UNIT ! – INTRODUCTION

Refrigeration deals with cooling of bodies or fluids to temperatures lower than those of surroundings. This involves absorption of heat at a lower temperature and rejection to higher temperature of the surroundings. In olden days, the main purpose of refrigeration was to produce ice, which was used for cooling beverages, food preservation and refrigerated transport etc. Now-a-days refrigeration and air conditioning find so many applications that they have become very essential for mankind. The most important applications of refrigeration is in cooling and dehumidification as required for summer air conditioning.

APPLICATIONS

The major applications of refrigeration can be grouped into following four major equally important areas.

1. Food processing, preservation and distribution
2. Chemical and process industries
3. Special Applications
4. Comfort air-conditioning

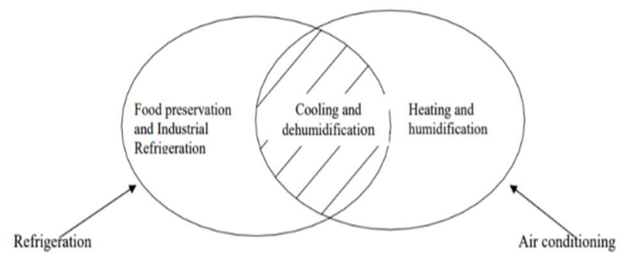


Fig.3.1. Relation between refrigeration and air conditioning

Special applications include Cold Treatment of Metals, Medical, Ice Skating Rinks, Construction, Desalination of Water, and Ice Manufacture

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

IMPORTANT TERMS AND LAWS

SYSTEM, SURROUNDINGS AND BOUNDARY

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.

Thermodynamic systems can be further classified into

- Closed systems,
- Open systems and
- Isolated systems.

A control volume, which may be considered as an open system, is defined as a specified region in space upon which attention is focused. The control volume is separated from the surroundings by a control surface. Both mass and energy can enter or leave the control volume. In case of closed system this mass transfer is not possible. Whereas in case of isolated system both mass and energy transfer is not possible.

PROCESS AND CYCLE

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 refrigeration and air conditioning one encounters a wide variety of processes. Understanding the nature of the process path is very important as heat and work depend on the path. 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

HEAT AND WORK

Heat is energy transferred between a system and its surroundings by virtue of a temperature difference only. The different modes of heat transfer are:

- Conduction,
- Convection and
- Radiation.

Heat is a way of changing the energy of a system. 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.

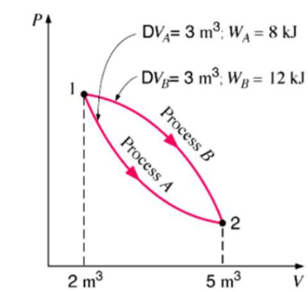


Fig. 4.1. Difference between point and path functions

Mechanical modes of work: In mechanics work is said to be done when a force 'F' moves through a distance 'dx'. When this force is a mechanical force, we call the work done as a mechanical mode of work. The classical examples of mechanical mode of work are:

1. Moving system boundary work
2. Rotating shaft work
3. Elastic work, and
4. Surface tension work

$$ {}_1W_2 = \int_1^2 p.dV $$

where 'p' is the pressure acting on the system boundary and 'dV' is the differential volume. It is assumed that the process is carried out very slowly so that at each instant of time the system is in equilibrium. Typically such a process is called a quasi-equilibrium process. For rigid containers, volume is constant, hence moving boundary work is zero in this case. For other systems, in order to find the work done one needs to know the relation between pressure p and volume V during the 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.

PATH AND POINT FUNCTIONS

Path function depends on history of the system (or path by which system arrived at a given state). Examples for path functions are work and heat.

Point function does not depend on the history (or path) of the system. It only depends on the state of the system. Examples of point functions are: temperature, pressure, density, mass, volume, enthalpy, entropy, internal energy etc.

Path functions are not properties of the system, while point functions are properties of the system. Change in point function can be obtained by from the initial and final values of the function, whereas path has to be defined in order to evaluate path functions.

PROPERTY

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)

Thermodynamic properties relevant to refrigeration and air conditioning systems are temperature, pressure, volume, density, specific heat, enthalpy, entropy etc. It is to be noted that heat and work are not properties of a system.

LAWS

Conservation of Mass is a fundamental concept, which states that mass is neither created nor destroyed.

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.

The First law of Thermodynamics is a statement of law of conservation of energy. It states that energy can neither be created not be destroyed but can be transferred from one form to the other. Also, according to this law, heat and work are interchangeable. Any system that violates the first law (i.e., creates or destroys energy) is known as a Perpetual Motion Machine (PMM) of first kind. For a system undergoing a cyclic process, the first law of thermodynamics is given by:

$$\oint \delta Q = \oint \delta W$$

where $\oint \delta Q$ = net heat transfer during the cycle

$\oint \delta W$ = net work transfer during the cycle

$$\oint(\delta Q-\delta W)=0$$

This implies that $(\delta Q-\delta W)$ must be a point function or property of the system. This property is termed as *internal energy,* **U**. Mathematically, internal energy can be written as:

$$dU=\delta Q-\delta W$$

The internal energy of a system represents a sum total of all forms of energy viz. thermal, molecular, lattice, nuclear, rotational, vibrational etc.

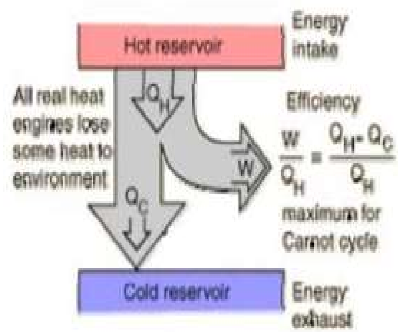
Clausius' statement of Second law

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

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. Mathematically, Kelvin-Planck statement can be written as:

$W_{cycle} \le 0$ (for a single reservoir)



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.

HEAT ENGINES, REFRIGERATORS, HEAT PUMPS:

A Heat Engine may be defined as a device that operates in a thermodynamic cycle and does a certain amount of net positive work through the transfer of heat from a high temperature body to a low temperature body. Example: A steam power plant

A Refrigerator may be defined as a device that operates in a thermodynamic cycle and transfers a certain amount of heat from a body at a lower temperature to a body at a higher temperature by consuming certain amount of external work. Example: Domestic refrigerators and room air conditioners.. In a refrigerator, the required output is the heat extracted from the low temperature body.

A Heat Pump is similar to a refrigerator, however, here the required output is the heat rejected to the high temperature body.

CARNOT'S THEOREMS FOR HEAT ENGINES:

Theorem 1: It is impossible to construct a heat engine that operates between two thermal reservoirs and is more efficient than a reversible engine operating between the same two reservoirs.

Theorem 2: All reversible heat engines operating between the same two thermal reservoirs have the same thermal efficiency.

The two theorems can be proved by carrying out a thought experiment and with the help of second law. Carnot's theorems can also be formed for refrigerators in a manner similar to heat engines.

Carnot Efficiency: The Carnot efficiencies are the efficiencies of completely reversible cycles operating between two thermal reservoirs. According to Carnot's theorems, for any given two thermal reservoirs, the Carnot efficiency represents the maximum possible efficiency.

Thermal efficiency for a heat engine, η_{HE} is defined as:

$$\eta_{HE} = \frac{W_{cycle}}{Q_H} = 1 - \frac{Q_C}{Q_H}$$

where W_{cycle} is the net work output, Q_C and Q_H and are the heat rejected to the low temperature reservoir and heat added (heat input) from the high temperature reservoir, respectively.

It follows from Carnot's theorems that for a reversible cycle ($\frac{Q_C}{Q_H}$) is a function of temperatures of the two reservoirs only. i.e. $\frac{Q_C}{Q_H} = \phi(T_C, T_H)$.

If we choose the absolute (*Kelvin*) temperature scale then:

$$\frac{Q_C}{Q_H} = \frac{T_C}{T_H}$$

$$\text{hence, } \eta_{Carnot,HE} = 1 - \frac{Q_C}{Q_H} = 1 - \frac{T_C}{T_H}$$

The efficiency of refrigerator and heat pump is called as Coefficient of Performance (COP). Similarly to heat engines, Carnot coefficient of performance for heat pump and refrigerators COP_{HP} and COP_R can be written as

$$\begin{aligned} COP_{Carnot,HP} &= \frac{Q_H}{W_{cycle}} = \frac{Q_H}{Q_H - Q_C} = \frac{T_H}{T_H - T_C} \\ COP_{Carnot,R} &= \frac{Q_C}{W_{cycle}} = \frac{Q_C}{Q_H - Q_C} = \frac{T_C}{T_H - T_C} \end{aligned} \tag{4.15}$$

where

W_{cycle} = work input to the reversible heat pump and refrigerator
Q_H = heat transferred between the system and the hot reservoir
Q_C = heat transferred between the system and cold reservoir
T_H = temperature of the hot reservoir
T_C = temperature of the cold reservoir

Third law of thermodynamics:

This law gives the definition of absolute value of entropy and also states that absolute zero cannot be achieved. Another version of this law is that "the entropy of perfect crystals is zero at absolute zero". This statement is attributed to Plank. This is in line with the concept that entropy is a measure of disorder of the system.

If 'ω' is the probability of achieving a particular state out of a large number of states; then entropy of the system is equal to ln(ω). The transitional movement of molecules ceases at absolute zero and position of atoms can be uniquely specified. In addition, if we have a perfect crystal, then all of its atoms are alike and their positions can be interchanged without changing the state. The probability of this state is unity, that is $\omega = 1$ and $\ln(\omega) = \ln(1) = 0$

For imperfect crystals however there is some entropy associated with configuration of molecules and atoms even when all motions cease, hence the entropy in this case does not tend to zero as $T \rightarrow 0$, but it tends to a constant called the entropy of configuration.

The third law allows absolute entropy to be determined with zero entropy at absolute zero as the reference state. In refrigeration systems we deal with entropy changes only, the absolute entropy is not of much use. Therefore entropy may be taken to be zero or a constant at any suitably chosen reference state.

Another consequence of third law is that absolute zero cannot be achieved. One tries to approach absolute zero by magnetization to align the molecules. This is followed by cooling and then demagnetization, which extracts energy from the substance and reduces its temperature. It can be shown that this process will require infinite number of cycles to achieve absolute zero. Infinitely large amount of work is required to maintain absolute zero if at all it can be achieved.

COMPARISON BETWEEN GAS CYCLES AND VAPOR CYCLES

Thermodynamic cycles can be categorized into gas cycles and vapour cycles. In a typical gas cycle, the working fluid (a gas) does not undergo phase change, consequently the operating cycle will be away from the vapour dome.
In Gas cycles, heat rejection and refrigeration take place as the gas undergoes sensible cooling and heating. In a Vapour cycle the working fluid undergoes phase change and refrigeration effect is due to the vaporization of refrigerant liquid. If the refrigerant is a pure substance then its temperature remains constant during the phase change processes. However, if a zeotropic mixture is used as a refrigerant, then there will be a temperature glide during vaporization and condensation. Since the refrigeration effect is produced during phase change, large amount of heat (latent heat) can be transferred per kilogram of refrigerant at a near constant temperature.

Hence, the required mass flow rates for a given refrigeration capacity will be much smaller compared to a gas cycle. Vapour cycles can be subdivided into

- Vapour compression systems
- Vapour absorption systems
- Vapour jet systems etc.

Among these the vapour compression refrigeration systems are predominant.

THE CARNOT REFRIGERATION CYCLE

Carnot refrigeration cycle is a completely reversible cycle, hence is used as a model of perfection for a refrigeration cycle operating between a constant temperature heat source and sink. It is used as reference against which the real cycles are compared.

The basic Carnot refrigeration system for pure vapour consists of four components: Compressor, Condenser, Turbine and Evaporator. Refrigeration effect (q4 - q1 = qe) is obtained at the evaporator as the refrigerant undergoes the process of vaporization (process 4-1) and extracts the latent heat from the low temperature heat source.

1. The low temperature, low pressure vapour is then compressed isentropically in the compressor to the heat sink temperature Tc. The refrigerant pressure increases from Pe to Pc during the compression process (process 1-2) and the exit vapour is saturated.
2. Next the high pressure, high temperature saturated refrigerant undergoes the process of condensation in the condenser (process 2-3) as it rejects the heat of condensation (q2-3 = qc) to an external heat sink at Tc.
3. The high pressure saturated liquid then flows through the turbine and undergoes isentropic expansion (process 3-4). During this process, the pressure and temperature fall from Pc,Tc to Pe, Te. Since a saturated liquid is expanded in the turbine, some amount of liquid flashes into vapour and the exit condition lies in the two-phase region.
4. This low temperature and low pressure liquid-vapour mixture then enters the evaporator completing the cycle. the cycle involves two isothermal heat transfer processes (processes 4-1 and 2-3) and two isentropic work transfer processes (processes 1-2 and 3-4).
5.

