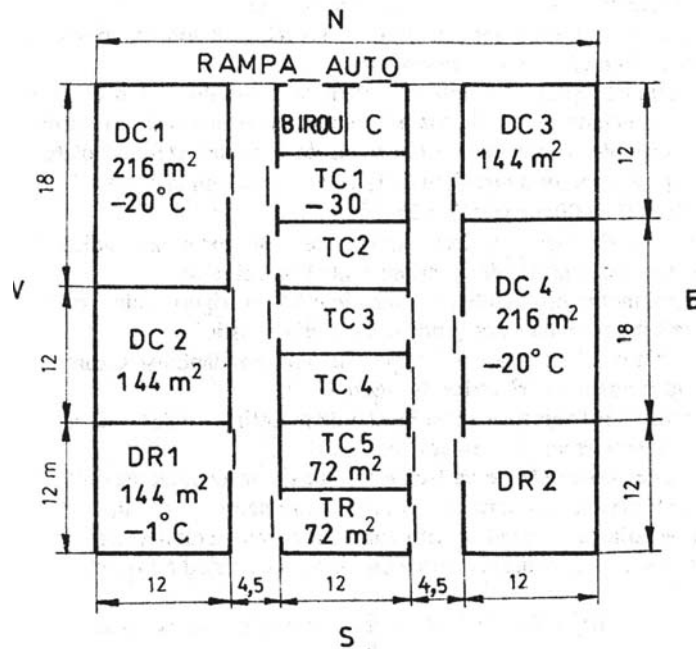


Fig. 16.1 – Schematic of the cold storage units:  
*DC* – frozen food storage unit;  
*DR* – refrigerated food storage unit;  
*TC* – freezing tunnel;  
*TR* – refrigeration tunnel.

<sup>1</sup> the evaporator of the cooling system cools the air and the air cools the products

**STEP 1.** The heat is transferred from the outside, through the walls of the unit, due to the temperature difference and solar radiation:

$$Q_1 = 86,4 \cdot \sum k_i \cdot S_i \cdot (\Delta t_i + \Delta t_r) \quad [kJ / 24h],$$

where:

- a)  $k_i$  – heat transfer coefficient for the structural element  $i$  (walls, floor, ceiling), considering a multilayer structure, as follows (fig. 16.2):

**Outer walls:**

- exterior plaster with  $\lambda_1 = 1 \text{ W/m.K}$  and  $\delta_1 = 2 \dots 3 \text{ cm}$ ;
- brick structure with  $\lambda_2 = 0,23 \dots 0,52 \text{ W/m.K}$  and  $\delta_2 = 25 \text{ cm}$ ;
- supporting plaster with  $\lambda_3 = 1,16 \text{ W/m.K}$  and  $\delta_3 = 2 \dots 3 \text{ cm}$ ;
- vapor barrier with  $\lambda_4 = 0,384 \text{ W/m.K}$  and  $\delta_4 = 3 \dots 5 \text{ mm}$ ;
- polystyrene insulation with  $\lambda_5 = 0,04 \text{ W/m.K}$  and  $\delta_5 = 10 \text{ cm}$ ;
- finishing plaster with  $\lambda_6 = 1,16 \text{ W/m.K}$  and  $\delta_6 = 2 \dots 3 \text{ cm}$ .

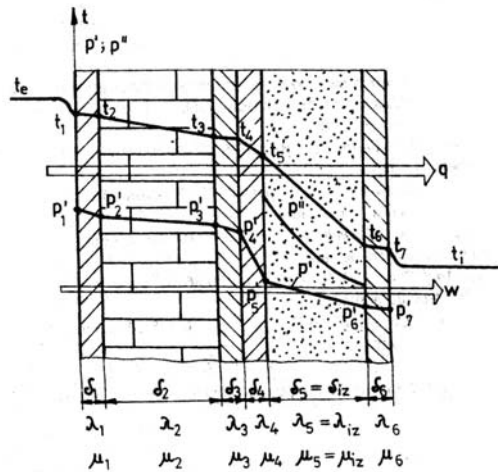


Fig. 16.2 – Multilayer structure of the walls

**Inner walls:** same as previous, but without layer 4 (vapor barrier).

**Ceiling** (from the inside out):

- finishing plaster with  $\lambda_6 = 1,16 \text{ W/m.K}$  and  $\delta_6 = 2 \dots 3 \text{ cm}$ ;
- polystyrene insulation with  $\lambda_5 = 0,04 \text{ W/m.K}$  and  $\delta_5 = 10 \text{ cm}$ ;
- concrete plate with  $\lambda = 2,03 \text{ W/m.K}$  and  $\delta = 25 \text{ cm}$ ;
- layer of bitumen with  $\lambda = 0,17 \text{ W/m.K}$  and  $\delta = 5 \text{ mm}$ ;
- gravel layer with  $\lambda = 0,7 \text{ W/m.K}$  and  $\delta = 4 \text{ cm}$ .

**Floor** (from the inside out):

- PVC cover with  $\lambda = 0,33 \text{ W/m.K}$  and  $\delta = 3 \text{ mm}$ ;
- water insulation with  $\lambda = 0,17 \text{ W/m.K}$  and  $\delta = 2 \dots 3 \text{ cm}$ ;
- concrete plate with  $\lambda = 2,03 \text{ W/m.K}$  and  $\delta = 25 \text{ cm}$ ;
- extruded polystyrene insulation with  $\lambda = 0,03 \text{ W/m.K}$  and  $\delta = 2,5 \text{ cm}$ ;
- gravel layer with  $\lambda = 0,7 \text{ W/m.K}$  and  $\delta = 4 \text{ cm}$ .

