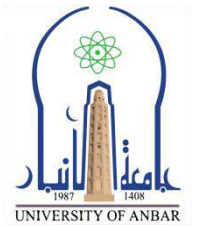


University of Anbar

College of Engineering

Mechanical Engineering Department



Subject: Refrigeration

Class: Fourth Year

Course Tutor: Assist.Prof.Dr.Obaid T.Fadhil

Introduction

Air conditioning deals with artificial tampering of the conditions of air that may involve cooling as well as heating coupled with ventilation, filtration, and air circulation. However, air conditioning is generally perceived as the process relating primarily to cooling of air. Control of environment through supply of heat, however, is generally treated as a process of heating.

Refrigeration, on the other hand, deals solely with cooling and one of its most important applications is air conditioning. Thus refrigeration and air conditioning are very closely interrelated as highlighted in Figure 1.1. For the same reason, very often the two subjects are treated in the same book. In the present book the discussion on air conditioning will focus more on providing a cool environment; the application of air conditioning to heating will be limited to that provided by a heat pump.

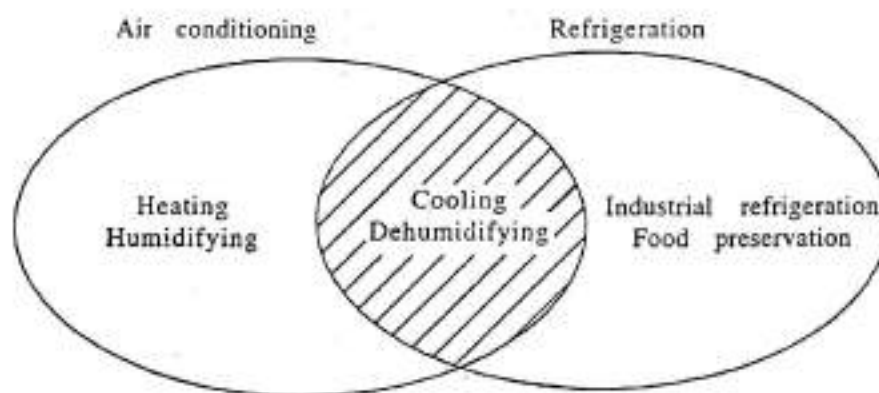


FIGURE 1.1 Relationship of refrigeration and air conditioning.

Refrigeration has played an important role in the growth and attainment of the present-day standard of living. Because of refrigeration, bulk of the perishable foodstuffs wasted hitherto can

not only be preserved but also marketed to far flung corners of the world. Its application to air-conditioning enabled taming the natural climate into creating comfortable environment even in blistering summer. There has also been dramatic and continuous growth in the air-conditioning industry since the turn of the century. Air conditioning is no longer considered a luxury. The benefits of air conditioning are well established so much so that today it has become a necessity and a tool for higher productivity.

1.1 HISTORY OF REFRIGERATION

Refrigeration means the artificial withdrawal of heat, producing in a substance or within a space a temperature lower than that which would exist under the natural influence of the surroundings. Cooling effect created by a machine or mechanical device is classified as mechanical refrigeration.

Since prehistoric times, artificial cooling has been recognized as desirable: food was kept in cold air in caves and wells to keep it fresh for longer periods.

Two physical phenomena were used in most remote times—without much understanding of the principles involved—*evaporation of water*, especially through vases of porous pottery (Figure 1.2) widely used in Egypt, India and China, and *terrestrial radiation* towards clear sky during the night. It is known that several centuries before the birth of Christ, Egyptians made ice by this means by putting porous earthen pots on the roof of the house during the night. Evaporation of water in cool dry air together with radiative heat transfer during a clear night caused ice formation even when the ambient temperature was above the freezing temperature.

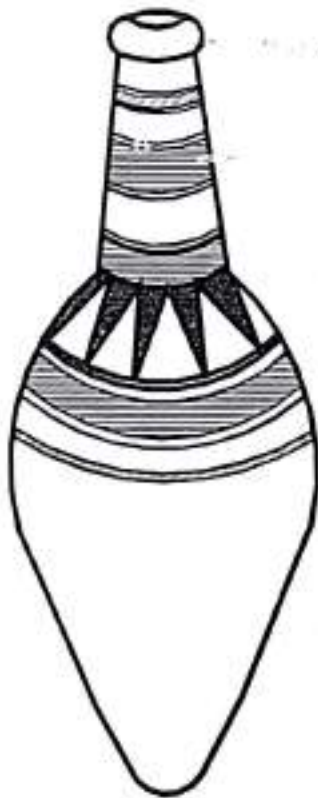


FIGURE 1.2 Earthen containers used by Egyptians for cooling water and making ice.

The first attempt to produce artificial cooling was made in 1755 when Cullen built an apparatus to make ice artificially by vaporization of water at reduced pressure. This experiment was followed by others over the years. A phenomenon called Peltier effect was discovered in 1834 which was commercialized after more than a century and culminated in the development of thermo-electric refrigerators. Refrigeration by this method is based on the fact that when an electric current is passed around a circuit composed of two different metals one junction becomes cool and the other becomes warm. Production of cold by mechanical methods is a relatively recent development. The first development took place in 1834 when Perkins proposed a hand-operated compressor machine working on ether. In 1851, Gorrie developed air refrigeration which was followed by the development of absorption systems and vapour compression systems using ammonia as refrigerant. Pictet liquefied oxygen in 1877 which signaled the era of cryogenics, a field of very low temperature refrigeration. By 1890 the refrigerating industry was fairly established, and within the next few years machines using ammonia, carbon dioxide and sulphur dioxide were developed. Vortex tube was discovered by Ranque in 1931, which is simply a straight length of tubing into which compressed air is admitted tangentially at the outer radius and so throttled that the central core of the resulting stream can be separated from the peripheral flow. The central core of air is found to be cold (as low as 40°C below the inlet temperature) relative to the hot air at the periphery.

With the advent of electric motors and halogenated hydrocarbon refrigerants during the first-half of the twentieth century, rapid developments in the application of refrigeration to commercial and industrial fields followed.

1.2 APPLICATIONS OF REFRIGERATION

Modern refrigeration is mostly applied in four general fields (Figure 1.3).

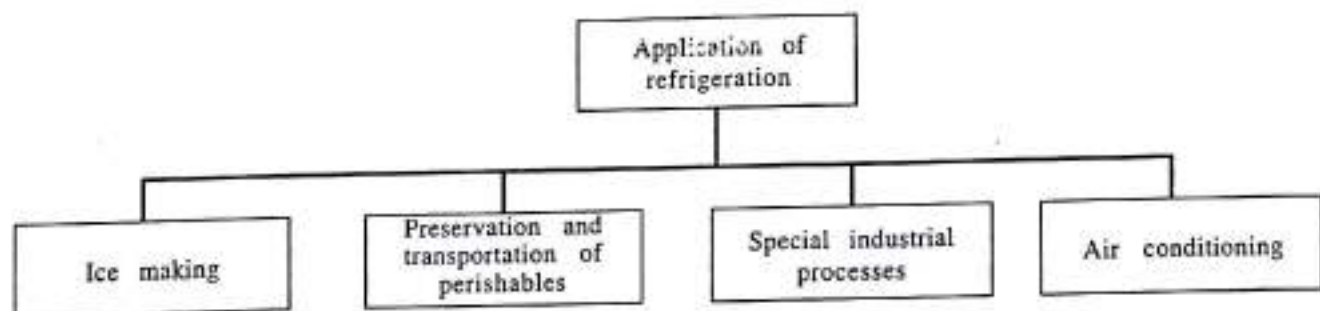


FIGURE 1.3 Applications of refrigeration.

Ice making

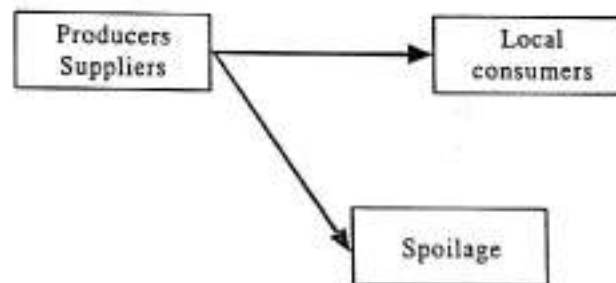
Over centuries natural ice was harvested for subsequent use or transported to warmer places lacking natural ice. The use of natural ice progressively increased as can be appreciated from the fact that millions of tons of natural ice were consumed in the United States in late nineteenth century. Thus the stage was set for the introduction of large-scale artificial refrigeration. With the introduction of refrigeration machinery, ice-plants were built near the consumers to produce ice on a commercial scale.

Preservation and transportation of perishables

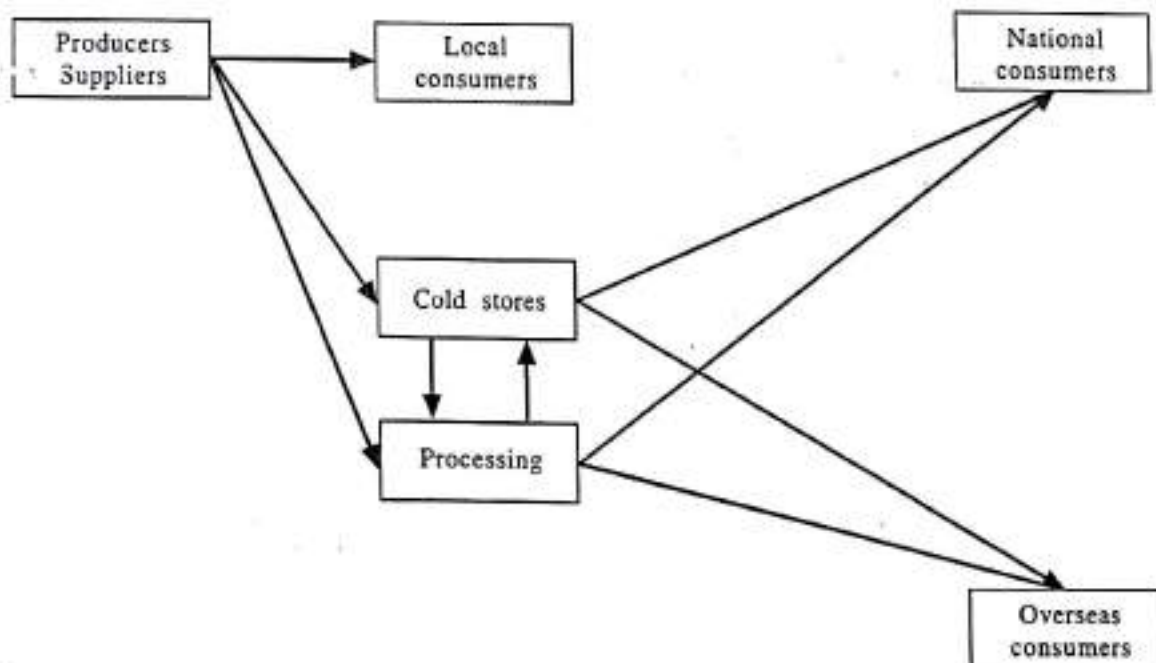
Refrigeration is widely used for the preservation of foodstuffs. Its purpose is to prevent or check spoilage, the more important causes of which are:

- Excessive growth of microorganisms, bacterial and fungal
- Changes due to oxidation, giving off flavours *تلفات*
- Enzymatic and fermentive processes, causing rancidity *الفساد*
- Drying.

Apart from dramatically reducing the spoilage of perishable foodstuffs, refrigeration has altered the pattern of consumption and movement of foodstuffs. With the parallel development of refrigerated transportation including reefer ships, the transformation of supply chain/system, as shown in Figure 1.4, has done away with the dependence on local markets as exporting to overseas markets became possible.



(a) Non-refrigerated supply system



(b) Integrated refrigerated supply system

FIGURE 1.4 Supply system transformations with the availability of refrigeration.

A global frozen food chain has been firmly established, opening the frontier of transcontinental transportation and distribution of perishable foodstuffs. Typically, the chain consists of the following links: refrigerated warehouses, refrigerated road transports, refrigerated (reefer) ships, port facilities and domestic refrigerators. Where a fully developed cold chain is in place, as illustrated in Figure 1.4(b), surplus agricultural and dairy produces are transported to production warehouses located near the growing areas. From there the processed food is transported to long-term warehouses for long-term storage. Canned food items are also transported from canning factories to the long-term warehouses. These products are then transported to distribution warehouses from where these are made available to consumers through supermarkets or grocers. The final link of the chain is the domestic freezer and refrigerator.

A brief description of the elements of cold chain (see Figure 1.5) will be useful to understand their specific roles and features.

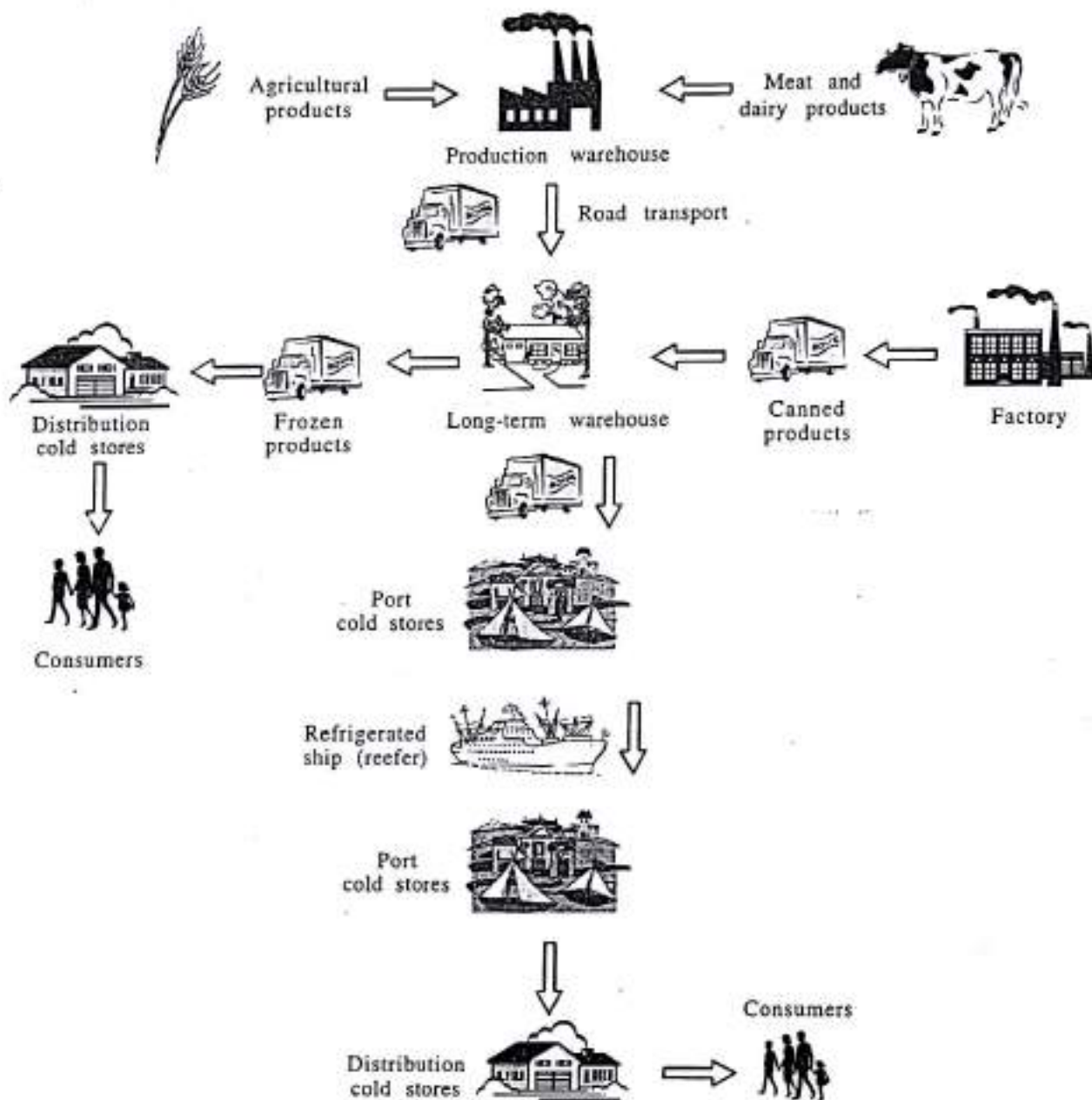


FIGURE 1.5 Elements of a frozen food chain.

Cold store is a building designed to store certain goods, particularly foodstuffs of perishable nature, in well-defined conditions of temperature and relative humidity. Based on types of activities, cold stores can be classified into (i) multi-purpose cold stores and (ii) specialized cold stores. The multi-purpose cold store is designed, built and equipped to store all varieties of products for short as well as longer periods. The specialized cold store is generally designed and equipped to store only a single type of product complementary to an agricultural or foodstuff industry, e.g. dairy, slaughterhouse, fruit packing station, etc.

Refrigerated transport is the main link of the frozen food chain. It can be broadly classified into refrigerated trucks and trailers, trains and refrigerated ships (reefers). A special feature of marine transport is the varying climate, ranging from extreme hot to extreme cold, through which the ship has to pass during the course of its journey. Since the cargo may vary from time to time, the refrigeration system should be capable of providing any temperature between -23.5 to 12.5°C .

Domestic refrigerators and freezers are the last link of the cold chain. These machines have become standard fixtures of most homes even in developing countries, particularly, since the prices have come down dramatically with the introduction of mass produced hermetically sealed compressors.

Special industrial processes

Refrigeration has applications in diverse industries, such as manufacturing chemicals, petroleum refineries, paper and pulp industries. It is also a necessary part of many chemical processes. It is needed frequently throughout the refining process. One of the main purposes is to crystallize wax and separate it out. It is also needed for fractional distillation of the lighter hydrocarbons and for desalination of water. It has also application in many heat treatment processes. Civil engineering projects are often aided by the use of refrigeration. Moisture laden soils are frozen to prevent cave-ins and to obtain soil samples. Ice and chilled water have been used in making concrete mix.

Air conditioning

Air conditioning is one of the most important applications of refrigeration and a rapidly expanding field of engineering. Further details have been elaborated later in the chapter as well as in subsequent chapters.

1.3 SCOPE OF AIR CONDITIONING

Climates in the different parts of the world vary with season of the year and with geographical locations and hence, in very few places the natural environment is comfortable for human beings or suitable for a specific product or process throughout the year. There is, thus, a need for the control of the environment for human comfort as well as for some specific equipment, material or process. Heating is essential in cold climates while cooling is generally desirable in hot climates. Figure 1.6 shows the climatic variation in different geographical zones of the world.

Apart from temperature, human comfort is greatly affected by humidity. Consequently in addition to cooling, dehumidification and humidification processes are necessary for humid and

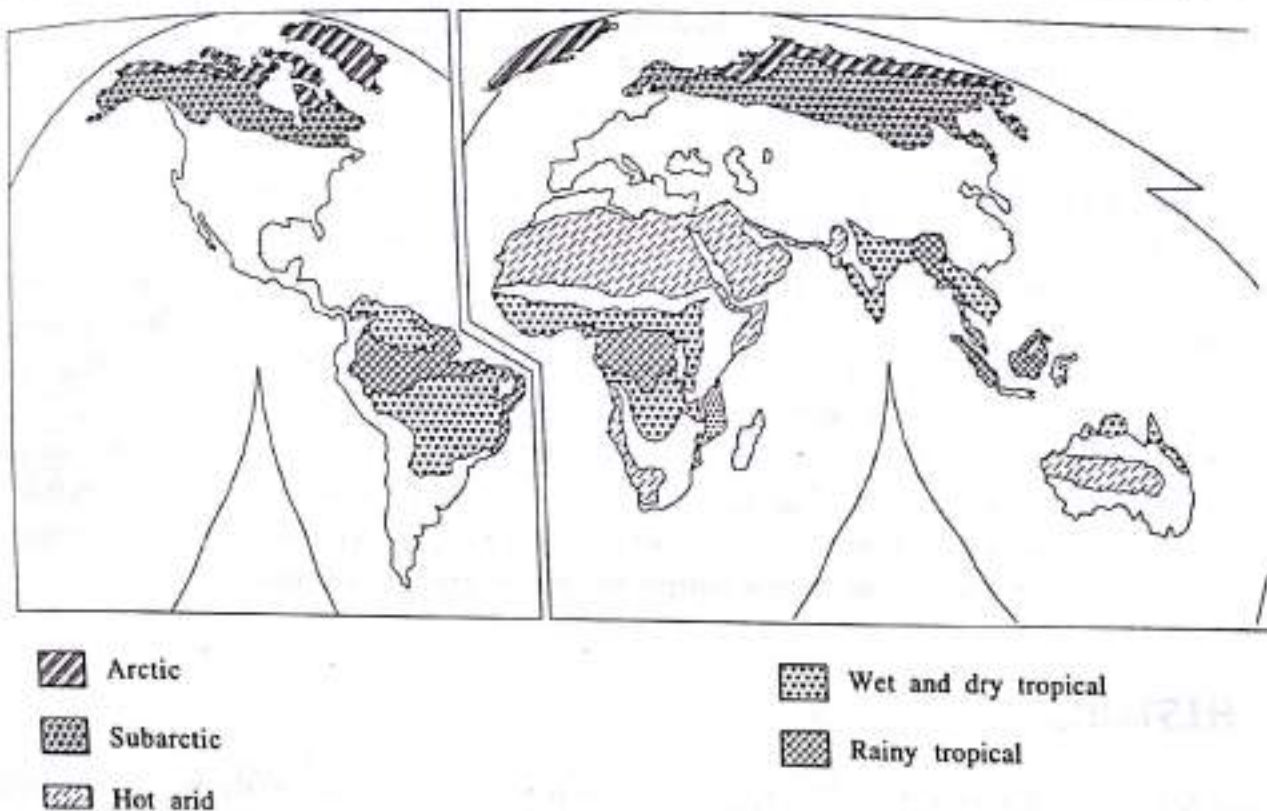


FIGURE 1.6 Different climatic zones of the world.

arid tropics respectively. Indoor air tends to get fouled due to absorption of pollutants from many different sources and the same needs to be controlled to an acceptable level. These processes involving artificial tampering of air and its proper distribution in an indoor environment, in fact, constitute air conditioning (Figure 1.7). More formally, *air conditioning may be defined as a form of air treatment which controls simultaneously the temperature, humidity, cleanliness and distribution of the air to meet the requirements of the conditioned space.* To provide these functions with most natural atmospheres, heating, cooling and regulation of

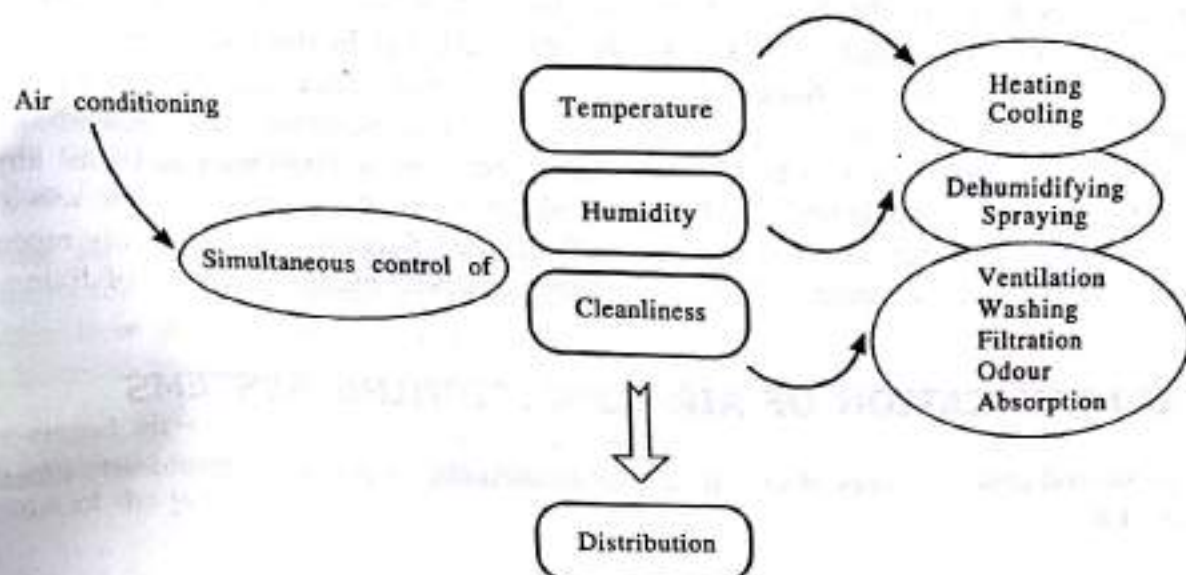


FIGURE 1.7 The functions of air conditioning.



thermal radiation will be required for temperature control; humidification and dehumidification will be required for humidity control and introduction of outside ventilation air, filtration, washing or odour absorption will be required for cleaning the air. Control of sound produced by the air-conditioning system itself is also important.

Besides controlling the temperature and humidity of the environment, air conditioning serves other useful purposes as well. By air conditioning, dust, bacteria, allergens such as pollens, noxious gases and odours can be removed from the air. It can assist in protection from such undesirable things as high levels of noise, and it can make possible the exclusion of harmful or annoying insects. However, the most important function of air conditioning remains the creation of an acceptable thermal environment.

The use of air conditioning is no longer confined to the earth. Man has been able to stretch his reach to the outer space as well as to the depth of ocean with the help of air conditioning. Man's landing on moon, prolonged stay in orbital spaceships, space shuttles and submarines have all been possible because of man's ability to create artificial environment.

1.4 HISTORICAL NOTES

The need for air-conditioning was first realized when it was found that certain products could be produced better in the right environment. With air-conditioning, there was no dependence of the product quality on the uncertainties of the weather and the factory sites were not limited to areas with a suitable climate. First to realize the importance of treatment of air was the textile industry which traditionally had to build its mills in areas with satisfactory humidity. Industrial air conditioning later spread to various other areas such as manufacture of rayon and plastics, colour printing, pharmaceuticals, tobacco industry, manufacture, development, reproduction of photographic materials, precision machine works involved in mass production, electrical equipment, and many hygroscopic products. These industrial applications were installed for the need of the products and not for personnel comfort.

Although sporadic attempts have been recorded for improving the ventilation and thermal conditions of indoor spaces, the first use of air-conditioning by mechanical means for human comfort was introduced in the United States of America at the turn of the twentieth century. There were few comfort cooling installations before 1920, but in the mid-twenties theatres all across the country were air conditioned to draw more customers. This was followed by the use of air conditioning in other commercial premises such as restaurants, night clubs, office blocks, etc. In the 1930s, air conditioning was introduced in transportation systems such as trains, airplanes, buses and trolleys. After the Second World War, with the rising family income there was a boom in domestic air conditioning. This trend of use in the United States was subsequently repeated in the developing countries as more and more countries achieved higher standards of living.

1.5 CLASSIFICATION OF AIR-CONDITIONING SYSTEMS

There are several different ways of classifying air-conditioning systems, as graphically illustrated in Figure 1.8.

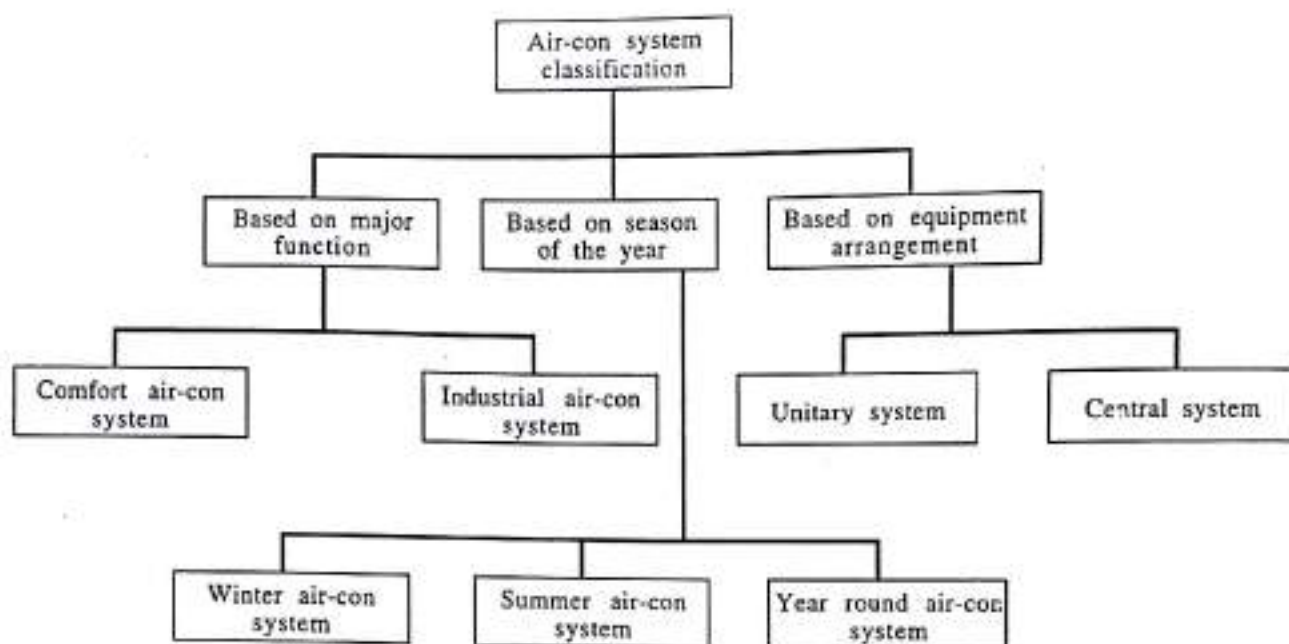


FIGURE 1.8 Classification tree of air-conditioning systems.

Based on major function

Comfort air-conditioning systems: Their purpose is to create atmospheric conditions conducive to human health, comfort, and efficiency. Air-conditioning systems in homes, offices, stores, restaurants, theatres, hospitals, and schools are of this type.

Industrial air-conditioning systems: The purpose of these systems is to control atmospheric conditions primarily for the proper conduct of research and manufacturing operations. Some of these systems also serve incidentally for human comfort, but some do not.

Based on season of the year

Winter air-conditioning systems: These systems when properly designed and installed, maintain indoor atmospheric conditions for winter comfort and involve heating of the air and bring the moisture content, i.e. humidity, up to an acceptable level. Heating is accomplished by furnaces or boilers fired with gas, oil, or coal.

Summer air-conditioning systems: These systems are meant for controlling the atmospheric conditions for summer comfort. In humid tropics, it involves cooling the air and removing excess moisture from it. For arid tropical climates, cooling is likely to be carried out in conjunction with humidification.

Year-round air-conditioning systems: These systems comprise heating and cooling equipment with automatic controls and associated components to produce conditions for human comfort for all time of the year.

ed on equipment arrangement

Unitary systems: These systems make use of air conditioners which are completely factory assembled and are designed to provide the functions of cooling, dehumidification, cleaning and movement of air.

Central-station systems: In these types of systems the various components are selected by the design engineer, procured from several manufacturers and erected on the job at site, usually in the central plant room.

1.6 APPLICATIONS OF AIR CONDITIONING

The main application of air conditioning is for human comfort. Summer cooling has become ubiquitous in large modern office and institutional buildings throughout the world. Even in climates where summer temperature is not very high, such buildings are air conditioned, primarily to offset the heat generated due to occupancy, lighting and equipment, computers and their paraphernalia.

The other applications, known as industrial air conditioning, are for maintaining optimum environments for specific products or processes in factories and industrial premises, which do not necessarily encompass complete air conditioning as in the case of comfort cooling. These applications will be discussed in greater detail in subsequent chapters of this book.

1.7 CURRENT STATUS AND FUTURE TRENDS

Currently, most of the air-conditioning systems use vapour compression refrigeration cycles. Vapour absorption cycles are competitive in those areas, where waste heat is available or an abundant supply of natural gas is available easily. Solar-powered absorption cycles have been built and commissioned but they are yet to be viable economically. Evaporative cooling is economical and viable in hot and dry climates. Uses of steam jet refrigeration cycle, vortex tube, thermo-electric cycle, or air and vapour expansion cycles are limited to special applications where economic considerations are secondary. For example, in aircraft, where the primary concerns are power-to-weight ratio and safety, air expansion cycles are used since the conventional vapour compression cycles are unsuitable. In the case of air conditioning of submarines, quietness is the primary concern to avoid detection by the enemies and, hence, thermo-electric cooling devices are employed, which meet the same criteria. Vortex tubes have similarly been useful in mine air conditioning due to their safety features.

Two of the dominating issues that confront today's world are the energy and environmental problems which are closely related. In the aftermath of the oil price boom in the early 1970s, energy conservation measures have become very important factors in the design and operation of air-conditioning systems. There are constant endeavours to introduce newer methods of air conditioning which are more energy efficient. Solar powered air conditioning is an attractive option from the environmental point of view, despite the fact that these systems are not currently economical. However, with further developments and more so if the energy prices soar, solar systems may become competitive in the tropical climates. Amongst other types of systems under development, desiccant air conditioning in the near future has some potential to become viable in the humid tropics. Desiccant cooling allows the atmospheric moisture to be removed by desiccants, followed by evaporative or mechanical cooling. Another interesting option being

explored is the radiant cooling in conjunction with chilled ceiling with or without desiccant dehumidification. The discovery of the depletion of ozone layer and global warming has become two of the dominating issues since the 1980s. Due to the harmful effects of the chlorofluorocarbon (CFC) refrigerants on the ozone layer, these are being phased out and replaced by environmentally-friendly refrigerants.