Heat is extracted isothermally at evaporator temperature Te during process 4-1, heat is rejected isothermally at condenser temperature Tc during process 2-3. Work is supplied to the compressor during the isentropic compression (1-2) of refrigerant vapour from evaporator pressure Pe to condenser pressure Pc, and work is produced by the system as refrigerant liquid expands isentropically in the turbine from condenser pressure Pc to evaporator pressure Pe. All the processes are both internally as well as externally reversible, i.e., net entropy generation for the system and environment is zero. Applying first and second laws of thermodynamics to the Carnot refrigeration cycle,

$$\oint \delta q = \oint \delta w$$

$$\oint \delta q = q_{4-1} - q_{2-3} = q_e - q_c$$

$$\oint \delta w = w_{3-4} - w_{1-2} = w_T - w_C = -w_{net}$$

$$\Rightarrow (q_c - q_e) = w_{net}$$

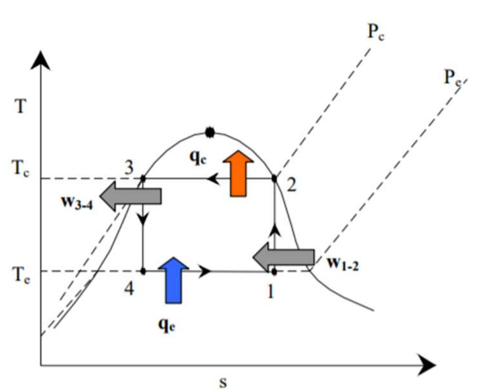
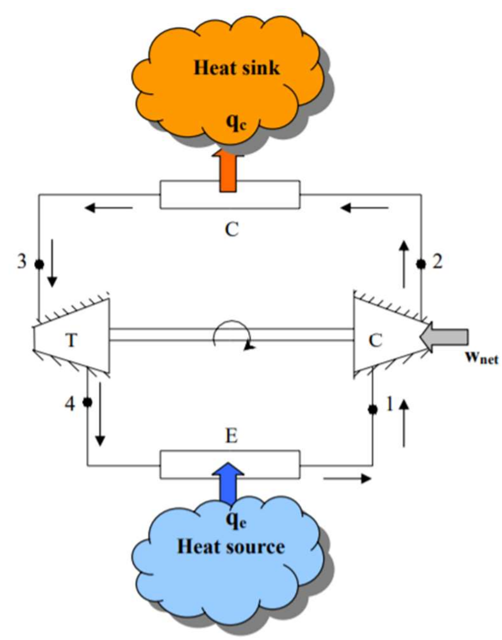


Fig. 10.1(b): Carnot refrigeration cycle on T-s diagram



Schematic of a Carnot refrigeration system now for the reversible, isothermal heat transfer processes 2-3 and 4-1, we can write:

$$q_c = -q_{2-3} = -\int_2^3 T.ds = T_c(s_2 - s_3)$$

$$q_e = q_{4-1} = \int_4^1 T.ds = T_e(s_1 - s_4)$$

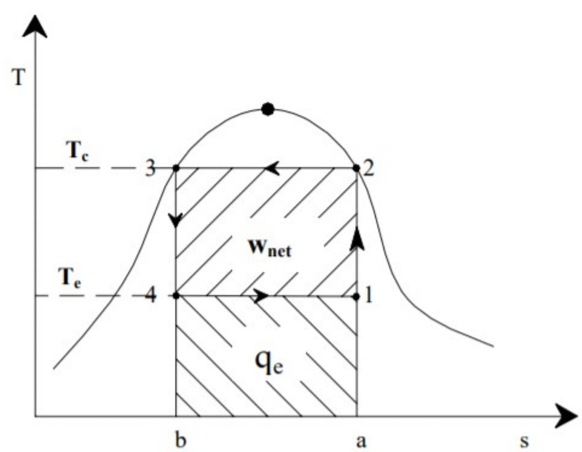
where Te and Tc are the evaporator and condenser temperatures, respectively, and,

$$s_1 = s_2 \text{ and } s_3 = s_4$$

the Coefficient of Performance (COP) is given by:

$$COP_{Carnot} = \frac{\text{refrigeration effect}}{\text{net work input}} = \frac{q_e}{w_{net}} = \frac{T_e(s_1 - s_4)}{T_c(s_2 - s_3) - T_e(s_1 - s_4)} = \left(\frac{T_e}{T_c - T_e}\right)$$

Thus the COP of Carnot refrigeration cycle is a function of evaporator and condenser temperatures only and is independent of the nature of the working substance. This is the reason why exactly the same expression was obtained for air cycle refrigeration systems operating on Carnot cycle The Carnot COP sets an upper limit for refrigeration systems operating between two constant temperature thermal reservoirs (heat source and sink). From Carnot's theorems, for the same heat source and sink temperatures, no irreversible cycle can have COP higher than that of Carnot COP.



Carnot refrigeration cycle represented in T-s plane

It can be seen from the above expression that the COP of a Carnot refrigeration system increases as the evaporator temperature increases and condenser temperature decreases. This can be explained very easily with the help of the T-s diagram

ANALYSIS OF STANDARD VAPOUR COMPRESSION REFRIGERATION SYSTEM

A simple analysis of standard vapor compression refrigeration system can be carried out by assuming a) Steady flow; b) negligible kinetic and potential energy changes across each component, and c) no heat transfer in connecting pipe lines. The steady flow energy equation is applied to each of the four components.

<u>Evaporator:</u> Heat transfer rate at evaporator or *refrigeration capacity*, $\dot{Q}_e$ is given by:

$$\dot{Q}_e = \dot{m}_r (h_1 - h_4)$$

where $\dot{m}_r$ is the refrigerant mass flow rate in kg/s, h_1 and h_4 are the specific enthalpies (kJ/kg) at the exit and inlet to the evaporator, respectively. $(h_1 - h_4)$ is known as specific refrigeration effect or simply *refrigeration effect*, which is equal to the heat transferred at the evaporator per kilogram of refrigerant. The evaporator pressure P_e is the saturation pressure corresponding to evaporator temperature T_e, i.e.,

$$P_e = P_{sat}(T_e)$$

<u>Compressor:</u> Power input to the compressor, $\dot{W}_c$ is given by:

$$\dot{W}_c = \dot{m}_r (h_2 - h_1)$$

where h_2 and h_1 are the specific enthalpies (kJ/kg) at the exit and inlet to the compressor, respectively. $(h_2 - h_1)$ is known as specific work of compression or simply *work of compression*, which is equal to the work input to the compressor per kilogram of refrigerant.

<u>Condenser:</u> Heat transfer rate at condenser, $\dot{Q}_c$ is given by:

$$Q_c = m_r (h_2 - h_3)$$

where h_3 and h_2 are the specific enthalpies (kJ/kg) at the exit and inlet to the condenser, respectively.

The condenser pressure P_c is the saturation pressure corresponding to evaporator temperature T_c, i.e.,

$$P_c = P_{sat}(T_c)$$

<u>Expansion device:</u> For the isenthalpic expansion process, the kinetic energy change across the expansion device could be considerable, however, if we take the control volume, well downstream of the expansion device, then the kinetic energy gets dissipated due to viscous effects, and

$$h_3 = h_4$$

The exit condition of the expansion device lies in the two-phase region, hence applying the definition of quality (or dryness fraction), we can write:

$$h_4 = (1 - x_4) h_{f,e} + x_4 h_{g,e} = h_f + x_4 h_{fg}$$

where x_4 is the quality of refrigerant at point 4, $h_{f,e}$, $h_{g,e}$, h_{fg} are the saturated liquid enthalpy, saturated vapour enthalpy and latent heat of vaporization at evaporator pressure, respectively.

The COP of the system is given by:

$$COP = \left(\frac{\dot{Q}_e}{\dot{W}_c} \right) = \left(\frac{\dot{m}_r (h_1 - h_4)}{\dot{m}_r (h_2 - h_1)} \right) = \frac{(h_1 - h_4)}{(h_2 - h_1)} \qquad (10.29)$$

At any point in the cycle, the mass flow rate of refrigerant $\dot{m}_r$ can be written in terms of volumetric flow rate and specific volume at that point, i.e.,

$$\dot{m}_r = \dot{V} / v$$

applying this equation to the inlet condition of the compressor,

$$\dot{m}_r = \dot{V}_1 / v_1$$

where $\dot{V}_1$ is the volumetric flow rate at compressor inlet and v_1 is the specific volume at compressor inlet. At a given compressor speed, $\dot{V}_1$ is an indication of the size of the compressor. We can also write, the refrigeration capacity in terms of volumetric flow rate as:

$$\dot{Q}_e = \dot{m}_r (h_1 - h_4) = \dot{V}_1 \left(\frac{h_1 - h_4}{v_1} \right)$$

where $\left(\frac{h_1 - h_4}{v_1} \right)$ is called as *volumetric refrigeration effect* (kJ/m^3 of refrigerant).

IDEAL PROPERTIES FOR A REFRIGERANT

It will be useful to remind ourselves of the requirements for a fluid used as a refrigerant.

- A high latent heat of vaporization
- A high density of suction gas
- Non-corrosive, non-toxic and non-flammable
- Critical temperature and triple point outside the working range
- Compatibility with component materials and lubricating oil
- Reasonable working pressures (not too high, or below atmospheric pressure)
- High dielectric strength (for compressors with integral motors)
- Low cost
- Ease of leak detection
- Environmentally friendly

No single fluid has all these properties, and meets the new environmental requirements.

Typical application	*Refrigerants recommended*
Domestic refrigerators and freezers	R12
Small retail and supermarkets	R12, R22, R502
Air-conditioning	R11, R114, R12, R22
Industrial	R717, R22, R502, R13B1
Transport	R12, R502

OZONE DEPLETION POTENTIAL (ODP)

The ozone layer in our upper atmosphere provides a filter for ultraviolet radiation, which can be harmful to our health. Research has found that the ozone layer is thinning, due to emissions into the atmosphere of chlorofluorocarbons (CFCs), halons and bromides. The Montreal Protocol in 1987 agreed that the production of these chemicals would be phased out by 1995 and alternative fluids were developed. From Table, R11, R12, R114 and R502 are all CFCs used as refrigerants, while R13B1 is a halon. They have all ceased production within those countries which are signatories to the Montreal Protocol. The situation is not so clear-cut, because there are countries like Russia, India, China etc. who are not signatories and who could still be producing these harmful chemicals.

It should be noted that prior to 1987, total CFC emissions were made up from aerosol sprays, solvents and foam insulation, and that refrigerant emissions were about 10% of the total. However, all the different users have replaced CFCs with alternatives. R22 is an HCFC and now regarded as a transitional refrigerant, in that it will be completely phased out of production by 2030, as agreed under the Montreal Protocol. A separate European Community decision has set the following dates

1/1/2000	CFCs banned for servicing existing plants
1/1/2000	HCFCs banned for new systems with a shaft input power greater than 150 kW
1/1/2001	HCFCs banned in all new systems except heat pumps and reversible systems
1/1/2004	HCFCs banned for *all* systems
1/1/2008	Virgin HCFCs banned for plant servicing

Comparison of new refrigerants

Refrigerant type/no.	*Substitute for*	*ODP*	*GWP*	*Cond. temp. at 26 bar (°C)*	*Sat. temp. at 1 bar abs °C*
HCFC (short term)					
R22	R502, R12	0.05	1700	63	−41
HFCFC/HFC service-blends (transitional alternatives)					
R401A	R12	0.03	1080	80	−33
R401B	R12	0.035	1190	77	−35
R409A	R12	0.05	1440	75	−34
HFC–Chlorine free (long-term alternative)					
R134A	R12, R22	0	1300	80	−26
HFC–Chlorine free–blends–(long-term alternatives)					
R404A	R502	0	3750	55	−47
R407A	R502	0	1920	56	−46
R407B	R502	0	2560	53	−48
R407C	R22	0	1610	58	−44
ISCEON 59	R22	0	2120	68	−43
R410A	R22, R13B1	0	1890	43	−51
R411B	R12, R22, R502	0.045	1602	65	−42
Halogen free (long-term alternatives)					
R717 ammonia	R22, R502	0	0	60	−33
R600a isobutane	R114	0	3	114	−12
R290 propane	R12, R22, R502	0	3	70	−42
R1270 propylene	R12, R22, R502	0	3	61	−48

GLOBAL WARMING POTENTIAL (GWP)

Global warming is the increasing of the world's temperatures, which results in melting of the polar ice caps and rising sea levels. It is caused by the release into the atmosphere of so-called 'greenhouse' gases, which form a blanket and reflect heat back to the earth's surface, or hold heat in the atmosphere.

The most infamous greenhouse gas is carbon dioxide (CO2), which once released remains in the atmosphere for 500 years, so there is a constant build-up as time progresses. The main cause of CO2 emission is in the generation of electricity at power stations. Each kWh of electricity used in the UK produces about 0.53 kg of CO_2 and it is estimated that refrigeration compressors in the UK consume 12.5 billion kWh per year. Table below shows that the newly developed refrigerant gases also have a global warming potential if released into the atmosphere.

For example, R134a has a GWP of 1300, which means that the emission of 1 kg of R134a is equivalent to 1300 kg of CO2. The choice of refrigerant affects the GWP of the plant, but other factors also contribute to the overall GWP and this has been represented by the term total equivalent warming impact (TEWI). This term shows the overall impact on the global warming effect, and includes refrigerant leakage, refrigerant recovery losses and energy consumption. It is a term which should be calculated for each refrigeration plant. Figures below show the equation used and an example for a medium temperature R134a plant.