The heat transfer coefficient is calculated for each structural element, with the relation:

$$k_i = \frac{1}{\frac{1}{\alpha_{\text{ext}}} + \sum_{j=1}^n \frac{\delta_j}{\lambda_j} + \frac{1}{\alpha_{\text{int}}}} \text{ [W/m}^2\text{K]},$$

where  $\alpha$  are coefficients for the convective heat transfer (for the outer surface of the outer walls  $\alpha_{\text{ext}} = 29 \text{ W/m}^2\cdot\text{K}$ ; for the outer surface of the inner walls  $\alpha_{\text{ext}} = 12 \text{ W/m}^2\cdot\text{K}$ ; for the inner surfaces of the walls  $\alpha_{\text{int}} = 8 \text{ W/m}^2\cdot\text{K}$ ; for the floor  $1/\alpha_{\text{ext}} = 0$ ).

b)  $S_i$  – surface of the respective element (wall, ceiling, floor) [ $\text{m}^2$ ];

c)  $\Delta t_i$  – temperature difference for the respective element, as follows:

- outer walls and platform ceilings  $\Delta t_i = \Delta t_c = 30 - t_i$  (an exterior temperature of  $30 \text{ }^\circ\text{C}$  is considered and  $t_i$  is the temperature inside the refrigerated space);

- inner walls, ceilings and floors separating a refrigerated space from a non-refrigerated space which communicates with the exterior:  $\Delta t_i = (0,7 \dots 0,8) \cdot \Delta t_c$ ;

- inner walls, ceilings and floors separating a refrigerated space from a non-refrigerated space which does not directly communicate with the exterior:  $\Delta t_i = 0,6 \cdot \Delta t_c$ ;

- inner walls, ceilings and floors separating a refrigerated space from an other refrigerated space  $\Delta t_i = 0,4 \cdot \Delta t_c$ ;

- floor:  $\Delta t_i = 15 - t_i$ .

d)  $\Delta t_r$  – temperature added in order to take into account the effect of solar radiation over the walls and ceiling:

-  $\Delta t_r = 0$  for outer walls facing N;

-  $\Delta t_r = 5 \dots 10 \text{ }^\circ\text{C}$  for outer walls facing E, V, SE, SV;

-  $\Delta t_r = 15 \text{ }^\circ\text{C}$  for outer walls facing S;

-  $\Delta t_r = 15 \dots 18 \text{ }^\circ\text{C}$  for ceilings.

The resulting data will be introduced in the following table.

Elemen	k [W/m <sup>2</sup> ·K]	S [m <sup>2</sup> ]	$\Delta t_i$ [°C]	$\Delta t_r$ [°C]	k·S·( $\Delta t_i + \Delta t_r$ )
wall 1					
wall 2					
wall 3					
wall 4					
ceiling					
floor					
<b>TOTAL [W]</b>					

**STEP 2.** The useful surface of the storage unit is:

$$S_{ui} = \frac{S}{\beta} \text{ [m}^2\text{]},$$

where S is the floor surface and  $\beta$  has the following values:

- 1,4 – refrigeration units with  $S < 80 \text{ m}^2$ ;
- 1,4 – freezing units;
- 1,3 – storage units for refrigerated products, with  $80 < S < 200 \text{ m}^2$ ;
- 1,3 – storage units for frozen products, with  $S < 300 \text{ m}^2$ .

**STEP 3.** *Table 16.1* presents the specific weight load of the storage unit [kg/m<sup>2</sup>] for different types of products; the total amount of food to be stored is calculated with the formula:

$$m = N \cdot S_{ui} \quad \text{[kg]}.$$

**STEP 4.** The cooling load results from the relation<sup>2</sup>:

$$Q_2 = m \cdot [(i_i - i_f) + 25 \cdot \Delta m] + Q_c \quad \text{[kJ / 24h]} \text{ - for refrigeration,}$$

$$Q_2 = m \cdot [(i_i - i_f) + 28,35 \cdot \Delta m] \quad \text{[kJ / 24h]} \text{ - for freezing,}$$

where:

- m – weight of the products [kg] – see above;
- $i_i, i_f$  – initial and final enthalpy of the products<sup>3</sup> (*table 16.3*), depending upon the initial and final temperature of the products<sup>4</sup> [kJ/kg];
- $\Delta m$  – weight loss in 24 hours [%] (*table 16.4*);
- $Q_c$  – heat due to the biochemical processes, for vegetables, fruits (*table 16.5*).

**STEP 5.** The cooling load needed in order to compensate the heat due to outside air ingress:

$$Q_3 = V \cdot a \cdot \rho_i \cdot (i_{\text{ext}} - i_{\text{int}}) \quad \text{[kJ / 24h]},$$

where:

- V – volume of the storage unit [m<sup>3</sup>];
- a = 2...4 air changes per day for meat, milk and milk products; 24...48 air changes per day for vegetables and fruits;
- $i_{\text{ext}}, i_{\text{int}}$  – enthalpy of exterior and interior air, respectively [kJ/kg]. The outside air has 60% humidity; the humidity of inside air is given in *Table 16.6* for refrigerated products and is 95% for frozen products. The enthalpies of air are calculated using the “Moist Air Properties” option from the “Cool Tools: Auxiliary” tab of CoolPack (fig. 16.3).
- $\rho_i$  – density of inside air, considered as a perfect gas, using the Clapeyron - Mendeleev equation:

<sup>2</sup>  $Q_2 = 0$  for refrigerated or frozen products storage units, because the products were already cooled.