Thermal energy storage for cooling or cool storage air conditioning is rapidly gaining popularity in the United States and Japan because it can significantly reduce the peak electrical consumption of buildings. The basic concept of cool storage involves operating the cooling system at or near the full capacity during the off-peak hours to generate and store chilled water or ice which would be later used during periods of peak electrical and/or cooling demand to supplement or replace conventional mechanical cooling. The current trend towards the use of cool storage systems for air conditioning has been accompanied by an increasing interest in the distribution of lower temperature air at as low as 5–6°C. The use of lower supply air temperature is attractive because smaller fans, ducts, pumps, and piping can be used, resulting in lower first costs, reduced electrical demand, and lower operating costs in comparison to conventional systems.

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REVIEW EXERCISES

- 1.1 Briefly describe how refrigeration and air conditioning are related to one another.
- 1.2 How did the Egyptians produce ice?
- 1.3 What are the major links of a frozen food chain?
- 1.4 What are the different methods of refrigeration employed in refrigerated transports?
- 1.5 Name the different types of cold stores and describe their purposes.
- 1.6 What are the different ways of classifying the air-conditioning systems?
- 1.7 What are the different functions of air conditioning?
- 1.8 Point out the differences between the requirements of summer and winter air conditioning.
- 1.9 What air-conditioning processes are involved in humid tropics and arid tropics?
- 1.10 What is the difference between industrial and comfort air conditioning?

2

Review of Basic Principles

We explained in Chapter 1 that *air conditioning* is concerned with controlling the temperature, humidity and velocity of air motion in an enclosed space to maintain a comfortable environment. Towards that end, *refrigeration* needs to be used to bring about cooling of the air. Anyone pursuing the study of refrigeration and air conditioning usually goes through courses on thermodynamics, fluid mechanics and heat transfer as necessary prerequisites. Hence, any detailed discussion of any topic related to those subjects is not considered necessary in this book. However, a brief review of a number of physical principles related to the study of refrigeration and air conditioning has been included to help students understand the subject and use the same as reference material.

2.1 UNITS

Unit is a relative idea, which is very important for measuring physical quantities. Unit may be defined as any specified amount of a quantity through which any other quantity of the same kind is measured. Metres, centimetres, kilometres, feet are all units of length.

There are two systems of units:

- (a) The International System of units, or the SI system
- (b) The British engineering system of units, or the British system

SI units

The International System of units (for the French words *Système Internationale d'Unites*) or SI is by far the most important and widely accepted system used in the world today. It is controlled by an international organization called the General Conference on Weights and Measures. The base units in SI are listed in Table 2.1.

TABLE 2.1 SI base units

Physical quantity	Name of unit	Symbol
length	metre	m
mass	kilogram	kg
time	second	s
electric current	ampere	A
thermodynamic temperature	kelvin	K
amount of substance	mole	mol

The SI unit of *length* is the metre (m) and is defined as 1,650,763.73 wavelengths in vacuum of the orange-red line of the spectrum of krypton-86 (Kr-86). The SI unit of *time* is the second (s). The second is defined as the duration of 9,192,631,770 cycles of the radiation associated with a specified transition of the cesium atom (Cs-133). The SI unit of *mass* is the kilogram (kg). It is defined as the mass of a particular cylinder, maintained under prescribed condition, of platinum-iridium alloy kept at the International Bureau of Weights and Measures in France.

Derived units: A derived unit involves combinations of the base (or fundamental) units. For example, the SI unit of *force* is a derived (or secondary) unit; it is called the newton (N) and defined from the Newton's second laws of motion, i.e.

$$F \propto ma \quad \text{or} \quad F = kma$$

In a coherent system of units such as SI, $k = 1$, hence:

$$F = ma$$

The newton (N) is defined as the force required to accelerate a mass of 1 kg at the rate of 1 m/s^2 , i.e.

$$1 \text{ N} = (1 \cdot \text{kg})(1 \text{ m/s}^2) = 1 \text{ kg} \cdot \text{m/s}^2$$

The weight of the body is the force of gravity. This is the force with which the body is attracted to the earth or some other body. The weight of an object can vary as it is calculated from the mass of the body and the local acceleration due to gravity. While a weight of a substance can vary, its mass remains constant.

Similarly, it follows that the SI unit of *work* (i.e. force \times distance) is the newton metre (N-m). A general unit for *energy*, more specifically heat and work, is introduced by giving the N-m the name joule (J). Thus,

$$1 \text{ joule (J)} = 1 \text{ newton (N)} \times 1 \text{ metre (m)} \quad \text{or} \quad 1 \text{ J} = 1 \text{ N} \cdot \text{m}$$

A quantity that is closely related to work is *power*, the rate at which work is done. A commonly used unit of power is watt (W) defined as

$$1 \text{ watt (W)} = 1 \text{ J/s} = 1 \text{ N} \cdot \text{m/s}$$

Another base unit that is frequently used in thermodynamics is the mole (mol). Mole is defined as the amount of substance containing as many elementary entities as there are atoms in 0.012 kg of carbon-12. These elementary entities may be atoms, molecules, electrons, ions or other particles or specified groups. For example, one mole of diatomic oxygen having a molecular weight of 32 (compared to 12 for carbon) has a mass of 32 g. The gram molecular weight of a substance is the number of grams per mole of that substance. Thus the gram

molecular weight of oxygen is 32 g/mol. But normally kmol is used. If the number of kilomoles of a substance is n , then it can be determined as

$$n = \frac{m}{M}$$

where

m is the mass (kg)

M is the molecular weight (kg/kmol).

2.2 THERMODYNAMICS

Thermodynamics is the science of the relationship between heat and work and the properties of substances in systems. Its development is based on some natural and experimental observations. Engineering thermodynamics is also the study of systems and the interaction of them with their surroundings. System can be studied from a macroscopic or a microscopic point of view. In the macroscopic approach the focus is on the gross or average effects of many molecules of a substance. These effects can be sensed and measured by instruments. This is called *classical thermodynamics*. On the other hand, in microscopic approach, the average value of all the molecules in a substance is considered. Thus it is concerned directly with the structure of the matter and is called *statistical thermodynamics*. Classical thermodynamics is a direct approach and needs fewer mathematical computations and is usually adopted for engineering analysis. But in this modern age, thermodynamics is perceived as the science that deals with energy.

Concept of systems

In thermodynamic analysis, the system is whatever chosen to be studied, i.e. it is the concept of identifying the subject. Normally the system is a specified region that can be separated from everything else by a well-defined surface. The defining surface is known as the *control surface* or *system boundary*. The boundary may be static (fixed) or dynamic (moveable). Anything external to the system is the *surrounding*. Thus a system may be defined as a collection of matter within the prescribed and identifiable boundaries. In the example of a piston-cylinder arrangement shown in Figure 2.1, the gas is the system as it is intended to be studied.

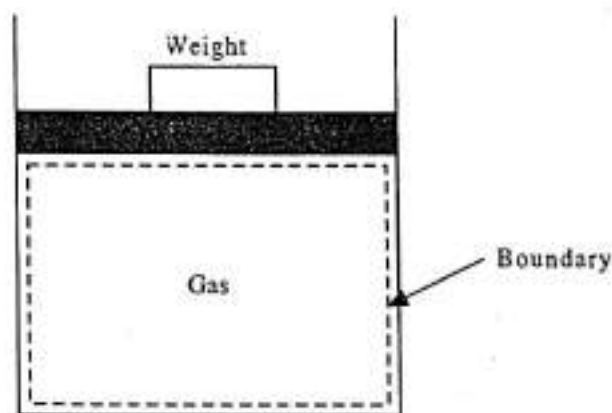


FIGURE 2.1 Example of a system.

There are two kinds of systems—*closed systems* and *open systems* which are sometimes referred to as *control volume*. A system of fixed mass is referred to as a *closed system*. When

there is flow of mass through the control surface, the system is called a *control volume* or *open system*. Thus in closed systems there is constant quantity of matter whereas the open systems deal with the variable quantity of matter. Consequently, in open systems mass may flow but in the closed systems mass is constant.

Characteristics of systems

The condition of a system at any instance is called its *state*. After defining a system, tools are needed to describe it and predict its behaviour. With properties we can describe system behaviour.

Properties are the macroscopic characteristics of a system having numerical values and are independent of the path or the process of the substance. Some examples of properties are mass, volume, energy, pressure, and temperature. There are two types of properties, *extensive* properties and *intensive* properties. Simply stated, an intensive property is independent of mass while an extensive property is mass dependent, i.e. its value varies with the mass and size of the system and can change with time. Mass, volume, and energy are extensive properties. An intensive property is not dependant on the size of the system and can vary from place to place within a system. Pressure and temperature are important intensive properties.

When we know the thermodynamic properties of a system, then we know the system behaviour or condition at any time. This system condition is known as *state*. More specifically, a state represents the condition of a phase. *Phase* is defined as a quantity of matter that is homogeneous throughout in both chemical composition and physical structure. Homogeneous in physical structure means whether the matter is all solid or all liquid or all vapour. A system may have different phases. A system of liquid water and water vapour contains two phases. In each phase the system may exist in different states. If more than one phase is present in the system at a time, the phases are separated from each other by *phase boundaries*. A *pure substance* is one that is uniform and invariable in chemical composition. It can have more than one phase but its chemical composition must be the same in each phase. For example, if liquid water and water vapour form a system of two phases, the system can be treated as a pure substance because each phase has the same composition as shown in Figure 2.2.

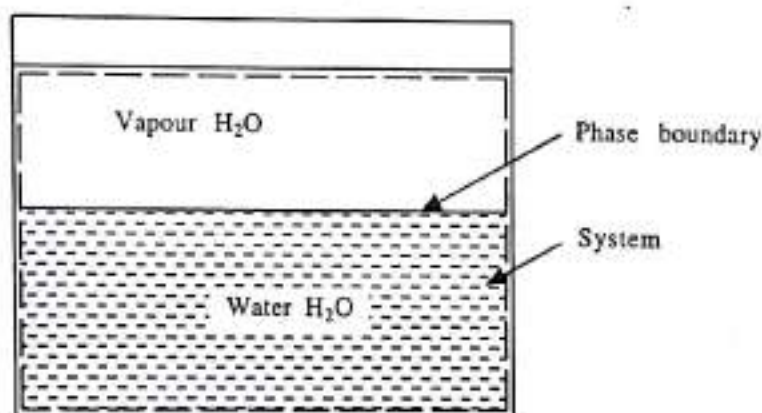


FIGURE 2.2 Example of a system of two phases.

Generally speaking, the term property is meant to be the property of a system and not the property of a substance. This is only possible when the system is in *equilibrium state*. Equilibrium means the condition of balance, i.e. where two opposite forces are equal.

Equilibrium can be in various forms. A system is in *thermal equilibrium* if the temperature of the system is same in every place of the system. And if there is no change in pressure, then it is called *mechanical equilibrium*. If a system involves two phases, then it is called *phase equilibrium*. And if the chemical composition is same throughout the system, then it is called *chemical equilibrium*. When a system is in equilibrium with all possible changes of state, then it is called *thermodynamic equilibrium*.

If a system changes from one equilibrium state to another equilibrium state, then it is called a *process*. The various states through which a system passes during a process constitute the *path* of the process as illustrated in Figure 2.3.

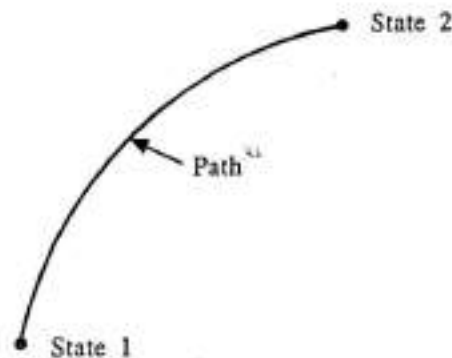


FIGURE 2.3 Path of a process.

The actual process which occurs in a non-equilibrium condition can be described using the quasi-equilibrium concept. A *quasi-equilibrium state* is one where deviation from thermodynamic equilibrium is infinitesimal. All the states of a quasi-equilibrium process may be considered as an equilibrium process. If the process is not possible to be described through the quasi-equilibrium process, then the states of the system are described both before the process occurs and after the process is completed, thus restoring equilibrium.

When a system goes through a number of changes of states from its initial state and finally returns to its initial state, it is called a *cycle* (Figure 2.4). A cycle may have different processes.

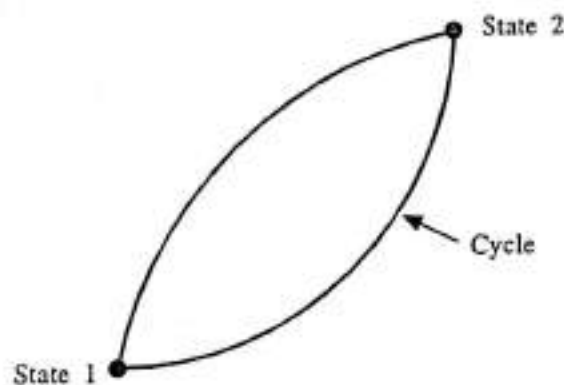


FIGURE 2.4 A cyclic process.

Intensive properties: specific volume, pressure, temperature

From the macroscopic point of view we are always concerned with large volume. We are not interested in molecular dimensions and the behaviour of individual molecules. So from the

macroscopic point of view, the substance is considered to be a continuous, homogeneous matter, that is a continuum. This concept of continuum is valid as long as we work with volumes, areas, and lengths that are large relative to their intermolecular space. With the continuum concept, it is possible to determine the intensive thermodynamic property at a point.

Specific volume: The specific volume of a substance is defined as the volume per unit of mass, i.e.

$$v = \frac{V}{m}$$

If a small volume δV of a system has a mass of δm , then the specific volume will be

$$v = \lim_{\delta V \rightarrow \delta V'} \frac{\delta V}{\delta m}$$

where $\delta V'$ is the smallest volume for which the system can be considered as a continuum. This is the specific volume at a point and can vary with elevation, but the systems we normally consider are relatively small and the change in specific volume with elevation is not significant. The units of the specific volume are m^3/kg and m^3/mol .

Density is the reciprocal of specific volume as it is defined as the mass per unit volume. Specific volume is represented by the Greek symbol v and density is represented by the Greek symbol ρ . The relation between specific volume and density is

$$v = \frac{1}{\rho}$$

Pressure: This intensive property is very important and significant when we consider gas and liquids. Pressure is defined as the normal component of force per unit of area. The pressure in a fluid at rest at a given point is the same in all directions. If δA is a small area and δF is the component of force normal to that small area, then pressure p is defined as

$$p = \lim_{\delta A \rightarrow \delta A'} \frac{\delta F}{\delta A}$$

where $\delta A'$ is the smallest area for which we can consider the fluid a continuum. The unit of pressure is the force of 1 newton acting on a square metre area. It is called pascal (Pa). Thus,

$$1 \text{ Pa} = 1 \text{ N/m}^2$$

Another unit of pressure is bar, where

$$1 \text{ bar} = 10^5 \text{ pascal}$$

Sometimes atmospheric pressure is used as a standard to express other pressures, where

$$1 \text{ atm} = 101,325 \text{ pascal}$$

Pressure as discussed above is called the absolute pressure. In measuring pressure, two types of pressures are frequently used. One is gauge pressure and the other is vacuum pressure. When the pressure in the system is greater than the local atmospheric pressure, then the gauge pressure term is used for measurement. Thus,

$$p(\text{gauge}) = p(\text{absolute}) - p_{\text{atm}}(\text{absolute})$$

When the local atmospheric pressure is greater than the pressure in the system, then the vacuum pressure term is used in the measurement. Thus,

$$p(\text{vacuum}) = p_{\text{atm}}(\text{absolute}) - p(\text{absolute})$$

The relationship between the various methods of pressure measurement is shown in Figure 2.5.

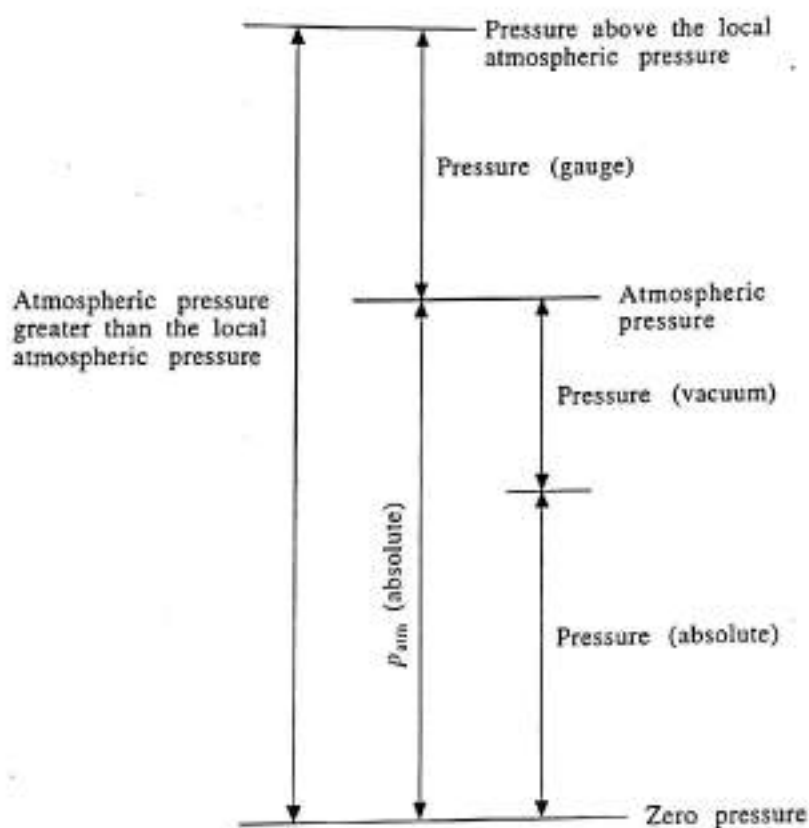


FIGURE 2.5 Relationship between different pressures.

The two commonly used devices for measuring pressure are manometer and bourdon tube. Manometers measure pressure differences in terms of the length of a column of liquid such as water, mercury.

The manometer shown in Figure 2.6 has one end opened to the atmosphere and the other end is attached to a closed vessel containing a fluid. From the length of the manometer fluid, the difference between the fluid pressure and that of the atmosphere can easily be measured from the equation

$$p - p_{\text{atm}} = \rho g L$$

where p is the fluid pressure, p_{atm} is the atmospheric pressure, ρ is the density of the manometer fluid, L is the difference in liquid levels, and g is the acceleration of gravity.

Temperature and zeroth law of thermodynamics

It is not easy to give an exact definition of temperature. We can sense whether a body is hot, cold or warm, but cannot assign any numerical value. And our sense may be wrong because sometimes different materials show different characteristics at the same temperature. For example,

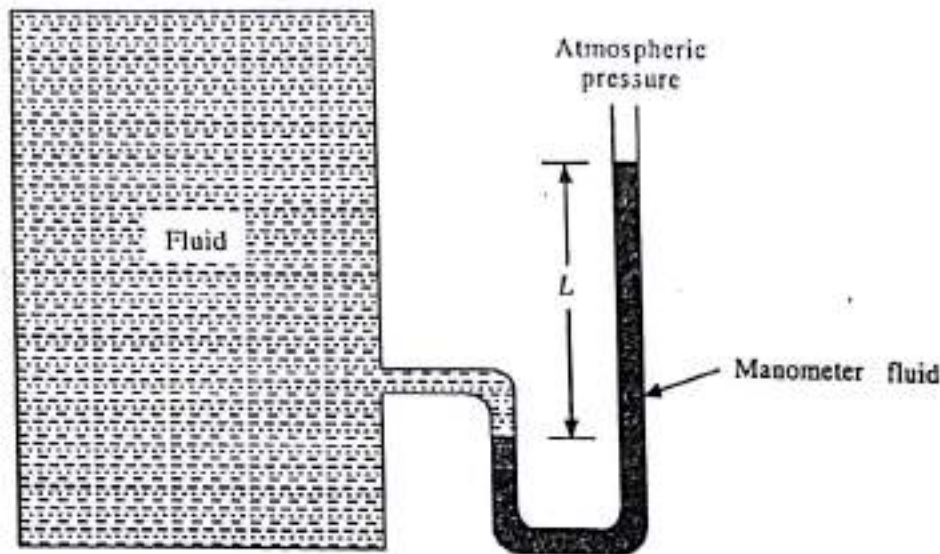


FIGURE 2.6 Manometer.

a metal chair will feel much cooler than a wooden chair at the same temperature. The basis of temperature measurement is the zeroth law of thermodynamics, which states that:

When two bodies have equality of temperature with a third body then the two bodies have equality of temperature with each other.

Equality of temperature means that they are in thermal equilibrium, i.e. there is no heat transfer between the two bodies when they are in contact. From the zeroth law if we consider the third body as a mercury thermometer, then we can measure the temperature of substances using the principle of thermal equilibrium.

Several properties of materials change with temperature, which form the basis of accurate temperature measurement. An example is the expansion of mercury with temperature. But now we need a temperature scale to relate between the different temperature-measuring devices. The temperature scale will be discussed with the second law of thermodynamics.

The first law of thermodynamics and energy

The first law of thermodynamics is called the law of the conservation of energy. Prior to defining this law, it is helpful to discuss work and heat as they are parts of the first law.

Work: When a body moves through a distance by the action of a force, work is said to be done. The direction of the force is in the direction of body movement. The expression of work can be written as follows:

$$W = \int_1^2 F \cdot ds \quad (2.1)$$

From the thermodynamic point of view, it is important to know the work in relation to systems, properties, and processes. Work done is observed to be energy in transition that is never contained in a body nor possessed by a body. The sign convention of work is as follows:

$W > 0$ work done by the system
 $W < 0$ work done on the system

In the SI system of units, the work is represented as joule (J). If 1 newton force is applied to a body and the body moves through a distance of 1 m as the result of force, then

$$1 \text{ J} = 1 \text{ N} \cdot \text{m}$$

Work done during a quasi-equilibrium process can be found by integrating the following equation:

$$W = \int_1^2 p dV \quad (2.2)$$

The shaded area in Figure 2.7 represents the work done on a p - V diagram.

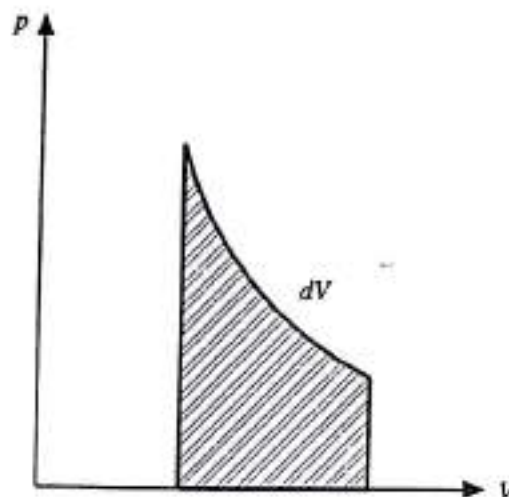


FIGURE 2.7 The p - V diagram.

Heat: It is defined as the form of energy that is transferred across the boundary as a result of temperature differences between the two systems. If $Q > 0$ then heat is transferred to the system and if $Q < 0$ then heat is transferred from the system. The units of heat and the units of work are same as both work and heat are a form of energy. In the SI system, the unit of heat is defined as joule.

The first law of thermodynamics for a thermodynamic cycle states that the integral of heat of a cycle is proportional to the integral of work of that cycle. Energy will be introduced as a new property as the first law is applied to a system that undergoes a change of state. It can be shown as follows:

$$\text{Heat} - \text{Work} = \text{Energy}$$

or

$$\delta Q - \delta W = dE \quad (2.3)$$

Energy E represents all the energy of the system in the given state. This energy can be present in a variety of forms such as potential energy, kinetic energy, and internal energy. Kinetic energy and potential energy of the system are associated with the motion and position of the molecules respectively. And all the other energy of a system is defined as internal energy. Thus,

$$\text{Energy} = \text{Internal energy} + \text{Kinetic energy} + \text{Potential energy}$$

The kinetic energy is expressed as $\frac{1}{2}m\bar{v}^2$ and potential energy as mgz , where z is the elevation of a system. Then the integral form of heat transfer between two states can be expressed as follows:

$${}_1Q_2 = U_2 - U_1 + \frac{m(\bar{v}_2^2 - \bar{v}_1^2)}{2} + mg(z_2 - z_1) + {}_1W_2 \quad (2.4)$$

Enthalpy: Sometimes, the combinations of thermodynamic properties also form a new thermodynamic property. If there is a constant pressure process and there are no changes in kinetic or potential energy and work is done only by boundary movement, then applying the first law we can write

$$\delta Q = \delta W + \delta E \quad (2.5)$$

By introducing the equation of work done $W = \int_1^2 p dV$, the above equation can be written in the following form:

$$Q = (U + pV)_{\text{state 1}} + (U + pV)_{\text{state 2}} \quad (2.6)$$

So the heat transfer during the process is given in terms of change of $U + pV$ between the initial and final states. All of these quantities are thermodynamic properties and only depend on the state. So it is defined as an extensive property called enthalpy (H). Therefore,

$$H = U + pV \quad (2.7)$$

Change of state

The prerequisite to conversion of a solid to a liquid, and liquid to a gas is to overcome the molecular forces. There are certain energy relationships for different substances to bring about these transitions from one state to another. When sufficient heat energy is supplied to a liquid, it boils and vapour is given off. When enough heat energy is taken away from a liquid, it freezes and becomes a solid. Figure 2.8 illustrates the energy-state-temperature relationship for water. The starting point is 1 kg of ice at condition point A (-10°C) in the diagram. As heat is slowly

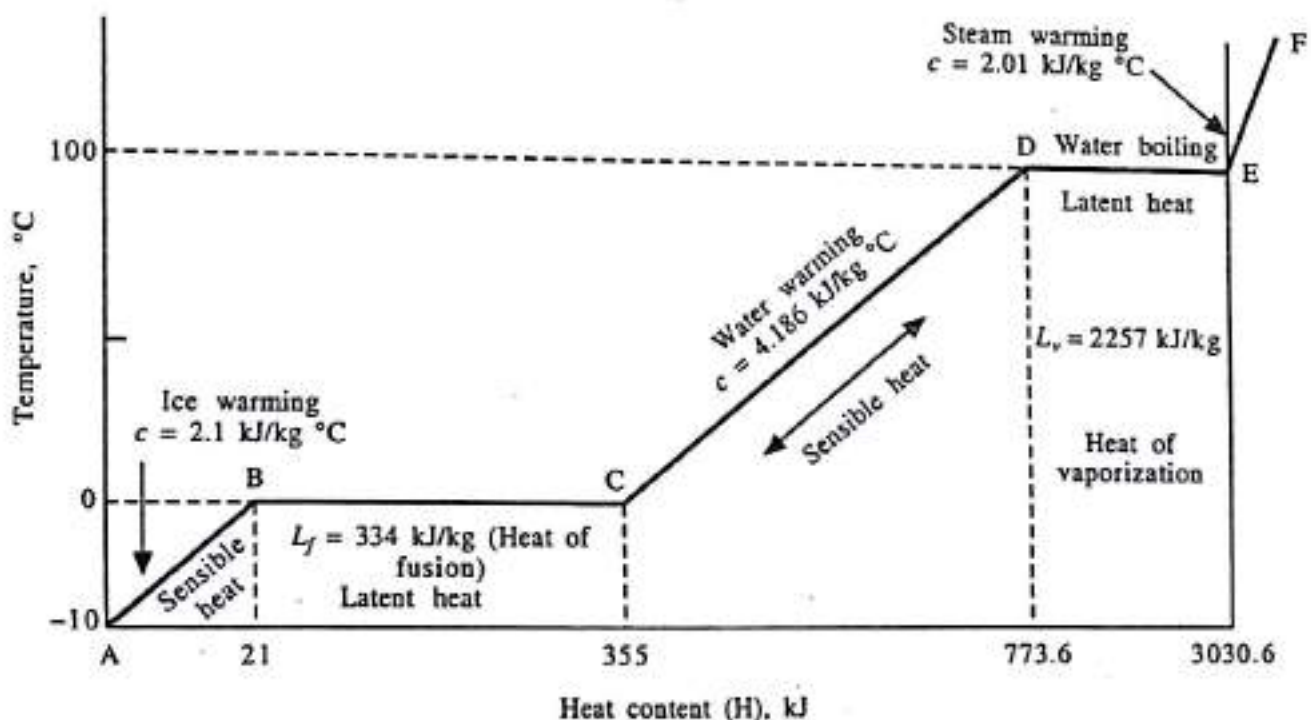


FIGURE 2.8 The temperature versus enthalpy diagram for water at various states.

added to the ice, the temperature is found to rise till it reaches 0°C . It may be noted that with the addition of heat the temperature rises. Since the rise in temperature can be detected by our senses, this is termed *sensible heat*. Referring again to the diagram, the water (1 kg) is now at condition point B still solid ice at a temperature of 0°C . If more heat is added to the ice, it is noted that no further temperature change occurs but the ice starts to melt. Since the additional heat during the melting process BC does not result in a temperature change, it is not sensible heat, as its effect cannot be measured by a thermometer. It is virtually hidden from our sensory perception, and so it is called *latent heat*. Latent heat is defined as the heat which, when supplied to or removed from a substance, produces a change of state without any change in temperature. The latent heat of fusion is the heat which must be added to 1 kg of solid already at its melting point temperature to change its state to liquid at the same temperature.

If we add more heat to water (point C), the temperature starts rising again, so this is again sensible heat. As more heat is slowly added to the water the temperature is found to rise till it reaches 100°C (Point D). Continuing to add heat, another change of state takes place along the DE segment of the diagram. The *latent heat of vaporization* is the heat which must be added to 1 kg of liquid (already at its boiling point temperature) to change its state to vapour at the same temperature. Further heating will result in increasing the temperature of the vapour (steam) and EF is, therefore, a sensible heat process.

The second law of thermodynamics

Like other physical laws the basis of the second law of thermodynamics is experimental evidence. There are many statements of the second law. Among them, the two classical statements of the second law are the Kelvin-Planck statement and Clausius statement.

According to Kelvin-Planck statement:

It is impossible for any system to operate in a thermodynamic cycle and deliver a net amount of energy by work to its surroundings while receiving energy by heat transfer from a single thermal reservoir.

Thus work can be done in a cycle by transferring heat only if there are two temperature levels. Thermal reservoir is a concept where energy remains constant even though energy is added or removed. So in the thermal reservoir, heat can be transferred indefinitely without change in the temperature of the reservoir. The reservoir from which heat is transferred is called *source* and the reservoir to which heat is transferred is called *sink*. The intensive property can be changed but the temperature remains constant.

According to Clausius Statement:

It is impossible to construct a device that operates in a cycle and produces no effect other than the transfer of heat from a cooler body to a hotter body.

From the Clausius statement it is impossible to construct a refrigerator that operates without an input of work.

Reversible and irreversible processes

The reversible process means an ideal process. Simply stated, a reversible process can easily be reversed to its initial state. In a reversible process there will be no change in surrounding and system. Generally, a process becomes irreversible by the following ways.

Friction is the main cause that makes a process irreversible. Unrestrained expansion of gas causes irreversibility. Heat transfer through a finite temperature difference is an irreversible process. But it becomes reversible if the temperature difference approaches zero. Electric current flow through a resistance causes irreversibility.

Irreversibility is divided into two parts—internal and external. Internal irreversibilities occur within the system and external irreversibilities occur within the surroundings. In a reversible process, the deviation from equilibrium is infinitesimal. But the actual process occurs at a finite rate, so deviation from equilibrium is finite. So the actual process is irreversible to some degree.

Based on the second law of thermodynamics the most efficient cycle is the Carnot cycle. The Carnot cycle operates between two reservoirs: (a) low-temperature reservoir and (b) high-temperature reservoir (Figure 2.9). If every process is reversible, the cycle is also reversible and it becomes a refrigerator when the cycle is reversed.

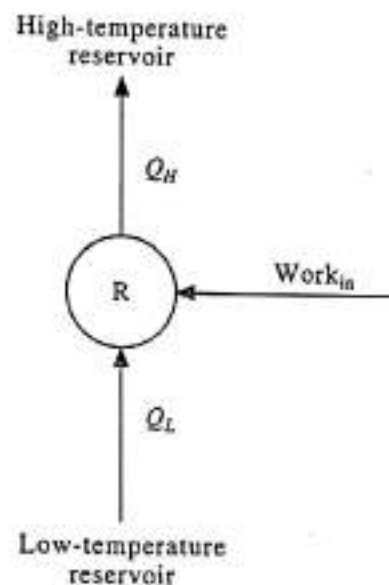


FIGURE 2.9 Carnot refrigeration cycle.

Two important conclusions can be made from the second law of thermodynamics. They are known as Carnot principles.

- The efficiency of an irreversible heat engine is always less than the efficiency of a reversible engine operating between the same two reservoirs.
- The efficiencies of all reversible heat engines operating between the same two reservoirs are the same.

The details of Carnot cycle and heat pump will be discussed in Chapter 3 on vapour compression refrigeration cycle.

Temperature scale

A temperature scale is necessary in order to use a common basis for measurement. The common bases are the freezing and the boiling points of water. They are also called the ice point and the steam point. The temperature scale used in SI system is the Celsius scale. Earlier it was called the



centigrade scale. Now it is known as the Celsius scale after the Swedish astronomer Anders Celsius. In the English system, the temperature scale is known as the Fahrenheit scale. Fahrenheit scale is named after the German instrument maker G. Fahrenheit. In the Celsius scale the ice and the steam points are 0°C and 100°C and in the Fahrenheit scale 32°F and 212°F respectively.