Environmental impact of some of the latest refrigerants

Refrigerant		*ODP (R11 = 1.0)*	*GWP* (CO_2 = 1.0)
R22	HCFC	0.05	1700
R134a	HFC	0	1300
R404a	HFC	0	3750
R407c	HFC	0	1610
R410a	HFC	0	1890
R411b	HCFC	0.045	1602
R717	ammonia	0	0
R290	propane	0	3
R600a	isobutane	0	3
R1270	propylene	0	3

$\boxed{\text{TEWI}}$ = TOTAL EQUIVALENT WARMING IMPACT

$$\text{TEWI} = \underbrace{\underbrace{(\text{GWP} \times L \times n)}_{\text{Leakage}} + \underbrace{(\text{GWP} \times m\,[1 - \alpha_{recovery}]}_{\text{Recovery losses}}}_{\text{direct global warming potential}} + \underbrace{\underbrace{(n \times E_{annual} \times \beta)}_{\text{Energy consumption}}}_{\text{indirect global warming potential}}$$

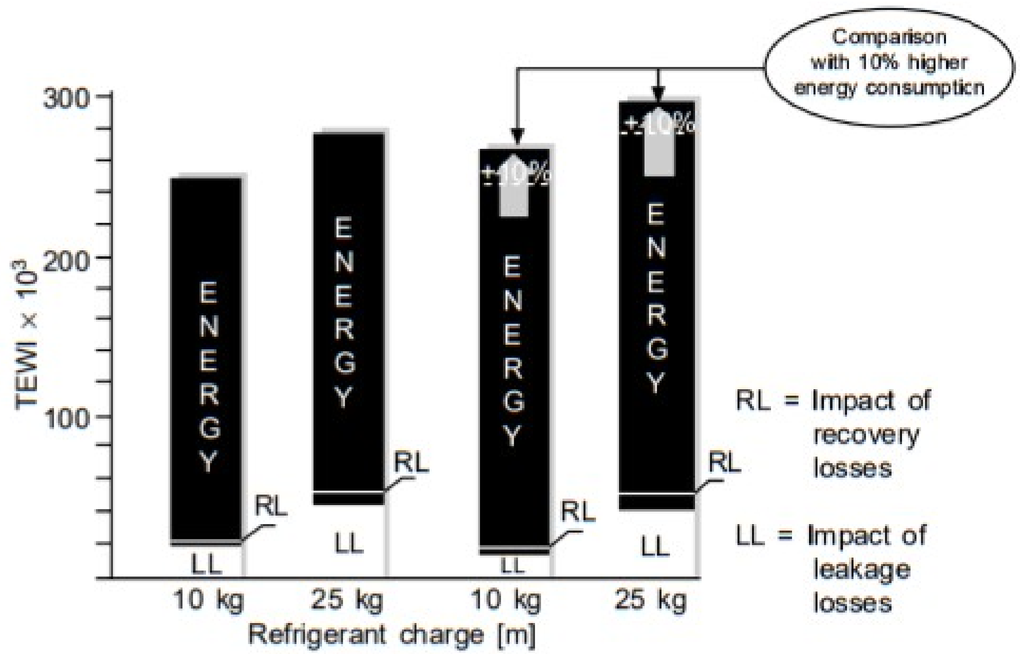
GWP	= Global warming potential	[CO_2-related]
L	= Leakage rate per year	[kg]
n	= System operating time	[Years]
m	= Refrigerant charge	[kg]
$\alpha_{recovery}$	= Recycling factor	
E_{annual}	= Energy consumption per year	[kWh]
β	= CO_2-Emission per kWh	(Energy-Mix)

Method for the calculation of TEWI figures

Example

Medium temperature R134a

t_o	−10°C
t_c	+40°C
m	10 kg // 25 kg
$L_{[10\%]}$	1 kg // 2,5 kg
Q_o	13,5 kW
E	5 kW × 5000 h/a
β	0,6 kg CO_2/kWh
α	0,75
n	15 years
GWP	1300 (CO_2 = 1) time horizon 100 years



Comparison of TEWI figures (example)

One thing that is certain is that the largest element of the TEWI is energy consumption, which contributes CO2 emission to the atmosphere. The choice of refrigerant is therefore about the efficiency of the refrigerant and the efficiency of the refrigeration system. The less the amount of energy needed to produce each kW of cooling, the less will be the effect on global warming

AMMONIA AND THE HYDROCARBONS

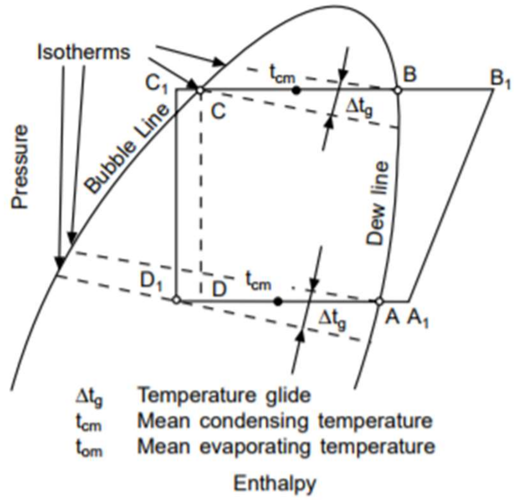
These fluids have virtually zero ODP and zero GWP when released into the atmosphere and therefore present a very friendly environmental picture.

Ammonia has long been used as a refrigerant for industrial applications. The engineering and servicing requirements are well established to deal with its high toxicity and flammability. There have been developments to produce packaged liquid chillers with ammonia as the refrigerant for use in air-conditioning in supermarkets, for example. Ammonia cannot be used with copper or copper alloys, so refrigerant piping and components have to be steel or aluminum. This may present difficulties for the air conditioning market where copper has been the base material for piping and plant. One property that is unique to ammonia compared to all other refrigerants is that it is less dense than air, so a leakage of ammonia results in it rising above the plant room and into the atmosphere. If the plant room is outside or on the roof of a building, the escaping ammonia will drift away from the refrigeration plant. The safety aspects of ammonia plants are well documented and there is reason to expect an increase in the use of ammonia as a refrigerant.

Hydrocarbons such as propane and butane are being successfully used as replacement and new refrigerants for R12 systems. They obviously have flammable characteristics which have to be taken into account by health and safety requirements. However, there is a market for their use in sealed refrigerant systems such as domestic refrigeration and unitary air-conditioners.

REFRIGERANT BLENDS

Many of the new, alternative refrigerants are ‘Blends’, which have two or three components, developed for existing and new plants as comparable alternatives to the refrigerants being replaced. They are ‘zeotropes’ with varying evaporating or condensing temperatures in the latent heat of vaporization phase, referred to as the ‘Temperature glide’.



Evaporating and condensing behaviour of zeotropic blends

Figure above shows the variation in evaporating and condensing temperatures. To compare the performance between single component refrigerants and blends it will be necessary to specify the evaporating temperature of the blend to point A on the diagram and the condensing temperature to point B. The temperature glide can be used to advantage in improving plant performance, by correct design of the heat exchangers. A problem associated with blends is that refrigerant leakage results in a change in the component concentration of the refrigerant. However, tests indicate that small changes in concentration (say less than 10%) have a negligible effect on plant performance. The following recommendations apply to the use of blends:

• The plant must always be charged with liquid refrigerant, or the component concentrations will shift.

• Since most blends contain at least one flammable component, the entry of air into the system must be avoided.

• Blends which have a large temperature glide, greater than 5K, should not be used for flooded-type evaporators.

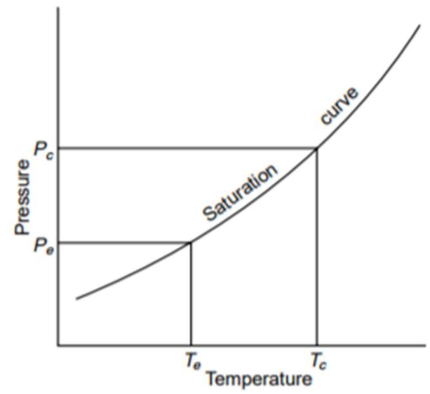
HEALTH AND SAFETY

When dealing with any refrigerant, personal safety and the safety of others are vitally important. Service and maintenance staff need to be familiar with safety procedures and what to do in the event of an emergency. Health and safety requirements are available from manufacturers of all refrigerants and should be obtained and studied. Safety codes are available from the Institute of Refrigeration in London, for HCFC/HFC refrigerants (A1 and A2), ammonia (B2) and hydrocarbons (A3). In the UK and most of Europe, it is illegal to dispose of refrigerant in any other way than through an authorized waste disposal company. The UK legislation expects that anyone handling refrigerants is competent to do so and has the correct equipment and containers. Disposal must be through an approved contractor and must be fully documented. Severe penalties may be imposed for failure to implement these laws

UNIT II - VAPOUR COMPRESSION REFRIGERATION SYSTEM

BASIC VAPOUR COMPRESSION CYCLE

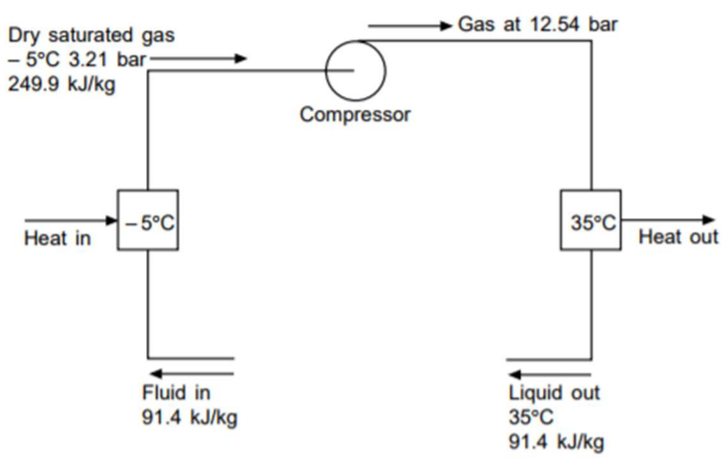
A liquid boils and condenses – the change between the liquid and gaseous states – at a temperature which depends on its pressure, within the limits of its freezing point and critical temperature. On boiling it must obtain the latent heat of evaporation and in condensing the latent heat must be given up again. The basic refrigeration cycle makes use of the boiling and condensing of a working fluid at different temperatures and, therefore, at different pressures.



Evaporation and condensation of a fluid

Heat is put into the fluid at the lower temperature and pressure and provides the latent heat to make it boil and change to a vapour. This vapour is then mechanically compressed to a higher pressure and a corresponding saturation temperature at which its latent heat can be rejected so that it changes back to a liquid.

The total cooling effect will be the heat transferred to the working fluid in the boiling or evaporating vessel, i.e. the change in enthalpies between the fluid entering and the vapour leaving the evaporator. For a typical circuit, using the working fluid Refrigerant 22, evaporating at – 5°C and condensing at 35°C, the pressures and enthalpies will be as shown



Basic Refrigeration Cycle

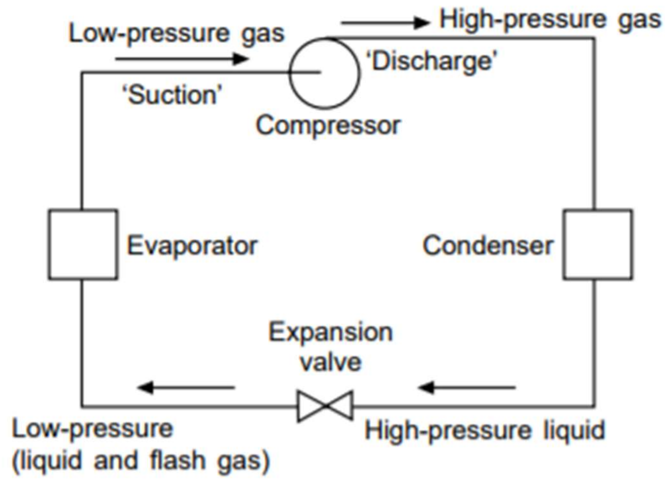
Enthalpy of fluid entering evaporator = 91.4 kJ/kg

Enthalpy of saturated gas leaving evaporator = 249.9 kJ/kg

Cooling effect = 249.9 – 91.4 = 158.5 kJ/kg

A working system will require a connection between the condenser and the inlet to the evaporator to complete the circuit. Since these are at different pressures this connection will require a pressure reducing and metering valve. Since the reduction in pressure at this valve must cause a corresponding drop in temperature, some of the fluid will flash off into vapour to remove the energy for this cooling. The volume of the working fluid therefore increases at the valve by this amount of flash gas, and gives rise to its name, the expansion valve

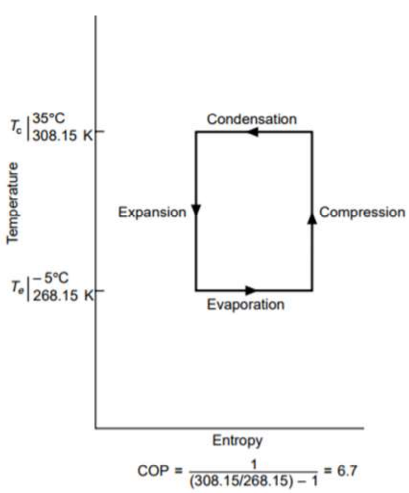
COEFFICIENT OF PERFORMANCE



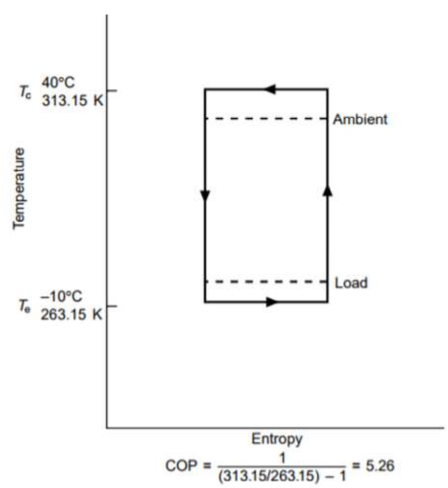
Complete Basic Cycle

Since the vapour compression cycle uses energy to move energy, the ratio of these two quantities can be used directly as a measure of the performance of the system. This ratio, the coefficient of performance, was first expressed by Sadi Carnot in 1824 for an ideal reversible cycle, and based on the two temperatures of the system, assuming that all heat is transferred at constant temperature. Since there are mechanical and thermal losses in a real circuit, the coefficient of performance (COP) will always be less than the ideal Carnot figure.