<sup>3</sup> at  $t_i < -20^\circ\text{C}$ ,  $i_f = 0$ .

<sup>4</sup>  $t_f = 0...4^\circ\text{C}$  when the products are refrigerated,  $t_f = -18...-25^\circ\text{C}$  when the products are frozen.



$$\rho_i = \frac{1,01325 \cdot 10^5}{287 \cdot (273 + t_i)} \left[ \frac{\text{kg}}{\text{m}^3} \right].$$

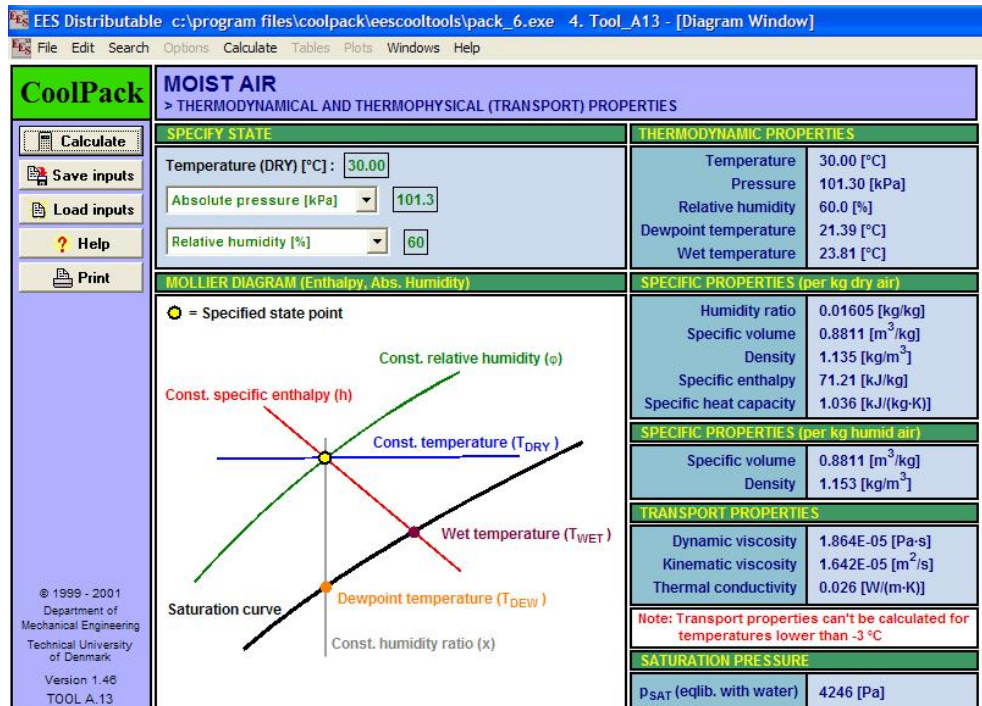


Fig. 16.3 - Calculation of air enthalpy

**STEP 6.** The supplementary cooling load takes into account the heat produced by the electric motors of the fans, the heat produced by the lightning equipment, the heat produced by the personal entering the unit:

$$Q_4 = \xi \cdot Q_1 [kJ / 24h] \quad ,$$

where  $\xi$  depends on the surface of unit floor:

- $S > 300 \text{ m}^2$ :  $\xi = 0,1$ ;
- $150 < S < 300 \text{ m}^2$ :  $\xi = 0,2$ ;
- $80 < S < 150 \text{ m}^2$ :  $\xi = 0,3$ ;
- $S < 80 \text{ m}^2$ :  $\xi = 0,4$ .

**STEP 7.** The total cooling load:

$$Q = \sum_{i=1}^4 Q_i [kJ / 24h].$$

**STEP 8.** Necessary power of the cooling system:

$$\Phi = \frac{Q}{\tau_c \cdot 3600} \cdot 1,05 [kW],$$

where the operating time of the cooling system is  $\tau_c = 20 \dots 21$  hours/day for refrigeration and  $21 \dots 22$  hours/day for freezing; 5% heat loss through the connecting pipes is considered.

Table 16.1

*Încărcări specifice pentru produsele depozitate în camerele frigorifice normale*