In thermodynamics, it is very important to have a temperature scale that is independent of the properties of any substance. It is developed in conjunction with the second law of thermodynamics. The second Carnot principle states that all reversible heat engines have the same thermal efficiency operating between the same two reservoirs. That is, the thermal efficiency of a reversible engine is independent of the working fluid and is a function of the reservoir temperatures only. Thus,

$$\eta_{th,rev} = 1 - \frac{Q_L}{Q_H} = f(T_H, T_L) \quad (2.8)$$

The thermal efficiency can also be expressed in the following form from the point of view of thermodynamic concept.

i.e.
$$\eta_{th} = 1 - \frac{Q_L}{Q_H} = 1 - \frac{T_L}{T_H} \quad (2.9)$$

From this if the efficiency of a Carnot cycle operating between the two given constant temperatures is known, the ratio of the reservoir temperatures can be determined. The temperature scale is not completely defined by the above equation since it gives only a ratio of temperatures. We need to know the magnitude of the scale. At the International Conference on Weights and Measures held in 1954, the triple point of water was assigned the value 273.16 K . The magnitude is defined as $1/273.16$ of the temperature interval between absolute zero and the triple point temperature of water.

Properties of a pure substance

A pure substance is one that has a homogeneous and invariable chemical composition and may have more than one phase. Water is a good example of a pure substance. Water can exist in different forms such as steam, ice, or a mixture of both. Their chemical composition is same in all the phases. The major properties of pure substances are pressure, specific volume, and temperature. Temperature and specific volume are known as independent properties and pressure is determined as a function of temperature and specific volume. We have to know some terms for proper understanding of the properties of pure substances. These are as follows:

Saturation temperature: It is defined as the temperature at which vaporization takes place at a given pressure.

Saturation pressure: The pressure at saturation temperature is called the saturation pressure.

Saturated liquid: A substance is called the saturated liquid if it exists as liquid at the saturation temperature.

Subcooled liquid: A substance is called subcooled liquid if it exists as liquid at saturation pressure but below the saturation temperature.

Quality of substance: If a substance exists as part liquid and part vapour at the saturation temperature, then its quality is defined as the ratio of the mass of vapour to the total mass.

Saturated vapour: A substance is called saturated vapour if it exists as vapour at saturation temperature.

There is a relationship between the saturation pressure and saturation temperature. A typical curve is shown in Figure 2.10.

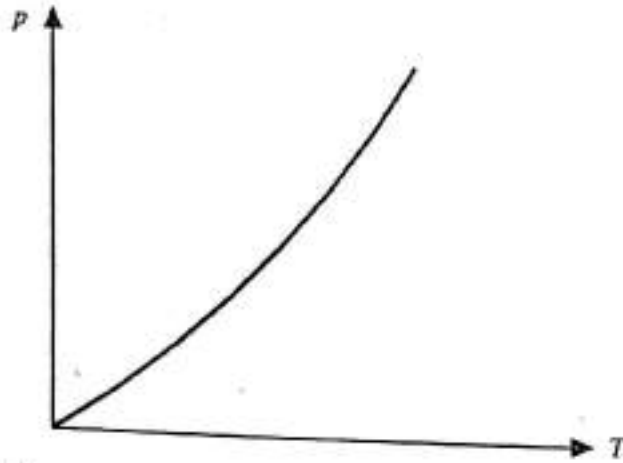


FIGURE 2.10 The pressure-temperature relationship of a pure substance.

If we determine the temperature and volume relationship under various pressures, the resulting curves will be like those shown in Figure 2.11.

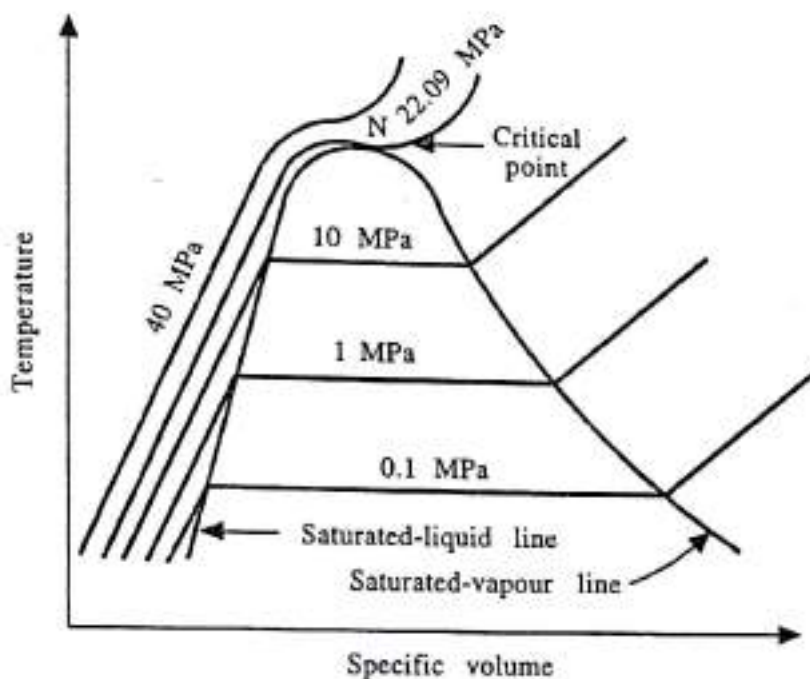


FIGURE 2.11 The temperature-volume curves for water.

Figures 2.11 and 2.12 show the p - v - T surface of a substance, such as water that expands on freezing. The coordinates of a point on the p - v - T surface represent the values that pressure, specific volume and temperature would assume when the substance is at equilibrium. There are regions on the p - v - T surfaces labelled solid, liquid and vapour. In these single-phase regions, the state is fixed by any two of the properties, i.e. pressure, specific volume, and temperature, since all of these are independent when there is a single phase present. Located between the

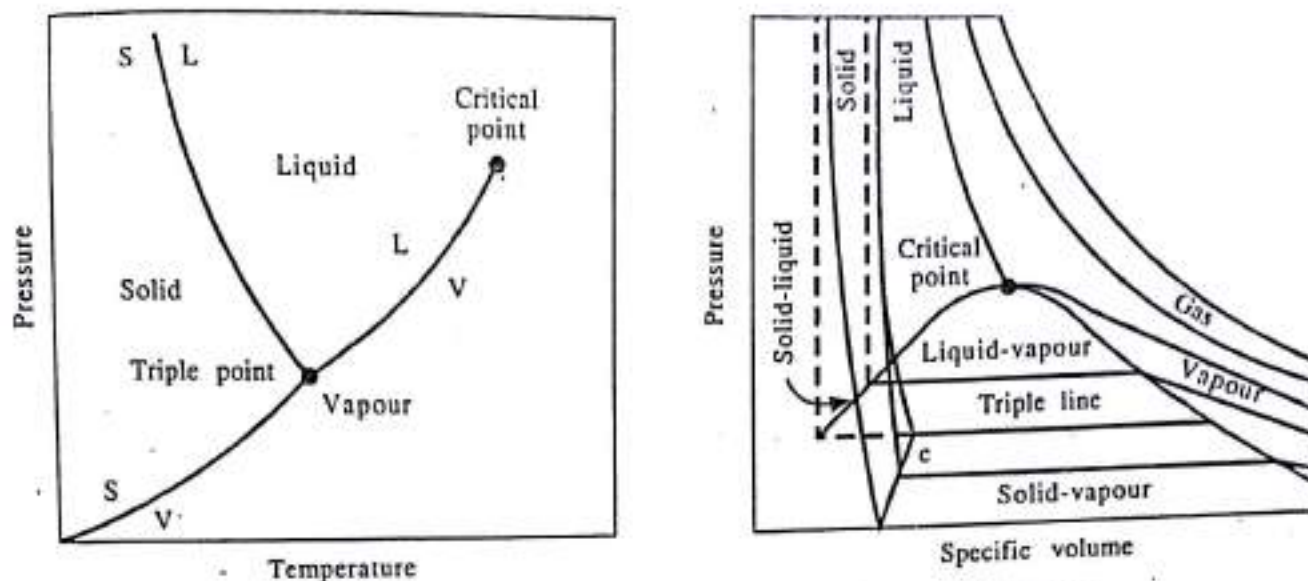


FIGURE 2.12 The pressure-volume-temperature relationship.

single-phase regions are the two-phase regions, where the two phases exist in equilibrium, i.e. liquid-vapour, solid-liquid, and solid-vapour. In these regions the state cannot be fixed by temperature and pressure alone; however, the state can be fixed by specific volume and either pressure or temperature. Three phases can exist in equilibrium along the line labelled triple line. At the top of the dome, where the saturated liquid and saturated vapour lines meet, is the critical point. The critical temperature of a pure substance is the maximum temperature at which liquid and vapour phases can coexist in equilibrium.

Properties of a perfect gas

Two regions are important in considering the properties of a perfect gas.

- The region of high temperature, i.e. the temperature above the critical temperature.
- The region of low pressure, i.e. the pressure of one atmosphere or less.

The permanent gases oxygen, hydrogen, etc. lie in the first region at ordinary working temperature. The water vapour in the atmosphere lies in the second region. The behaviour of substances in these two regions of states is very close to that of perfect gas. The perfect gas can easily be represented by simple equations relating primary properties, i.e. pressure, specific volume, and temperature.

The relation between pressure, specific volume and temperature or the equation of state was first developed by Robert Boyle. It is often known as Boyle's law. According to Boyle's law, the product pV remains constant when varying the pressure and volume of a gas while keeping the temperature constant. At low pressure (near to zero) the relation (pV/T) equals to R (Universal gas constant). Mathematically the equation of state can be expressed as:

$$\frac{pV}{T} = R \quad (2.10)$$

Taking mass into account, the equation of state of a perfect gas becomes

$$pV = mRT \quad (2.11)$$

2.3 HEAT TRANSFER

From thermodynamics we can calculate the amount of energy required to change a system from one equilibrium state to another equilibrium state since thermodynamics deals with systems in equilibrium. But if we want to know the rate of heat transfer or how fast a system will change from one equilibrium state to another then we have to know the subject of heat transfer. Heat transfer occurs by three basic mechanisms, i.e. conduction, convection and radiation.

Conduction

When heat transfer occurs through a substance without any motion of the substance, then the mode of heat transfer is called *conduction*. Like thermodynamics, the equation representing the rate of conduction is established from physical phenomena and from experiments. Fourier presented a mathematical model for the conduction phenomena as follows:

$$q = -kA \frac{\partial T}{\partial l} \quad (2.12)$$

where

q is the heat transfer rate

k is the thermal conductivity

A is the cross-sectional area

$\partial T / \partial l$ is the temperature gradient in the x -direction.

The same has been illustrated in Figure 2.13

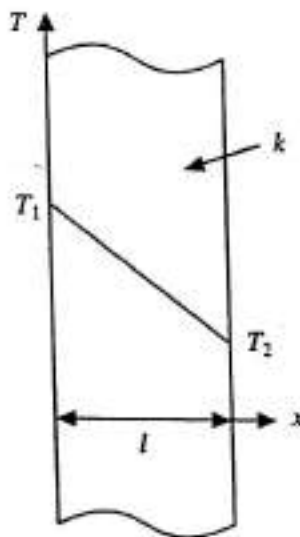


FIGURE 2.13 Representation of Fourier principle.

Equation (2.12) is negative because heat must flow downside according to the second law of thermodynamics (Figure 2.13). *Thermal conductivity* is an important property of a material. Thermal conductivity gives us a macroscopic representation of molecular effects of substances that contribute to the conduction of heat. From the Fourier equation, thermal conductivity can be determined for various substances. The thermal conductivity of a substance gives us the knowledge of how fast heat will be conducted through that substance. Table 2.2 shows the thermal conductivity of some materials.

TABLE 2.2 Thermal conductivity of various metals at 0°C

Material	Thermal conductivity (W/m·K)
Silver	410
Copper	385
Aluminium	202
Carbon steel	43
Sandstone	1.83
Glass	0.78
Ice	2.22
Water	0.556
Ammonia	0.540
Air	0.024
Water vapour	0.0206

Convection

If heat transfer through a substance is due to the movements of the substance, then it is called convection heat transfer. Convection is divided into two parts—natural convection and forced convection. Natural convection is caused by the density differences resulting from the process of heat transfer. But in forced convection external devices are used to increase the convection phenomena. The equation for representing convective heat transfer also came from the observation of physical phenomena. Heat transfer per square area by convection is observed to be proportional to the temperature difference. It can be expressed as follows.

$$\frac{Q_c}{A} \propto (T_w - T_a) \quad (2.13)$$

In Figure 2.14, T_w is the uniform wall temperature of a heated plate. The uniform flow velocity is U_a and the ambient fluid temperature is T_a . If we introduce the proportionality constant then the equation will be

$$\frac{Q_c}{A} = \bar{h}_c (T_w - T_a) \quad (2.14)$$

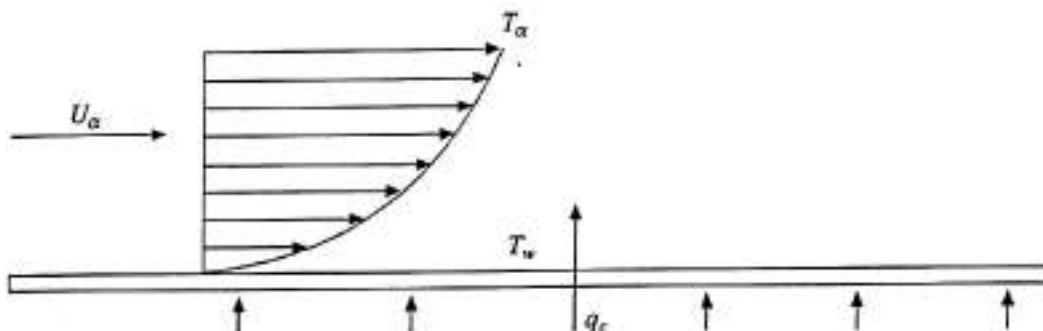


FIGURE 2.14 Convection heat transfer.

where \bar{h}_c is the average convection heat transfer coefficient. It is also called the *film conductance*. The above equation is called the Newton's law of cooling. For some systems the coefficient can be calculated analytically but for complex cases it must be determined empirically.

Radiation

Heat can also be transferred without any medium, i.e. in the vacuum. This phenomenon is described as electromagnetic radiation. There are many types of electromagnetic radiations, for example, X-rays, gamma rays, visible spectrum, etc. The net heat transfer by radiation phenomenon occurs due to temperature difference. To demonstrate the heat transfer rate by radiation, it is useful to introduce a substance that emits radiation ideally, i.e. a blackbody. The radiation heat transfer rate for a blackbody is proportional to the fourth power of its absolute temperature, i.e.

$$\frac{Q_r}{A} \propto T^4 \quad (2.15)$$

Introducing a proportionality constant, the equation becomes

$$Q_r = \sigma AT^4 \quad (2.16)$$

where σ is called the Stefan-Boltzmann constant and its value is

$$\sigma = 5.67 \times 10^{-8} \text{ W/m}^2\text{K}^4$$

But most real material surfaces do not emit electromagnetic radiation ideally. And these materials are called gray substances. For gray substances a new dimensionless factor called *emissivity* is introduced; emissivity for an ideal blackbody is 1 and but for a gray body it is always less than unity. There is another important factor called the *shape factor*. The shape factor tells us how much energy will leave from one surface and how much energy will arrive directly on the other surface. Like emissivity, the shape factor is also dimensionless and less than unity. Now introducing these two new factors (emissivity and shape factor), the net radiant exchange between two surfaces at temperatures T_1 and T_2 respectively can be expressed as

$$Q_r = \sigma A_1 F_{1-2} (T_1^4 - T_2^4) \quad (2.17)$$

where A_1 and F_{1-2} are the area of the body and the shape factor respectively.

Thermal circuit

Thermal circuit or network is a useful concept for the analysis of heat transfer problems. In the thermal circuit concept, equations of thermal resistance are introduced for different modes of heat transfer. For conduction, the heat transfer equation can be written as

$$Q_c = \frac{T_1 - T_2}{l/kA} \quad (2.18)$$

In analogy with electronic circuits, $(T_1 - T_2)$ can be considered as the potential that causes heat to flow against the resistance l/kA . So for conduction, the thermal resistance is

$$R_k = \frac{l}{kA} \quad (2.19)$$

The Newton's law of cooling, or the convection heat transfer equation, can be rearranged as follows:

$$Q_c = \frac{T_w - T_a}{1/(\bar{h}_c A)} \quad (2.20)$$

The thermal resistance for convection heat transfer is, therefore, given by

$$R_c = \frac{1}{h_c A} \quad (2.21)$$

Overall heat transfer coefficient

For the case of combined heat transfer the term overall heat transfer coefficient is very important. For example, if convection and conduction is present then the overall heat transfer will be the ratio of the overall temperature difference to the sum of the thermal resistances, i.e.

$$Q = \frac{T_1 - T_2}{(1/hA) + (l/kA)} \quad (2.22)$$

where $1/hA$ represents the thermal resistance for convection and l/kA represents the thermal resistance for conduction. The overall heat transfer coefficient is denoted by U and expressed by the following relation

$$Q = UA\Delta T \quad (2.23)$$

where

$$U = \frac{1}{(1/hA) + (l/kA)} \quad (2.24)$$

Materials in series

For the materials placed in series like in Figure 2.15, the overall heat transfer will be the ratio of the overall temperature difference and the sum of the thermal resistances, i.e.

$$Q = \frac{T_0 - T_2}{\frac{l_1}{k_1 A} + \frac{l_2}{k_2 A}} \quad (2.25)$$

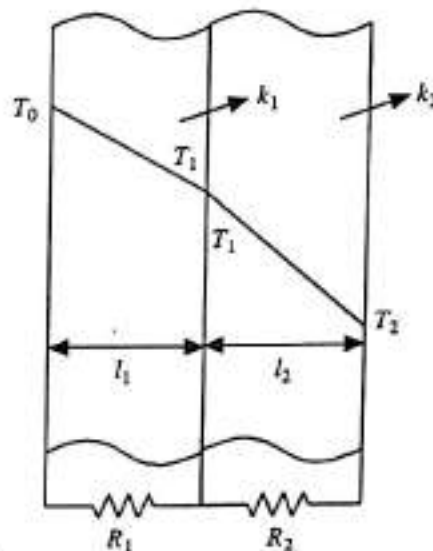


FIGURE 2.15 Materials in series arrangement.

Materials in parallel

For the materials placed in parallel like in Figure 2.16, the overall heat transfer will be

$$Q = \frac{T_0 - T_2}{\frac{l_1}{k_1 A_1} + \frac{l_2}{k_{2a} A_{2a} + k_{2b} A_{2b}}} \quad (2.26)$$

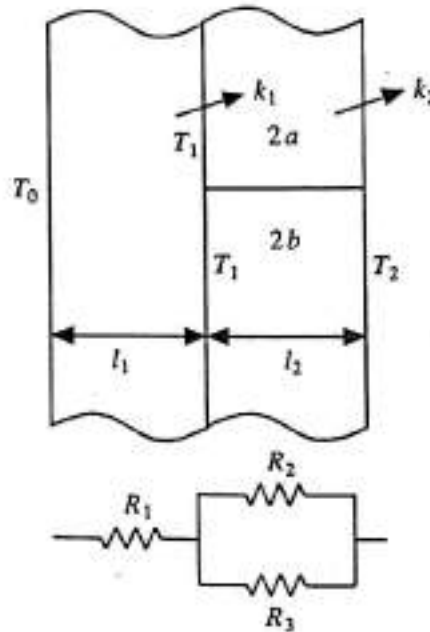


FIGURE 2.16 Materials in parallel arrangement.

Heat exchanger

Heat exchanger is a device which transfers heat between two fluids. Based on the mode of heat transfer, heat exchangers are classified as follows:

- Regenerators**, where a hot fluid, then a cold fluid, flow alternately through the same space containing solid particles. There are two types of regenerators—fixed and rotary. Heat transfer is affected by the type of surface material, fluid type, and system geometry.
- Open type exchangers**, where fluid streams flow into an open chamber in which complete physical mixing takes place. Examples include cooling towers and jet condenser.
- Closed type exchanger (recuperators)**, in which heat transfer occurs between two fluid streams that do not mix or come in physical contact with each other.

2.4 FLUID MECHANICS

Fluid mechanics is a field of applied mechanics and discusses the behaviour of liquids and gases at rest or in motion. For any thermal system, fluid is necessary for the transfer of energy. A fluid is defined as a substance that deforms continuously when acted on by any magnitude of shearing stress. In refrigeration and air conditioning, we also use different kinds of fluids for transfer of energy.



Viscosity is the most important property of a fluid and shows the resistance to being moved from place to place. Consider a situation as shown in Figure 2.17 where a fluid is flowing between two parallel plates.

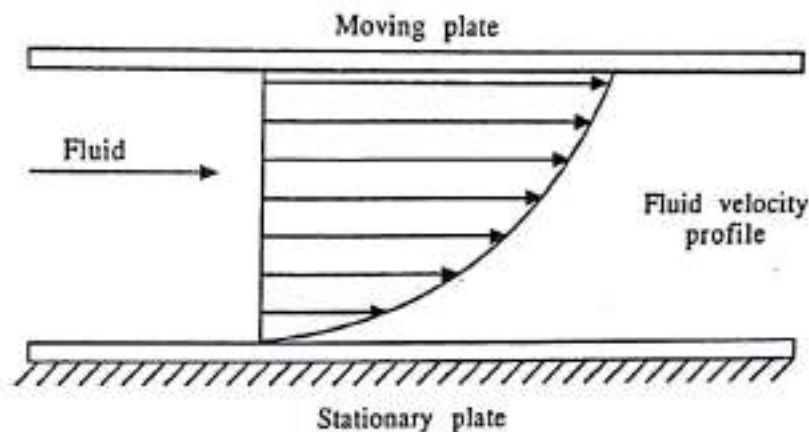


FIGURE 2.17 Fluid flow between two parallel plates.

Here the upper plate is moving with a velocity \bar{v} and the lower plate is stationary. Physically it can be understood that fluid molecules will face resistance adjacent to the lower stationary surface. While the upper plate is moving, so it creates a force on the lower layer of the fluid along the cross-section. The shear stress developed by the fluid is given by

$$\tau \propto \frac{d\bar{v}}{dy} \quad (2.27)$$

If we introduce a constant then the equation looks like the following.

$$\tau = \mu \frac{d\bar{v}}{dy} \quad (2.28)$$

The proportionality constant μ is called the *dynamic viscosity* (units kg/m.s). There is another form of viscosity called the *kinematic viscosity*, represented by ν (units m²/s). The relationship between dynamic and kinematic viscosities is expressed by

$$\nu = \frac{\mu}{\rho} \quad (2.29)$$

where ρ is the density.

In air conditioning and refrigeration the main application of fluid mechanics is in the modelling of piping and duct systems. The Bernoulli equation which is widely used for relating pressure, velocity and elevation of a fluid is significant in designing piping and ducts of thermal systems. For steady frictionless and incompressible flow, the Bernoulli equation is stated as follows:

Pressure head + Velocity head + Elevation head = Constant

$$\text{i.e.} \quad \frac{p_2 - p_1}{\rho} + \frac{\bar{v}_2^2 - \bar{v}_1^2}{2} + g(z_2 - z_1) = 0 \quad (2.30)$$

$$\text{or} \quad \frac{p_2}{\rho} + \frac{\bar{v}_2^2}{2} + gz_2 = \frac{p_1}{\rho} + \frac{\bar{v}_1^2}{2} + gz_1 = \text{Constant} \quad (2.31)$$

If we consider a flow system where a fluid flows through a pipe with work and friction, then the equation will be

$$\frac{p_1}{\rho} + \frac{\bar{v}_1^2}{2} + gz_1 = \frac{p_2}{\rho} + \frac{\bar{v}_2^2}{2} + gz_2 + \text{Work} + \text{Loss} = \text{Constant} \quad (2.32)$$

The last term in LHS of Eq. (2.32) is also termed *head loss* and can be expressed with a dimensionless parameter f which is called the *Darcy friction factor*. Thus,

$$\text{Head loss} = f \frac{L\bar{v}^2}{d2g} \quad (2.33)$$

where

f is the friction factor

L is the distance along the duct

d is the inner diameter

\bar{v} is the mean velocity.

The friction factor is determined from the flow Reynolds number and the relative roughness.

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3

Vapour Compression Cycle and Heat Pump

SECTION A: VAPOUR COMPRESSION CYCLES

Although different types of refrigeration systems may be employed to carry out the cooling and dehumidification processes, two methods of refrigeration currently in common use in majority of the systems are *mechanical vapour compression cycle* and *vapour absorption cycle*. Up to the present time, the use of other cycles such as thermo-electric refrigeration, air and gas expansion refrigeration, vortex tube, steam jet refrigeration cycles has been limited more to such special applications as in air conditioning of aircraft, ships, submarines, and mines, etc.

3.1 REFRIGERATOR AND HEAT PUMP

In the mechanical vapour compression cycle the specific working fluid used, known as refrigerant, is compressed by a compressor. The compressed vapour is then condensed to a liquid, following which the pressure is reduced through throttling so that the fluid can evaporate at a low pressure. There are two main applications of the vapour compression cycle, namely *refrigerators* and *heat pumps*. They are essentially the same device differing in their use and objectives. When the cooling produced by the cycle by removing heat from a cold space is used, for example, to cool air in an air-conditioning application, as shown in Figure 3.1(a), the device used is termed refrigerator. When the heat rejected by the refrigeration cycle is gainfully used such as for providing winter heating, as shown in Figure 3.1(b), the device used is known as heat pump.

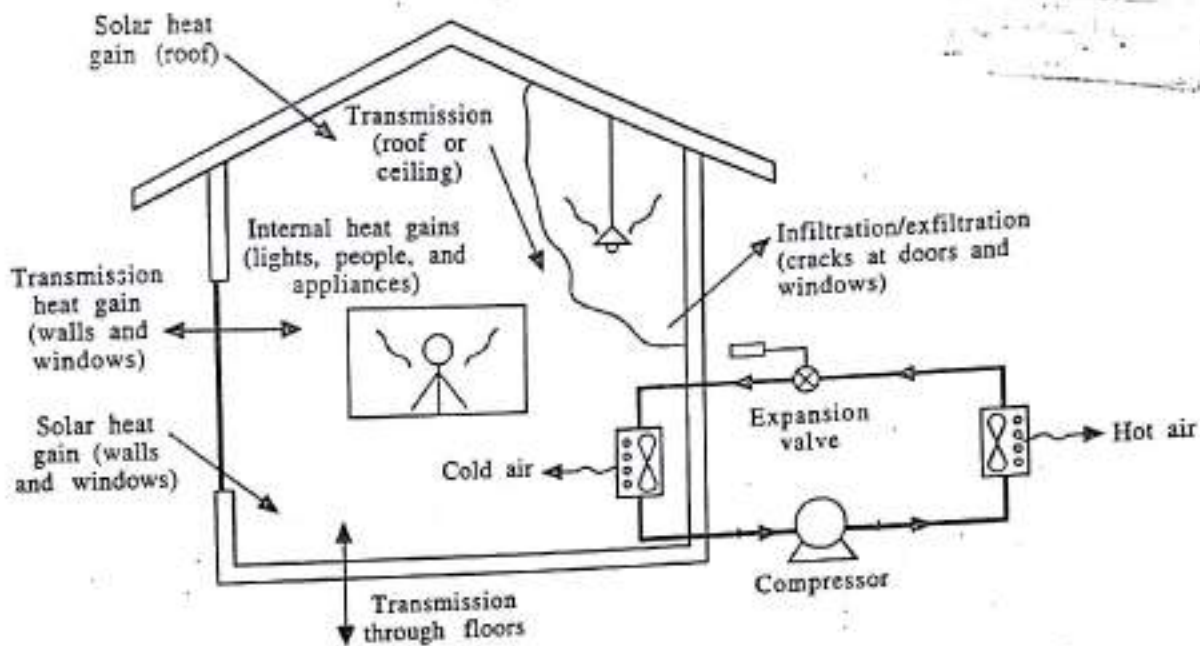


FIGURE 3.1(a) Refrigerator mode of the vapour compression cycle.

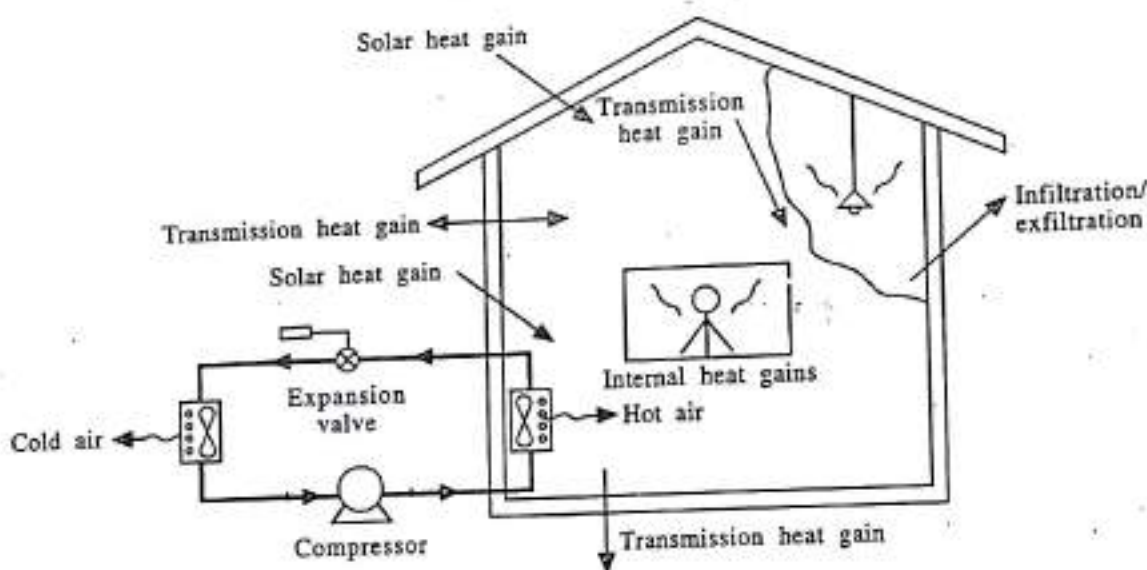


FIGURE 3.1(b) Heat pump mode of the vapour compression cycle.

In Figure 3.2, the source and sink temperatures of both the refrigerator and heat pump applications have been depicted in relation to ambient temperature. In the refrigerator mode, Q_L is the amount of heat removed from the refrigerated space at temperature T_L . Q_H is the amount of heat rejected to the relatively warm ambient environment at temperature T_{ambient} and W_{in} is the net work input to the refrigerator. In the case of heat pump, Q_H , generally rejected to atmosphere is being utilized for space heating, while the cooling produced is unutilized.

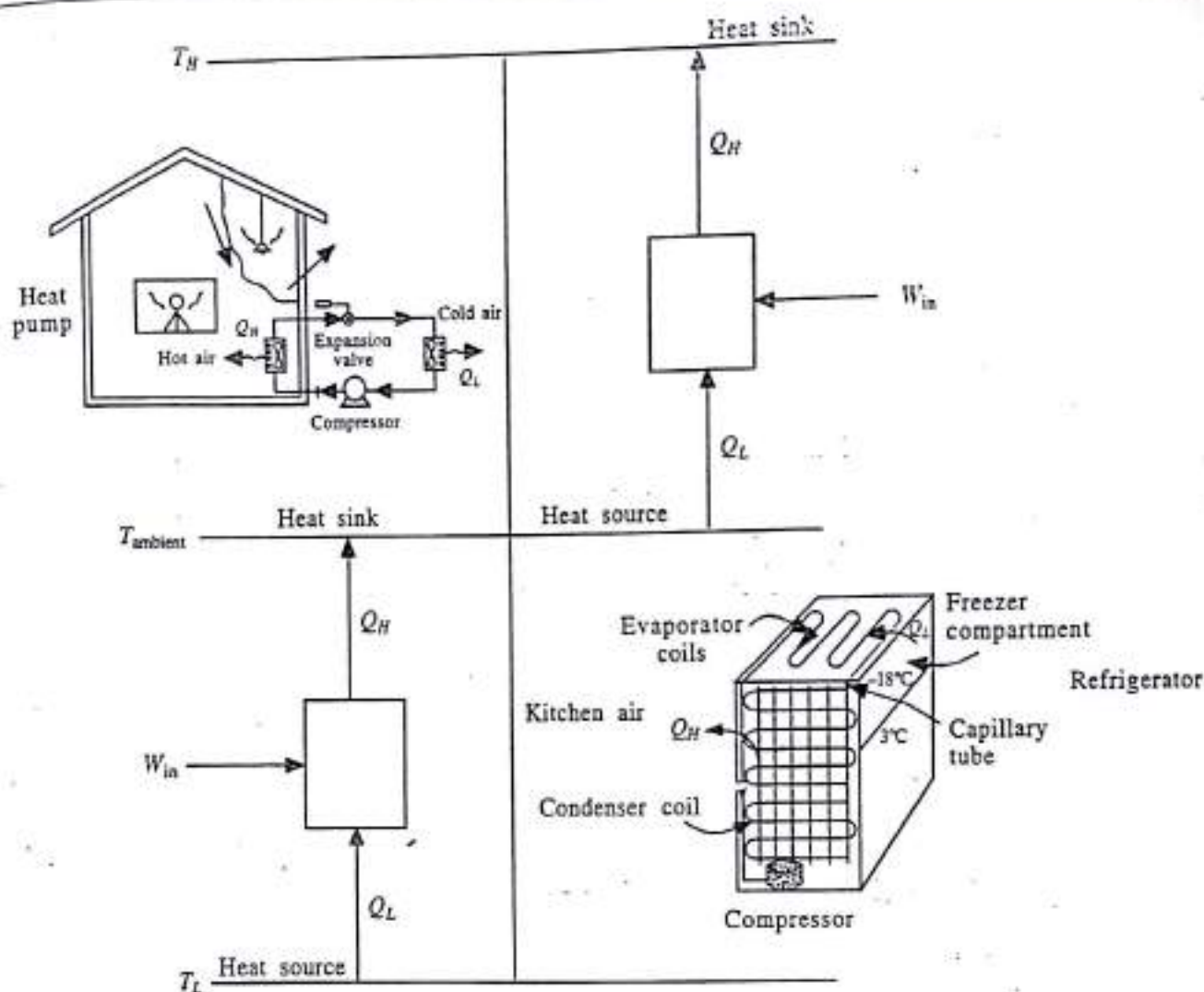


FIGURE 3.2 Relationship between refrigerator and heat pump modes.