For practical purposes in working systems, it is the ratio of the cooling effect to the input compressor power. At the conditions, evaporating at – 5°C and condensing at 35°C (268.15 K and 308.15 K), the Carnot coefficient of performance is 6.7. Transfer of heat through the walls of the evaporator and condenser requires a temperature difference. This is shown on the modified reversed Carnot cycle. For temperature differences of 5 K on both the evaporator and condenser, the fluid operating temperatures would be 263.15 K and 313.15 K, and the coefficient of performance falls to 5.26.

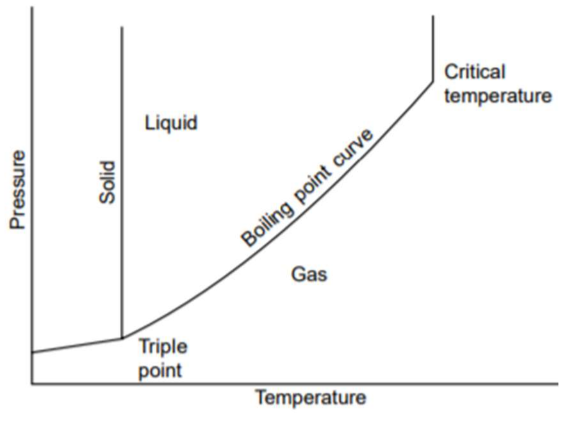


Ideal reversed Carnot cycle



Modified reversed Carnot cycle

SUBCOOLING AND SUPERHEATING



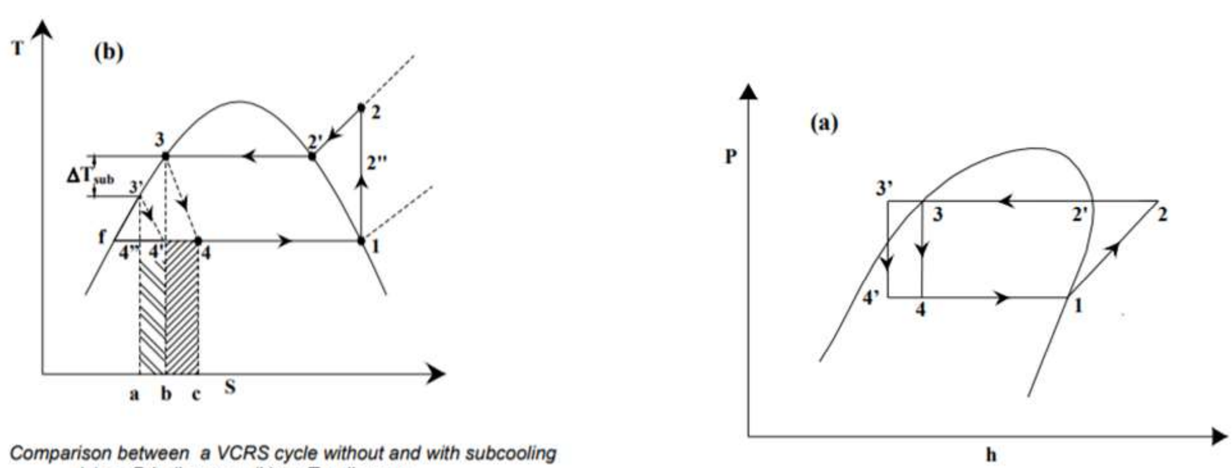
Change of state with pressure and temperature

The boiling point is limited by the critical temperature at the upper end, beyond which it cannot exist as a liquid, and by the triple point at the lower end, which is at the freezing temperature. Between these two limits, if the liquid is at a pressure higher than its boiling pressure, it will remain a liquid and will be subcooled below the saturation condition, while if the temperature is higher than saturation, it will be a gas and superheated. If both liquid and vapour are at rest in the same enclosure, and no other volatile substance is present, the condition must lie on the saturation line. At a pressure below the triple point pressure, the solid can change directly to a gas (sublimation) and the gas can change directly to a solid, as in the formation of carbon dioxide snow from the released gas. The liquid zone to the left of the boiling point line is subcooled liquid. The gas under this line is superheated gas

In actual refrigeration cycles, the temperature of the heat sink will be several degrees lower than the condensing temperature to facilitate heat transfer. Hence it is possible to cool the refrigerant liquid in the condenser to a few degrees lower than the condensing temperature by adding extra area for heat transfer. In such a case, the exit condition of the condenser will be in the subcooled liquid region.

Similarly, the temperature of heat source will be a few degrees higher than the evaporator temperature, hence the vapour at the exit of the evaporator can be superheated by a few degrees. If the superheating of refrigerant takes place due to heat transfer with the refrigerated space (low temperature heat source) then it is called as useful superheating as it increases the refrigeration effect. On the other hand, it is possible for the refrigerant vapour to become superheated by exchanging heat with the surroundings as it flows through the connecting pipelines. Such a superheating is called as useless superheating as it does not increase refrigeration effect.

Subcooling is beneficial as it increases the refrigeration effect by reducing the throttling loss at no additional specific work input. Also subcooling ensures that only liquid enters into the throttling device leading to its efficient operation. Below Figure shows the VCRS cycle without and with subcooling on P-h and T-s coordinates



Comparison between a VCRS cycle without and with subcooling (a) on P-h diagram (b) on T-s diagram

A minimum amount of superheat is desirable as it prevents the entry of liquid droplets into the compressor Whether the volumic refrigeration effect (ratio of refrigeration effect by specific volume at compressor inlet) and COP increase or not depends upon the relative increase in refrigeration effect and work of compression, which in turn depends upon the nature of the refrigerant used. The temperature of refrigerant at the exit of the compressor increases with superheat as the isentropes in the vapour region gradually diverge.

MULTIPRESSURE SYSTEMS

A single stage vapour 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.

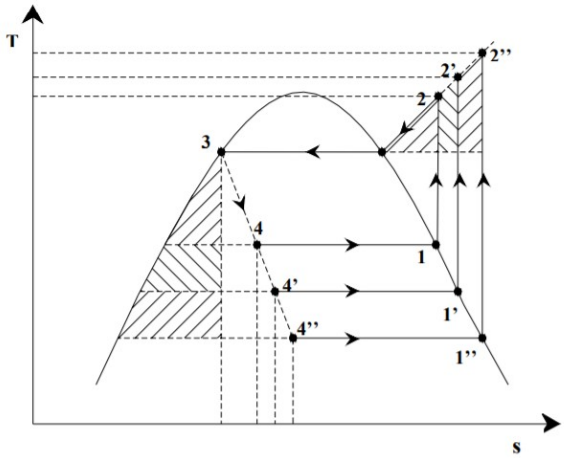
However, there are many applications where the temperature lift can be quite high. The temperature lift can become large either due to the requirement of very low evaporator temperatures and/or due to the requirement of very high condensing temperatures. For example, in frozen food industries the required evaporator can be as low as –40ºC, while in chemical industries temperatures as low as –150ºC 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. For example, the following figures shows the effect of decreasing evaporator temperatures on T s and P h diagrams. It can be seen from the T s diagrams that for a given condenser temperature, as evaporator temperature decreases:

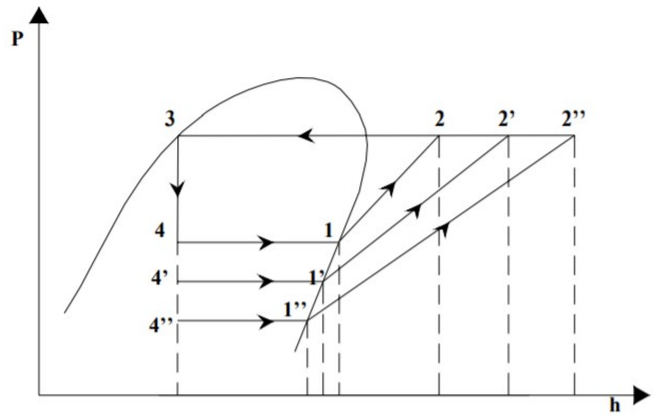
i. Throttling losses increase
ii. Superheat losses increase
iii. Compressor discharge temperature increases
iv. Quality of the vapour at the inlet to the evaporator increases
v. Specific volume at the inlet to the compressor increases

As a result of this, the refrigeration effect decreases and work of compression increases as shown in the P h diagram. The volumic refrigeration effect also decreases rapidly as the specific volume increases with decreasing evaporator temperature. Similar effects will occur, though not in the same proportion when the condenser temperature increases for a given evaporator temperature. 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° C. A two-stage system is used up to –60° C and a three-stage system is used for temperatures below –60°C. Apart from high temperature lift applications, multi-stage systems are also used in applications requiring refrigeration at different temperatures. For example, in a dairy plant refrigeration may be required at –30° C for making ice cream and at 2° C for chilling milk. In such cases it may be advantageous to use a multi-evaporator system with the low temperature evaporator operating at –30° C and the high temperature evaporator operating at 2° C



Effect of evaporator temperature on cycle performance (T-s diagram)



A

Effect of evaporator temperature on cycle performance (P-h diagram)

Multi-stage system is a refrigeration system with two or more low-side pressures. Multistage systems can be classified into:

a) Multi-compression systems

b) Multi-evaporator systems

c) Cascade systems, etc.

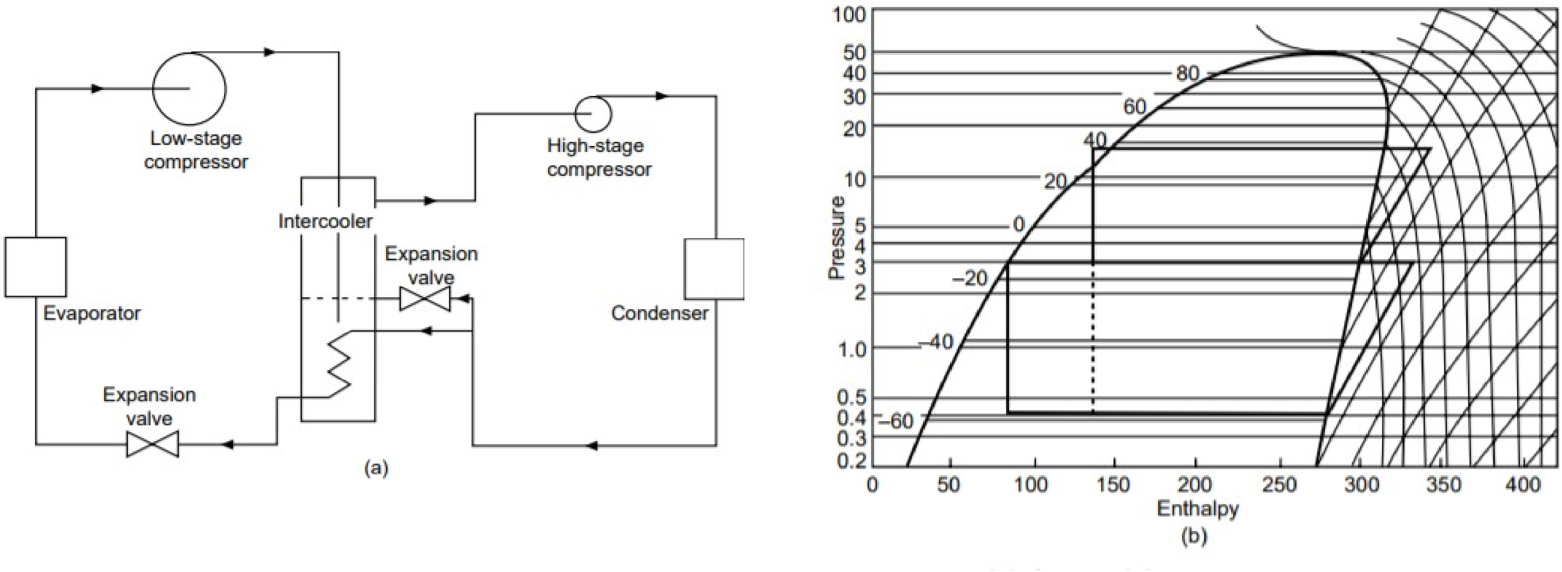
Two concepts which are normally integral to multi-pressure systems are,

i) Flash gas removal and ii) Intercooling.