Produsul	Încărcare kg / m <sup>2</sup> util	Observații
<b>CARNE</b>		
Vită refrigerată	180 – 250	suspendații
Porc refrigerat	170 – 220	suspendată
Oaie refrigerată	140 – 160	suspendată
Vită congelată	700 – 900	stivuire manuală
Porc congelat	900 – 1000	stivuire manuală
Oaie congelată	700 – 800	stivuire manuală
Slănină sărată	800 – 900	stive
Mezeluri	150 – 200	atârnat
Vânat	100 – 150	atârnat
Conserve de carne	600 – 700	lăzi
<b>PĂSĂRI</b>		
refrigerate	180 – 200	cutii – lăzi
pentru congelare	200 – 250	cutii – lăzi
congelate	550 – 600	cutii – lăzi
ouă refrigerate	500 – 600	cutii – lăzi
ouă congelate	600 – 800	bidoane
<b>PEȘTE</b>		
prospăt (cu gheață)	400 – 500	lăzi
sărat	750 – 900	lăzi – butoaie
afumat	500 – 600	lădițe
congelat	700 – 900	stive – lăzi
conserve	600 – 700	lăzi
icre	450 – 700	butoaie – lădițe
<b>LACTATE</b>		
unt congelat	1000 – 1200	butoaie – lăzi
brânzeturi tari	600 – 750	pe stelaje
brânzeturi moi	500 – 600	cutii – lădițe
telemea	750 – 800	butoaie
lapte	300 – 400	navete – sticle
<b>FRUCTE ȘI LEGUME</b>		
fructe proaspete	450 – 500	lăzi – coșuri
legume proaspete	400 – 700	lăzi – coșuri
fructe și legume uscate	400 – 500	lăzi
fructe și legume congelate	500 – 800	lăzi – cutii
<b>BĂUTURI</b>		
bere	300 – 500	sticle – butoaie
vin	300 – 500	sticle – butoaie
sucuri	300 – 500	sticle – butoaie

Table 16.2

*Parametrii caracteristici refrigerării cărnii*

Specia	Caracteristici tehnologice	Refrigerare cu zvântare prealabilă	Refrigerare directă						
			lentă	rapidă într-o fază		rapidă în două faze (prin convecție)			
				radiație (cameră)	convecție (tunel)	în aceeași cameră		în camere diferite	
						faza I	faza II	faza I	faza II
Carne de vită	Temperatura aerului [°C]	0...4	0...1	-	0...1	-4...-5	0	-4...-5	0
	Umiditatea relativă [%]	90...95							
	Viteza aerului [m/s]	-	0,25	-	1...3	2...3	0,3...0,5	2...3	0,3...0,5
	Durata [h]	36	36	-	20...22	4	16...18	4	16...18
	Pierderi în greutate [%]	-	1,25..2,31	-	1,06..1,96	-	-	-	-
	Puterea frigorifică instalată [kJ/h·t]	-	6279	-	12558	33488	2093	20930	2093
Carne de porc fără slănină	Temperatura aerului [°C]	0...4	0...1	2	0...1	-8...-10	0	-8...-10	0
	Umiditatea relativă [%]	90...95							
	Viteza aerului [m/s]	-	0,25	convecție naturală	1...3	2...3	0,3...0,5	2...3	0,3...0,5
	Durata [h]	24	24	16	12...14	1,5..2,5	10...12	1,5...2,5	10...12
	Pierderi în greutate [%]	2,5...3,5	2,4	1,25..1,3	1,4...2,1	0,7..1,25	0,7..1,25	1,25	1,25
	Puterea frigorifică instalată [kJ/h·t]	-	6279	-	12558... 13794	41860	2903	25116	2093

Table 16.3

*Entalpia specifică a unor produse alimentare în funcție de temperatură*

Produsul	Entalpia h [kJ / kg]														
	-18	-15	-10	-5	-1	0	1	3	5	10	15	20	25	30	35
Carne vită, păsări	4,6	12,97	30,13	57,34	186,2	232,3	235,6	241	248,2	264,5	280,4	296,7	312,6	329,0	354,5
Carne oie	4,6	12,55	29,70	55,60	179,5	223,9	227,2	233,5	239,8	255,3	271,2	287,7	302,6	318,5	344,0
Carne porc	4,6	12,10	28,80	54,40	169,9	211,8	214,7	221,0	226,8	241,9	257,0	272,5	287,9	313,9	339,9
Subproduse porc	5,0	13,80	33,00	62,70	204,2	261,2	306,4	271,6	278,7	295,9	313,5	331,1	348,6	365,8	383,0
Carne desosată	5,0	13,30	31,30	59,80	194,2	242,7	246,1	252,8	259,5	275,8	292,6	309,3	325,6	342,4	367,9
Pește slab	5,0	14,20	33,40	64,00	212,2	265,8	264,5	276,6	283,3	300,9	318,5	336,1	353,7	-	-
Pește gras	5,0	14,20	32,60	61,50	199,6	249,0	252,8	259,5	266,2	283,3	300,5	317,2	334,4	-	-
Filou pește	5,4	14,60	34,70	66,90	224,7	281,7	285,4	292,6	300,1	318,1	336,5	354,9	372,9	-	-
Ouă în coajă	4,1	10,40	22,60	41,40	233,9	237,3	240,2	246,3	252,8	268,7	284,6	300,1	316,0	-	-
Ouă melanj	4,6	11,30	20,00	44,70	141,9	264,1	267,4	274,1	218,2	298,8	315,6	332,7	349,9	-	-
Lapte	5,4	14,20	32,60	62,70	184,1	318,9	322,7	330,6	338,6	358,3	378,4	398,5	418,1	438,2	457,9
Unt	4,1	10,80	22,60	36,80	83,70	92,90	95,40	102,1	108,4	126,4	146,9	171,6	-	-	-
Unt topit	2,5	6,20	13,80	22,60	34,70	37,20	39,70	45,20	51,4	69,0	89,90	114,2	-	-	-
Struguri, caise, cireșe	7,5	20,50	49,80	115,9	232,7	236,0	239,8	246,9	254,0	271,6	289,1	307,3	325,2	-	-
Alte fructe	6,6	17,10	39,30	82,80	267,9	271,6	275,4	282,9	290,5	309,3	328,1	347,0	365,8	-	-
Smântână, iaurt, chefir	-	-	-	-	-	0	3,80	9,45	19,37	38,36	55,20	73,70	95,80	110,6	-
nghețată	7,1	19,70	46,90	105,3	224,4	227,4	230,8	238,1	245,5	264,0	277,8	294,3	311,0	328,0	344,6