3.2 BASIC CYCLE THEORY

The theoretical basis of all practical refrigeration systems is the reversed Carnot cycle. First the Carnot cycle will be introduced, followed by the discussion on the reversed Carnot cycle. The Carnot cycle is a reversible process operating between two given temperatures as shown in Figure 3.3.

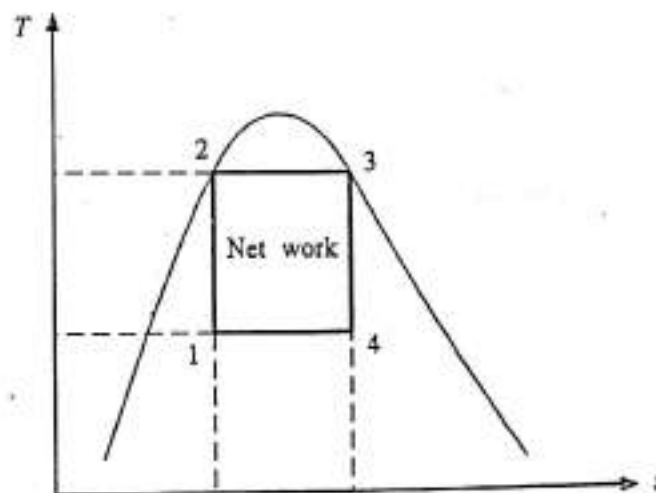


FIGURE 3.3 The $T-s$ diagram of Carnot cycle.

The processes of the Carnot cycle are:

- 1-2 Isentropic compression
- 2-3 Isothermal heat addition
- 3-4 Isentropic expansion
- 4-1 Isothermal heat rejection

The efficiency of the Carnot cycle is the highest among cycles operating between two given temperatures. This is because all the processes of the Carnot cycle are reversible, that is, impossible to attain in actual cycles.

A reversed Carnot cycle is illustrated in Figure 3.4. The cycle consists of the following processes:

- 1-2 Isothermal heat addition at lower temperature
- 2-3 Isentropic compression to higher temperature
- 3-4 Isothermal heat rejection at higher temperature
- 4-1 Isentropic expansion to lower temperature

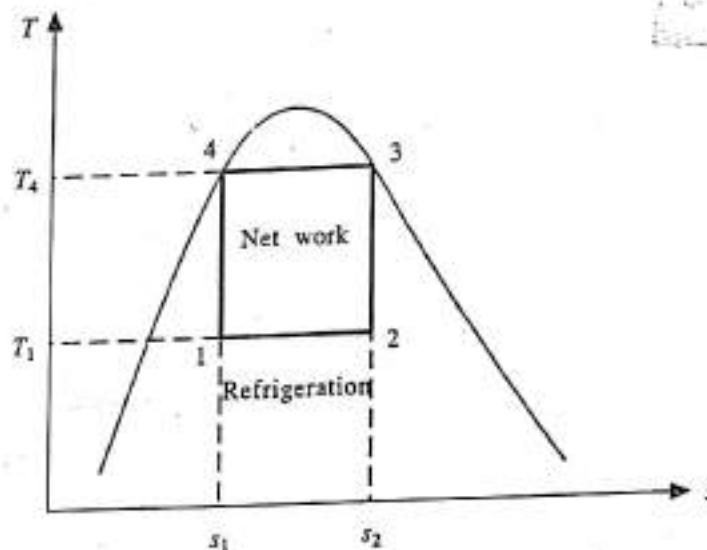


FIGURE 3.4 The T - s diagram of reversed Carnot cycle.

Heat absorbed by the refrigerant in process 1-2 is the heat removed from the space or material being cooled and is called the *refrigerating effect* (RE). The index of performance of a refrigeration cycle is the *coefficient of performance* (COP) defined as the dimensionless ratio of the refrigerating effect and the work energy required for the same, i.e.

$$\text{COP}_{\text{rev}} = \text{COP}_{\text{Carnot}} = \frac{\text{refrigerating effect}}{\text{net work energy}}$$

Therefore,

$$\begin{aligned} \text{COP}_{\text{Carnot}} &= \frac{T_1(s_2 - s_1)}{(T_4 - T_1)(s_2 - s_1)} \\ &= \frac{T_1}{T_4 - T_1} \end{aligned}$$

The COP of the Carnot cycle is entirely a function of the temperature limits and can vary from zero to infinity. Since all the processes in the Carnot cycle are thermodynamically reversible, the COP obtained from the above equation is higher than that achievable in an actual cycle and is the maximum possible for a given temperature limit. Hence, this value of COP can be used as a yardstick for the performance of an actual cycle. In the case of irreversible cycle, the COP is less than that for the reversible one, i.e.

$$\text{COP}_{\text{irrev}} < \frac{T_1}{T_4 - T_1}$$

The COPs of most practical vapour compression heat pump cycles usually have a value about half of that of Carnot COP.

3.3 IDEAL VAPOUR COMPRESSION REFRIGERATION CYCLE

The mechanical vapour compression cycle utilizes the evaporation of a liquid refrigerant to absorb heat and lower the temperature of its surroundings at the evaporator. The simple cycle, as shown in Figure 3.5, comprises four main components: compressor, condenser, evaporator, and the expansion device.

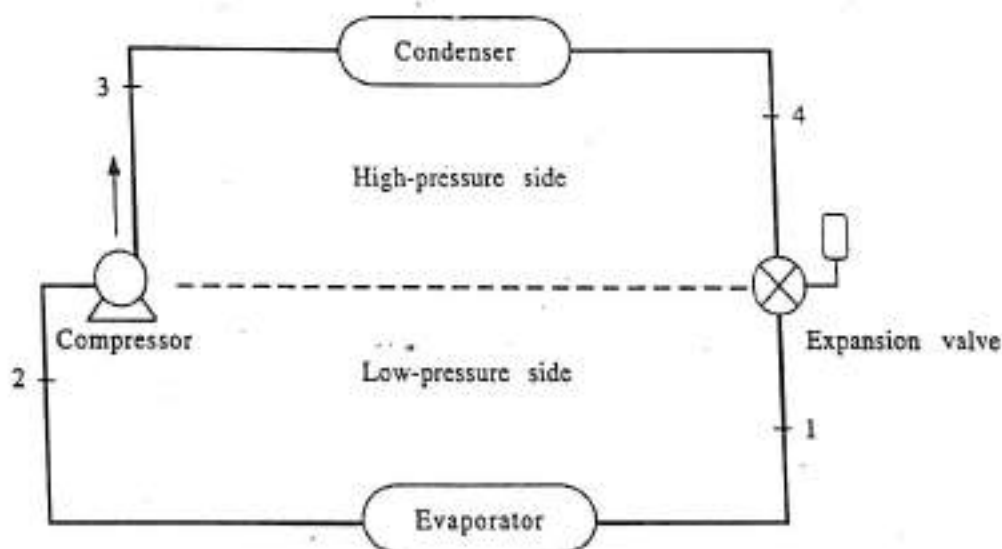


FIGURE 3.5 Simple vapour compression cycle.

The vapour compression refrigeration cycle operates by continuously changing the physical properties of a fluid through a cycle, as illustrated in Figures 3.5 and 3.6. The processes constituting the standard vapour compression cycle are:

- 1-2 Isothermal heat absorption of low temperature heat energy in the evaporator
- 2-3 Isentropic compression by the compressor to condenser pressure
- 3-4 Heat rejection through constant pressure cooling in the condenser
- 4-1 Adiabatic expansion through a throttling device which meters the flow of refrigerant to the evaporator in response to the load.

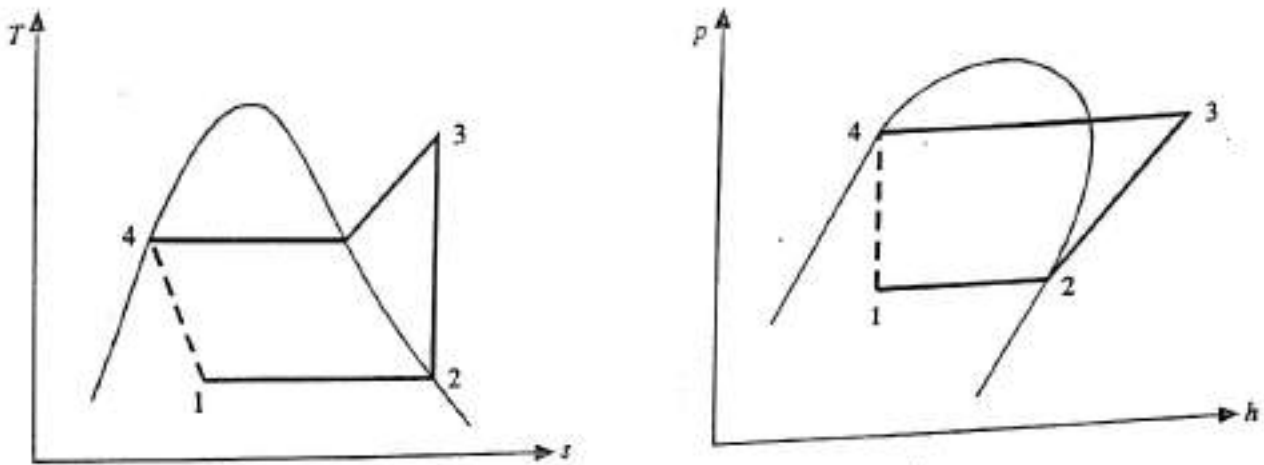


FIGURE 3.6 Ideal vapour compression cycle.

The working substance i.e. the refrigerant in the same refrigeration system, enters the compressor as saturated vapour at state 2 which is then compressed isentropically to state 3. The refrigerant then enters the condenser at state 3 in superheated condition and leaves as saturated liquid at state 4 following heat rejection to the surroundings. Hot refrigerant vapour is thus liquefied after rejecting the heat to the condensing media such as air or water. The saturated liquid at state 4 then expands to evaporator pressure at state 1 through a throttling device such as an expansion valve or capillary tube. Consequent to the adiabatic expansion, the refrigerant temperature drops below the ambient temperature, which is a prerequisite of heat flow from the surroundings to the refrigerant in the evaporator. The refrigerant enters the evaporator at state 1 as wet vapour and leaves the evaporator at state 2 as dry saturated vapour and re-enters the compressor, thus completing the cycle.

From the analysis of the cycle, it is obtained that:

$$\text{Refrigerating effect (RE)} \quad Q_r = Q_{1-2} = \dot{m}_r(h_2 - h_1)$$

$$\text{Work input} \quad \dot{W} = \dot{W}_{2-3} = \dot{m}_r(h_3 - h_2)$$

$$\text{Heat rejection} \quad Q_c = Q_{3-4} = \dot{m}_r(h_3 - h_4)$$

and

$$\text{COP} = \frac{\dot{m}_r(h_2 - h_1)}{\dot{m}_r(h_3 - h_2)} = \frac{h_2 - h_1}{h_3 - h_2}$$

where

\dot{m}_r is the mass flow rate of the refrigerant

h_1 is the enthalpy of the refrigerant at the evaporator inlet

h_2 is the enthalpy of the refrigerant at the evaporator outlet and compressor inlet

h_3 is the enthalpy of the refrigerant at the compressor outlet and condenser inlet

h_4 is the enthalpy of the refrigerant at the condenser outlet.

The ability to use the latent heats of vaporization and condensation of a refrigerant enables a vapour compression cycle to obtain greater refrigeration effect per unit mass of refrigerant.

In choosing the evaporator and condenser temperatures, it has to be ensured that the temperature of the refrigerant in the condenser is higher than that of the surrounding to effect heat rejection. Similarly, the evaporator temperature would have to be lower than the required cooling temperature (Figure 3.7). A temperature differential of around 5 to 7°C is commonly practised.

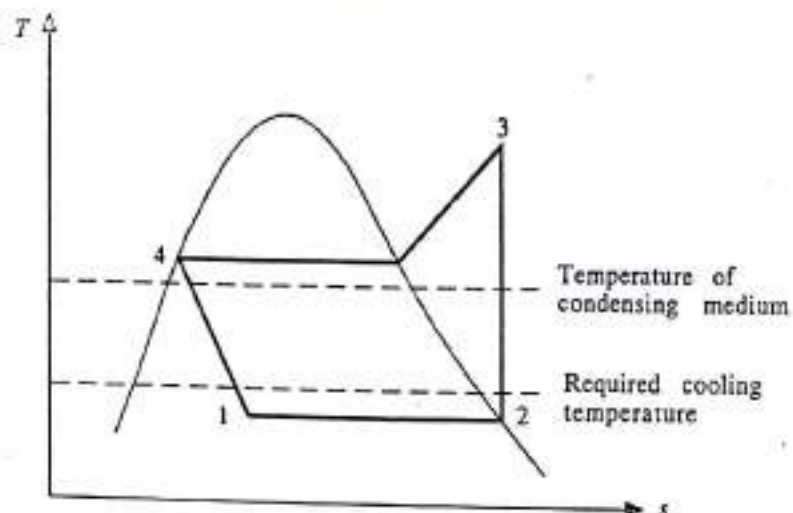


FIGURE 3.7 Evaporator and condenser pressure limitations in practical cycles.

3.4 PRACTICAL VAPOUR COMPRESSION REFRIGERATION CYCLE

In practical systems, however, the refrigerant follows the modified Rankine cycle, shown in Figures 3.7 and 3.8. The actual refrigeration cycle differs fundamentally from the ideal cycle in respect of pressure drops in refrigerant flow and heat transfer to or from the surroundings. There are other differences between this cycle and the theoretical cycle.

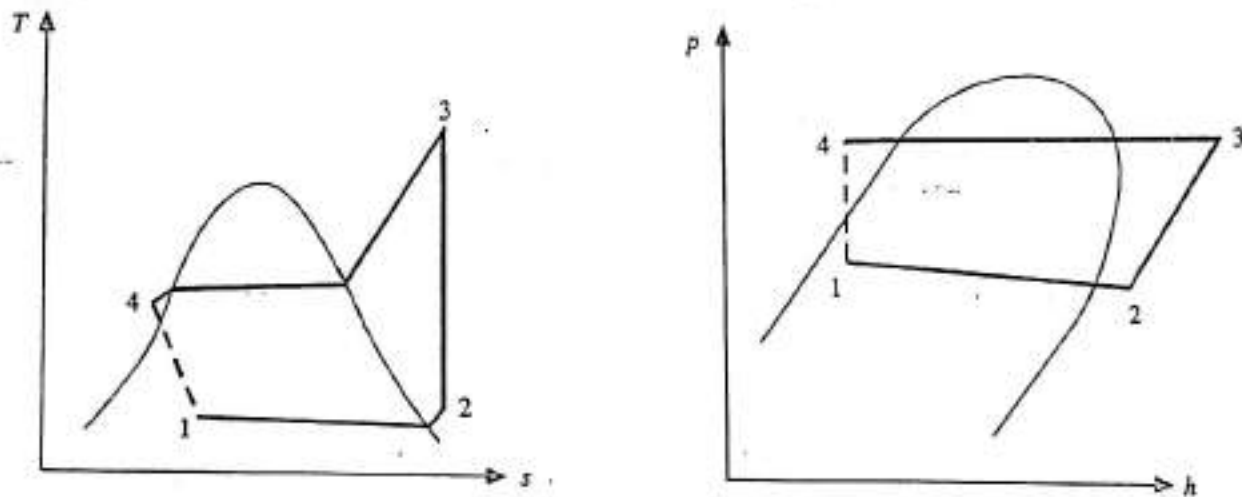


FIGURE 3.8 Simplified practical cycles.

Normally in the practical cycles the temperature of the liquid refrigerant leaving the condenser is lower than the saturation temperature, and this difference is termed *degree of subcooling*. Subcooling ensures that no vapour enters the expansion valve and furthermore it increases the refrigerating effect. The refrigerant is also superheated before leaving the evaporator to make sure that only the dry vapour will enter the compressor. Because of the extremely small clearance between the piston and cylinder head, it is necessary to ensure that there is no liquid in the gas entering the compressor as that may cause damage to the piston. The compression is not isentropic and the expansion is close to isenthalpic. During the compression process there are

irreversibilities and heat transfer either to or from the surroundings. The heat transfer mode depends on the temperature of the refrigerant and the surroundings. Consequently, the entropy might increase or decrease as represented by 1-2' and 1-2 lines in Figure 3.9. Due to friction the pressure of the liquid leaving the condenser will be less than the pressure of the vapour entering it. There is also some pressure drop in the evaporator. Being an irreversible cycle, practical cycles have a lower COP than the reversed Carnot cycle when operating between the same temperature limits.

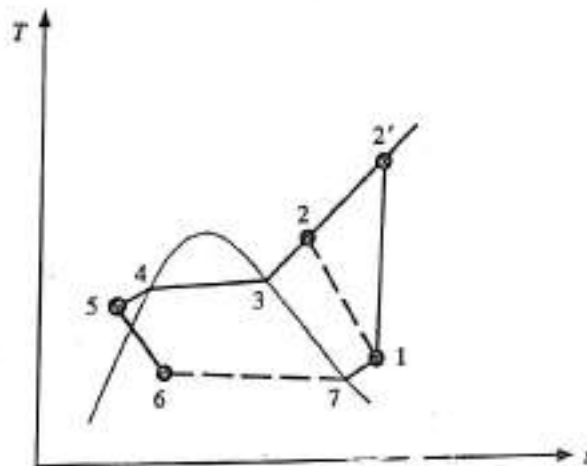


FIGURE 3.9 Actual cycle details.

Figure 3.10 shows a practical vapour compression refrigeration cycle with typical operating conditions.

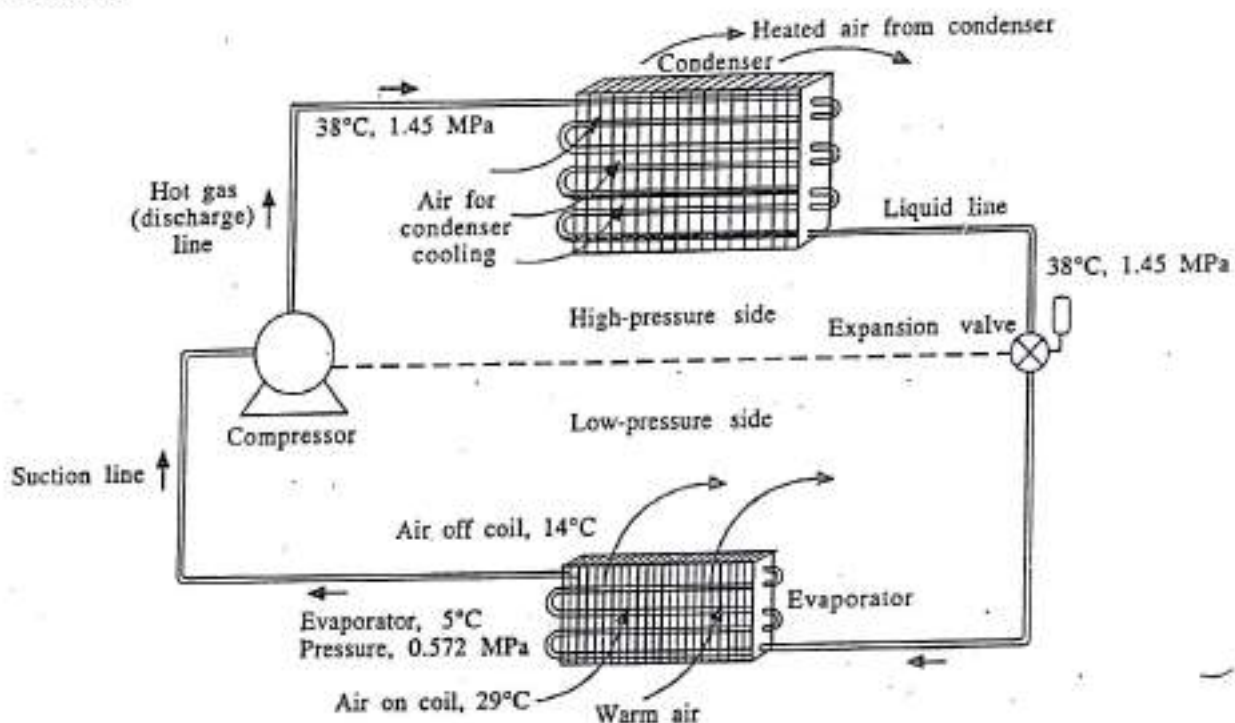


FIGURE 3.10 Schematic diagram of a refrigeration cycle showing typical temperatures and pressures for a summer cooling application.

The capacity of a refrigeration cycle can be measured in any one of several units such as Btu/h or kW or kcal/h. Another unit very commonly used is ton of refrigeration (TR), which is the amount of cooling produced by one US ton (2000 lb) of ice in melting over a period of 24 hours. Therefore,

$$1 \text{ TR} = \frac{1 \times 2000 \times 144}{24} = 12,000 \text{ Btu/h}$$

where the heat of fusion of ice is 144 Btu/lb.

1 TR is thus equal to 12,000 Btu/h or 200 Btu/min or 3.516 kW of useful heat removal.

Vapour compression cycle calculation

Basically in the calculation of performance of a vapour compression cycle, the equations given in Section 3.3 can be used. As regards the enthalpies at the various points in the system, the same can be obtained either from the Tables of the respective refrigerants or from the p - h diagrams. In a typical table of thermophysical properties of a refrigerant, for example that of refrigerant R134a (see the Appendix B), the enthalpy values of the refrigerant in saturated liquid and saturated vapour state are provided against a particular temperature/pressure. Additionally, the same tables also provide the values of specific heat, entropy, viscosity, thermal conductivity, etc. The properties of superheated vapour state can also be obtained from the tables of 'properties of superheated vapour'.

The same enthalpy values can also be obtained from the pressure-enthalpy (p - h) diagrams of the respective refrigerants. A skeleton diagram of a typical refrigerant is shown in Figure 3.11. By plotting the different points in the cycle from the known pressures and temperatures, the corresponding enthalpy values can be readily obtained.

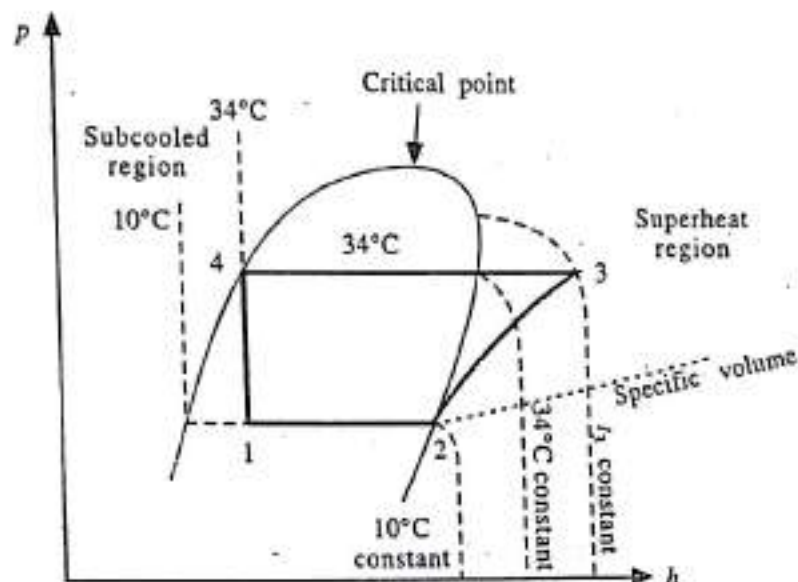


FIGURE 3.11 Skeleton pressure-enthalpy diagram.

The values obtained from the Tables are generally more accurate than those obtained from the p - h diagrams. However, practising engineers find it easier to use the p - h diagrams.

EXAMPLE 3.1

A vapour compression cycle using refrigerant R22 operates at condensing temperature of 34°C and evaporative temperature of -10°C as shown in Figure 3.12. For a mass flow rate of the refrigerant equal to 0.33 kg/s , determine the following:

- The compressor power
- The refrigerating effect
- The coefficient of performance.

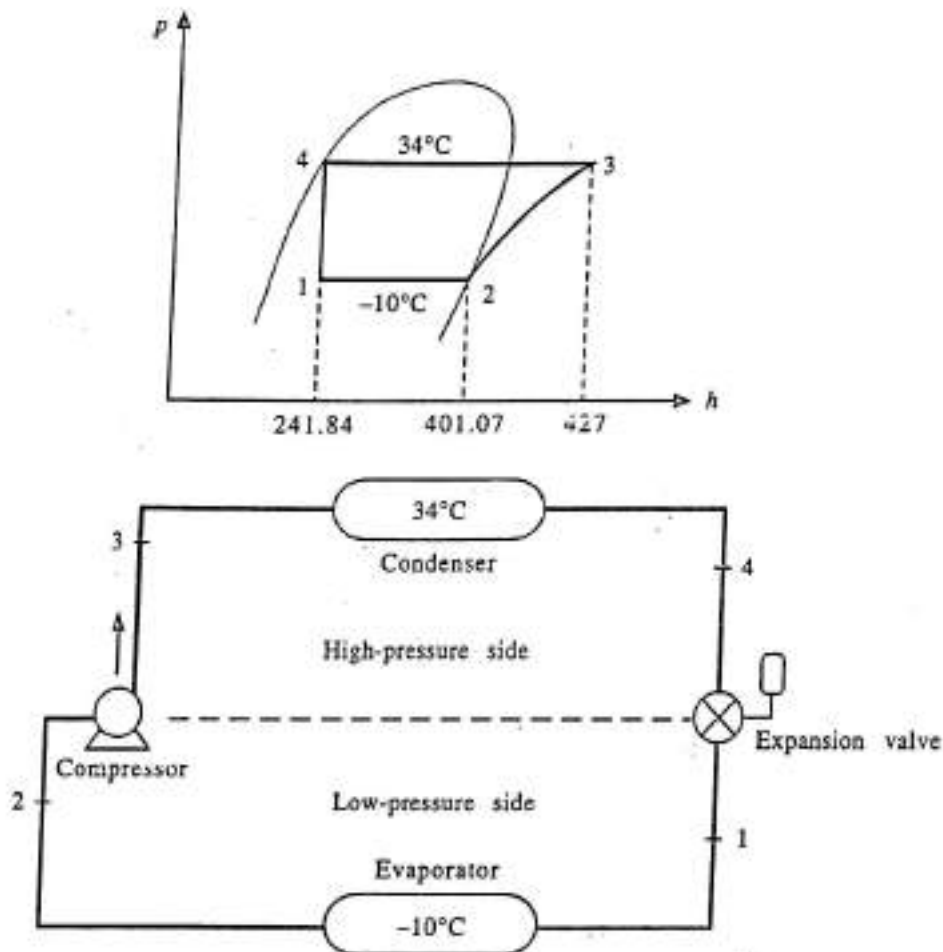


FIGURE 3.12 Example 3.1.

Solution

From the $p-h$ diagram (Figure 3.12) and tables of R22, we get

$$h_2 = 401.07 \text{ kJ/kg} \quad h_3 = 427 \text{ kJ/kg} \quad h_4 = 241.84 \text{ kJ/kg} = h_1$$

$$\text{Compressor work, } \dot{W} = \dot{m}(h_3 - h_2) = 0.33 \text{ kg/s} \times (427 - 401.07) \text{ kJ/kg} = 8.55 \text{ kW}$$

$$\text{Refrigerating capacity} = \dot{m}(h_2 - h_1) = 0.33(401.07 - 241.84) = 52.54 \text{ kW}$$

$$\text{Coefficient of performance, COP} = \frac{\text{RE}}{\dot{W}} = \frac{52.54}{8.55} = 6.14$$

EXAMPLE 3.2

A vapour compression cycle using refrigerant R22 operates at condensing temperature of 36°C and evaporative temperature of -16°C as shown in Figure 3.13. For a system capacity of 55 kW, determine the following:

- The mass flow rate
- The compressor power
- The refrigerating effect
- The coefficient of performance.

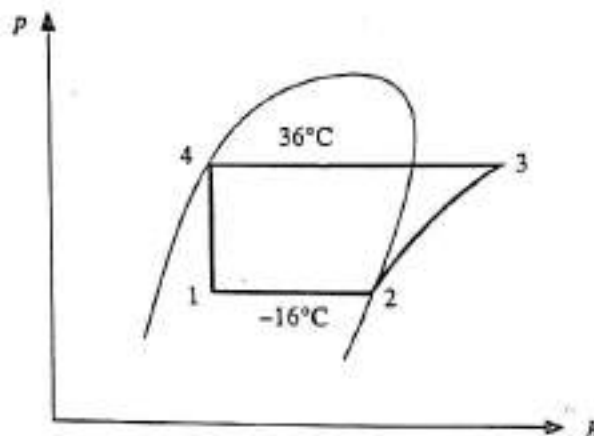


FIGURE 3.13 Example 3.2.

Solution

From the p - h diagram and tables of R22, we get

$$h_2 = 398.64 \text{ kJ/kg} \quad h_3 = 428 \text{ kJ/kg} \quad h_4 = 244.44 \text{ kJ/kg} = h_1$$

$$\text{Refrigerating capacity} = \dot{m}(h_2 - h_1) = \dot{m}(398.64 - 244.44) = 55 \text{ kW}$$

$$\text{Therefore, } \dot{m} = 0.36 \text{ kg/s}$$

$$\text{Compressor work, } \dot{W} = \dot{m}(h_3 - h_2) = 0.36 \text{ kg/s} \times (428 - 398.64) \text{ kJ/kg} = 10.57 \text{ kW}$$

$$\text{Coefficient of performance, COP} = \frac{\text{RE}}{\dot{W}} = \frac{55}{10.57} = 5.20$$

EXAMPLE 3.3

An ice plant operates on the ideal vapour compression cycle using refrigerant R134a. The refrigerant enters the compressor as saturated vapour at 0.15 MPa and leaves the condenser as saturated liquid at 0.7 MPa. Water enters the ice machine at 30°C and leaves as ice at -5°C . For an ice production rate of 10 kg per hour, determine the power input to the ice plant. The specific heats of ice and water are 2.1 and 4.18 kJ/kg·K, respectively, and the latent heat of fusion of ice is 334 kJ/kg.

Solution

Heat needed to be removed per kg of water to convert it to ice at -5°C is given by

$$Q = \dot{m}c_{\text{water}} \times (t_{\text{water}} - 0) + mL_f + mc_{\text{ice}} (0 - t_{\text{ice}})$$

where

m is the mass of water/ice (kg)

L_f is the latent heat of fusion of ice (kJ/kg)

$$\begin{aligned} \text{or } Q &= 1 \times 4.18 \times (30 - 0) + 334 + 1 \times 2.10 \times (0 - (-5)) \\ &= 125.4 + 334 + 10.5 = 469.9 \text{ kJ/kg} \end{aligned}$$

Therefore, to produce 10 kg/h of ice, the refrigerating capacity required is

$$\frac{10}{3600} \times 469.9 = 1.30 \text{ kW}$$

From the p - h diagram and tables of R134a, we get

$$h_2 = 389.11 \text{ kJ/kg} \quad h_3 = 425 \text{ kJ/kg} \quad h_4 = 236 \text{ kJ/kg} = h_1$$

$$\text{Refrigerating capacity} = \dot{m}(h_2 - h_1) = \dot{m}(389.11 - 236) = 1.30 \text{ kW}$$

$$\text{Therefore, } \dot{m} = 0.0084 \text{ kg/s}$$

$$\begin{aligned} \text{Compressor work, } \dot{W} &= \dot{m}(h_3 - h_2) = 0.0084 \text{ kg/s} \times (425 - 389.11) \text{ kJ/kg} \\ &= 0.301 \text{ kW} \end{aligned}$$

$$\text{Coefficient of performance, COP} = \frac{\text{RE}}{\dot{W}} = \frac{1.30}{0.301} = 4.31$$

EXAMPLE 3.4

A refrigerator using refrigerant R134a as a working fluid operates on an ideal vapour compression refrigeration cycle between 0.12 and 0.7 MPa as shown in Figure 3.14. The mass flow rate of the refrigerant is 0.06 kg/s. Show the cycle on a T - s diagram, and determine the following:

- The rate of heat removal of the refrigerated space and the power input to the compressor
- The rate of heat rejection to the environment
- The coefficient of performance.