MULTISTAGE CYCLES

Where the ratio of suction to discharge pressure is high enough to cause a serious drop in volumetric efficiency or an unacceptably high discharge temperature, vapour compression must be carried out in two or more stages.

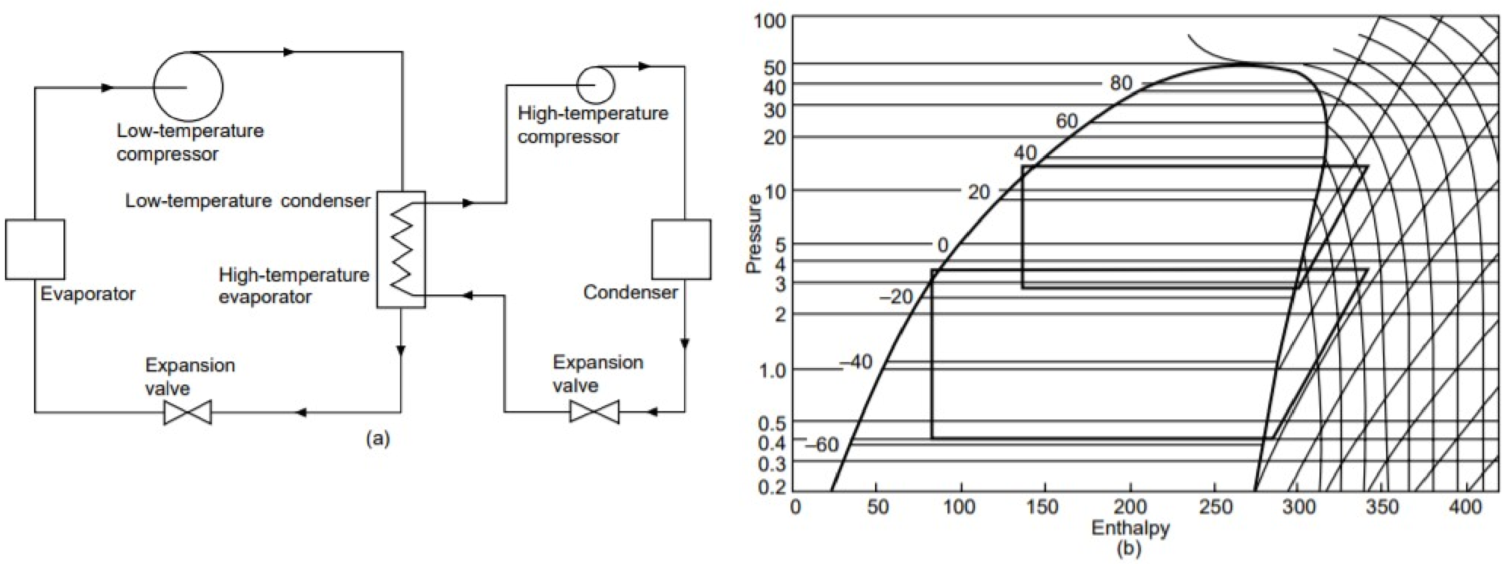
Two basic systems are in use. Compound systems use the same refrigerant throughout a common circuit, compressing in two or more stages. Discharge gas from the first compression stage will be too hot to pass directly to the high-stage compressor, so it is cooled in an intercooler, using some of the available refrigerant from the condenser. The opportunity is also taken to subcool liquid passing to the evaporator. Small compound systems may cool the inter-stage gas by direct injection of liquid refrigerant into the pipe.



Compound cycle: a) Cycle b) Mollier diagram (compound)

The cascade cycle has two separate refrigeration systems, one acting as a condenser to the other. This arrangement permits the use of different refrigerants in the two systems, and high pressure refrigerants such as R.13 are common in the lower stage. The Mollier

diagrams for compound and cascade systems indicate the enthalpy change per kilogram of circulated refrigerant, but it should be borne in mind that the mass flows are different for the low and high stages.



Cascade cycle: a) Circuit b) Mollier Diagram

COMPRESSORS

The purpose of the compressor in the vapour compression cycle is to accept the low-pressure dry gas from the evaporator and raise its pressure to that of the condenser. Compressors may be of the

- Positive Displacement or
- Dynamic type(or Roto – Dynamic type)

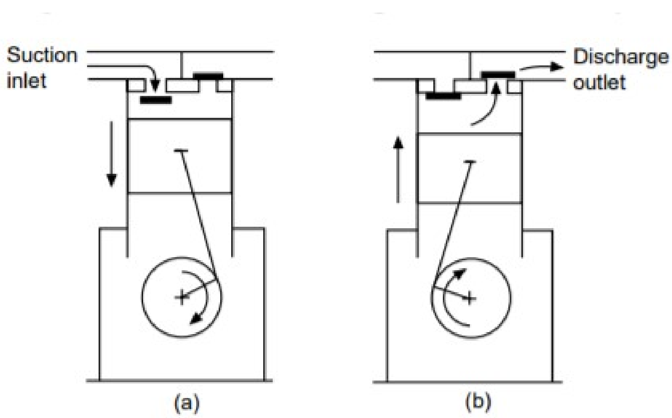
Depending upon the construction, positive displacement type compressors used in refrigeration and air conditioning can be classified into:

i. Reciprocating type
ii. Rotary type with sliding vanes (rolling piston type or multiple vane type)
iii. Rotary screw type (single screw or twin-screw type)
iv. Orbital compressors, and
v. Acoustic compressors

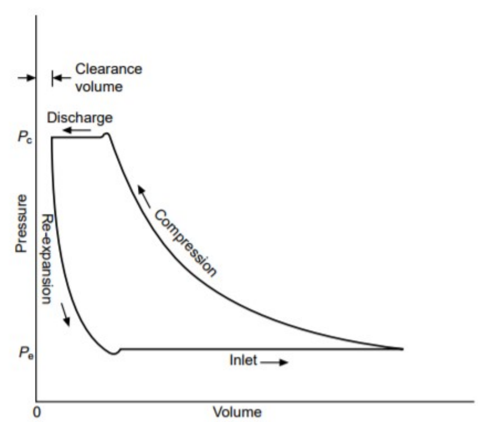
Depending upon the construction, roto-dynamic type compressors can be classified into:

i. Radial flow type, or ii. Axial flow type

The general form of positive displacement compressor is the piston type, being adaptable in size, number of cylinders, speed and method of drive. It works on the two-stroke cycle. As the piston descends on the suction stroke, the internal pressure falls until it is lower than that in the suction inlet pipe, and the suction valve opens to admit gas from the evaporator. At the bottom of the stroke, this valve closes again and the compression stroke begins. When the cylinder pressure is higher than that in the discharge pipe, the discharge valve opens and the compressed gas passes to the condenser. Clearance gas left at the top of the stroke must re-expand before a fresh charge can enter the cylinder



Reciprocating compressor. (a) Suction stroke.(b) Discharge stroke



Reciprocating compressor, indicator diagram

The first commercial piston compressors were built in the middle of the last century, and evolved from the steam engines which provided the prime mover. Construction at first was double acting, but there was difficulty in maintaining gas-tightness at the piston rod, so the design evolved further into a single-acting machine with the crankcase at suction inlet pressure, leaving only the rotating shaft as a possible source of leakage, and this was sealed with a packed gland

MULTICYLINDER COMPRESSORS

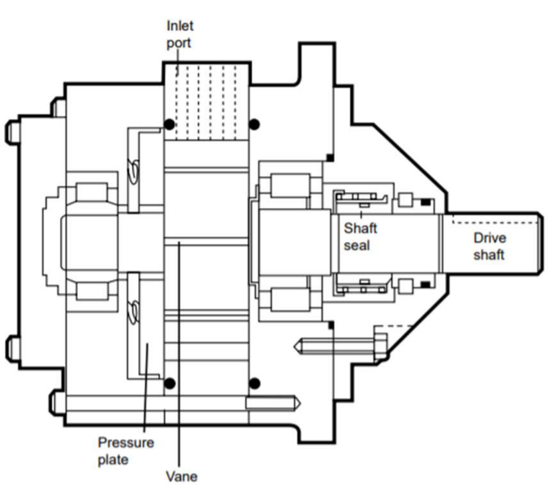
In the first century of development, compressors for higher capacity were made larger, having cylinder bores up to 375 mm, and running at speeds up to 400 rev/min. The resulting component parts were heavy and cumbersome. To take advantage of larger-scale production methods and provide interchangeability of parts, modern compressors tend to be multicylinder, with bores not larger than 175 mm and running at higher shaft speeds. Machines of four, six and eight cylinders are common. These are arranged in a multibank configuration with two, three or four connecting rods on the same throw of the crankshaft to give a short, rigid machine This construction gives a large number of common parts – pistons, connecting rods, loose liners and valves – through a range of compressors, and such parts can be replaced if worn or damaged without removing the compressor body from its installation. Compressors for small systems will be simpler, of two, three or four cylinders

SLIDING AND ROTARY VANE COMPRESSORS

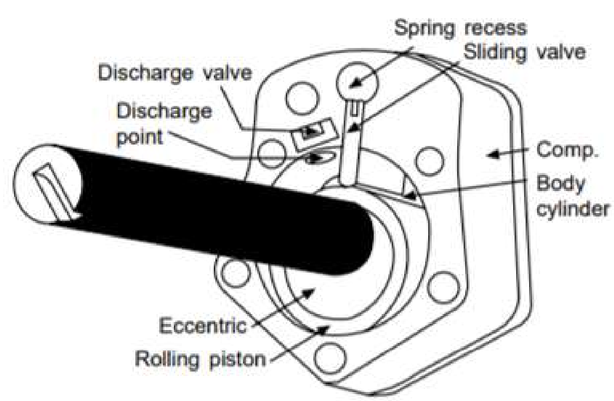
The volumes between an eccentric rotor and sliding vanes will vary with angular position, to provide a form of positive displacement compressor. Larger models have eight or more blades and do not require inlet or outlet valves. The blades are held in close contact with the outer shell by centrifugal force, and sealing is improved by the injection of lubricating oil along the length of the blades. Rotating vane machines have no clearance volume and can work at high pressure ratios.

Larger rotating vane compressors are limited in application by the stresses set up by the thrust on the tips of the blades, and are used at low discharge pressures such as the first stage of a compound cycle. Smaller compressors, up to 110 kW cooling capacity, are now available for the full range of working pressures. These also incorporate a spring-loaded safety plate to relieve excess pressure if liquid refrigerant enters

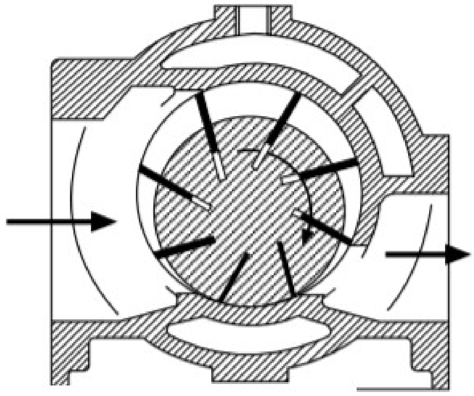
Sliding vane or rolling piston compressors have one or two blades, which do not rotate, but are held by springs against an eccentric rotating roller. These compressors require discharge valves. This type has been developed extensively for domestic appliances, packaged air-conditioners and similar applications, up to a cooling duty of 15 kW



Rotary vane compressor



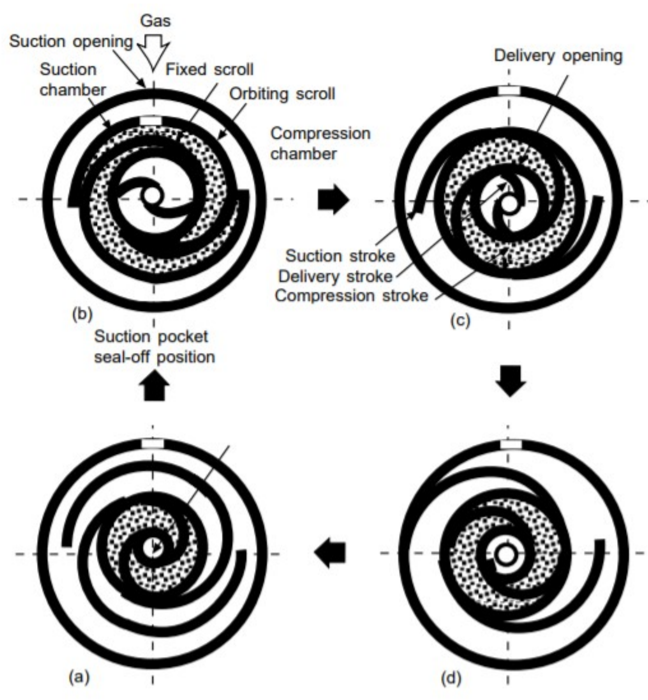
Rolling piston compressor



Rotary vane compressor

SCROLL COMPRESSOR

A positive displacement gas compressor can be constructed with a pair of nesting volutes, one stationary and one orbiting. Gas enters from the surrounding enclosure, is trapped between the volutes and moved inwards (a, b, c, d etc.) until it is finally forced out through the central discharge port. Owing to the close manufacturing tolerances the scroll compressor is built only in hermetic enclosed models. The dynamic and gas pressure loads are balanced so that it is free of vibration. It is currently available in cooling capacities up to 60 kW, and is being made in larger sizes as development proceeds. Capacity control of these compressors is achieved by varying the compressor speed by means of an inverter motor.