Obs:  $h = 0$  la  $t = -20^{\circ}\text{C}$ , excepție pentru smântână, iaurt, chefir la care  $h = 0$  pentru  $t = 0^{\circ}\text{C}$ .

Table 16.4

*Pierderi în greutate în procesul de depozitare a cărnii*

Produsul	Pierderi în greutate (%) după:		
	24 h	48 h	72 h
Carne de bovină îngrășată	0,35...0,40	0,55...0,62	0,65...0,72
Carne de bovină neîngrășată	0,45...0,50	0,65...0,62	0,75...0,80
Carne de vițel	0,50	0,70	0,80
Carne de porc neșupuit	0,20...0,30	0,40...0,50	0,50...0,60
Carne de porc șupuit, cu slănină	0,30	0,50	0,60
Carne de porc fără slănină	0,40	0,50...0,60	0,70...0,80
Carne de ovine	0,40...0,50	0,60...0,70	0,70...0,80
Carne de miel	0,60	0,80	1,00

*Pierderi în greutate în procesul de transport al cărnii*

Produsul	Pierderi în timpul transportului (%) după:				
	6 h	12 h	24 h	48 h	72 h
Carne de bovine	0,4	0,4	0,5	0,7	0,8
Carne de porc	0,3	0,3	0,4	0,6	0,7
Carne de ovine	0,4	0,5	0,5	0,7	0,8
Carne de miel	0,8	1,0	1,2	1,4	1,5

*Pierderi în greutate la congelarea cărnii cu refrigerare prealabilă*

Specia	Temperatura aerului [°C]		Durata de congelare [h]	Pierderi în greutate [%]		
	medie	finală		la refrigerare		totală
				la refrigerare	la congelare	
Carne de vită	- 20	- 35	20 - 30	2,06	1,06	3,12
Carne de porc cu slănină	- 18	- 30	15 - 17	1,1	0,6	1,7

*Loss of weight for horticultural products, in storage units*

Product	Weight loss (%/24h)	Product	Weight loss (%/24h)
sweet cherries	1,2...2,0	citrus	0,1
sour cherries	2,6...3,0	strawberries	3,6...4,0
apricots	0,25...0,35	carrots	0,04
apples	0,03	parsley	0,05
pears	0,05	fallow	
water melons	2,0...3,0	onion	0,05...0,07

Table 16.5

## Căldura de respirație pentru fructe și legume

Denumire	kJ / (tone · 24 h) la temperatura [°C] de:					
	0	2	5	10	15	20
<b>FRUCTE:</b>						
Cireșe, vișine	1880	2970	4600	9200	15880	20900
Caise	1463	2300	4807	8780	13380	17140
Piersici	1630	1880	3510	7940	11290	15675
Prune	1840	3010	5643	10870	15880	20060
Pere timpurii	1254	2260	3970	5430	13800	23000
Pere târzii	920	1920	3550	8400	8360	18800
Mere timpurii	1590	1797	2720	5225	7940	10450
Mere târzii	920	1170	1800	2675	5016	6270
Struguri	840	1460	2090	3135	4180	6690
Căpșuni	4013	5430	7940	15050	20900	25900
Zmeură	7940	10030	14210	24240	50160	62700
Mure	5950	8780	11700	24240	37200	50160
Lămâi	840	1130	1670	2800	4050	5020
Portocale	920	1090	1630	3010	4800	5935
Grapefruit	920	1090	1300	2170	3680	4800
Banane verzi	-	-	4390	8440	11290	13380
Banane coapte	-	-	5020	10030	14210	20900
<b>LEGUME:</b>						
Sparanghel	5430	6270	7315	13800	24035	31350
Spanac	7106	10240	17140	26960	45140	77330
Salată verde	3344	3760	4390	8780	16300	29260
Fasole verde	6900	7100	10450	17760	35530	49740
Mazăre verde	8990	12330	16300	23000	39700	55600
Castraveți	1756	2090	2930	5225	10450	15050
Roșii	1505	1670	2300	3550	7520	8780
Conopidă	5430	6060	6700	11910	22360	34700
Ciuperci	10450	11290	13800	21730	41800	54760

Table 16.6

## Storage conditions for refrigerated foods

Product	Temperature [°C]	Relative humidity [%]	Duration
Apples	1...4	85...90	2...8 months
Lilk	0...1	95	2...4 months
Cheese	1...4	65...70	6...18 months
Eggs	-1...0	80...85	5...6 months
Grapes	0...1	90...95	2...5 months
Fish	1...2	90...95	5...15 days
Veal	-1...1	85...90	1...6 weeks
Pork meat	0...1	85...90	3...7 days
Chicken meat	-1...0	85...90	1 week
lamb meat	0...1	85...90	5...14 days
Ciuperci	0	90	1...4 days
Pears	-1...1	90...95	2...6 months
Plums	0...1	85...90	2...8 week
Onion	0...1	65...70	1...8 months
Potatoes	1...3	90...95	6...10 months
Strawberries	0...1	90...95	5...7 days

## 16.2. Operating parameters of the cooling system

The refrigeration unit uses a vapor compression cooling system, operating with the refrigerant R-22 for freezing and with R-134a for refrigeration. The condenser of the cooling system is air cooled. Fig. 16.4 and 16.5 present the schematics and the operating cycle of the cooling system.