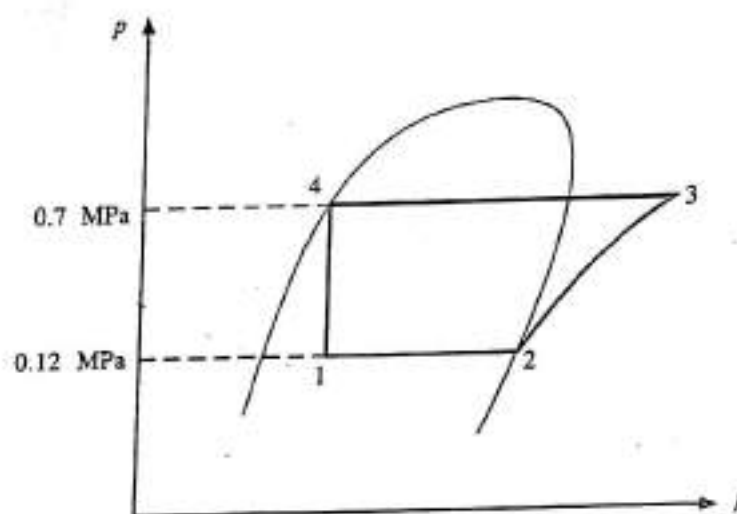


FIGURE 3.14 Example 3.4.

Solution

From the p - h diagram and tables of R134a, we get

$$h_2 = 385.43 \text{ kJ/kg} \quad h_3 = 428 \text{ kJ/kg} \quad h_4 = 236 \text{ kJ/kg} = h_1$$

$$\text{Refrigerating capacity} = \dot{m}(h_2 - h_1) = 0.06(385.43 - 236) = 8.96 \text{ kW}$$

$$\text{Compressor work, } \dot{W} = \dot{m}(h_3 - h_2) = 0.06 \text{ kg/s} \times (428 - 385.43) \text{ kJ/kg} \\ = 2.55 \text{ kW}$$

$$\text{Rate of heat rejection} = 8.96 + 2.55 = 11.51 \text{ kW}$$

$$\text{Coefficient of performance of the refrigerator, COP} = \frac{\text{RE}}{\dot{W}} = \frac{8.96}{2.55} = 3.51$$

EXAMPLE 3.5

In a refrigeration system, would you recommend condensing the refrigerant R134a at a pressure of 0.8 MPa or 1 MPa if heat is rejected to a cooling medium at 30°C?

Solution

Allowing a temperature difference of 7°C for effective heat transfer, the condensation temperature of the refrigerant should be 37°C or higher. The saturation temperature corresponding to 37°C is 0.937 MPa. Therefore, the recommended pressure would be 1 MPa.

EXAMPLE 3.6

A refrigeration system operating on a vapour compression cycle with refrigerant R134a as a working fluid has a cooling capacity of 10 kW. The refrigerant enters the compressor as saturated vapour at 0.14 MPa and is compressed isentropically to 1 MPa. The refrigerant leaves the condenser as saturated liquid and expands adiabatically through an expansion device. Determine the following:

- The quality of the refrigerant at the end of the throttling process
- The power input to the compressor
- The coefficient of performance.

Solution

From the p - h diagram (Figure 3.15) and tables of R134a, we get

$$h_2 = 387.89 \text{ kJ/kg} \quad h_3 = 435 \text{ kJ/kg} \quad h_4 = 256.35 \text{ kJ/kg} = h_1$$

- The quality (x) of the refrigerant at the end of the throttling process is

$$x = \frac{h_1 - h_f}{h_{fg}} = \frac{256.35 - 176.39}{211.5} = 0.37$$

where x , the quality or dryness fraction, denotes the fraction of dry refrigerant in the two-phase mixture at a particular point.

- Refrigerating capacity = $\dot{m}(h_2 - h_1) = \dot{m}(387.89 - 256.35) = 10 \text{ kW}$
or mass flow rate, $\dot{m} = 0.076 \text{ kg/s}$

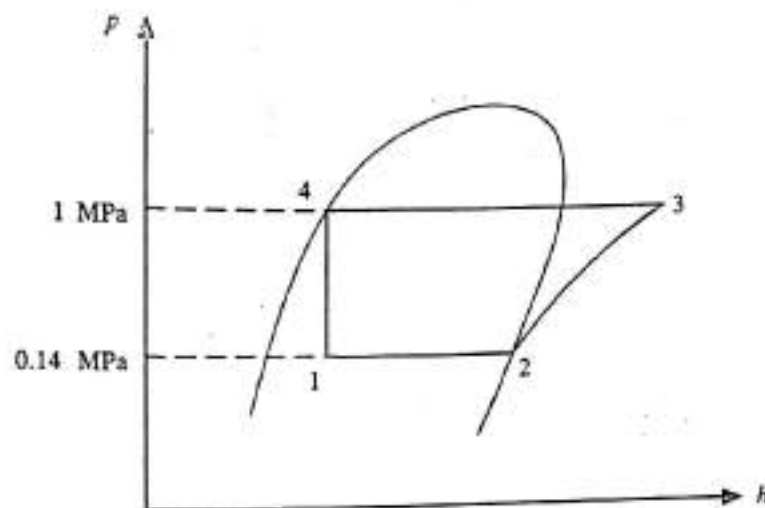


FIGURE 3.15 Example 3.6.

$$\text{Compressor work, } \dot{W} = \dot{m}(h_3 - h_2) = 0.076 \text{ kg/s} \times (435 - 387.89) \text{ kJ/kg} = 3.58 \text{ kW}$$

(c) Coefficient of performance of the refrigerator, $\text{COP} = \frac{\text{RE}}{\dot{W}} = \frac{10}{3.58} = 2.79$

EXAMPLE 3.7

Refrigerant R134a enters the compressor of a refrigerator as superheated vapour at 0.14 MPa and -12°C at a rate of 0.076 kg/s and leaves at 1 MPa and 70°C . The refrigerant is cooled in the condenser to 36°C and 1 MPa and is throttled to 0.15 MPa. Ignoring any heat transfer and pressure drop in the connecting lines between the components, determine the following:

- The rate of heat removal from the refrigerated space
- The power input to the compressor
- The isentropic efficiency of the compressor
- The coefficient of performance.

Solution

The enthalpies of the refrigerant at various states are determined from the refrigerant tables and/or p - h diagrams (Figure 3.16) of R134a

$$\begin{aligned} p_2 &= 0.14 \text{ MPa,} & T_2 &= -12^\circ\text{C,} & h_2 &= 398 \text{ kJ/kg} \\ p_3 &= 1 \text{ MPa,} & T_3 &= 70^\circ\text{C,} & h_3 &= 450 \text{ kJ/kg} \\ p_4 &= 1 \text{ MPa,} & T_4 &= 36^\circ\text{C,} & h_4 &= 255 \text{ kJ/kg} \\ h_4 &\equiv h_1(\text{throttling}) & h_1 &= 255 \text{ kJ/kg} \end{aligned}$$

- (a) Heat removal from the refrigerated space

$$\dot{m}(h_2 - h_1) = 0.076(398 - 255) = 10.86 \text{ kW}$$

- (b) Compressor work, $\dot{W} = \dot{m}(h_3 - h_2) = 0.076 \text{ kg/s} \times (450 - 398) \text{ kJ/kg} = 3.95 \text{ kW}$

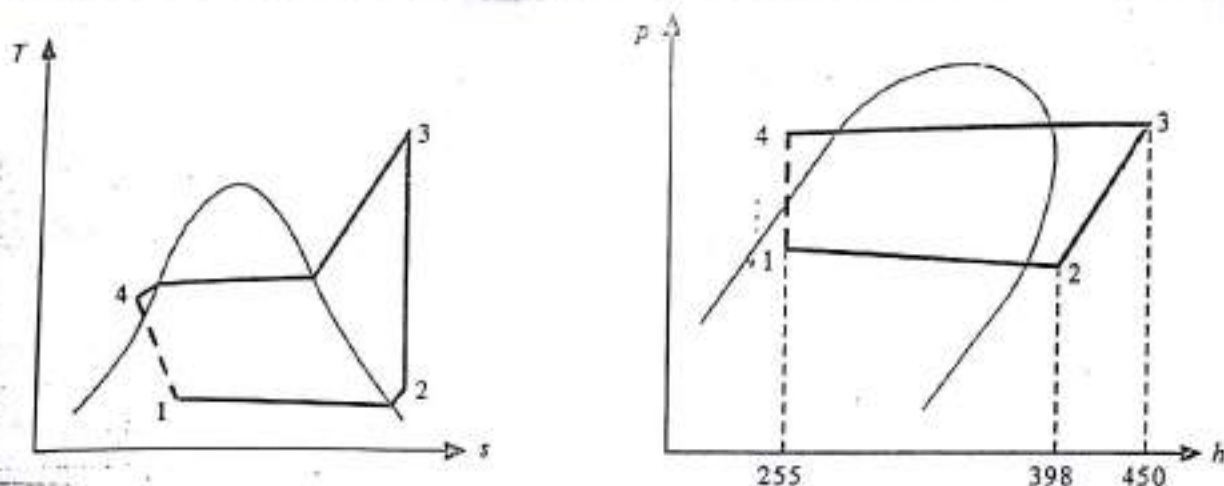


FIGURE 3.16 Example 3.7.

- (c) Isentropic efficiency η_c of the compressor is determined from,

$$\eta_c = \frac{h_{3s} - h_2}{h_3 - h_2} = \frac{442 - 398}{450 - 398} = 0.84 \text{ or } 84\%$$

- (d) Coefficient of performance of the refrigerator, $\text{COP} = \frac{\text{RE}}{\dot{W}} = \frac{10.86}{3.95} = 2.75$

SECTION B: MULTIPRESSURE VAPOUR COMPRESSION CYCLES

The simple vapour compression system that has been dealt with up to this point is a two-pressure system. However, there are many installations that require refrigerated spaces to be maintained at various temperature levels like those in a multi-product cold store storing different products at different temperatures. Ideally, this can be achieved by employing two independent systems, which however, is not a desirable choice because of the obvious investment of additional initial cost.

3.5 MULTIPRESSURE SYSTEMS

A multipressure system is a refrigeration system that has two or more low-side pressures. Systems with more than two pressures may be necessary due to (a) multistage or compound compression or (b) due to supplying refrigerant to a multi-evaporator system. Cascade systems which employ more than one refrigerant also have multipressures, although each refrigerant circuit operates on a two-pressure system only.

Figure 3.17 illustrates the classification of vapour compression refrigeration cycles.

Multistage vapour compression cycles

For high condensing and low evaporating temperature applications, staging of compressors is necessary because of reduced volumetric efficiency. By employing multiple compressors combined with intercooling, this handicap may be overcome. Generally, compound compression is employed when the pressure ratio exceeds 4 or 5. While with a single-stage compression the

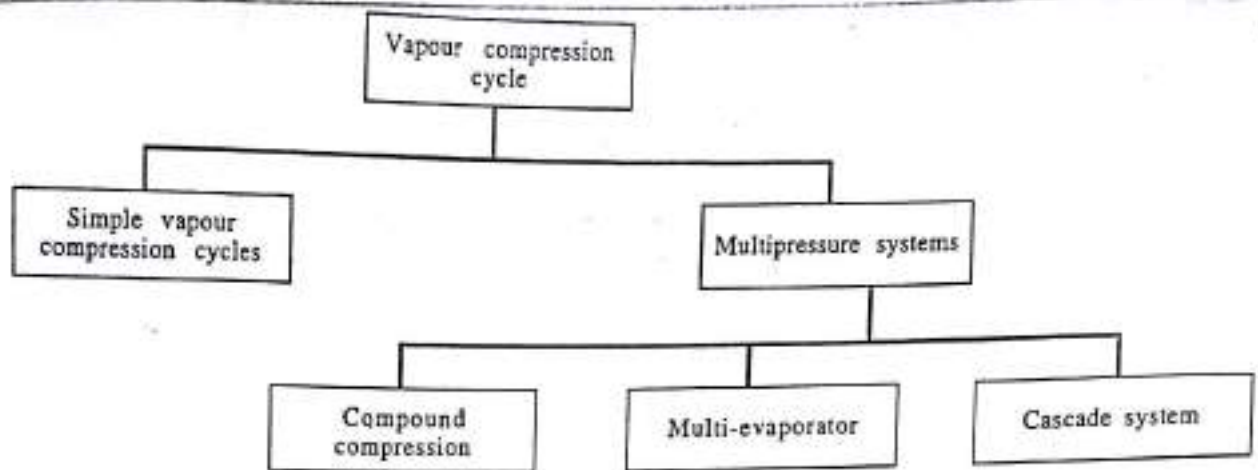


FIGURE 3.17 Multipressure systems.

lowest temperature achieved could be in the region of -30°C , with two-stage compression a temperature as low as -55°C , and with three-stage compression a temperature down to -80°C is achievable.

Figure 3.18 shows a schematic diagram of a two-stage cycle using ammonia as the refrigerant while Figure 3.19 shows the thermodynamic processes of the same cycle. The high-pressure side of the cycle represented by states 3-4-5-6 is theoretically comparable to a

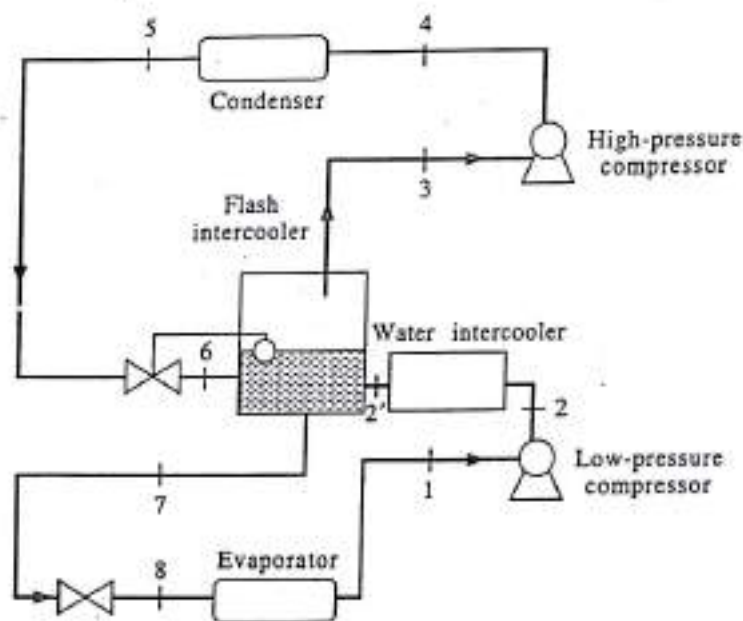


FIGURE 3.18 Schematic diagram of a two-stage compression system with water and flash intercooling.

single-stage system, with the exception that the evaporator has been substituted by the flash intercooler and the remaining components constitute the low pressure system. The vapour is compressed isentropically by the low-pressure compressor to the intermediate pressure (state 2). The refrigerant is then precooled in a water intercooler to state 2'. Superheated discharge gas

from the low-stage compressor at state 2' bubbles through the liquid in the intercooler and thus gets cooled to the saturation temperature at the pressure of the flash chamber (state 7) and a part of the liquid evaporates which goes to the higher stage along with the vapour from the lower stage (state 3). Complete desuperheating has been assumed in the case shown in Figures 3.18 and 3.19, which is not practical though. The refrigerant at state 3 enters the compressor and is compressed to state 4. It then enters the condenser and after condensing leaves the condenser in saturated liquid condition (state 5). It then expands through the expansion valve to state 6. Saturated liquid at the cycle intermediate pressure and temperature is drawn from the flash intercooler and throttled through the low-stage expansion valve. The liquid and vapour mixture enter the evaporator at state 8 and leave at state 1.

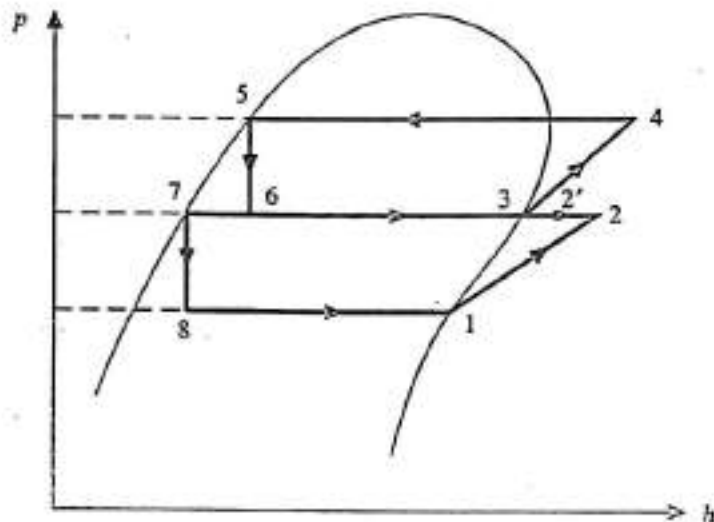


FIGURE 3.19 Thermodynamic cycle for the system in Figure 3.18.

Intercooling: With certain refrigerants, such as NH_3 and CO_2 , the refrigerant effect is greatly improved and power requirement decreased by precooling the liquid just before the final expansion into the evaporator or flash intercooler. The flash intercooler is a pressure vessel or tank in which a fixed liquid level is maintained by a float valve. The water-cooled intercooler may be satisfactory for multistage air compression, but water is usually not cold enough to be suitable for refrigerant compression. Figure 3.20 shows how compression with intercooling appears on the p - h diagram of a refrigerant. Processes 1-2-3 and 2'-4 are on lines of constant entropy but the slope of the process 2-3 is flatter than that of 2'-4, indicating reduced work input. In the case of R12 and R22, intercooling is not effective since the slopes before and after the intercooler are about the same and, hence, there is no savings in work. It may be noted that a part of the heat of low stage compression can be removed by water intercooling in case the discharge temperature from the stage is substantially higher than the cooler water temperature.

For two-stage compression of a refrigerant with complete intercooling, for minimum work, the interstage pressure p_i is given by

$$p_i = (p_1 p_2)^{1/2}$$

where p_1 is the suction pressure of the low-pressure compressor and p_2 is the discharge pressure of the high-pressure compressor.

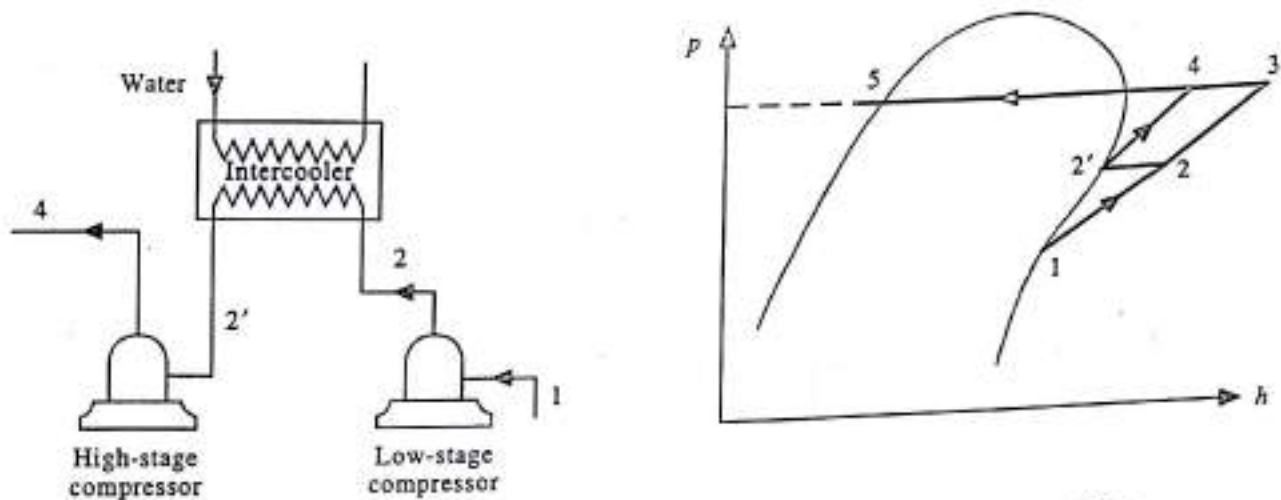


FIGURE 3.20 Intercooling of a refrigerant in two-stage compression.

Multi-evaporator systems

There are many installations that require refrigerated spaces to be maintained at various temperature levels. For example, in a cold store there may be different rooms for different products maintained at different temperatures. Figure 3.21 shows the schematic layout of a typical multi-evaporator system. Figure 3.22 is the p - h diagram of the multi-evaporator system of Figure 3.21.

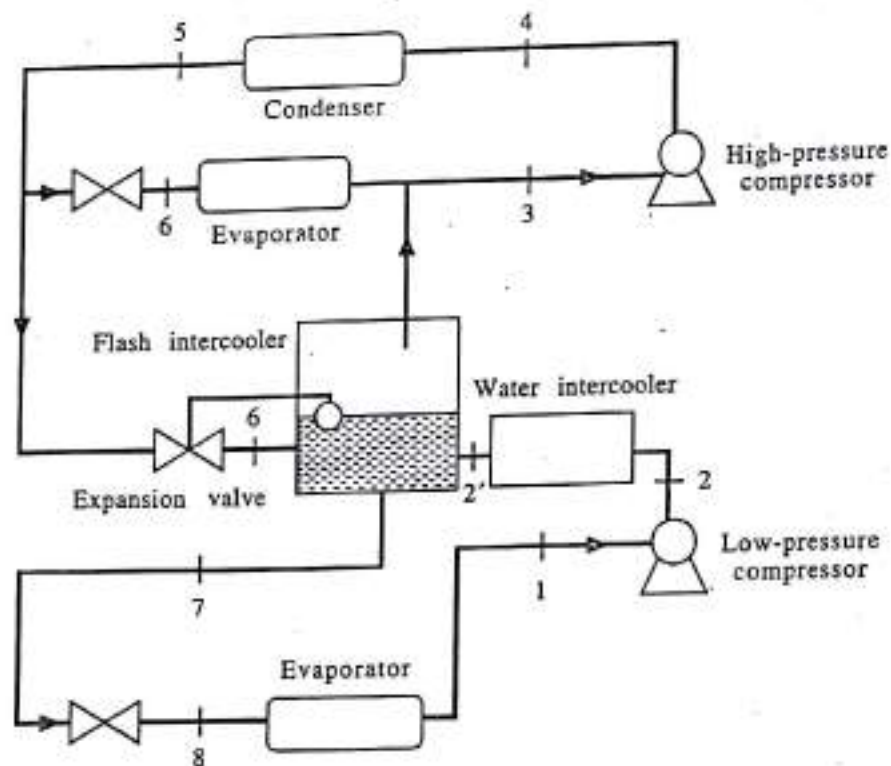


FIGURE 3.21 Schematic diagram of a multi-evaporator system.

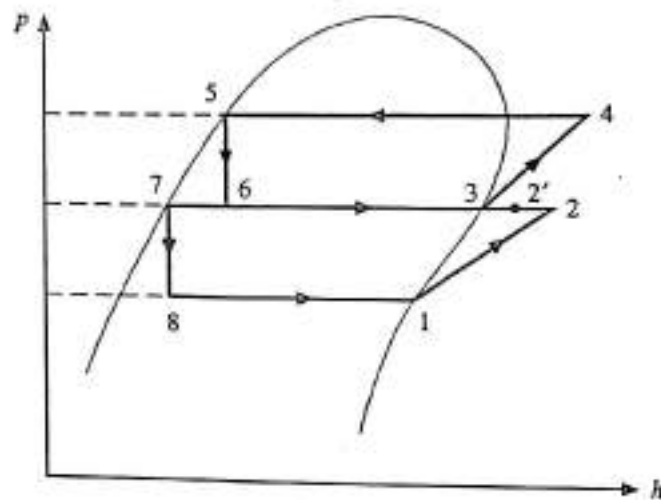


FIGURE 3.22 The p - h diagram of the multi-evaporator system of Figure 3.21.

Cascade system

For the production of low temperatures (-75°C) with vapour compression cycles, the common alternative to multistage compression is the *cascade* system. In these systems a series of refrigerants with progressively lower boiling points may be used in a series of single-stage units. Figure 3.23 shows a schematic diagram of a simple cascade system using two single-stage cycles

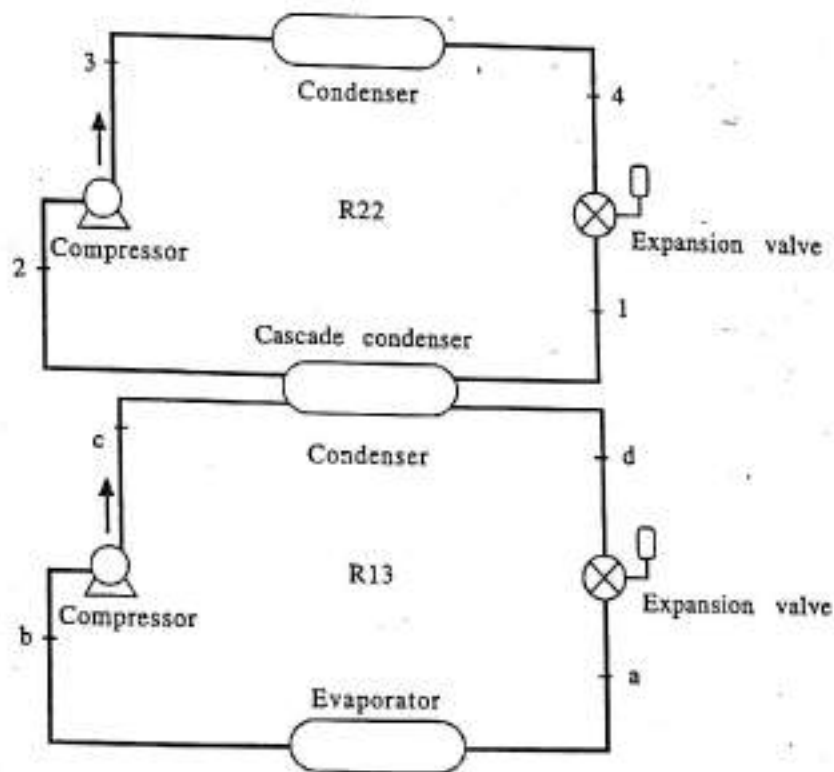


FIGURE 3.23 Schematic diagram of a cascade system.

using refrigerants R13 and R22 respectively. Figure 3.24 shows the T - s diagram of the cascade system. The cascade condenser is the common intermediate heat exchanger which serves as the condenser of the low-temperature stage and evaporator of the high temperature stage. In this case R13 is condensed and R22 is evaporated in the cascade condenser. Thus each refrigerant circuit

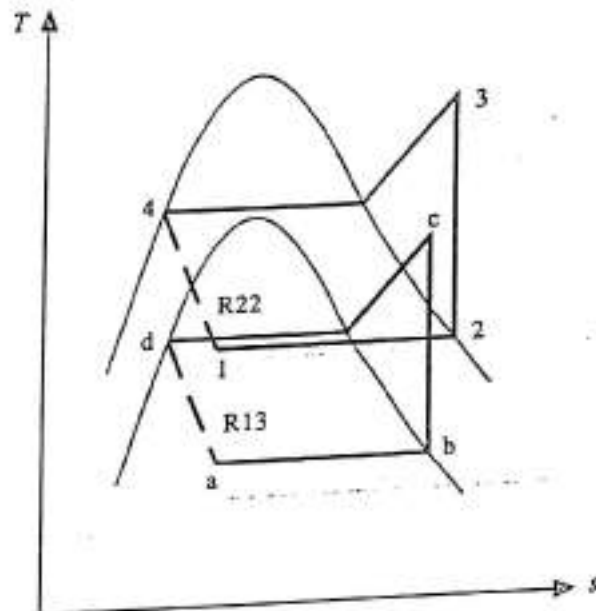


FIGURE 3.24 The T - s diagram of the cascade system of Figure 3.23.

is comparatively simple and is a system in itself, and each refrigerant can be chosen that operates best within the required temperature and pressure range. The cascade system may not be more efficient than the multi-stage systems but its use is primarily to obtain a lower temperature. Using a single refrigerant has a number of limitations because of which multiple refrigerants are used in a cascade system. Some of the limitations are (a) solidification of refrigerants at low evaporating temperature; (b) with high boiling refrigerant, very low pressure in evaporator and large suction volume; (c) with low boiling refrigerant, extremely high pressure in condenser; (d) low COP with high pressure ratio; and (e) operational difficulties of mechanical equipment at low temperatures.

EXAMPLE 3.8

In an NH_3 refrigeration system installed in a cold store (Figure 3.25), the evaporator provides 300 kW of refrigeration at -30°C . The system uses two-stage compression as shown in the figure, with intercooling and removal of flash gas. The condensing temperature is 35°C . Draw the cycle on a p - h diagram and calculate the following:

- The power required by the compressors
- The coefficient of performance.

Solution

$$p_s = \text{saturation pressure at } -30^\circ\text{C} = 119.9 \text{ kPa}$$

$$p_d = \text{saturation pressure at } 35^\circ\text{C} = 1352 \text{ kPa}$$

$$p_i = \sqrt{119.2 \times 1352} = 402.7 \text{ kPa}$$

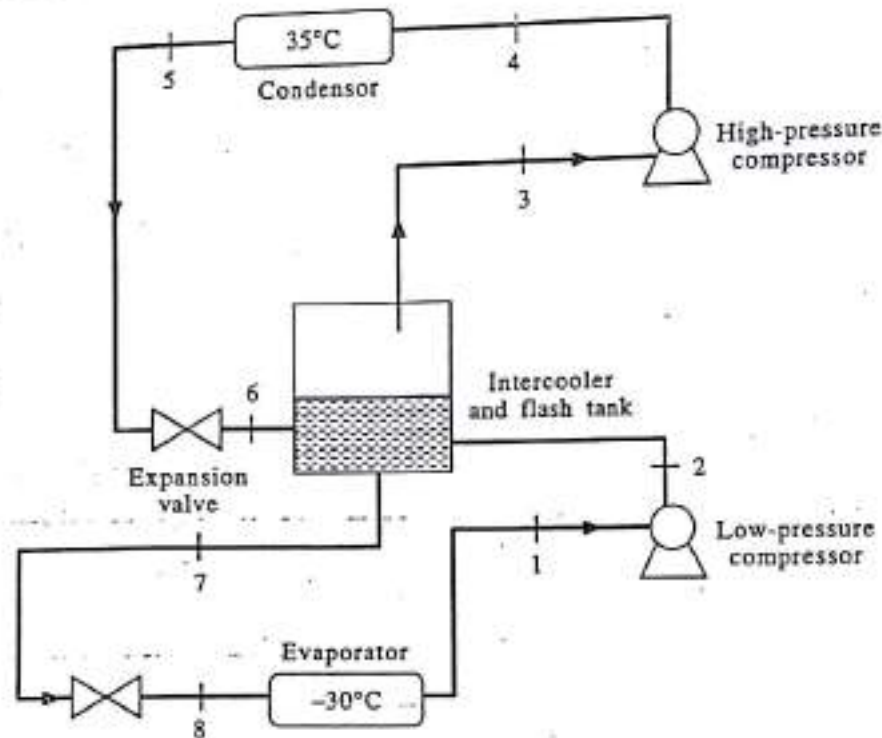


FIGURE 3.25 Example 3.8.

$$h_1 = h_g @ -30^\circ\text{C} = 1423 \text{ kJ/kg}$$

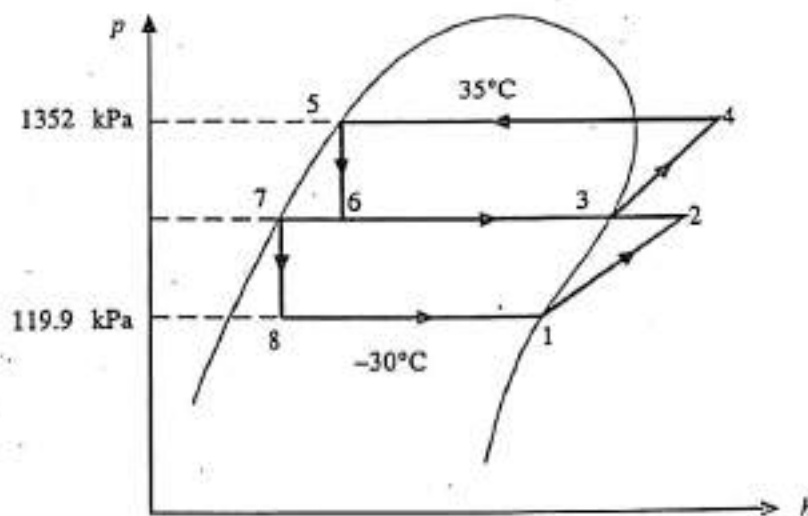
$$h_2 = 1582 \text{ kJ/kg from the } p-h \text{ diagram of } \text{NH}_3$$

$$h_3 = h_g \text{ at } 402.7 \text{ kPa} = 1459.84 \text{ kJ/kg}$$

$$h_4 = 1632 \text{ kJ/kg}$$

$$h_5 = h_f \text{ at } 35^\circ\text{C} = 366 \text{ kJ/kg} = h_6$$

$$h_7 = h_f \text{ at } 402.7 \text{ kPa} = 191.8 \text{ kJ/kg}$$

FIGURE 3.26 The p - h diagram for Example 3.8.

$$\dot{m}_{r1} = \frac{300 \text{ kW}}{h_1 - h_8} = \frac{300 \text{ kW}}{1423 - 191.8} = 0.244 \text{ kg/s}$$

$$\dot{m}_{r1} = \dot{m}_{r2} = \dot{m}_{r7} = \dot{m}_{r8}$$

Heat and mass balance about the intercooler gives

$$\dot{m}_{r2}h_2 + \dot{m}_{r6}h_6 = \dot{m}_{r7}h_7 + \dot{m}_{r3}h_3 \quad (\dot{m}_{r2} = \dot{m}_{r3}, \dot{m}_{r7} = \dot{m}_{r2})$$

$$(0.244 \times 1582) + \dot{m}_{r3} \times 366 = 0.244 \times 191.8 + \dot{m}_{r3} \times 1459.84$$

Solving, $\dot{m}_{r3} = 0.31 \text{ kg/s}$

$$(a) \text{ Low-stage power} = \dot{m}_{r2}(h_2 - h_1) = 0.244 \times (1582 - 1423) = 38.8 \text{ kW}$$

$$\text{High-stage power} = \dot{m}_{r3}(h_4 - h_3) = 0.31 \times (1632 - 1459.84) = 53.4 \text{ kW}$$

$$\text{Total power} = 38.8 + 53.4 = 92.2 \text{ kW}$$

$$(b) \text{ COP} = \frac{\text{refrigeration effect}}{W_{\text{in}}} = \frac{300 \text{ kW}}{92.2} = 3.25$$

EXAMPLE 3.9

In a R22 refrigeration system, as shown in Figure 3.27 one evaporator is required to provide 160 kW of refrigeration at -30°C and another evaporator is required to provide 190 kW at 5°C . The system uses two-stage compression with intercooling and is arranged as shown in the figure. The condensing temperature is 30°C .