Scroll compressor

DYNAMIC COMPRESSORS

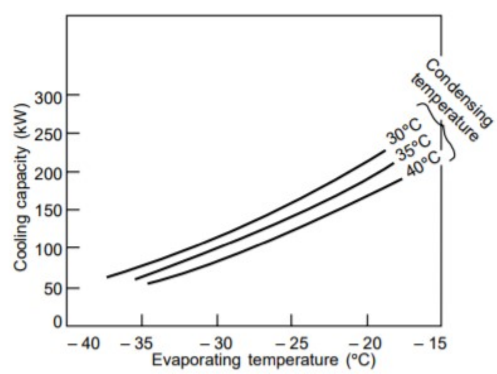
Dynamic compressors impart energy to the gas by velocity or centrifugal force and then convert this to pressure energy. The most common type is the centrifugal compressor. Suction gas enters axially into the eye of a rotor which has curved blades, and is thrown out tangentially from the blade circumference. The energy given to gas passing through such a machine depends on the velocity and density of the gas. Since the density is already fixed by the working conditions, the design performance of a centrifugal compressor will be decided by the rotor tip speed.

Owing to the low density of gases used, tip speeds up to 300 m/s are common. At an electric motor speed of 2900 rev/min, a single-stage machine would require an impeller 2 m in diameter. To reduce this to a more manageable size, drives are geared up from standard speed motors or the supply frequency is changed to get higher motor speeds.

The drive motor is integral with the compressor assembly, and may be of the open or hermetic type. On single-stage centrifugal compressors for air-conditioning duty, rotor speeds are usually about 10 000 rev/min. Gas may be compressed in two or more stages. The impellers are on the same shaft, giving a compact tandem arrangement with the gas from one stage passing directly to the next. The steps of compression are not very great and, if two-stage is used, the gas may pass from the first to the second without any intercooling of the gas. Centrifugal machines can be built for industrial use with ammonia and other refrigerants, and these may have up to seven compression stages. With the high tip speeds in use, it is not practical to build a small machine, and the smallest available centrifugal compressor for refrigeration duty has a capacity of some 260 kW.

Semi-hermetic compressors are made up to 7000 kW and open drive machines up to 21 000 kW capacity. Systems of this size require large-diameter refrigerant suction and discharge pipes to connect the components of the complete system. As a result, and apart from large-scale industrial plants, they are almost invariably built up as liquid-cooling, water-cooled packages with the condenser and evaporator complete as part of a factory built package. The main refrigerant for packaged water chillers of the centrifugal type are R123 and R134a. Since centrifugal machines are too big to control by frequent stopping and restarting, some form of capacity reduction must be inbuilt. The general method is to throttle or deflect the flow of suction gas into the impeller. With most models it is possible to reduce the pumping capacity down to 10–15% of full flow. There are no components which require lubrication, with the exception of the main bearings. As a result, the machine can run almost oil free.

The pumping characteristic of the centrifugal machine differs from the positive displacement compressor since, at excessively high discharge pressure, gas can slip backwards past the rotor. This characteristic makes the centrifugal compressor sensitive to the condensing condition, giving higher duty and a better coefficient of performance if the head pressure drops, while heavily penalizing performance if the head pressure rises



Compressor capacity ratings in graph form

CONDENSER

The purpose of the condenser in a vapour compression cycle is to accept the hot, high-pressure gas from the compressor and cool it to remove first the superheat and then the latent heat, so that the refrigerant will condense back to a liquid. In addition, the liquid is usually slightly subcooled. In nearly all cases, the cooling medium will be air or water.

HEAT TO BE REMOVED

The total heat to be removed in the condenser is shown in the p–h diagram and, apart from comparatively small heat losses and gains through the circuit, will be Heat taken in by evaporator + heat of compression This latter, again ignoring small heat gains and losses, will be the net shaft power into the compressor, giving Evaporator load + compressor power = condenser load Condenser rating is correctly stated as the rate of heat rejection. Some manufacturers give ratings in terms of the evaporator load, together with a 'de-rating' factor, which depends on the evaporating and condensing temperatures. Evaporator load × factor = condenser load

AIR-COOLED CONDENSERS

The simplest air-cooled condenser consists of a plain tube containing the refrigerant, placed in still air and relying on natural air circulation. An example is the condenser of the domestic refrigerator, which may also have some secondary surface in the form of supporting and spacer wires. Above this size, the flow of air over the condenser surface will be by forced convection, i.e. fans. The high thermal resistance of the boundary layer on the air side of the heat exchanger leads to the use, in all but the very smallest condensers, of an extended surface. This takes the form of plate fins mechanically bonded onto the refrigerant tubes in most commercial patterns. The ratio of outside to inside surface will be between 5 : 1 and 10 : 1. Flow of the liquefied refrigerant will be assisted by gravity, so the inlet will be at the top of the condenser and the outlet at the bottom. Rising pipes should be avoided in the design, and care is needed in installation to get the pipes level. The flow of air may be vertically upwards or horizontal, and the configuration of the condenser will follow from this Small cylindrical matrices are also used, the air flowing radially inwards and out through a fan at the top. Forced convection of the large volumes of air at low resistance leads to the general use of propeller or single-stage axial flow fans. Where a single fan would be too big, multiple smaller fans give the advantages of lower tip speed and noise, and flexibility of operation in winter.In residential areas slower-speed fans may be specified to reduce noise levels. A smaller air flow will de-rate the condenser, and manufacturers will give ratings for 'standard' and 'quiet' products. It will be recognized that the low specific heat capacity and high specific volume of air implies a large volume to remove the condenser heat. If the mass flow is reduced, the temperature rise must increase, raising the condensing temperature and pressure to give lower plant efficiency. In practice, the temperature rise of the air is kept between 9 and 12 K. The mass flow, assuming a rise of 10.5 K, is then

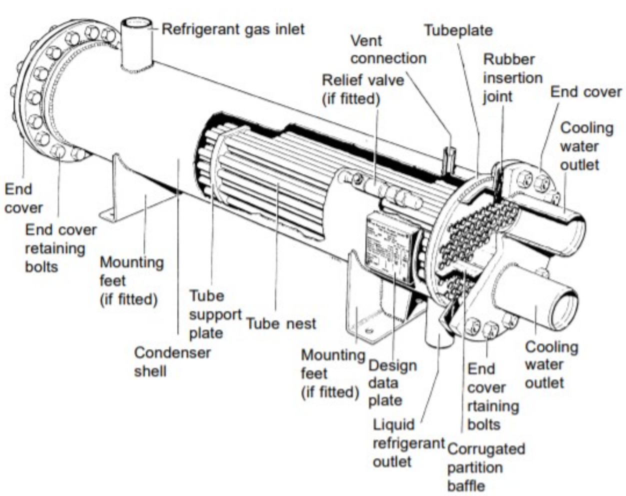
$$\frac{1}{10.5 \times 1.02} = 0.093 \text{ kg/(s kW)}$$

where 1.02 is the specific heat capacity of ambient air. As an example of these large air flows required, the condenser for an air-conditioning plant for a small office block, having a cooling capacity of 350 kW and rejecting 430 kW, would need 40.85 kg/s or about 36 m^3/s of air. This cooling air should be as cold as possible, so the condenser needs to be mounted where such a flow of fresh ambient air is available without recirculation. The large air flows needed, the power to move them, and the resulting noise levels are the factors limiting the use of air-cooled condensers. Materials of construction are aluminium fins on stainless steel tube for ammonia, or aluminium or copper fins on aluminium or copper tube for the halocarbons. Aluminium tube is not yet common, but its use is expected to increase. In view of the high material cost for air-cooled condensers compared with other types, a higher ln MTD is usually accepted, and condensing temperatures may be 5–8 K higher for a given cooling medium temperature. Air-cooled condensers must, of course, be used on land transport systems. They will also be used in desert areas where the supply of cooling water is unreliable.

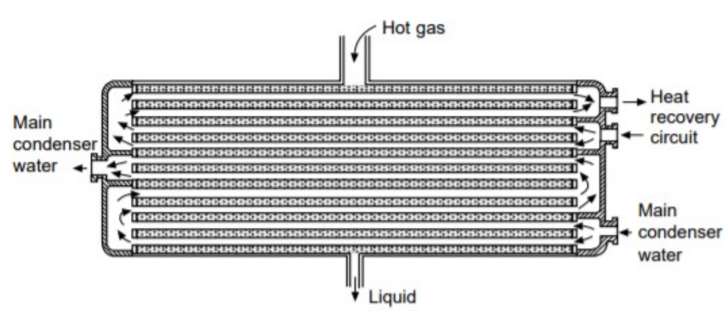
WATER-COOLED CONDENSERS

The higher heat capacity and density of water make it an ideal medium for condenser cooling and, by comparison with the 350 kW plant cited above, the flow is only 9.8 liter/s. Small water-cooled condensers may comprise two concentric pipes ('double pipe'), the refrigerant being in either the inner tube or the annulus. Configurations may be straight, with return bends or headers, or coiled. The double-pipe condenser is circuited in counter-flow (media flowing in opposite directions) to get the most subcooling, since the coldest water will meet the outgoing liquid refrigerant.

Larger sizes of water-cooled condenser require closer packing of the tubes to minimize the overall size, and the general form is shell and-tube, having the water in the tubes. This construction is a very adaptable mechanical design and is found in all sizes from 100 mm to 1.5 m diameter and in lengths from 600 mm to 6 m, the latter being the length of commercially available tubing. Materials can be selected for the application and refrigerant, but all mild steel is common for fresh water, with cupronickel or aluminium brass tubes for salt water. Some economy in size can be effected by extended surfaces on the refrigerant side, usually in the form of low integral fins formed on the tubes. On the water side, swirl strips can be fitted to promote turbulence, but these interfere with maintenance cleaning and are not much in favor. Water velocity within the tubes is of the order of 1 m/s, depending on the bore size. To maintain this velocity, baffles are arranged within the end covers to direct the water flow to a number of tubes in each 'pass'. Some condensers have two separate water circuits (double bundle), using the warmed water from one circuit as reclaimed heat in another part of the system. The main bundle rejects the unwanted heat. Where the mass flow of water is unlimited (sea, lake, river or cooling tower), the temperature rise through the condenser may be kept as low as 5 K, since this will reduce the ln MTD with a lowering of head pressure at the cost only of larger water pumps and pipes.



Shell-and-tube condenser



Double-bundle shell-and-tube condenser

COOLING TOWERS

In a cooling tower, cooling of the main mass of water is obtained by the evaporation of a small proportion into the airstream. Cooled water leaving the tower will be 3–8 K warmer than the incoming air wet bulb temperature. The quantity of water evaporated will take up its latent heat equal to the condenser duty, at the rate of about 2430 kJ/kg evaporated, and will be approximately $\frac{1}{2430} = 0.41 \times 10^{-3}$ kg/(s kW)

Cooled water from the drain tank is taken by the pump and passed through the condenser, which may be built up with the compressor as part of a compressor–condenser package (condensing unit). The warmed water then passes back to sprays or distribution troughs at the top of the tower and falls in the up going airstream, passing over packings which present a large surface to the air. Evaporation takes place, the vapour obtaining its latent heat from the body of the water, which is therefore cooled

EVAPORATIVE CONDENSERS

This cooling effect of the evaporation of water can be applied directly to the condenser refrigerant pipes in the evaporative condenser. The mass flow of water over the condenser tubes must be enough to ensure wetting of the tube surface, and will be of the order of 80–160 times the quantity evaporated. The mass flow of air must be sufficient to carry away the water vapour formed, and a compromise must be reached with expected variations in ambient conditions. An average figure is 0.06 kg/(s kW).

The atmospheric condenser is a simplified form of evaporative condenser, having plain tubes over a collecting tank and relying only on natural air draught. This will be located on an open roof or large open space to ensure a good flow of air. The space required is of the order of 0.2 m^2 /kW, and such condensers are not much used because of this large space requirement. Atmospheric condensers can still be seen on the roofs of old breweries. They are in current use where space is plentiful.

EVAPORATOR

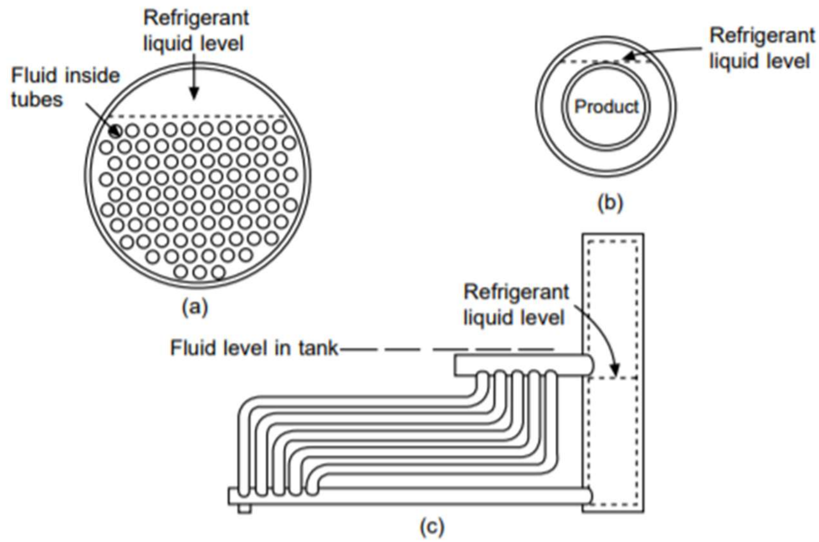
The purpose of the evaporator is to receive low-pressure, low temperature fluid from the expansion valve and to bring it in close thermal contact with the load. The refrigerant takes up its latent heat from the load and leaves the evaporator as a dry gas. Evaporators are classified according to their refrigerant flow pattern and their function.