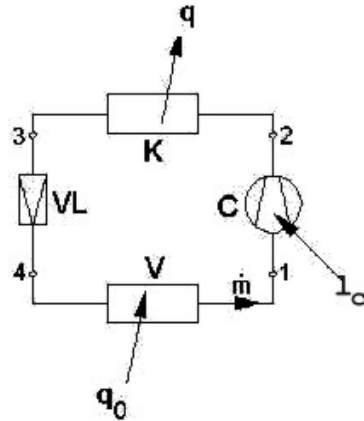


Fig. 16.4 – Schematics of the cooling system

V-evaporator;  
C-compressor;  
K-condenser;  
VL-expansion valve.

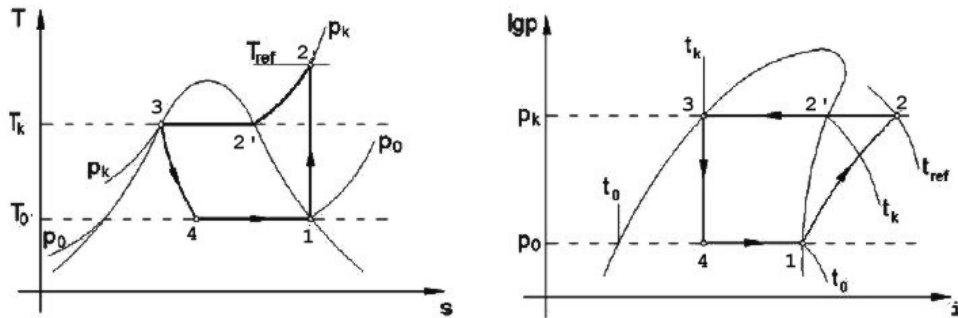


Fig. 16.5 – Operating cycle of the cooling system  
 $t_{ref}$ -refrigerant temperature at the compressor outlet

### STEP 1. Evaporating and condensing temperature

Fig. 16.6 presents the temperatures for the evaporator;  $t_{ai}$  is the temperature of the air entering the evaporator, equal to the temperature inside the unit ( $t_{ai} = t_i$ ). For industrial applications:

- $\Delta t_{a0} = 3 \dots 5 \text{ } ^\circ\text{C}$ ;
- $\Delta t_{tot0} = 4 \dots 10 \text{ } ^\circ\text{C}$  (table 16.7, as a function of the relative humidity of the air inside the unit).

Finally, we get  $t_0 = t_{ai} - \Delta t_{tot0}$ .

The corresponding evaporating pressure  $p_0$  is obtained by the means of the option “Saturation table” within the “Refrigeration utilities” tab of CoolPack (fig. 16.7).

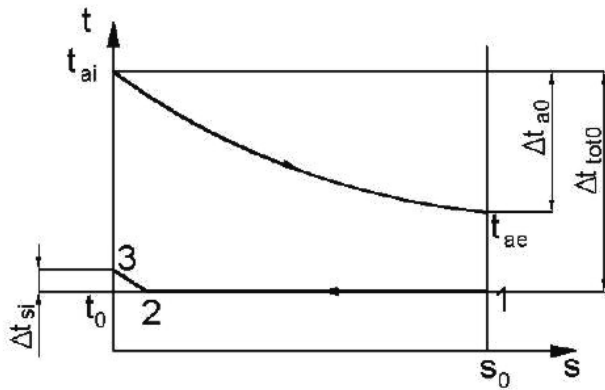


Fig. 16.6 – Temperatures for the evaporator:

$t_{ai}$  – temperature of air entering the evaporator;  
 $t_{ae}$  – air temperature at the outlet of the evaporator;  
 $t_0$  – evaporating temperature of the refrigerant;  
 1-2 – refrigerant evaporation process.

Table 16.7

Total temperature difference in the evaporator  $\Delta t_{tot0}$

$\Delta t_{tot0}, ^\circ\text{C}$	4,0...5,5	5,5...6,5	6,5...8,0	8,0...9,0	9,0...10,0
relative humidity, %	95...91	90...86	85...81	80...76	75...70