Draw the cycle on a p - h diagram and calculate the following:

- The power required by the compressor
- The coefficient of performance.

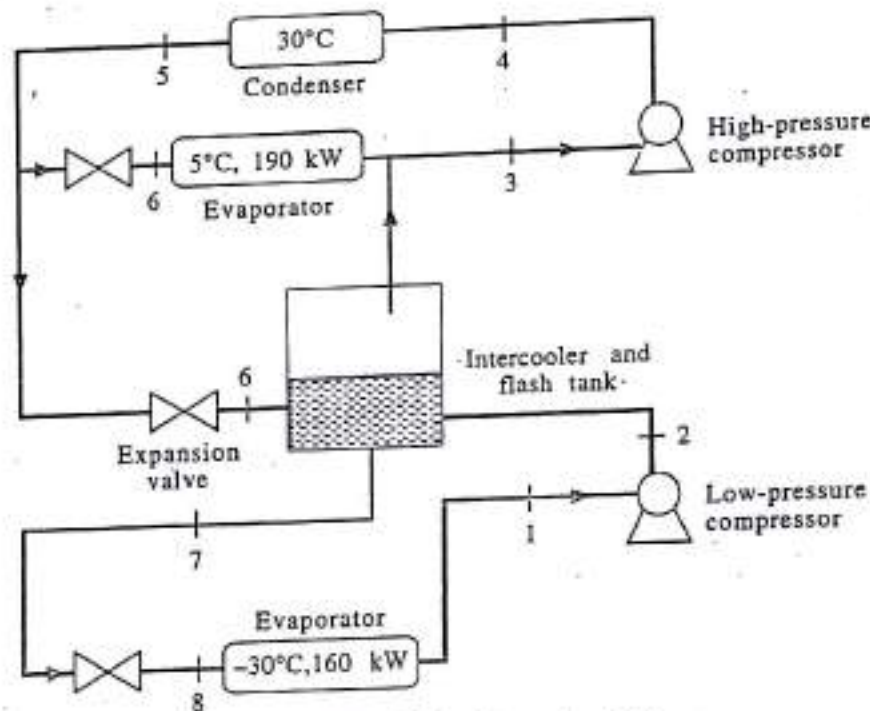


FIGURE 3.27 Example 3.9.

Solution

$$h_1 = h_g @ -30^\circ\text{C} = 393.14 \text{ kJ/kg}$$

$$h_2 = 424 \text{ kJ/kg}$$

$$h_3 = 407.14 \text{ kJ/kg}$$

$$h_4 = 425 \text{ kJ/kg}$$

$$h_5 = h_f @ 30^\circ\text{C} = 236.66 \text{ kJ/kg} = h_6$$

$$h_7 = 205.9 = h_8$$

$$\dot{m}_1 = \frac{160 \text{ kW}}{h_1 - h_8} = \frac{160 \text{ kW}}{393.14 - 205.9} = 0.855 \text{ kg/s}$$

$$\dot{m}_1 = \dot{m}_7 = \dot{m}_2 = \dot{m}_8$$

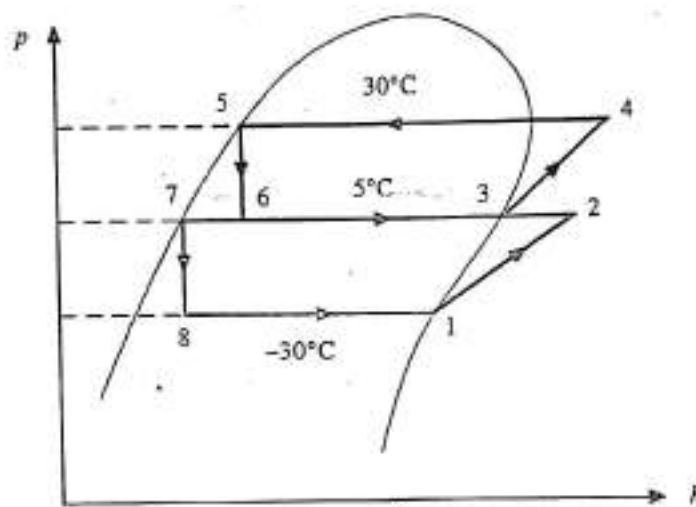


FIGURE 3.28 The p - h diagram for Example 3.9.

Heat and mass balance equations about both the high temperature evaporator and the intercooler, are as follows:

$$\dot{m}_5 h_5 + 190 + \dot{m}_2 h_2 = \dot{m}_3 h_3 + \dot{m}_7 h_7$$

$$\dot{m}_2 = \dot{m}_7 = 0.855, \quad \dot{m}_5 = \dot{m}_3$$

$$\dot{m}_3 \times 236.66 + 190 + 0.855 \times 424 = \dot{m}_3 \times 407.14 + 0.855 \times 205.9$$

$$\text{or} \quad \dot{m}_3 (407.14 - 236.66) = 190 + 362.52 - 176.04$$

$$\text{or} \quad \dot{m}_3 \times 170.48 = 376.48$$

$$\therefore \quad \dot{m}_3 = \frac{376.48}{170.48} = 2.208 \text{ kg/s}$$

The power required by the compressors:

$$(a) \text{ Low-stage power} = 0.855(424 - 393.14) = 26.39 \text{ kW}$$

$$\text{High-stage power} = 2.208(425 - 407.14) = 39.44 \text{ kW}$$

$$\text{Total power} = 26.39 + 39.44 = 65.83 \text{ kW}$$

$$(b) \text{ COP} = \frac{\text{Total cooling produced}}{W_{in}} = \frac{190 + 160}{65.83} = 5.32$$

EXAMPLE 3.10

A two-stage ammonia system designed for a fish freezing plant, uses flash gas removal and intercooling operation (Figure 3.29). The condensing temperature is 35°C. The saturation temperature of the high-temperature evaporator is 0°C and its capacity is 90 kW. The saturation temperature of the low-temperature evaporator is -40°C and its capacity is 170 kW of refrigeration.

Draw the cycle on a p - h diagram. Determine the following:

- The rate of refrigerant compressed by the high-stage compressor
- The power required by the compressors
- The coefficient of performance.

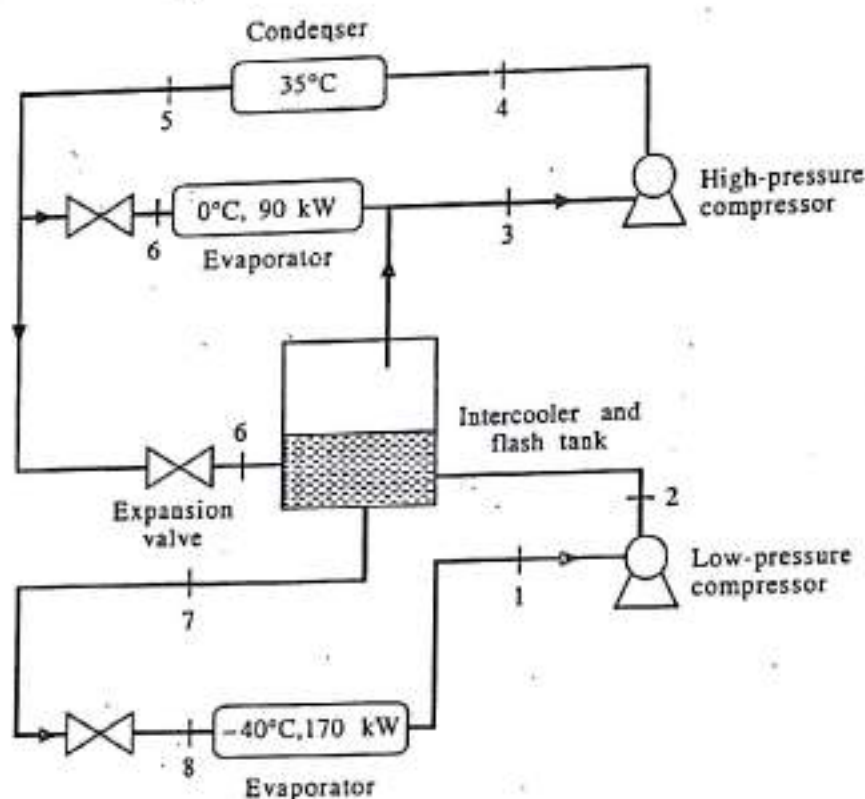


FIGURE 3.29 Example 3.10.

Solution

Here,

$$h_1 = h_g @ -40^\circ\text{C} = 1408.41 \text{ kJ/kg}$$

$$h_2 = 1655 \text{ kJ/kg}$$

$$h_3 = 1461.70 \text{ kJ/kg}$$

$$h_4 = 1626 \text{ kJ/kg}$$

$$h_5 = h_f @ 35^\circ\text{C} = 366.07 \text{ kJ/kg} = h_6$$

$$h_7 = h_8 = 200 \text{ kJ/kg}$$

$$\dot{m}_1 = \frac{170 \text{ kW}}{h_1 - h_8} = \frac{170 \text{ kW}}{1408.41 - 200} = 0.1407 \text{ kg/s}$$

$$\dot{m}_1 = \dot{m}_7 = \dot{m}_2 = \dot{m}_8$$

$$\dot{m}_5 = \dot{m}_3$$

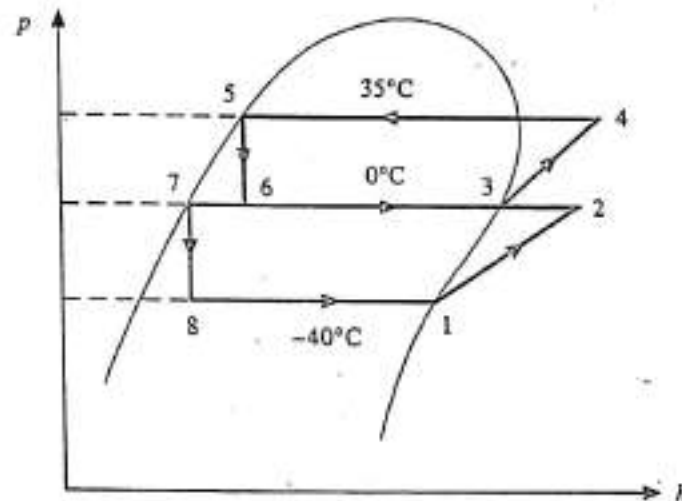


FIGURE 3.30 The p - h diagram for Example 3.10.

- (a) Heat and mass balance equations about both the high temperature evaporator and the intercooler are given below:

$$\dot{m}_5 h_5 + 90 + \dot{m}_2 h_2 = \dot{m}_3 h_3 + \dot{m}_7 h_7$$

$$\dot{m}_2 = \dot{m}_7 = 0.1407 \text{ kg/s}$$

$$\therefore \dot{m}_5 \times 366.07 + 90 + 0.1407 \times 1655 = \dot{m}_3 \times 1461.70 + 0.1407 \times 200$$

$$\text{or } \dot{m}_3 (1461.70 - 366.07) = 90 + 0.1407(1655 - 200)$$

$$\text{or } \dot{m}_3 \times 1095.63 = 294.72$$

$$\therefore \dot{m}_3 = 0.269 \text{ kg/s}$$

(b) The power required by the compressors:

$$(i) \text{ Low-stage power} = 0.1407(h_2 - h_1) = 0.1407(1655 - 1408.41) = 34.69 \text{ kW}$$

$$\text{High-stage power} = 0.269(h_4 - h_3) = 0.269(1626 - 1461.70) = 44.20 \text{ kW}$$

$$\text{Total power} = 34.69 + 44.20 = 78.89 \text{ kW}$$

$$(ii) \text{ COP} = \frac{\text{Total cooling produced}}{W_{in}} = \frac{90 + 170}{78.89} = 3.30$$

SECTION C: HEAT PUMPS

Heat pump is the expression for a refrigerating system in which the heat discharged at the condenser is of principal interest and importance (Figure 3.31). The cooling effect produced by the evaporator is secondary and is not necessarily utilized. The medium being cooled serves as a heat source, and the compressor pumps the heat, picked up by the refrigerant in the evaporator, to the higher level in the condenser for practical use. However, in a combined heating and cooling system there is a reversible valve enabling the cycle to operate as an air cooler or as an air heater as shown in Figure 3.31. The three basic heat sources that have been used are air, water, and the earth.

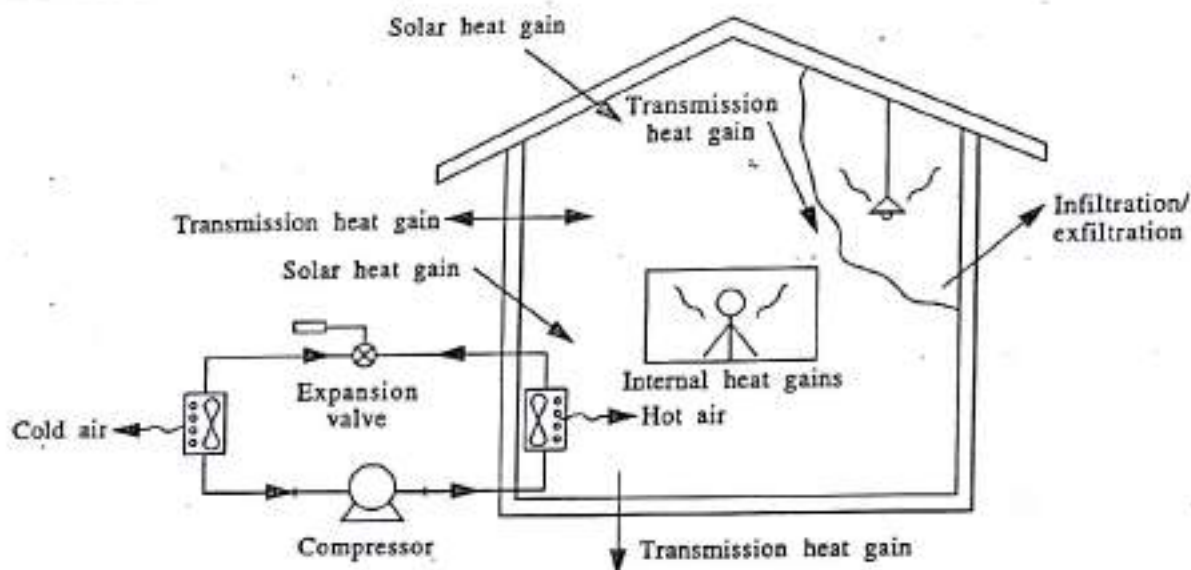


FIGURE 3.31 Heat pump.

3.6 HEAT PUMP CYCLE ANALYSIS

Referring to the same cycle as shown in Figure 3.6 and repeated below as, Figure 3.32, the COP of the heat pump can be deduced which is greater than that of the refrigeration cycle.

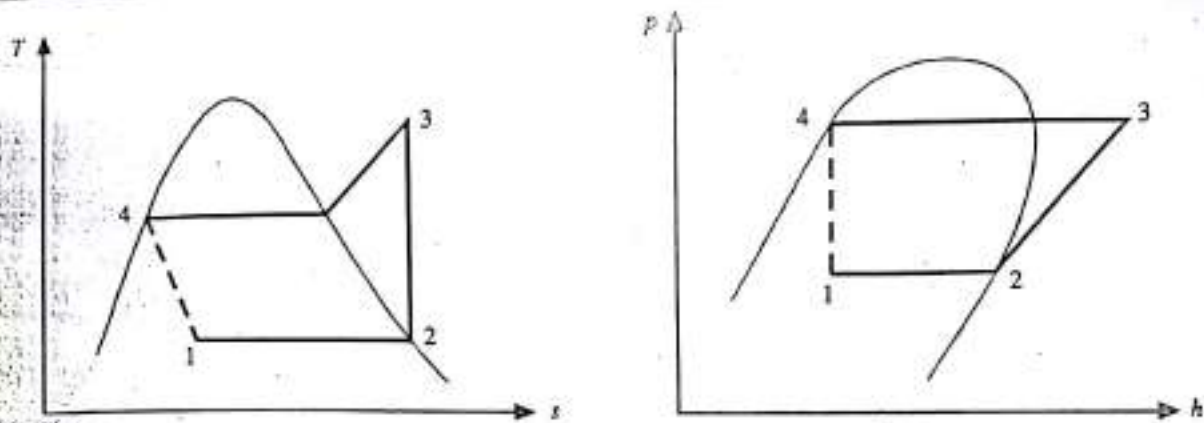


FIGURE 3.32 Ideal vapour compression cycle.

Refrigerating effect, $Q_{1-2} = \dot{m}_r(h_2 - h_1)$

Work input, $W_{2-3} = \dot{m}_r(h_3 - h_2)$

Heat rejection, $Q_{3-4} = \dot{m}_r(h_3 - h_4)$

Coefficient of performance, $COP_{bp} = \frac{\dot{m}_r(h_3 - h_4)}{\dot{m}_r(h_3 - h_2)} = 1 + COP_r$

A refrigerator may be converted into a heat pump by reversing the direction of flow of the refrigerant when the roles of the evaporator and the condenser are reversed. To accomplish that, a reversing valve is needed as shown in Figure 3.33.

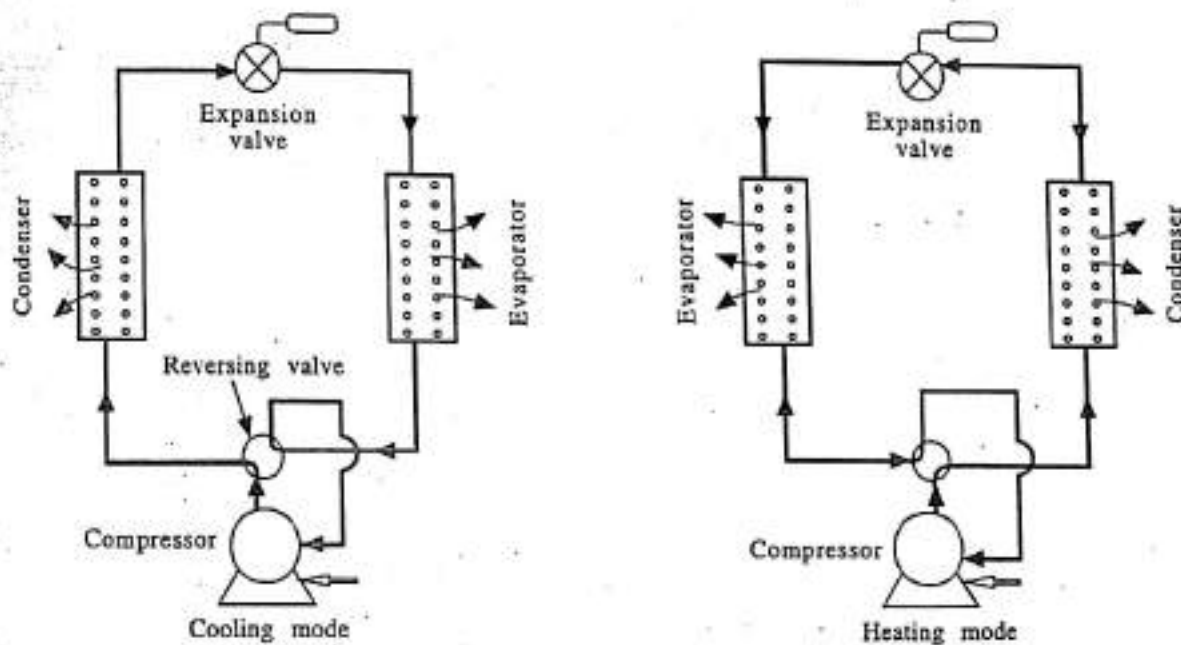


FIGURE 3.33 The reversing valve operation in a refrigeration cycle.

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- Moran, M.J. and H.N. Shapiro, *Fundamentals of Engineering Thermodynamics*, 3rd ed., Wiley, New York, 1995.
- Van Wylen, G.J., R.E. Sonntag, C. Borgnakke, *Fundamentals of Classical Thermodynamics*, 4th ed., John Wiley & Sons, New York, 1993.

REVIEW EXERCISES

- 3.1 What modifications are necessary to convert a domestic air-conditioner meant for summer cooling for use in winter as well?
- 3.2 Sketch an ideal vapour compression cycle on p - h and T - s diagrams. Label each point on the diagram and show the comparable locations on a sketch of the refrigeration system.
- 3.3 Sketch an ideal refrigeration cycle on p - h diagram and show the refrigerating effect, heat of compression and heat of rejection. What is their arithmetic relationship?
- 3.4 Describe the effect of changing the evaporating temperature on system performance characteristics. Use a sketch of the p - h diagram as an aid.
- 3.5 What is subcooling? What are the effects of subcooling?
- 3.6 A refrigeration system operating on a vapour compression refrigeration cycle using refrigerant R134a as the working fluid has a cooling capacity of 6 kW. The refrigerant enters the compressor as saturated vapour at 0.14 MPa and is compressed isentropically to 0.8 MPa. The refrigerant leaves the condenser as saturated liquid and expands adiabatically through an expansion device.
Show the cycle on a T - s diagram with respect to saturation lines, and determine the following:
 - (a) The quality of the refrigerant at the end of the throttling process
 - (b) The coefficient of performance.
- 3.7 An NH_3 refrigeration system designed for a refrigerated warehouse (Figure 3.27) has one evaporator that provides 200 kW of refrigeration at -20°C . The system uses two-stage compression as shown in the figure, with intercooling and removal of flash gas. The condensing temperature is 35°C . Draw the cycle on a p - h diagram and calculate the following:
 - (a) The ratio of the power required by the high-stage and low-stage compressor.
 - (b) The coefficient of performance.
- 3.8 In an R22 refrigeration system, one evaporator is required to provide 150 kW of refrigeration at -30°C and another evaporator is required to provide 200 kW at 5°C . The system uses two-stage compression with intercooling and is arranged as shown in Figure 3.27. The condensing temperature is 36°C . Calculate the coefficient of performance.

4

Refrigerants

A refrigerant is any body or substance, which acts as a cooling agent by absorbing heat from another body or substance. With regard to the vapour compression cycle, the refrigerant is the working fluid of the cycle, which alternately vaporises and condenses as it absorbs and gives off heat. Theoretically, any reasonably volatile substance which is liquid at the temperature desired in the evaporator can be used as a refrigerant, but in practice the choice is limited by factors such as toxicity, cost, flammability, chemical stability, etc.

A *primary refrigerant* is one, which is used in a recirculating cycle and is accompanied by a change in state. During the cyclic process it undergoes compression, condensation, expansion, and evaporation. A *secondary refrigerant* is one, which is used as a heat transfer medium without a change of state but with a change in temperature. Chilled water used in a conventional air-conditioning plant and circulated through the air-cooling coils is an example of a secondary refrigerant. Figure 4.1 shows an air-conditioning system that uses both primary and secondary refrigerants.

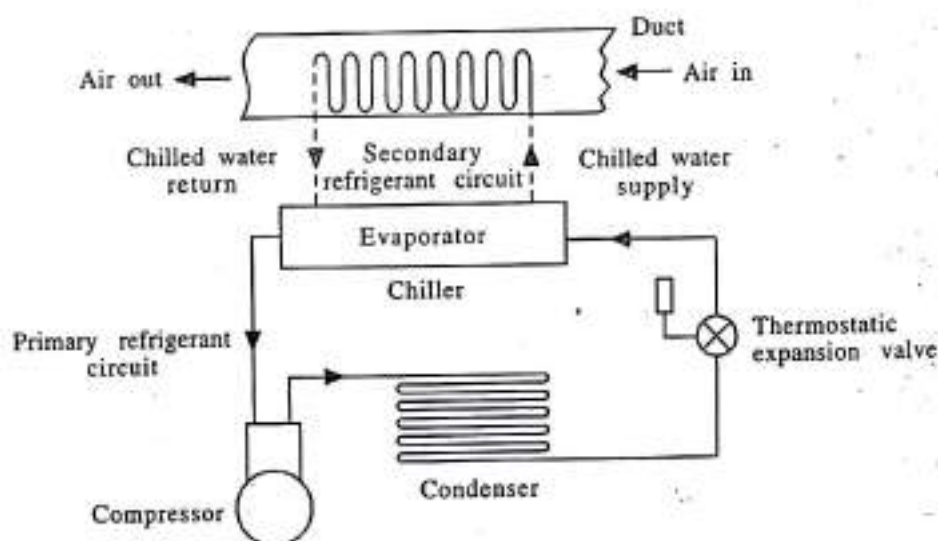


FIGURE 4.1 Air-conditioning system using both primary and secondary refrigerants.

4.1 CLASSIFICATION

Many refrigerants have been used over the years and some of them have become obsolete and since replaced by newer refrigerants. Some currently popular refrigerants are being phased out due to their damaging effect on ozone layer.

The generic classification of refrigerants is shown in Figure 4.2. Based on this classification, some of the popular refrigerants have been listed in Tables 4.1 to 4.4. The detailed properties of some of these refrigerants have been discussed at length later in this chapter.

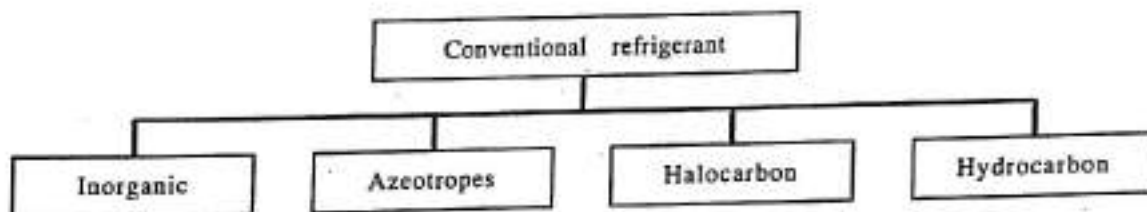


FIGURE 4.2 Classification of refrigerants.

TABLE 4.1 Halocarbon refrigerants

Designation	Chemical name	Chemical formula	Use	Remarks
R11	Trichloromonofluoromethane	CCl_3F	Air conditioning	
R12	Dichlorodifluoromethane	CCl_2F_2	Do	Being phased out
R22	Monochlorodifluoromethane	CHClF_2	Refrigeration and air conditioning	Still acceptable
R134a	Tetrafluoroethane	$\text{CF}_3\text{CH}_2\text{F}$	Air conditioning	Substituting R12
R40	Methyl chloride	CH_3Cl	Refrigeration	Obsolete

TABLE 4.2 Inorganic refrigerants

Designation	Chemical name	Chemical formula	Use	Remarks
R717	Ammonia	NH_3	Industrial refrigeration	Still popular
R718	Water	H_2O	Absorption cycles	—
R729	Air		Refrigeration and air conditioning	Still acceptable
R744	Carbon dioxide	CO_2	Marine refrigeration, Ice making	Discontinued
R764	Sulphur dioxide	SO_2	Air conditioning	Discontinued

TABLE 4.3 Hydrocarbon refrigerants

Designation	Chemical name	Chemical formula	Use	Remarks
R50	Methane	CH_4	No direct use	—
R170	Ethane	C_2H_6	Gas liquefaction	—
R290	Propane	$\text{CH}_3\text{CH}_2\text{CH}_3$	Gas liquefaction	—

TABLE 4.4 Common azeotropes

Group	Designation	Composition	Chemical formula	Use	Remarks
Azeotropes	R500	R12	73.8%	CCl_2F_2	—
		R115	26.2%	CH_3CHF_2	—
	R501	R22	75%	CHClF_2	Cascade
		R12	25%	CCl_2F_2	systems
	R502	R22	48.8%	CHClF_2	—
		R115	51.2%	CClF_2CF_3	—

Halocarbon refrigerants, listed in Table 4.1, contain one or more of the three halogens—chlorine, fluorine, and bromine. These refrigerants are obtained by replacing one or more hydrogen atoms in a hydrocarbon molecule such as ethane or methane with the above-mentioned halogens. The halocarbon family is comprised of refrigerants which are non-flammable and most of them are non-toxic too. Thus a refrigerant can be found in this group to fit practically any application. Because of their overall performance, these synthetic refrigerants have replaced the classic refrigerants like NH_3 , CO_2 , and water etc. listed in Table 4.2.

However, in recent years some of the refrigerants in the halocarbon family have lost favour because of their negative impact on environmental issues. Because of their great stability, fully halogenated compounds, such as chlorofluorocarbons (CFCs) persist in the atmosphere for many years and eventually diffuse into the stratosphere. The molecules of CFCs, such as R11 and R12, contain only carbon and the halogens contain chlorine and fluorine. Once in the upper atmosphere, the CFC molecules break down releasing chlorine, which destroys ozone and causes ozone depletion. In the lower atmosphere, the CFC molecules absorb infrared radiation, which may contribute to the warming of the earth, i.e. **global warming**. Substitution of a hydrogen atom by one or more of the halogens in a CFC molecule greatly reduces its lifetime in atmosphere and consequently its negative impact. These compounds are referred to as hydrochlorofluorocarbons (HCFCs). The ozone depleting CFC refrigerants will be phased out as per the decision of the Montreal protocol, an international treaty.

Like halocarbon refrigerants, many *hydrocarbon refrigerants*, listed in Table 4.3, are suitable as refrigerants. They are commonly used as refrigerants in liquefaction of gases.

Azeotropes, listed in Table 4.4 are those mixtures of two or more substances which behave as if they were compounds, for they cannot be separated into their components by distillation. An azeotrope evaporates and condenses as a single substance with properties that are different from those of its constituents.

4.2 DESIRABLE PROPERTIES OF REFRIGERANTS

No single refrigerant satisfies all the attributes desired of a refrigerant for all operating conditions. So a refrigerant is chosen that has the greatest number of advantages and the least number of disadvantages for the specific application involved. Obviously, in most cases there must be a compromise. The best refrigerant for producing low temperatures is not necessarily the best one for producing moderate temperatures. Similarly, the refrigerant chosen for a reciprocating compressor is usually different from the one that is chosen for a centrifugal compressor.

The important properties that relate to the overall performance of a refrigeration system are tabulated in Table 4.5, followed by a review of some of the refrigerant properties which affect the refrigerant cycle performance.

TABLE 4.5 Comparative refrigerant performance per kilowatt of refrigeration^a

Refrigerant	Evaporator pressure, MPa	Condenser pressure, MPa	Compression ratio	Net refrigerating effect, kJ/kg	Refrigerant circulated, kg/s	Liquid circulated, L/s	Specific volume of suction gas, m ³ /kg	Compressor displacement, L/s	Power consumption, kW	Coefficient of performance	Comp. discharge temperature, K
No. Chemical name or composition (% by mass)											
744 Carbon dioxide	2.291	7.208	3.15	134.24	0.00745	0.0123	0.0087	0.065	0.338	2.96	343
290 Propane	0.291	1.077	3.71	279.88	0.00357	0.0074	0.1542	0.551	0.211	4.74	320
502 R22/115 (48.8/51.2)	0.349	1.319	3.78	104.39	0.00958	0.0080	0.0500	0.479	0.226	4.43	310
22 Dichlorodifluoromethane	0.296	1.192	4.03	162.46	0.00616	0.0053	0.0774	0.476	0.210	4.75	326
717 Ammonia	0.236	1.164	4.94	1102.13	0.00091	0.0015	0.5106	0.463	0.207	4.34	371
500 R12/152a (73.8/26.2)	0.214	0.879	4.11	140.95	0.00709	0.0062	0.0938	0.665	0.213	4.69	314
12 Dichlorodifluoromethane	0.183	0.745	4.07	116.58	0.00858	0.0066	0.0914	0.784	0.213	4.69	311
134a Tetrafluoroethane	0.160	0.770	4.81	150.71	0.00664	0.0056	0.1224	0.812	0.226	4.42	316
600 Butane	0.056	0.283	5.05	292.01	0.00342	0.0060	0.6641	2.274	0.214	4.68	318

Adapted by permission from *ASHRAE Fundamentals* 1989. Notes: ^aBased on 258 K evaporation and 303 K condensation.

Evaporator and condenser pressure

It is desirable that both these pressures be positive, yet not too high above the atmospheric pressure. Positive pressure prevents leakage of air and moisture into the system and also makes it easier to detect leaks. The pressure at which the refrigerant vapour will condense at ordinary air or water temperatures should be low. A high condensing pressure would require heavy material for compressors, piping and coils, and also increase the tendency to leakage.

Critical temperature

High critical temperature is desirable as it is impossible to condense the refrigerant at a temperature above the critical, no matter how much the pressure is increased. The critical temperature of all refrigerants except CO_2 is sufficiently high and such refrigerants, therefore, present no problem. In the case of CO_2 the pressure at the end of compression would be too large, necessitating heavy construction to withstand such high pressure. The critical temperatures and pressures of some common refrigerants are listed in Table 4.6.