FLOW PATTERN AND FUNCTION

The refrigerant flow pattern is dependent on the method of ensuring oil removal from the evaporator and, possibly, its return to the crankcase. Flooded evaporators have a body of fluid boiling in a random manner, the vapour leaving at the top. In the case of ammonia, any oil present will fall to the bottom and be drawn off from the drain pot or oil drain connection. With the halocarbons, a proportion of the fluid is bled off and rectified. Evaporators which keep the oil moving by means of continuous fluid velocity, until it gets back to the compressor suction, are termed dry expansion. In these, the refrigerant is totally evaporated. The function of the evaporator will be to cool gas, liquid or other product load. In most cases air or a liquid is first cooled, and this is then used to cool the load. For example, in a cold room air is cooled and this air cools the stored produce and carries away heat leaking through the structure; in a water chiller system, the water is circulated to cool the load, etc.

AIR COOLING EVAPORATORS

Air cooling evaporators for cold rooms, blast freezers, air-conditioning



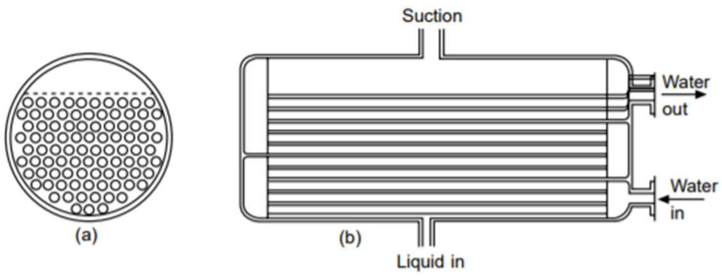
Flooded evaporators. (a) Shell-and-tube. (b) Jacketted. (c) Raceway

etc., will have finned pipe coils. In all but very small coolers, there will be fans to blow the air over the coil. Construction materials will be the same as for air-cooled condensers. Aluminum fins on copper tube are the most common for the halocarbons, with stainless steel or aluminum tube for ammonia. Frost or condensed water will form on the fin surface and must be drained away. To permit this, fins will be vertical and the air flow horizontal, with a drain tray provided under. The size of the tube will be such that the velocity of the boiling fluid within it will cause turbulence to promote heat transfer. Tube diameters will vary from 9 mm to 32 mm, according to the size of coil. Fin spacing will be a compromise between compactness (and cost) and the tendency for the inter fin spaces to block with condensed moisture or frost. Spacing will vary from 2 mm on a compact air conditioner to 12 mm on a low-temperature cold room coil [8].

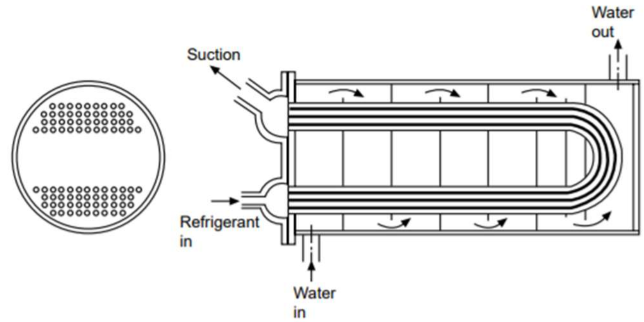
LIQUID COOLING EVAPORATORS

Liquid cooling is mostly in shell-and-tube or shell-and-coil evaporators. In the shell-and-tube type, the liquid is usually in the pipes and the shell is some three-quarters full of the liquid, boiling refrigerant. A number of tubes is omitted at the top of the shell to give space for the suction gas to escape clear of the surface without entraining liquid. Further features such as multiple outlet headers, suction trap domes and baffles will help to avoid liquid droplets entering the main suction pipe. Gas velocities should not exceed 3 m/s and lower figures are used by some designers.

Operated in this manner, the shell-and-tube type is a flooded evaporator and has oil drainage pots if using ammonia, or a mixture bleed system if the refrigerant is one of the halocarbons. The speed of the liquid within the tubes should be about 1 m/s or more, to promote internal turbulence for good heat transfer. End cover baffles will constrain the flow to a number of passes, as with the shell-and-tube condenser



Shell-and-tube evaporator, flooded



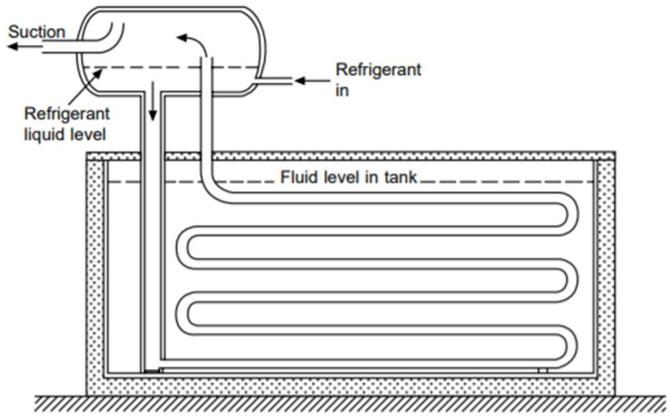
Shell-and-coil evaporator

Evaporators of this general type with dry expansion circuits will have the refrigerant within the tubes, in order to maintain a suitable continuous velocity for oil transport, and the liquid in the shell. These can be made as shell-and-tube, with the refrigerant constrained to a number of passes, or may be shell-and-coil. In both these configurations, baffles are needed on the water side to improve the turbulence, and the tubes may be finned on the outside. Internal swirl strips or wires will help to keep liquid refrigerant in contact with the tube wall.

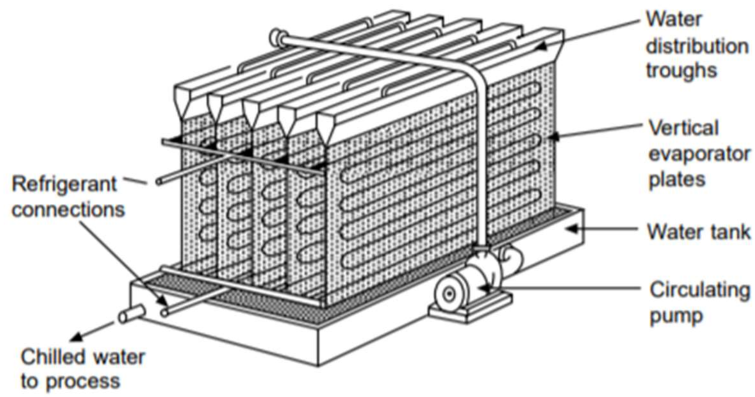
Liquid cooling evaporators may comprise a pipe coil in an open tank, and can have flooded or dry expansion circuitry. Flooded coils will be connected to a combined liquid accumulator and suction separator (usually termed the surge drum), in the form of a horizontal or vertical drum. The expansion valve maintains a liquid level in this drum and a natural circulation is set up by the bubbles escaping from the liquid refrigerant at the heat exchanger surface. Dry expansion coils for immersion in an open tank will be in a continuous circuit or a number of parallel circuits Liquid velocity over such coils can be increased by tank baffles and there may be special purpose agitators, as in an ice making tank. Coils within an open tank can be allowed to collect a layer of ice during off-load periods, thus providing thermal storage and giving a reserve of cooling capacity at peak load times

Another type comprises a bank of corrugated plates, forming alternative paths for refrigerant and liquid, similar to that shown in, of brazed or welded construction. Where water is to be cooled close to its freezing point without risk of damage to the evaporator, the latter is commonly arranged above the water-collection tank and a thin film of water runs over the tubes. Heat transfer is very high with a thin moving film of liquid and, if any ice forms, it will be on the outside, free to expand, and it will not damage the tube. Such an evaporator is termed a Baudelot cooler. It may be open, enclosed in dust-tight shields to avoid contamination of the product (as in surface milk and cream coolers), or may be enclosed in a pressure vessel as in the Mojonniér cooler for soft drinks, which pressurizes with carbon dioxide at the same time

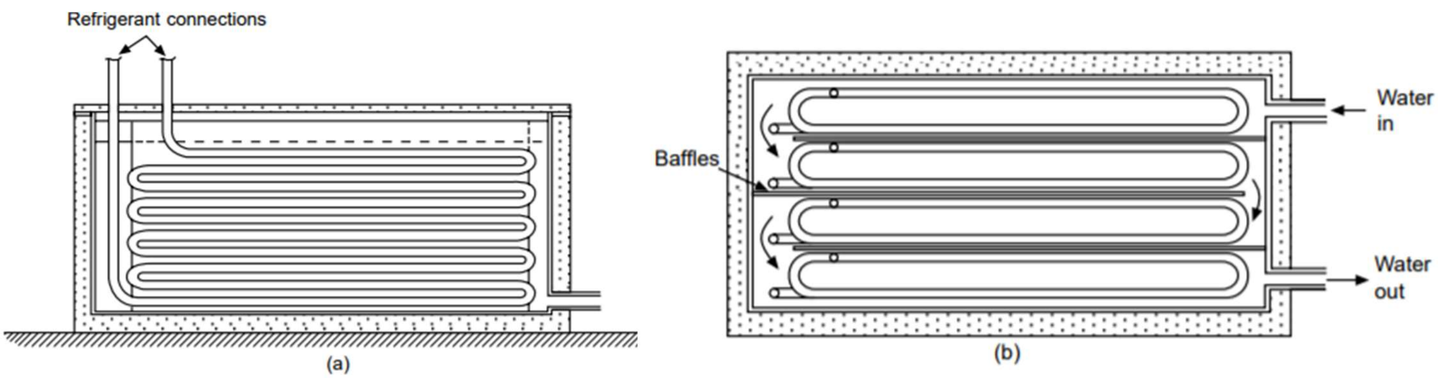
Some liquids, such as vegetable fats and ice-cream mixes, increase considerably in viscosity as they are cooled, sticking to the heat exchanger surface. Evaporators for this duty are arranged in the form of a hollow drum surrounded by the refrigerant and having internal rotating blades which scrape the product off as it thickens, presenting a clean surface to the flow of product and impelling the cold paste towards the outlet.



Flooded tank evaporator



Baudelot cooler



Dry Expansion tank evaporator a) Section b) Elevation

PLATE EVAPORATORS

Plate evaporators are formed by cladding a tubular coil with sheet metal, welding together two embossed plates, or from aluminum extrusions. The extended flat face may be used for air cooling, for liquid cooling if immersed in a tank, or as a Baudelot cooler. The major use for flat plate evaporators is to cool a solid product by conduction, the product being formed in rectangular packages and held close between a pair of adjacent plates. In the horizontal plate freezer, the plates are arranged in a stack on slides, so that the intermediate spaces can be opened and closed. Trays, boxes or cartons of the product are loaded between the plates and the stack is closed to give good contact on both sides. When the necessary cooling is complete, the plates are opened and the product removed. The vertical plate freezer is used to form solid blocks of a wet product, typically fish. When frozen solid, the surfaces are thawed and the blocks pushed up and out of the bank. To ensure good heat transfer on the inner surface of the plates and achieve a high rate of usage, liquid refrigerant is circulated by a pump at a rate 5–12 times the rate of evaporation. If a plate evaporator is partially filled with brine this can be frozen down while the plate is on light load, and the reserve of cooling capacity used at other times. The freezing point of the brine can be formulated according to the particular application and the plate can be made as thick as may be required for the thermal storage needed. The major application of this device is the cooling of vehicles. The plates are frozen down at night, or other times when the vehicle is not in use, and the frozen brine keeps the surface of the plate cold while the vehicle is on the road. The refrigeration machinery may be on the vehicle or static.

DEFROSTING

Air cooling evaporators working below 0°C will accumulate frost which must be removed periodically, since it will obstruct heat transfer.

Evaporators of suitable and robust construction can be defrosted by brushing, scraping or chipping, but these methods are labor intensive and may lead to damage of the plant.

Where the surrounding air is always at + 4°C or higher, it will be sufficient to stop the refrigerant for a period and allow the frost to melt off (as in the auto-defrost domestic refrigerator). This method can be used for cold rooms, packaged air-conditioners etc., where the service period can be interrupted. For lower temperatures, heat must be applied to melt the frost within a reasonable time and ensure that it drains away. Methods used are as follows:

1. Electric resistance heaters. Elements are within the coil or directly under it.

2. Hot gas. A branch pipe from the compressor discharge feeds superheated gas to the coil. The compressor must still be working on another evaporator to make hot gas available. Heat storage capsules can be built into the circuit to provide a limited reserve of heat for a small installation.

3. Reverse cycle. The direction of flow of the refrigerant is reversed to make the evaporator act as a condenser. Heat storage or another evaporator are needed as a heat source.