Refrigeration utilities - [Saturated values for: R134a, CH2FCF3, 1,1,1,2-tetrafluoroethane]								
File Options Window Help								
T	p	$v_l$	$v_g$	$h_l$	$h_g$	R	$s_l$	$s_g$
$^\circ\text{C}$	Bar	$\text{dm}^3/\text{kg}$	$\text{m}^3/\text{kg}$	$\text{kJ}/\text{kg}$	$\text{kJ}/\text{kg}$	$\text{kJ}/\text{kg}$	$\text{kJ}/(\text{kg K})$	$\text{kJ}/(\text{kg K})$
-22.00	1.219	0.7328	0.15896	171.74	384.05	212.31	0.8927	1.7380
-21.00	1.274	0.7345	0.15253	172.99	384.67	211.68	0.8976	1.7371
-20.00	1.330	0.7361	0.14641	174.24	385.28	211.04	0.9025	1.7362
-19.00	1.388	0.7378	0.14059	175.49	385.89	210.40	0.9075	1.7353
-18.00	1.448	0.7394	0.13504	176.75	386.50	209.75	0.9124	1.7345
-17.00	1.511	0.7411	0.12975	178.01	387.11	209.10	0.9173	1.7336
-16.00	1.575	0.7428	0.12471	179.27	387.71	208.44	0.9222	1.7328
-15.00	1.641	0.7445	0.11991	180.54	388.32	207.78	0.9271	1.7320
-14.00	1.710	0.7463	0.11533	181.81	388.92	207.11	0.9320	1.7312
-13.00	1.781	0.7480	0.11095	183.09	389.52	206.44	0.9369	1.7304
-12.00	1.854	0.7498	0.10678	184.36	390.12	205.76	0.9418	1.7297
-11.00	1.929	0.7515	0.10279	185.65	390.72	205.08	0.9467	1.7289
-10.00	2.007	0.7533	0.09898	186.93	391.32	204.39	0.9515	1.7282
-9.00	2.088	0.7551	0.09534	188.22	391.92	203.69	0.9564	1.7275
-8.00	2.170	0.7569	0.09186	189.52	392.51	202.99	0.9613	1.7269
-7.00	2.256	0.7588	0.08853	190.82	393.10	202.29	0.9661	1.7262
-6.00	2.344	0.7606	0.08535	192.12	393.70	201.58	0.9710	1.7255
-5.00	2.434	0.7625	0.08230	193.42	394.28	200.86	0.9758	1.7249
-4.00	2.527	0.7644	0.07938	194.73	394.87	200.14	0.9807	1.7243
-3.00	2.623	0.7663	0.07659	196.04	395.46	199.42	0.9855	1.7237
-2.00	2.722	0.7682	0.07391	197.36	396.04	198.68	0.9903	1.7231
-1.00	2.824	0.7701	0.07135	198.68	396.62	197.95	0.9952	1.7225
0.00	2.928	0.7721	0.06889	200.00	397.20	197.20	1.0000	1.7220
1.00	3.036	0.7740	0.06653	201.33	397.78	196.45	1.0048	1.7214
2.00	3.146	0.7760	0.06427	202.66	398.36	195.70	1.0096	1.7209
3.00	3.260	0.7781	0.06210	203.99	398.93	194.94	1.0144	1.7204
4.00	3.376	0.7801	0.06001	205.33	399.50	194.17	1.0192	1.7199
5.00	3.496	0.7821	0.05801	206.67	400.07	193.40	1.0240	1.7194

Fig. 16.7 – Saturation table in CoolPack



Fig. 16.8 presents the temperatures for the condenser;  $t_{aik}$  is the temperature of outside air entering the condenser, equal to the outside temperature ( $t_{aik} = 30\text{ }^{\circ}\text{C}$ )

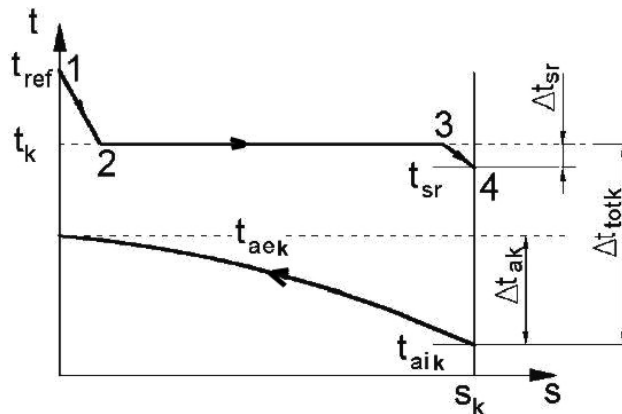


Fig. 16.8 – **Temperatures for the condenser**

$t_{aik}$  – temperature of the air entering the condenser;  
 $t_{aek}$  – air temperature at the outlet of the condenser;  
 $t_k$  – refrigerant condensing temperature;  
 $t_{ref}$  – refrigerant temperature at the compressor outlet;  
 2-3 – refrigerant condensing process.

For industrial applications  $\Delta t_{ak} = 5...10\text{ }^{\circ}\text{C}$ ,  $\Delta t_{totk} = 10...20\text{ }^{\circ}\text{C}$ ; the condensing temperature is therefore  $t_k = t_{aik} + \Delta t_{totk}$ .

The corresponding condensing pressure  $p_k$  is also obtained using the option “Saturation table” within the “Refrigeration utilities” tab of CoolPack.

We now calculate the pressure ratio:

$$\varepsilon = \frac{p_k}{p_0}$$

For  $\varepsilon < 6...8$ , a single stage compressor should be used; for  $\varepsilon > 8$ , a two stage compressor is needed.

## STEP 2. Operating cycle of the refrigeration system

In order to draw the operating cycle of the refrigeration system we use again the CoolPack package (**Refrigeration utilities** → **log(p)-h diagram** → **Cycle input**); the cycle will be drawn in  $\lg p - h$  (pressure – enthalpy) coordinates.