TABLE 4.6 Critical temperature and pressure of various refrigerants

Refrigerants	Critical temperature ($^{\circ}\text{C}$)	Critical pressure in bar
R11	197.96	44.0
R12	111.78	41.24
R22	96.14	49.90
R134a	101.03	40.56
R502	82.20	40.75
NH_3	132.22	113
H_2O	373.99	220
CO_2	30.98	74
CH_4	190.55	45.9

Boiling and freezing temperatures

A refrigerant should have a low boiling temperature, otherwise it would become necessary to operate the compressor at high vacuums with the resulting lowered efficiency and capacity. The refrigerant chosen must have a freezing point well below any temperature that would be encountered in operation.

Density

Low refrigerant densities are usually preferable, since they permit the use of small suction and discharge lines without excessive pressure drops. For larger capacity machines, however, a high vapour density is desirable since this results in a need for a smaller compressor and a smaller gas piping.

Latent heat of vaporization

A high latent heat at the evaporator temperature is desirable because it is usually associated with a high refrigerating effect per unit mass of refrigerant circulated. The values of enthalpy of vaporization of a number of common refrigerants are listed in Table 4.7.

TABLE 4.7 Enthalpy of vaporization at 1 atmosphere pressure (1.01 bar)

Refrigerant	Saturation temperature (°C) at 1 atm.	Enthalpy of vaporisation (kJ/kg)
R11	23.71	181.36
R12	-29.80	165.91
R22	-40.80	233.79
R134a	-26.07	216.83
R114	3.6	131.45
R502	-45.42	172.48
R717	-33.33	1369
R744	-78.4	571.5
R718	100	2257

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Stability and inertness

An ideal refrigerant should not decompose at any temperature normally encountered. Disintegration of refrigerants may result in non-condensable gases in the system as well as sludge.

Corrosive properties

The refrigerant should not react with any of the materials normally used in system construction. Any such reaction would lead to corrosion products, contaminating the system and possibly resulting in leakages.

Specific volume

A refrigerant should have low volume per kg when in gaseous state. This not only reduces the size of the equipment but also means higher compressor efficiency. Figure 4.3 shows the relative displacement volume required by different refrigerants. As high displacement volume of

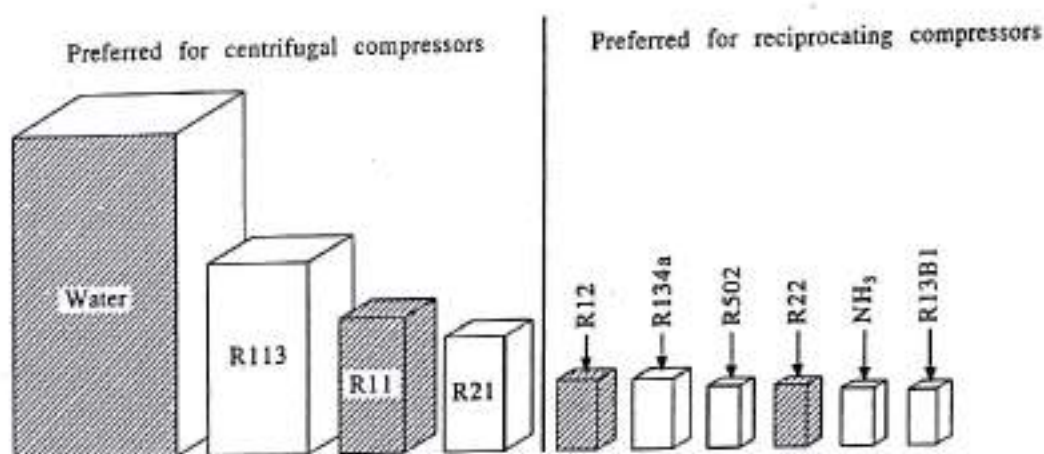


FIGURE 4.3 Graphic representation of displacement volume required by different refrigerants.

refrigerants is desirable for centrifugal compressors, the refrigerants shown on the left-hand side of the vertical line are preferred for this duty. Likewise, the refrigerants on the right of the vertical line are desirable for use in reciprocating compressors.

Viscosity

It is desirable that both the liquid and the vapour refrigerants have low viscosities because of the lower pressure drops in passing through liquid and suction lines. Heat transfer is also improved in the evaporator and the condenser due to low viscosity.

Thermal conductivity

For efficient use of evaporator and condenser surfaces, a refrigerant should possess a high value of thermal conductivity.

Dielectric strength

The electrical resistance of a refrigerant is important if it is to be used in a hermetically sealed unit with the motor exposed to the refrigerant.

Oil effect and miscibility with oil

The refrigerant should have no chemical reaction with the lubricating oil. Such reaction would modify the lubricating properties, and hence, affect lubrication. Certain reactions can even acidify oils. As regards miscibility, oil miscible refrigerants are advantageous in that they give better lubrication because the refrigerant acts as a carrier of oil to the moving parts. They also eliminate oil separation problems and aid in the return of oil from the evaporator.

Toxicity and explosive properties

The refrigerant should be non-poisonous and non-irritating. In any air-conditioning application, particularly, there should be no odour. An ideal refrigerant should present no danger of explosion or fire, either in combination with air or in association with lubricating oil.

Effects on perishable products

In case of leakage, an ideal refrigerant should have no effect on any perishable materials with which it may come into contact.

Leak detection

A leak should be susceptible to detection by some mechanical or chemical means. The detection of leaks just by odour can be very dangerous as well as deceiving.

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Cost

In very small plants requiring a small charge of refrigerant, the cost factor is unimportant. But the cost factor assumes significance in large plants, and the tightness of the construction and leakage minimization are looked after with much care. The rate of leakage is inversely proportional to the square root of molecular weight. It also depends on the pressure differential, velocity, density, and capillarity.

4.3 COMMON REFRIGERANTS

The characteristics and properties of a number of commonly used refrigerants are discussed below.

Ammonia

Ammonia is one of the oldest and most widely used of all refrigerants. It is extensively used in large refrigeration plants such as ice plants, breweries, and industrial processing plants. Ammonia gives a large amount of refrigeration per cubic metre of compressor displacement, which results in low compressor cost. The cost of ammonia itself is less than that of any refrigerant except that of carbon dioxide and water. Ammonia is a low-density refrigerant which creates smaller pressure drops for a given size line. For these reasons, an ammonia system is the most economical system available for large-sized reciprocating refrigerating systems. The freezing point of ammonia, -60°C , is low enough so that there is no problem with ordinary refrigeration temperatures. The odour of ammonia gives instant warning of a leak. A small amount of sulphur dioxide causes a dense white smoke to appear at an ammonia leak, indicating the source of leak. Sulphur candles made of wood or metal pieces dipped in molten sulphur are used for leak detection.

Against the above-mentioned advantages, ammonia is highly toxic and highly irritating. Ammonia is flammable in mixtures of between 16 and 25 per cent in air. Because of its affinity for water, it is difficult to keep ammonia perfectly dry. When ammonia contain any water, it is corrosive to copper and most copper alloys. Accordingly, copper, brass or bronze should not be used in pipes, fittings and bearings in ammonia systems.

At high discharge temperature of ammonia, unless controlled, discharge temperatures can carbonize or otherwise damage lubricating oil. At the high discharge temperatures generated by ammonia, there is a tendency for ammonia to dissociate, i.e. breakdown into its constituents of nitrogen and hydrogen. These gases, unless disposed off, collect in the condenser. Their pressures are added to the normal condensing pressure, which increases the total head pressure and power required. Because of high discharge temperatures, the cylinders of ammonia compressors are water-jacketed. Water-cooling is necessary to keep the cylinder walls cool enough to make satisfactory lubrication possible.

In spite of these disadvantages, ammonia gives excellent economical service in large compression refrigeration systems when the systems are under the supervision of competent operating personnel. Ammonia is used fairly extensively in semi-automatic and automatic refrigeration plants like cold stores, fish freezing plants, etc. Because of the negative impact of CFC refrigerants on the environment, the industry is showing renewed interest in environmentally safe refrigerants like ammonia.

Carbon dioxide

Carbon dioxide is one of the first refrigerants to be used in mechanical refrigerating systems. It is odourless, non-toxic, non-flammable, non-explosive, and non-corrosive. Because of its safe properties, it has been widely used in the past for marine service and for air conditioning in hospitals, theatres, and hotels and in other places where safety is the prime consideration. At the present time, the use of carbon dioxide is limited to some extremely low temperature applications, particularly in the production of dry ice, i.e. solid CO_2 .

One of the main disadvantages of carbon dioxide is its high operating pressure, which under standard conditions of -15°C and 30°C is 22.2 bar and 71 bars respectively. The high suction pressure gives a small piston displacement resulting in smaller compressors, but this requires the use of extra heavy equipment and piping. Normal condensing temperatures are near the critical temperature of 31°C for carbon dioxide. This leads to excessive power costs, nearly three times those of other refrigerants. Carbon dioxide is non-miscible in oil and, therefore, does not dilute the oil in the crankcase of the compressor. Like ammonia, carbon dioxide is lighter than oil.

Next to water, carbon dioxide is the cheapest refrigerant. But its high pressure, and the difficulty in leak detection, results in higher replacement cost compared to that associated with ammonia. Because of these disadvantages, carbon dioxide was replaced when other safe refrigerants became available. Leak detection is carried out by soap solution only.

Water

The principal advantage of water as a refrigerant is its availability at virtually no cost. Another obvious advantage of water is that it is non-toxic and non-flammable. The two disadvantages of water are its high freezing point and high vacuum required. The very large volume of water vapour at such low operating pressures makes the use of ordinary compression equipment impossible.

In the past the steam-jet system with water as the refrigerant has had some use in large air-conditioning systems. The lithium bromide absorption refrigeration system, using water as refrigerant, is extensively used in large air-conditioning applications.

Sulphur dioxide

Sulphur dioxide was the most used domestic refrigerant in the 1920s and 1930s, having been replaced first by methyl chloride and later by the more desirable fluorocarbon refrigerants. It is highly toxic, but non-flammable and nonexplosive. As sulphur dioxide is not oil miscible, it simplifies the problem of oil return, which accounts for the relative popularity of sulphur dioxide for small automatic equipment used in the earlier days.

Like most common refrigerants, sulphur dioxide in the pure state is non-corrosive to metals normally used in the refrigerating systems. However, it combines with moisture to form sulphurous acids (H_2SO_3) and sulphuric acid (H_2SO_4), both of which are highly corrosive.

Methyl chloride

Methyl chloride is a halocarbon of the methane series. It has many of the properties desirable in a refrigerant, which accounts for its wide use in the past in both domestic and commercial

applications. Methyl chloride is corrosive to aluminium, zinc, magnesium and the compounds formed in combinations with these materials. It is also both flammable and explosive. In the presence of moisture, methyl chloride forms a weak hydrochloric acid, which is corrosive to both ferrous and non-ferrous metals. Since natural rubber and synthetic neoprene are dissolved by methyl chloride, neither is a suitable gasket material for use in methyl chloride systems.

Halocarbon refrigerants

Due to limitations of the other refrigerants, studies were made to investigate the possibility of using synthetic chemicals as refrigerants. The outcome of these studies was the development of the halocarbon group of refrigerants. The refrigerants of this group are non-flammable and most of them are non-toxic. They are available with evaporating temperature at atmospheric pressure from approximately 100°C below zero to 100°C above zero. Thus, a refrigerant can be chosen from this group that will fit practically any application.

All refrigerants in this family are derivatives of the hydrocarbons, methane and ethane. Refrigerants of this family are called halogenated hydrocarbons, or more simply halocarbons, because chlorine and fluorine are part of a family of chemicals called halogens.

Leaks in halocarbon systems may be detected in two ways—by a special torch or by an electronic leak detector. The presence of any halogen will change the colour of a blue gas flame in the presence of copper to green or purple. Another halogen leak detection device depends on a cell whose electrical characteristics change in the presence of halocarbon gases. A change in electric current through this cell triggers a relay, which operates a light or buzzer to indicate a leak. One unsafe characteristic of all halocarbons is, that they break down at high temperature to give very irritating and toxic compounds. The temperature at which this happens is high enough and does not occur in an operating system.

Refrigerant-11 (CCl_3F): It is non-corrosive, non-toxic and non-flammable refrigerant. The low operating pressures and the relatively high compressor displacement required necessitate the use of a centrifugal compressor. Like other fluorocarbon refrigerants, Refrigerant-11 dissolves natural rubber. Refrigerant-11 is used mainly in the air-conditioning of small office buildings, factories, stores, theatres, etc. A halide torch may be used for leak detection.

Refrigerant-12: It has been by far the most widely used refrigerant until recent years. It is a completely safe refrigerant in that it is non-toxic, non-flammable, and non-explosive. Moreover, it is a highly stable compound and, therefore, difficult to break down even under extreme operating conditions. Refrigerant-12 condenses at moderate pressures under normal atmospheric conditions and is a suitable refrigerant for use in high, medium and low temperature applications and with all the three types of compressors. Refrigerant-12 is oil miscible under all operating conditions that not only simplifies the problem of oil return but also tends to increase the efficiency and capacity of the system. The horsepower required per kilowatt of capacity compares favourably with that required for other commonly used refrigerants. However, this refrigerant is being phased out because of its damaging effect on ozone layer.

Refrigerant-22: This refrigerant was developed primarily as a low temperature refrigerant and it is extensively used in domestic and farm freezers and in commercial and industrial low temperature systems down to evaporator temperatures as low as -70°C . It is also widely used in packaged air conditioners, where, because of space limitations, the relatively small compressor displacement required is a definite advantage. Both the operating pressures and the adiabatic discharge temperature are higher for Refrigerant-22 than for Refrigerant-12. Horsepower

requirements are approximately the same. Although miscible with oil at temperatures prevailing in the condensing sections, Refrigerant-22 will often separate from the oil in the evaporator. With proper piping design this problem may be surmounted. Oil separators should always be used in low temperature applications.

The ability of Refrigerant-22 to absorb moisture is considerably greater than that of Refrigerant-12 and therefore less trouble is experienced with freeze ups in Refrigerant-22 systems. Being a fluorocarbon, Refrigerant-22 is a safe refrigerant. A halide torch may be used for leak detection. This refrigerant is also being phased out.

Refrigerant-134a: Due to phasing out of Refrigerant-12, which was the most widely used refrigerant up to recent years, there was frantic search for a suitable replacement. At this point in time, R134a is considered to be the most direct substitute for R12 and as a replacement in higher temperature applications. Its net refrigerating effect (based on -15°C evaporator and 30°C condenser temperature) is 151 kJ/kg against 116.58 kJ/kg for R12.

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REVIEW EXERCISES

- 4.1 Describe the desirable and undesirable features of ammonia for use as a refrigerant.
- 4.2 Why is it that most of the currently popular halocarbon refrigerants are being phased out?
- 4.3 What piping material would you recommend for ammonia systems?
- 4.4 What is an azeotrope? Which popular halocarbon refrigerant is an azeotrope? For what reason it is often used instead of R22?
- 4.5 Why are positive condenser and evaporator pressures desirable?
- 4.6 Briefly describe the damages done to ozone layers by chlorofluorocarbon (CFC) refrigerants.

5

Vapour Absorption Cycle

Vapour compression refrigeration cycles discussed in Chapter 4 are heat pumps in which the energy input to operate the system is in the form of mechanical work. In certain circumstances it is desirable to use heat directly as the operating energy, for instance in a plant where there is waste heat from co-generation or process steam plants. The absorption refrigeration system developed by Ferdinand Carré in France, and later patented in the United States in 1860 during the period of civil war, is such a heat operated cycle. These cycles are also economical where inexpensive heat energy sources such as geothermal energy, solar heat and cheap natural gas are available in abundance.

5.1 SIMPLE VAPOUR ABSORPTION CYCLE

The simple vapour absorption refrigeration cycle, as shown in Figure 5.1, is similar to the vapour compression cycle in many ways, however, it differs in the method employed for compressing the fluid. In this cycle the compressor is substituted by a generator, an absorber and a pump. There is another difference, that is, in addition to the *refrigerant* another fluid known as *absorbent* is used in vapour absorption cycles.

Two common combinations of refrigerant-absorbent are: (a) a solution of lithium bromide in water, water being refrigerant and lithium bromide (LiBr), which is a highly hygroscopic salt, being the absorbent, and (b) aqua-ammonia, where ammonia (NH_3) is the refrigerant and water or aqua is the absorbent. The former combination is used in air conditioning but is not suitable for low temperature applications. Due to the toxicity of ammonia the latter combination is not used in air-conditioning applications. The absorbent should have a greater affinity for the refrigerant and the two must be mutually soluble over the desired range of operating conditions. Ideally, the absorbent should have a low volatility and the refrigerant should have a reasonably high latent heat.

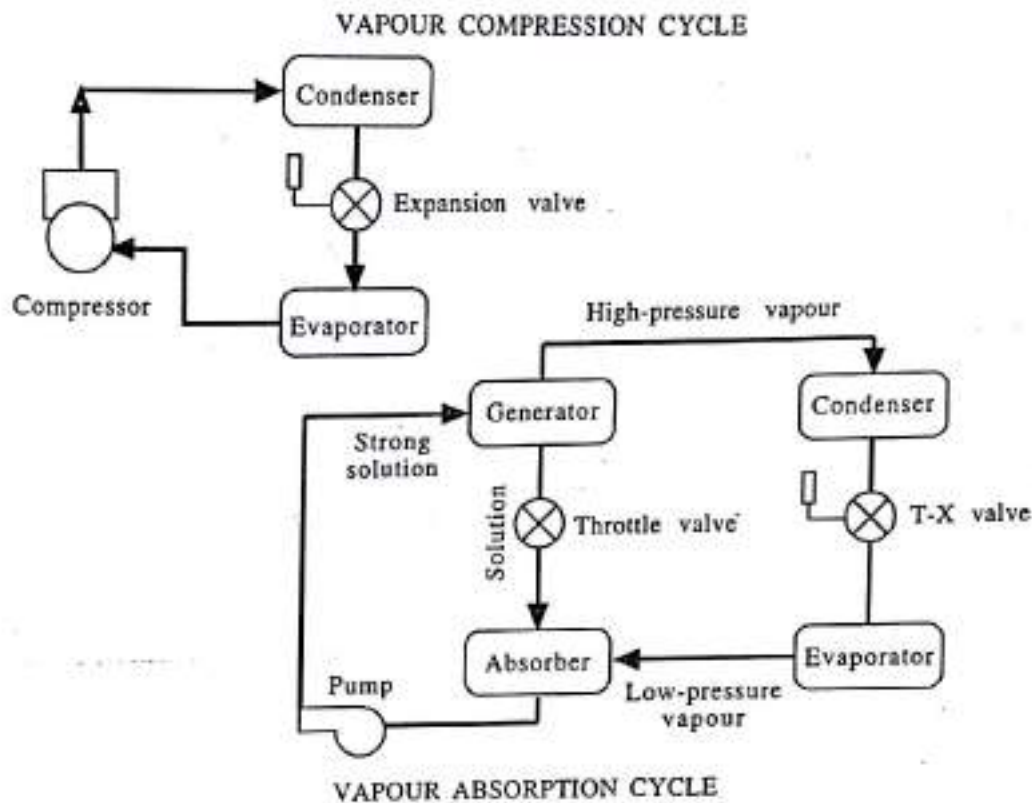


FIGURE 5.1 Comparison of simple vapour compression and vapour absorption cycles.

The operating principle is explained in terms of the various processes that the absorption cycle undergoes in the course of an aqua ammonia cycle as shown in Figure 5.2. In this cycle, water (aqua) is the absorbent and ammonia is the refrigerant. After leaving the evaporator the

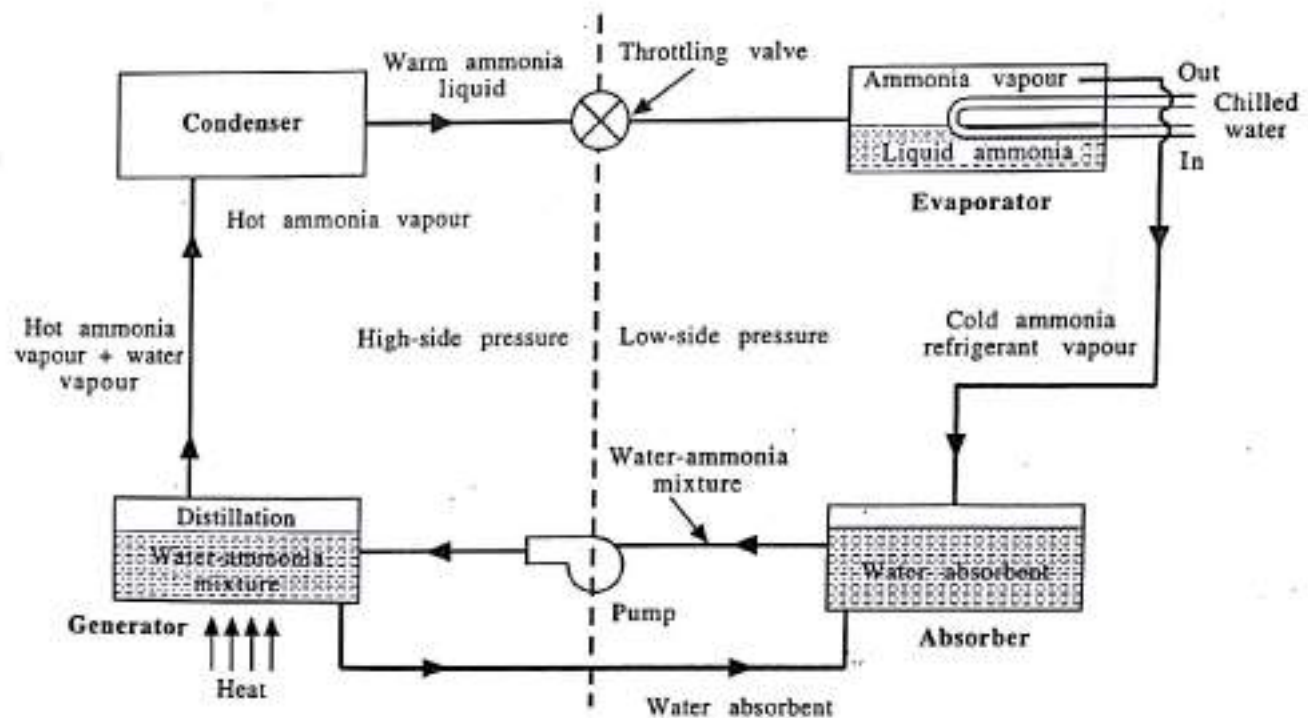


FIGURE 5.2 Schematic representation of an aqua ammonia absorption cycle.

refrigerant (ammonia) is absorbed by the liquid solution that has a very high affinity for the refrigerant in the absorber. The strong, i.e. the refrigerant-rich solution formed in the absorber is then pumped mechanically into the generator at a higher pressure. The refrigerant is distilled from the solution by the application of heat in the generator, and is then passed to the condenser, after which the weakened solution returns to the absorber to repeat its cycle. The refrigerant that goes to the condenser gets condensed and then expands through the expansion valve before entering into the evaporator.

The work done on compression is less than that done in the vapour compression cycle since pumping a liquid requires much less work than compressing a vapour between the same pressures. But in this cycle, a heat input to the generator is required. But unlike vapour compression cycles, the heat, however, may be supplied by any convenient form, i.e. gas, steam, or electricity.

5.2 PRACTICAL ABSORPTION SYSTEMS

The basic components of a practical vapour absorption cycle are described below. In order to improve efficiency, some items of additional auxiliary equipment are added in practical systems. However, detailed studies need to be carried out to ensure the most desirable balance between the initial cost and the operating cost. The purposes and locations of the main components and auxiliaries are discussed in the following paragraphs.

Generator

The generator unit is basically a shell partly filled with a strong solution and automatically maintained at a constant level covering the submerged steam or the hot gas/water tubes. Energy is provided from an external source such as process/waste steam or even solar-heated water.

Absorber

In industrial systems the absorber may consist of one or more shells similar to the shell-and-tube type condensers containing water cooling tubes. The weak solution from the generator and the ~~strong solution~~ ^{refrigerant} from the evaporator are introduced to the absorber and the resulting strong solution is drawn off from the absorber. The purpose of the cooling water in the tubes is to remove the heat of condensation and heat of absorption as the vapour condenses and is absorbed by the weak solution. The lower aqua ammonia temperature in the absorber allows more absorption of vapour, leading to a refrigerant rich solution.

Heat exchanger

The most common auxiliary is the heat exchanger (Figure 5.3) located between the absorber and the generator to heat the cool solution from the absorber on its way to the generator and to cool the solution returning from the generator to the absorber. This operation economises on the amount of cooling needed for the absorber and the amount of heat needed for the generator. Shell-and-tube heat exchangers are commonly used in large installations. Sometimes a second heat exchanger may be used to subcool the refrigerant entering the expansion valve as shown later in Figure 5.5.

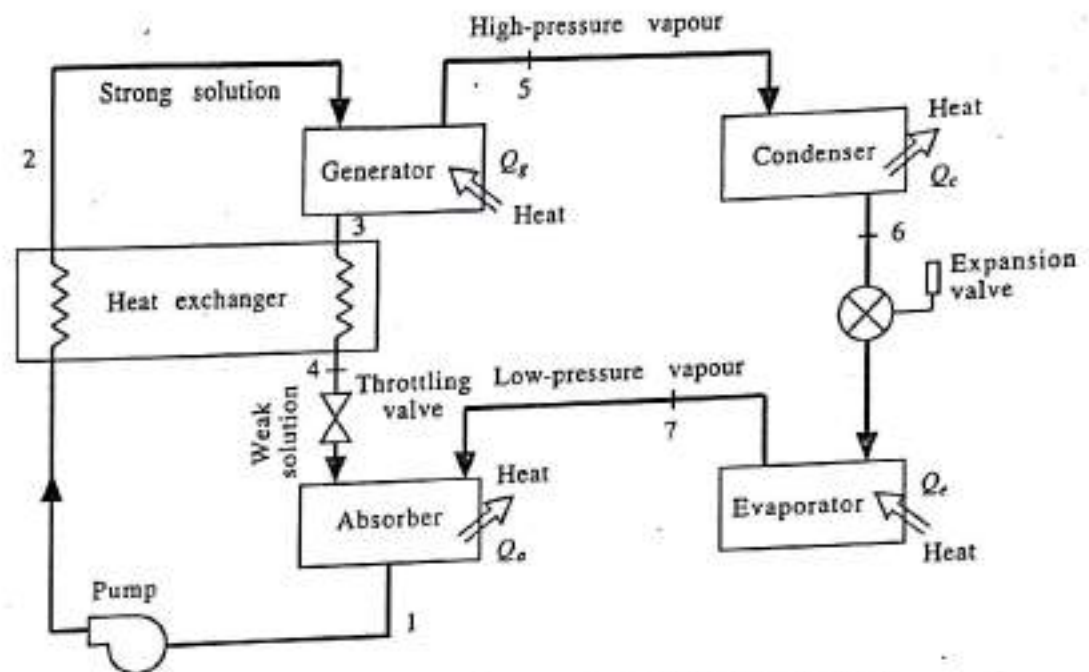


FIGURE 5.3 Use of heat exchanger in an absorption cycle.

For aqua ammonia systems (Figure 5.4) two additional components—^{اناليزر} analyzer and ^{مستطيل} rectifier—are used in addition to the generator, absorber, condenser, evaporator and heat

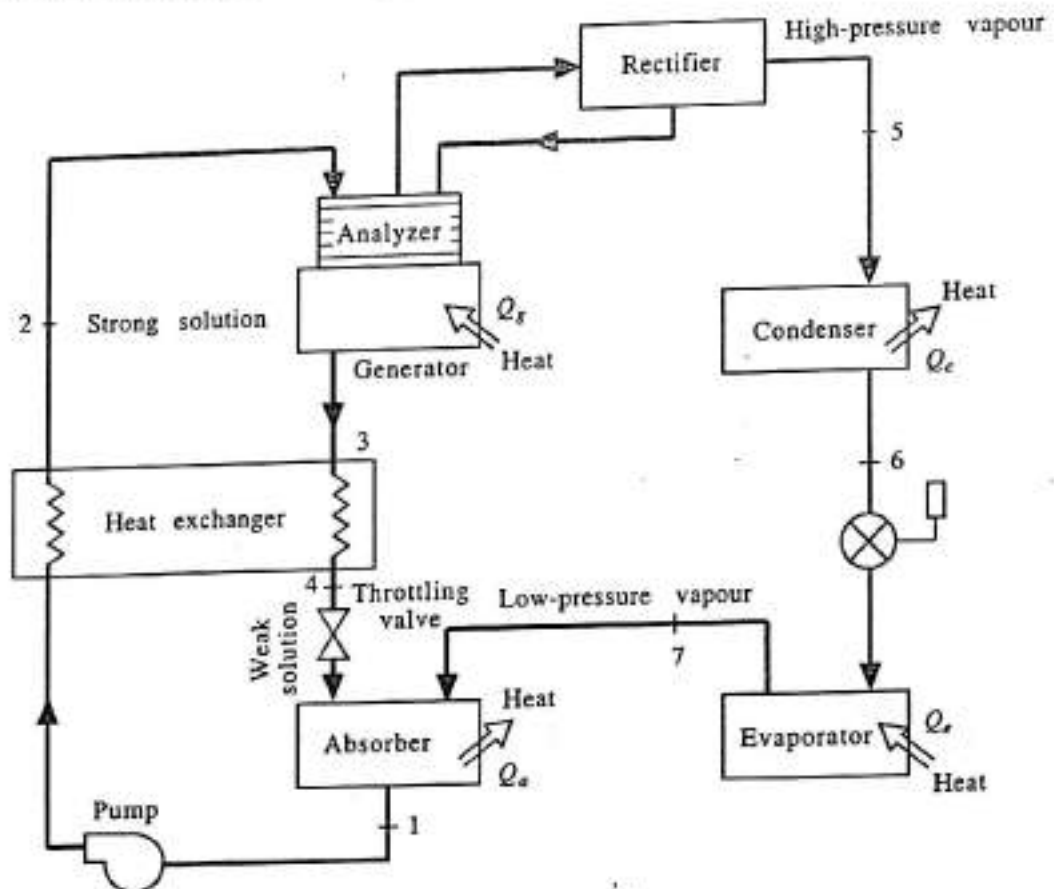


FIGURE 5.4 Aqua ammonia absorption cycle with analyzer and rectifier.

exchanger(s). Both the analyzer and the rectifier are used to remove water vapours from the ammonia going to the evaporator. When ammonia is evaporated in the generator, some water is also vaporized and tends to flow into the condenser along with the refrigerant. The analyzer and rectifier are added to the system to separate the water vapours leaving the generator so as to send dry ammonia to the condenser.

Analyzer

The analyzer may be built as an integral part of the generator or as a separate equipment. Both the strong aqua from the absorber and the returning aqua from the rectifier are introduced from the top and these flow downward into the generator. In this way the escaping water vapour is cooled by the incoming solution and most of the vapour is condensed. As the aqua is heated by the vapour, less heat is required in the generator, thus effecting economy.

Rectifier

The purpose of a rectifier is to cool further the vapour leaving the analyzer so that the residual vapour is condensed, leaving the dehydrated ammonia vapour to flow into the condenser. The rectifier is generally water-cooled and may be similar in construction to conventional water-cooled condensers. Temperatures of 40 to 50°C will usually dehydrate ammonia sufficiently.

Figure 5.5 shows an aqua ammonia absorption cycle—complete with the basic components and all the auxiliaries including two heat exchangers.

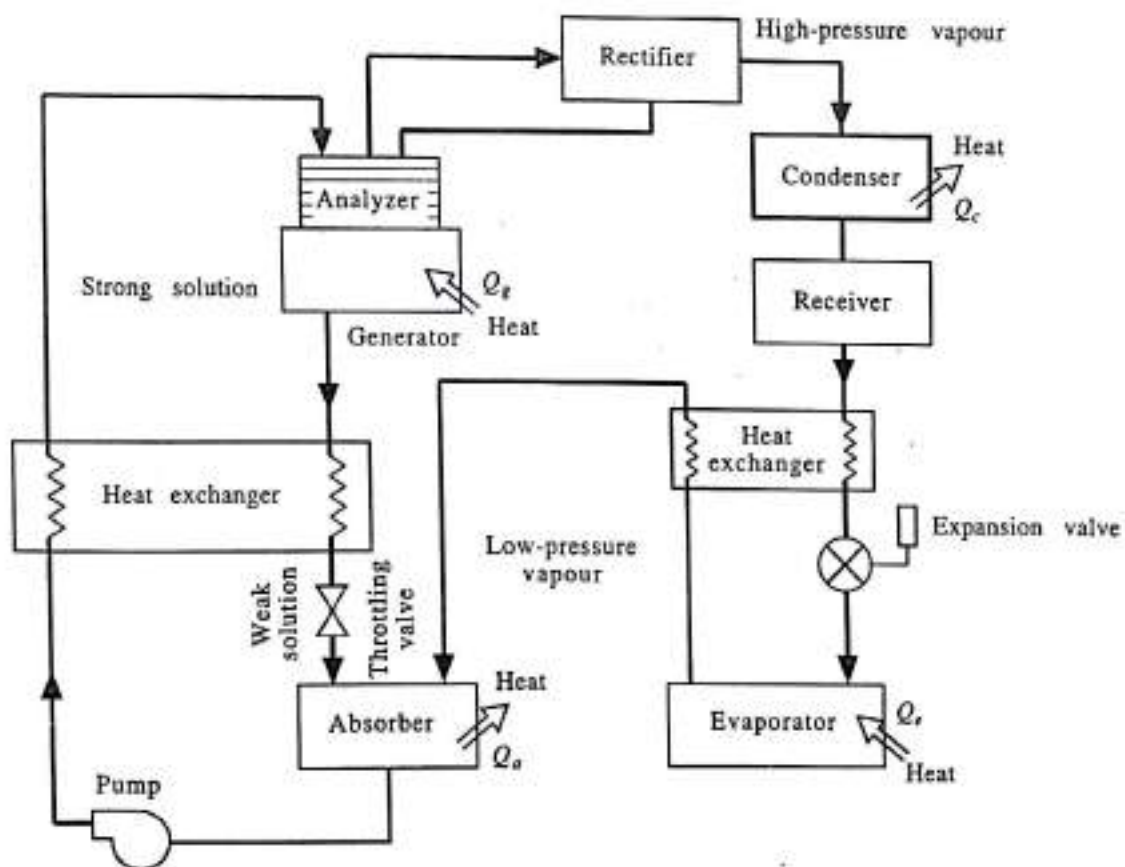


FIGURE 5.5 Absorption cycle with auxiliaries and two heat exchangers.