In each of these cases, arrangements must be made to remove cold refrigerant from the coil while defrosting is in progress. Drip trays and drain pipes may require supplementary heating

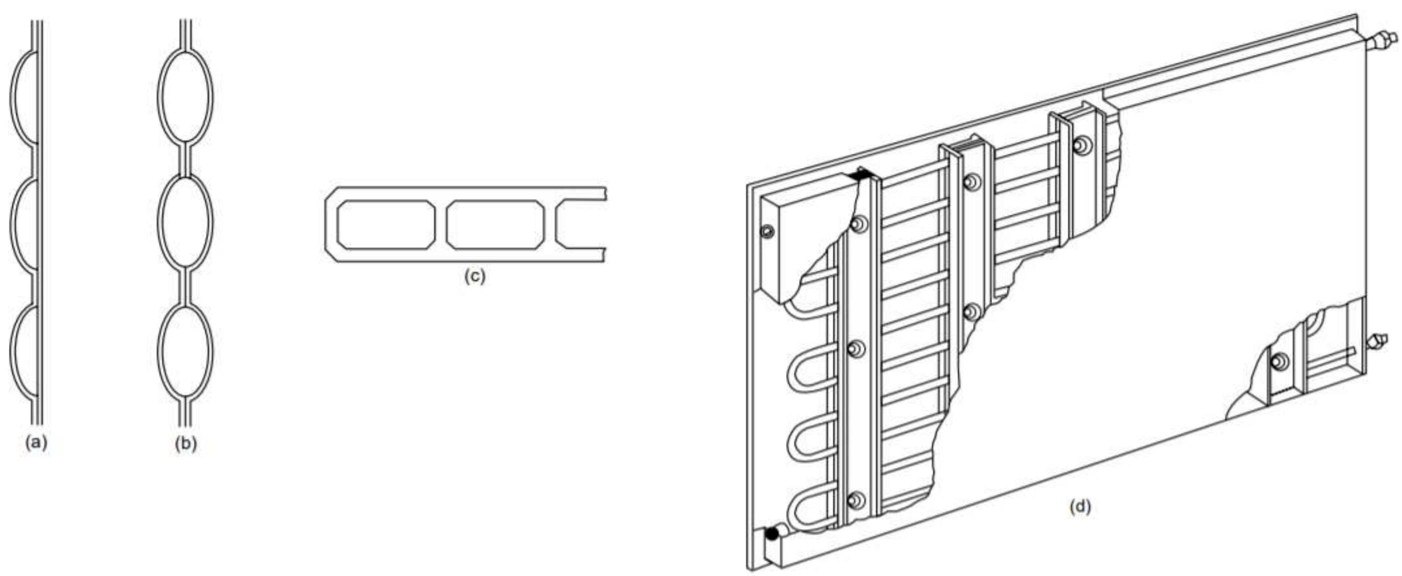


Plate Evaporator a) Single Embossed b) Double Embossed c) Extruded d) Hold-over (brine-filled)

EXPANSION VALVES

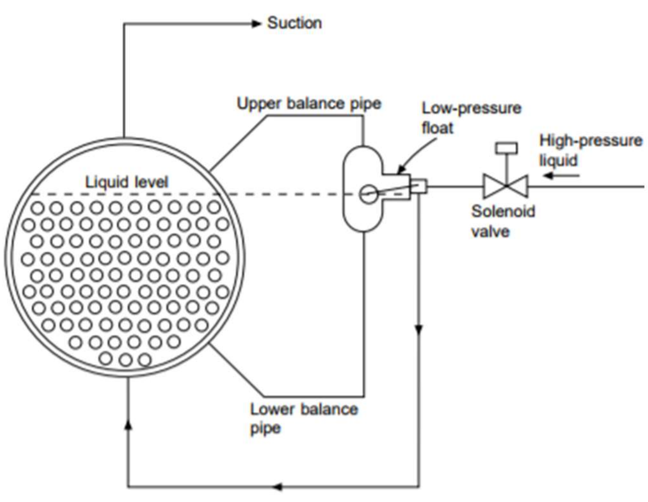
The purpose of the expansion valve is to control the flow of refrigerant from the high-pressure condensing side of the system into the low pressure evaporator. In most cases, the pressure reduction is achieved through a variable flow orifice, either modulating or two-position. Expansion valves may be classified according to the method of control.

LOW-PRESSURE FLOAT VALVES

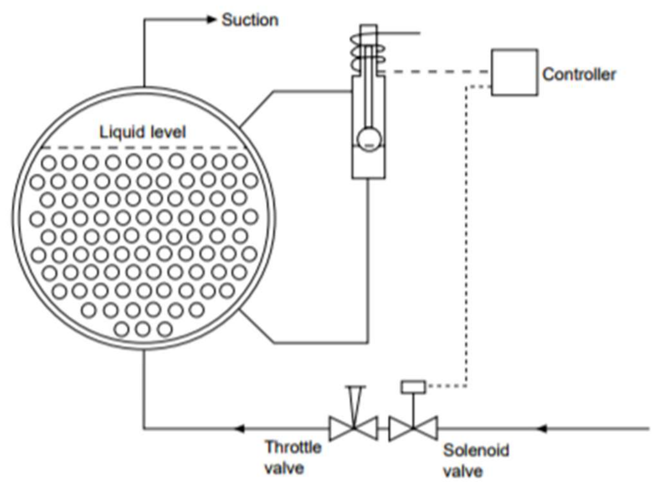
Flooded evaporators require a constant liquid level, so that the tubes remain wetted. A simple float valve suffices, but must be located with the float outside the evaporator shell, since the surface of the boiling liquid is agitated and the constant movement would cause excessive wear in the mechanism. The float is therefore contained within a separate chamber, coupled with balance lines to the shell. Such a valve is a metering device and may not provide positive shut-off when the compressor is stopped. Under these circumstances, refrigerant will continue to leak into the evaporator until pressures have equalized, and the liquid level might rise too close to the suction outlet. To provide this shut-off, a solenoid valve is needed in the liquid line.

LOW-PRESSURE FLOAT SWITCHES

Since the low-pressure float needs a solenoid valve for tight closure, this valve can be used as an on–off control in conjunction with a pre-set orifice and controlled by a float switch. The commonest form of level detector is a metallic float carrying an iron core which rises and falls within a sealing sleeve. An induction coil surrounds the sleeve and is used to detect the position of the core. The resulting signal is amplified to switch the solenoid valve, and can be adjusted for level and sensitivity. A throttle valve is fitted to provide the pressure-reducing device. Should a float control fail, the level in the shell may rise and liquid pass into the compressor suction. To warn of this, a second float switch is usually fitted at a higher level, to operate an alarm and cut-out. Where a flooded coil is located in a liquid tank, the refrigerant level will be within the tank, making it difficult to position the level control. In such cases, a gas trap or siphon can be formed in the lower balance pipe to give an indirect level in the float chamber. Siphons or traps can also be arranged to contain a non-volatile fluid such as oil, so that the balance pipes remain free from frost.



Low-pressure float valve on flooded cooler

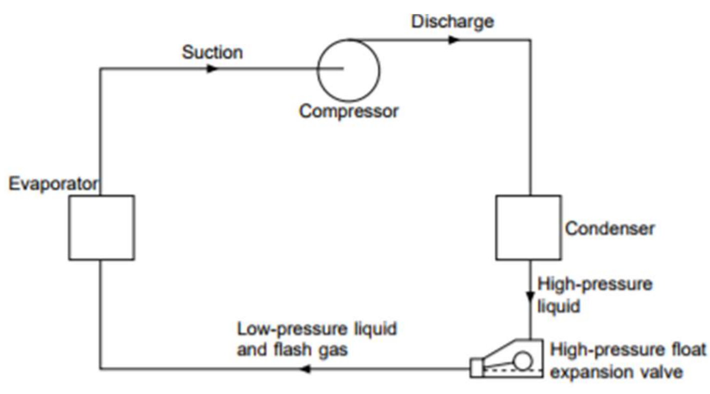


Low-pressure float switch

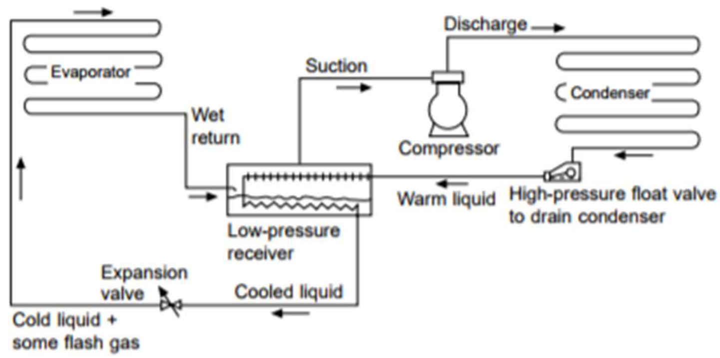
HIGH-PRESSURE FLOAT VALVE

On a single-evaporator flooded system, a float valve can be fitted which will pass any drained liquid from the condenser direct to the evaporator. The action is the same as that of a steam trap. The float chamber is at condenser pressure and the control is termed a high pressure float

The refrigerant charge of such a system is critical, since it must not exceed the working capacity of the evaporator. It is not possible to have a receiver in circuit and this control cannot feed more than one evaporator, since it cannot detect the needs of either. The difficulty of the critical charge can be overcome by allowing any surplus liquid refrigerant leaving the evaporator to spill over into a receiver or accumulator in the suction line, and boiling this off with the warm liquid leaving the condenser. In this system, the low-pressure receiver circuit, liquid is drained from the condenser through the high-pressure float, but the final step of pressure drop takes place in a secondary expansion valve after the warm liquid has passed through coils within the receiver. In this way, heat is available to boil off surplus liquid leaving the evaporator. Two heat exchangers carry the warm liquid from the



High-pressure float valve



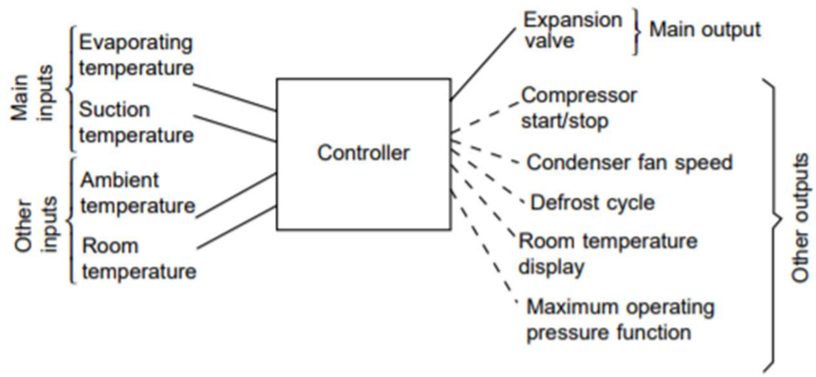
Low-pressure receiver circuit

condenser within this vessel. The first coil is in the upper part of the receiver, and provides enough superheat to ensure that gas enters the compressor in a dry condition. The lower coil boils off surplus liquid leaving the evaporator itself. With this method of refrigerant feed, the evaporator has a better internal wetted surface, with an improvement in heat transfer.

The low-pressure receiver system can be adapted to compound compression and can be fitted with hot gas defrost by reverse gas flow. In both circuits the low-pressure receiver provides the safety vessel to prevent liquid entering the compressor. Providing the highpressure float is correctly sized, this system can operate at low condenser pressures, saving compressor energy in cool weather. Where the halocarbon refrigerants are used in this system, an oil distillation device is fitted, working on the same principle

ELECTRONIC EXPANSION VALVE

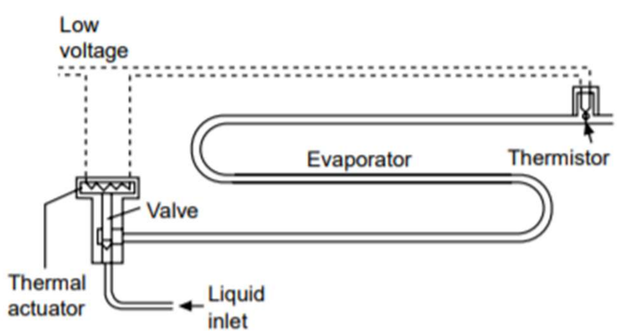
Evaporator superheat can be sensed by two thermistors, one on the main pipes of the evaporator and the other on the suction outlet, and the signal used to control refrigerant flow. The final control element is a pulsing or modulating solenoid valve. The controller can also accept other signals, such as load temperature, discharge temperature, condensing pressure and motor current, and use these to provide optimum coil effectiveness for minimum input power. The electronic expansion valve has been fitted for some years onto factory-built packages but is now available for field installations, and its use will become more general. The extent of its future success will depend on the ability of installation mechanics to set the controller correctly. Electronic expansion valves are now widely used on small automatic systems, mainly as the refrigerant flow control device (evaporating or condensing) in an integrated control circuit



Electronic expansion valve

THERMAL ELECTRIC EXPANSION VALVE

The signal from a suitable thermistor placed at the evaporator outlet will vary, depending on whether it senses dry refrigerant gas or traces of liquid. This can be used directly to control the current through a thermal element to modulate the expansion valve. This device usually has no separate adjustable controller and so cannot be incorrectly set



Thermal electric expansion valve