The window presented in fig. 16.9 allows the introduction of the necessary parameters:

- type of compressor: one stage or two stage;
- evaporating temperature  $t_0$ ;
- condensing temperature  $t_k$ ;
- isentropic efficiency: 0,6..0,8.

In order to simplify the calculation process, all pressure drops (in evaporator, condenser, connecting lines) are set to zero. After pressing the button “Draw cycle” the operating cycle will be drawn (fig. 16.10).

Supplementary operating parameters of the cooling system are obtained with the option

“Cycle info” (fig. 1.611); the previously calculated cooling power is

introduced in the “*Dimensioning*” section and a value of 0.85 is used for the “*Volumetric efficiency*” section.

Fig. 16.9 – Window for introducing the working parameters

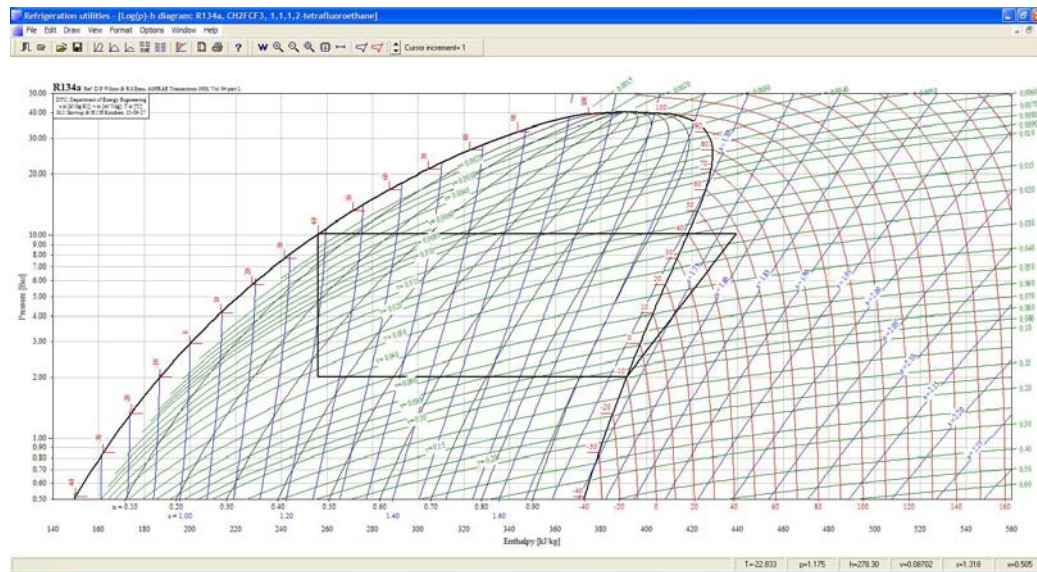


Fig. 16.10 – Operating cycle of the refrigeration system

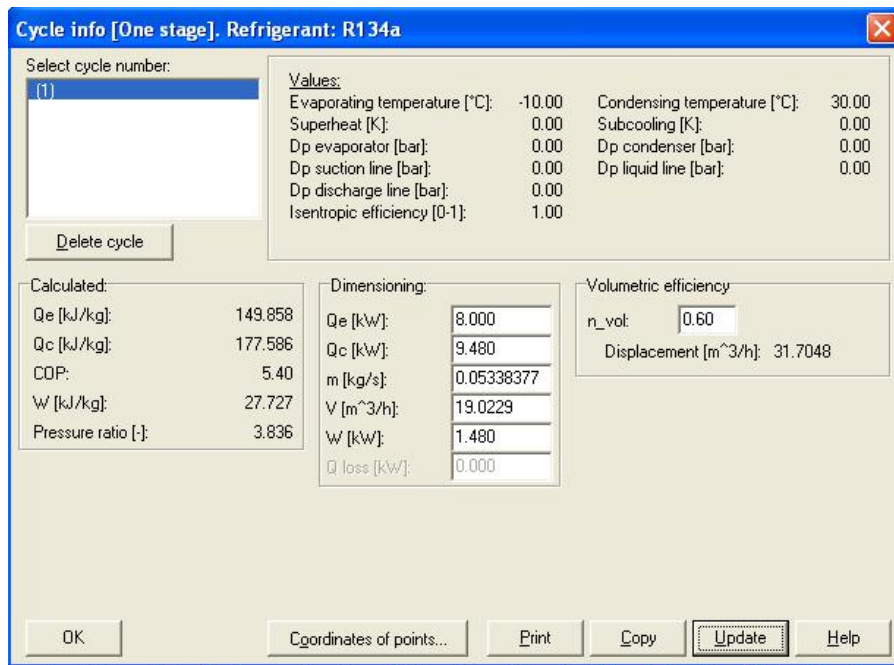


Fig. 16.11 - "Cycle info" window

The program calculates the rest of the operating parameters when the "Update" button is pressed, as follows:

- refrigerant mass flow,  $\mathbf{m}$  [kg/s];
- heat load of the condenser,  $\mathbf{Q_c}$  [kW];
- refrigerant volume flow,  $\mathbf{V}$  [m³/h]<sup>5</sup>;
- compressor power,  $\mathbf{W}$  [kW];
- compressor displacement (theoretical volume flow), "**Displacement**" [m³/h];
- the system's coefficient of performance, **COP**.

<sup>5</sup>  $V = n_{vol} \cdot \text{Displacement}$