One of the promising areas of development is the exploitation of solar energy for absorption cycles. Solar collectors can be used to heat water and in turn supply the stored energy in the generator. Figure 5.6 shows a schematic layout of a solar-powered absorption cycle.

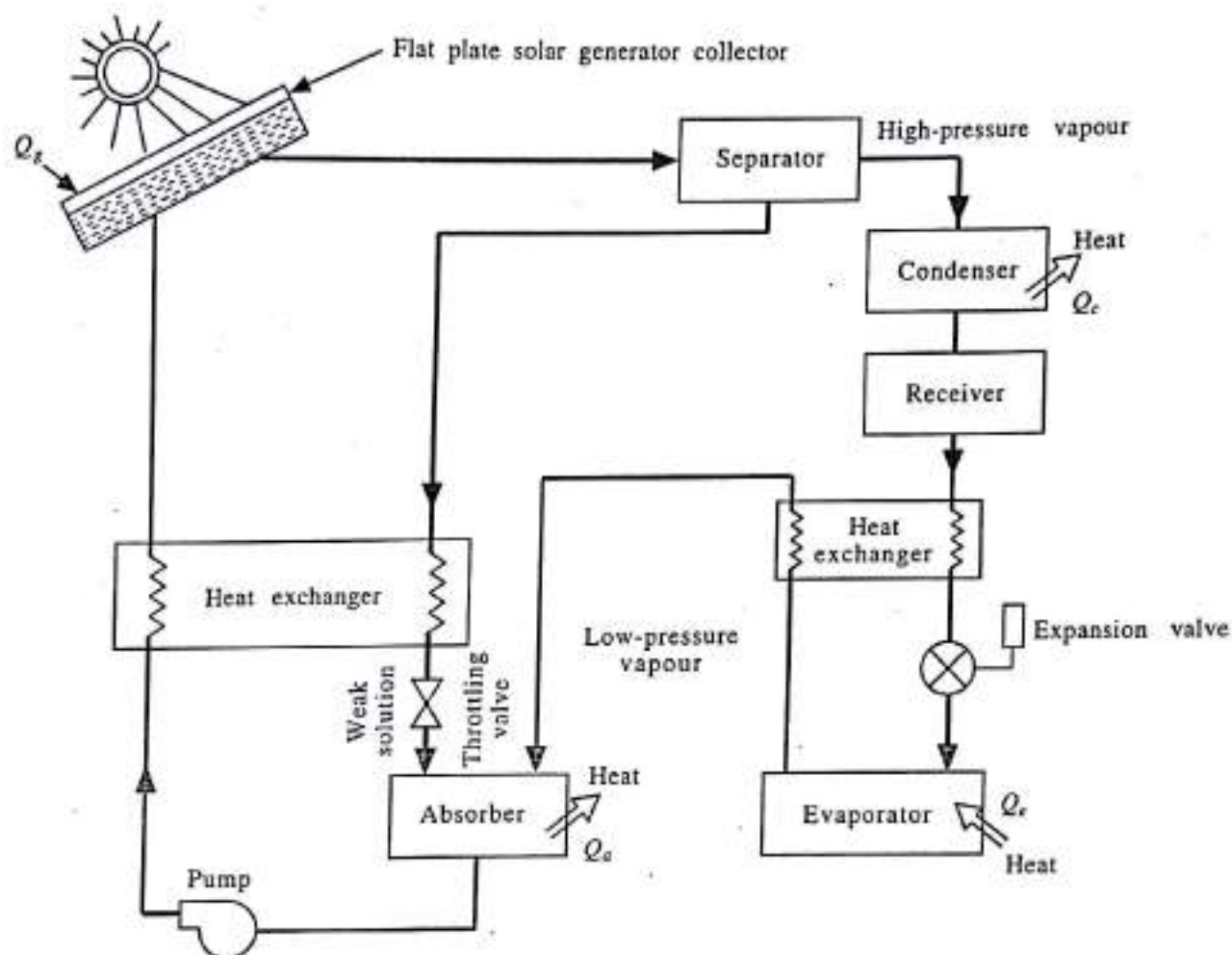


FIGURE 5.6 Solar powered absorption cycle.

5.3 PROPERTIES OF REFRIGERANTS AND ABSORBENTS

The properties of refrigerants were discussed in Chapter 4. Here the desirable properties of absorbent and refrigerant-absorbent combinations will be discussed.

Absorbents

One of the important properties of an absorbent is that it should have a greater affinity for the refrigerant than the ordinary law of solubility would indicate. It should have negligible vapour pressure at generator temperature compared to the refrigerant. The absorbent should be stable and capable of withstanding the heating operation at the maximum temperatures encountered in the generator. It should remain liquid throughout the cycle and should have low specific heat, low surface tension and low viscosity. Lithium bromide (LiBr) is a highly hygroscopic salt with great affinity for water. Similarly, ammonia is highly soluble in water, but the solubility decreases with temperature rise.

Refrigerant-absorbent combinations

The desirable refrigerant-absorbent combination should be highly soluble at conditions in the absorber but should have low solubility at conditions in the generator. The combination must not have non-reversible chemical reaction in the operating temperature range. The two substances should have a large difference in their normal boiling points, as high as around 200°C , so that the absorbent exerts negligible vapour pressure at the generator temperature. The viscosity should be low to minimize pump work. The lithium bromide-water system is operated under high vacuum that necessitates the use of a purge unit to remove air and non-condensable gases. For these systems, the temperature needed to be maintained in the generator should be less than 120°C to avoid corrosion. In the case of ammonia-water system, the difference in their boiling temperatures is relatively low (138°C) resulting in the evaporation of some water vapour that is carried away by the refrigerant, i.e. ammonia.

Although ammonia-water and lithium bromide-water are the most popular combinations, other combinations have been tried such as methylene chloride and dimethoxy tetraethylene glycol. In the lithium bromide-water system, water being the refrigerant it cannot be used in a low-temperature application because water will freeze at 0°C . Consequently, its use is limited to air-conditioning applications only.

5.4 COMPARISON OF VAPOUR COMPRESSION AND ABSORPTION CYCLES

The major differences between the two cycles are as follows:

1. Unlike the vapour compression cycle, the vapour absorption cycle is a heat-operated cycle. The need for electric power is only to operate a pump, which is a small fraction of the power needed to drive the compressor in a vapour compression cycle. Use of direct low grade heat compared to electric power, which is high grade energy, is inherently economical.
2. The vapour compression cycle transforms the low-pressure vapour to high-pressure vapour by a compressor, whereas in the absorption cycle the same is accomplished by a generator, an absorber and a pump.
3. In an absorption system the only moving part is the pump, and that too is of small capacity. Hence, essentially the operation of such a system is very quiet and noise-free. For the same reason, the wear and tear is also minimal resulting in a longer life span compared to that of a vapour compression system.
4. The absorption units generally take up more space than that required by the compression systems; however, the equipment can be located outdoors and assembled as a vertical unit.
5. Absorption cycles are almost as efficient at part load as at full load. Load fluctuations are taken care of by controlling the quantity of the liquid solution and the quantity of steam supplied to the generator.

5.5 ABSORPTION CYCLE ANALYSIS

In order to perform thermal calculations on the absorption refrigeration cycle, in addition to temperature-pressure-concentration diagram for lithium bromide-water solutions, the enthalpy data must be available for the working substances at all important points of the cycle.

Neglecting the pump work the coefficient of performance of the absorption cycle COP_{abs} is defined as

$$COP_{abs} = \frac{\text{rate of refrigeration}}{\text{rate of heat supply to the generator}} \quad (5.1)$$

and considering pump work, the COP_{abs} becomes

$$COP_{abs} = \frac{RE}{Q + W_p} \quad (5.2)$$

where Q is the heat supplied to the generator and W_p is the pump work.

The COP of an absorption cycle is usually less than 1 compared to 3 and above for the vapour compression cycle, however, the absorption cycle being a heat-operated one, the cost of energy input is considerably less (approximately one-third) than that in a work-operated compression cycle. Absorption cycles become economic when waste steam is available for the operation of the system.

Heat balances across the main components, i.e. the condenser, the evaporator, the generator and the absorber are fairly straightforward and the same have been explained in the worked-out examples.

Vapour absorption cycle calculation

Basically in the calculation of performance of a vapour absorption cycle, two charts need to be used—(a) the equilibrium chart for aqueous lithium bromide solutions (Figure B3, Appendix B) and (b) the enthalpy-concentration diagram for lithium bromide-water solutions (Figure B4, Appendix B). With the known solution and refrigerant temperatures the corresponding refrigerant concentration (percentage) in the mixture can be obtained from the equilibrium chart for aqueous lithium bromide solutions. Then with the known refrigerant concentration (mass percentage) and solution temperature we can get the enthalpy of the solution from the concentration diagram for lithium bromide-water solutions.

As regards the enthalpy values of water, the same can be obtained from the Table of saturated water.

Commercial absorption units

In commercial absorption plants, the condenser and generator are combined into one vessel since these components operate at the same pressure. Similarly, since the evaporator and the absorber operate at the same pressure, these components may also be combined into one vessel with an internal separator between the high- and low-pressure chambers.

Cooling water from the cooling tower passes in series through the absorber and condenser, extracting heat from both the components.

5.6 OPERATION OF ABSORPTION CYCLES

For the efficient operation of a water-lithium bromide cycle, crystallization of lithium bromide needs to be avoided. Crystallization results in the formation of a slush, which can block the flow in a pipe, and interrupt the operation of the absorption unit.

With the reduction in refrigeration load capacity, control is necessary to avoid freezing of refrigerant water. The three common methods used are:

- (a) Reducing the refrigerant water solution flow rate by the pump
- (b) Reducing the generator temperature
- (c) Increasing the condensing temperature.

EXAMPLE 5.1

The operating temperatures of a lithium bromide-water absorption system, as shown in Figure 5.7, are as follows:

Generator	100°C
Condenser	35°C
Evaporator	5°C
Absorber	30°C

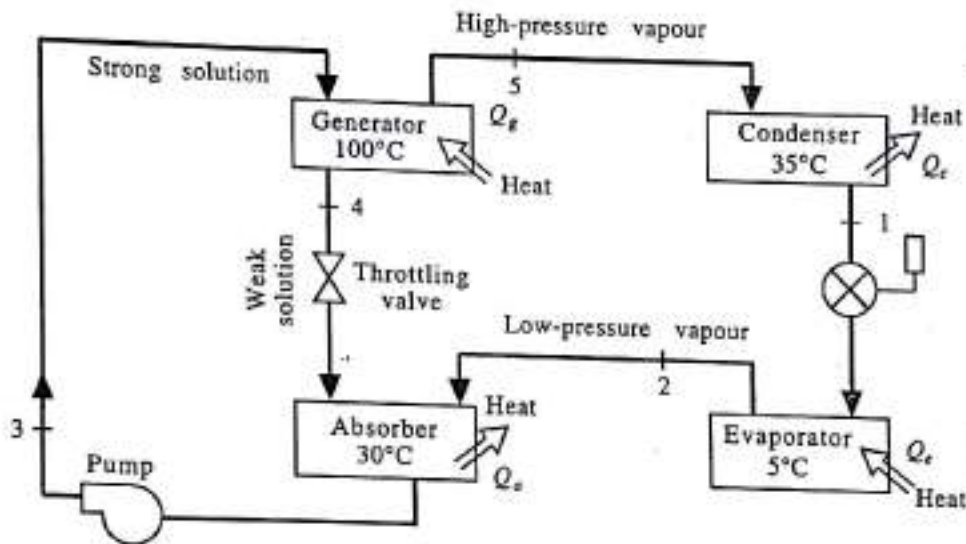


FIGURE 5.7 Example 5.1: Lithium bromide-water absorption system.

The mass flow rate delivered by the aqua pump is 0.4 kg/s. Calculate:

- The heat supplied to the generator
- The heat rejected at the condenser and absorber
- The cooling produced by the evaporator
- The coefficient of performance of the cycle.

Solution

Total mass flow balance:

$$\dot{m}_3 = \dot{m}_4 + \dot{m}_5 = 0.4 \text{ kg/s}$$

Lithium bromide balance:

$$\dot{m}_3 x_3 = \dot{m}_4 x_4 \quad \text{or} \quad 0.4 \times 0.53 = \dot{m}_4 \times 0.69$$

$$\therefore \dot{m}_4 = 0.307 \text{ kg/s}$$

and

$$\dot{m}_5 = \dot{m}_3 - \dot{m}_4 = 0.4 - 0.307 = 0.093 = \dot{m}_1 = \dot{m}_2$$

where x_3 and x_4 are the concentrations of lithium bromide in the solution and calculated from the equilibrium chart (Figure 5.8) for the aqueous lithium bromide solution as follows:

- For 30°C absorber temperature and 5°C evaporator temperature, $x_3 = 0.53$
- For 100°C generator temperature and 35°C condenser temperature, $x_4 = 0.69$

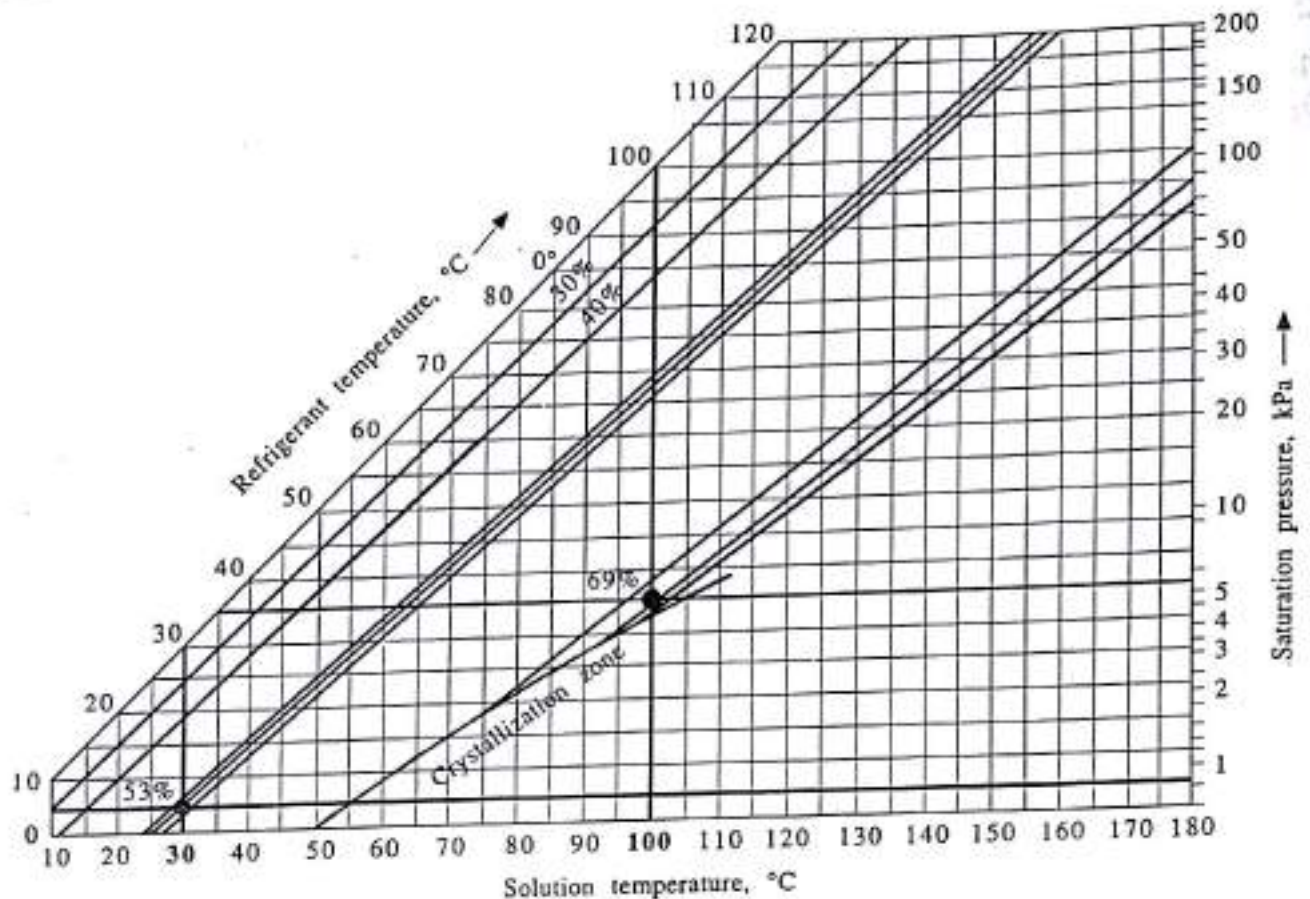


FIGURE 5.8 Example 5.1: chart.

From the enthalpy of lithium bromide-water solution chart:

$$h_3 = h \text{ at } 30^\circ\text{C} \quad \text{and} \quad x = 0.53 = 65 \text{ kJ/kg}$$

$$h_4 = h \text{ at } 100^\circ\text{C}, \quad x = 0.69 = 275 \text{ kJ/kg}$$

Enthalpies of water and liquid vapour are:

$$h_5 = h_{100^\circ\text{C}} = h_g = 2676 \text{ kJ/kg}$$

$$h_1 = h_{35^\circ\text{C}} = h_f = 146.68 \text{ kJ/kg}$$

$$h_2 = h_{5^\circ\text{C}} = h_g = 2510.6 \text{ kJ/kg}$$

(a) Heat transfer rate to the generator:

$$\begin{aligned} Q_g &= \dot{m}_5 h_5 + \dot{m}_4 h_4 - \dot{m}_3 h_3 \\ &= 0.093(2676) + 0.307(275) - 0.4(65) \\ &= 248.8 + 84.42 - 26 \\ &= 307.22 \text{ kW} \end{aligned}$$

(b) Heat transfer rate from the condenser

$$\begin{aligned} Q_c &= \dot{m}_5 h_5 - \dot{m}_1 h_1 \\ &= 0.093 \times 2676 - 0.093 \times 146.68 = 248.87 - 13.64 = 235.23 \text{ kW} \end{aligned}$$

Heat transfer rate from the absorber:

$$\begin{aligned} Q_a &= \dot{m}_4 h_4 + \dot{m}_2 h_2 - \dot{m}_3 h_3 \\ &= 0.307 \times 275 + 0.093 \times 2510.6 - 0.4 \times 65 \\ &= 84.43 + 233.48 - 26 \\ &= 291.91 \text{ kW} \end{aligned}$$

(c) Heat transfer rate to the evaporator:

$$\begin{aligned} Q_e &= \dot{m}_2 h_2 - \dot{m}_1 h_1 \\ &= 0.093 \times 2510.6 - 0.093 \times 146.68 \\ &= 233.48 - 13.64 = 219.84 \text{ kW} \end{aligned}$$

(d) Coefficient of performance:

$$\begin{aligned} \text{COP} &= \frac{219.84}{307.22} \\ &= 0.715 \end{aligned}$$

EXAMPLE 5.2

A lithium bromide-water absorption system incorporating a heat exchanger, as shown in Figure 5.9, operates at the following temperatures:

Generator temperature	= 100°C
Condenser temperature	= 36°C
Absorber temperature	= 28°C
Evaporator temperature	= 10°C
Mass flow rate delivered by the pump	= 0.55 kg/s
Solution temperature entering the generator	= 50°C

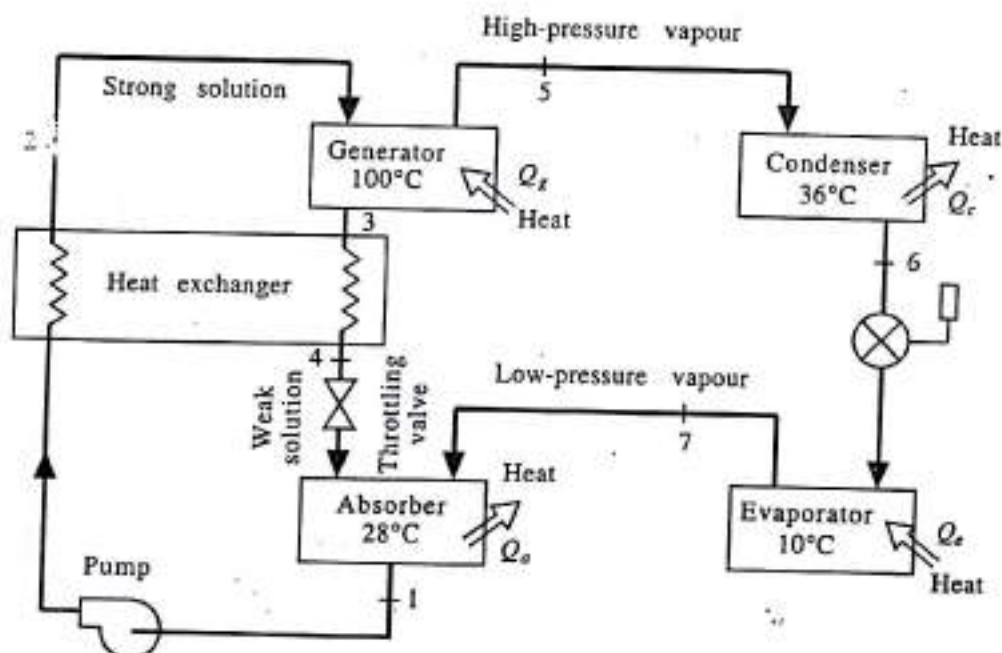


FIGURE 5.9 Example 5.2: Lithium bromide-water absorption system.

Calculate:

- The rates of heat transfer to or from the generator, the condenser, the evaporator, and the absorber
- The coefficient of performance of the cycle.

Solution

Here,

$$\dot{m}_1 = \dot{m}_2 = 0.55 \text{ kg/s}$$

Total mass flow balance about the generator:

$$\dot{m}_2 = \dot{m}_3 + \dot{m}_5 = \dot{m}_1 = 0.55 \text{ kg/s}$$

From the equilibrium chart (Figure 5.10) for the aqueous lithium bromide solution:

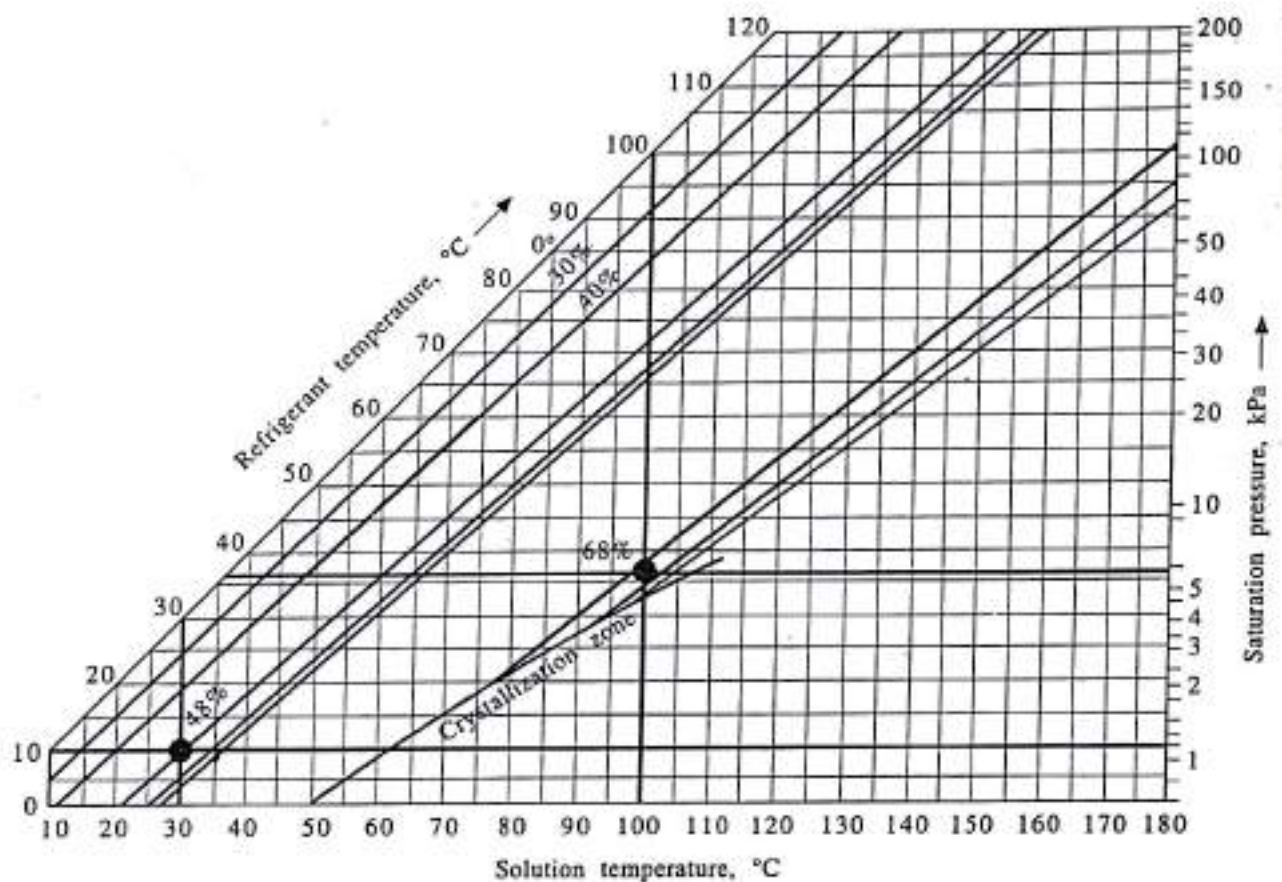


FIGURE 5.10 Example 5.2: chart.

For 28°C absorber temperature and 10°C evaporator temperature, $x_2 = 0.48$

For 100°C generator temperature and 36°C condenser temperature, $x_3 = 0.68$

Lithium bromide balance:

$$\dot{m}_2 x_2 = \dot{m}_3 x_3 \quad \text{or} \quad 0.55 \times 0.48 = \dot{m}_3 \times 0.68$$

$$\dot{m}_3 = 0.388 \text{ kg/s} = \dot{m}_4$$

and $\dot{m}_5 = \dot{m}_6 = \dot{m}_7 = 0.55 - 0.388 = 0.162 \text{ kg/s}$

From the enthalpy of lithium bromide-water solution chart:

$$h_1 = 51 \text{ kJ/kg (} h \text{ at } 28^\circ\text{C, } x = 0.48) \text{ from the enthalpy concentration diagram for lithium bromide-water solutions}$$

$$h_3 = 275 \text{ kJ/kg (} h \text{ at } 100^\circ\text{C, } x = 0.68)$$

Enthalpies of water and liquid vapour are:

$$h_5 = h_{100^\circ\text{C}} = h_g = 2676 \text{ kJ/kg}$$

$$h_6 = h_{36^\circ\text{C}} = h_f = 150.86 \text{ kJ/kg}$$

$$h_7 = h_{10^\circ\text{C}} = h_g = 2519.8 \text{ kJ/kg}$$

Heat transfer rate from the condenser:

$$\begin{aligned} Q_c &= \dot{m}_5 h_5 - \dot{m}_6 h_6 \\ &= 0.162 \times 2676 - 0.162 \times 150.86 \\ &= 433.51 - 24.44 \\ &= 409.07 \text{ kW} \end{aligned}$$

Heat transfer rate to the evaporator:

$$\begin{aligned} Q_e &= \dot{m}_7 h_7 - \dot{m}_6 h_6 \\ &= 0.162 \times 2519.8 - 0.162 \times 150.86 \\ &= 408.2 - 24.44 = 383.76 \text{ kW} \end{aligned}$$

The temperature of 48% solution leaving the heat exchanger at 2 is 50°C . From the enthalpy concentration diagram, $h_2 = 101 \text{ kJ/kg}$.

$$\begin{aligned} Q_{\text{heat exchanger}} &= \dot{m}_1 (h_2 - h_1) \\ &= 0.55 (101 - 51) = 28.05 \text{ kW} \end{aligned}$$

or $28.05 = \dot{m}_3 (h_3 - h_4) = 0.388 (275 - h_4)$

$$\therefore h_4 = 202.71 \text{ kJ/kg}$$

From the chart, 68% solution with an enthalpy of 202.71 kJ/kg has a temperature of 58°C .
Heat transfer rate to the generator:

$$\begin{aligned} Q_g &= \dot{m}_5 h_5 + \dot{m}_3 h_3 - \dot{m}_2 h_2 \\ &= 0.162 \times 2676 + 0.388 \times 275 - 0.55 \times 101 \\ &= 433.51 + 106.70 - 55.55 \\ &= 484.66 \text{ kW} \end{aligned}$$

Heat transfer rate from the absorber:

$$\begin{aligned}
 Q_a &= \dot{m}_7 h_7 + \dot{m}_4 h_4 - \dot{m}_1 h_1 \\
 &= 0.162 \times 2519.8 + 0.388 \times 202.71 - 0.55 \times 51 \\
 &= 408.2 + 78.65 - 28.05 \\
 &= 458.8 \text{ kW}
 \end{aligned}$$

Coefficient of performance,

$$\begin{aligned}
 \text{COP} &= \frac{383.76}{484.66} \\
 &= 0.792
 \end{aligned}$$

EXAMPLE 5.3

In the planning stage for a cooling project, two options were considered: (a) vapour compression system and (b) vapour absorption system. The cost of fuel on a heating value basis is Rs 0.5 per mega joule. The COP and the relevant efficiencies for the two systems are tabulated below. At what cost of electricity would the operating cost for both the cycles be equal? Assume 100% steam cycle efficiency.

Vapour compression	Vapour absorption
COP = 3.2	COP = 0.75
Motor efficiency, $\eta_m = 85\%$	Boiler efficiency, $\eta_b = 80\%$

Solution

Per kWh of refrigeration:

$$\text{Input to absorption machine from steam} = \left(\frac{1}{0.75} \right) \text{ kWh}$$

$$\text{Input to steam from boiler} = \frac{1}{0.75} \times \frac{1}{\eta_s}, \text{ where } \eta_s (=100\%) \text{ is the steam cycle efficiency}$$

$$\text{Input to boiler from gas} = \frac{1}{0.75} \times \frac{1}{\eta_s} \times \frac{1}{0.8} = \frac{1.667}{\eta_s} = \frac{1.667}{1} = 1.667 \text{ kWh}$$

$$1 \text{ kWh} = 3600 \text{ kJ} = 3.6 \text{ MJ}$$

$$\text{Cost of gas} = 0.5 \times 3.6 \times 1.667 = \text{Rs } 3$$

$$\text{Input to vapour compression machine from motor} = \left(\frac{1}{3.2} \right) \text{ kWh}$$

$$\text{Input to motor from electricity} = \frac{1}{3.2} \times \frac{1}{0.85} = 0.368 \text{ kWh}$$

$$\therefore \text{Cost of 0.368 kWh of electricity is to be Rs } 3$$

$$\therefore \text{Cost of 1 kWh of electricity can be } \frac{3}{0.368} = \text{Rs } 8.15$$

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- Stoecker, W.F. and J.W. Jones, *Refrigeration and Air Conditioning*, McGraw-Hill, New York, 1982.

REVIEW EXERCISES

- 5.1 Name any two combinations of refrigerant-absorbent and their applications. Which combination is used in the air-conditioning application? Give reasons in support of your answer.
- 5.2 Why are analyzer and rectifier employed in the ammonia-water absorption system but not in a lithium bromide-water system?
- 5.3 What are the desirable characteristics of absorbent and the absorbent-refrigerant combinations in a vapour absorption refrigeration cycle?
- 5.4 What is the role of a heat exchanger in a vapour absorption cycle? Draw a schematic diagram of an absorption cycle with heat exchanger and explain how it improves the operational efficiency?
- 5.5 The operating temperatures of a lithium bromide-water absorption system, as shown in Figure 5.1, are as follows:

Generator	100°C
Condenser	35°C
Evaporator	10°C
Absorber	30°C

The mass flow rate delivered by the aqua pump is 0.5 kg/s. Calculate:

- (a) The heat supplied to the generator
 - (b) The heat rejected at the condenser and absorber
 - (c) The cooling produced by the evaporator
 - (d) The coefficient of performance of the cycle.
- 5.6 A lithium bromide-water absorption system incorporating a heat exchanger, as shown in Figure 5.3, operates at the following temperatures:

Generator temperature	= 100°C
Condenser temperature	= 36°C
Absorber temperature	= 28°C
Evaporator temperature	= 10°C
Mass flow rate delivered by the pump	= 0.65 kg/s
Solution temperature entering the generator	= 52°C

Calculate:

- (a) The rates of heat transfer to or from the generator, the condenser, the evaporator, and the absorber
- (b) The coefficient of performance of the cycle.

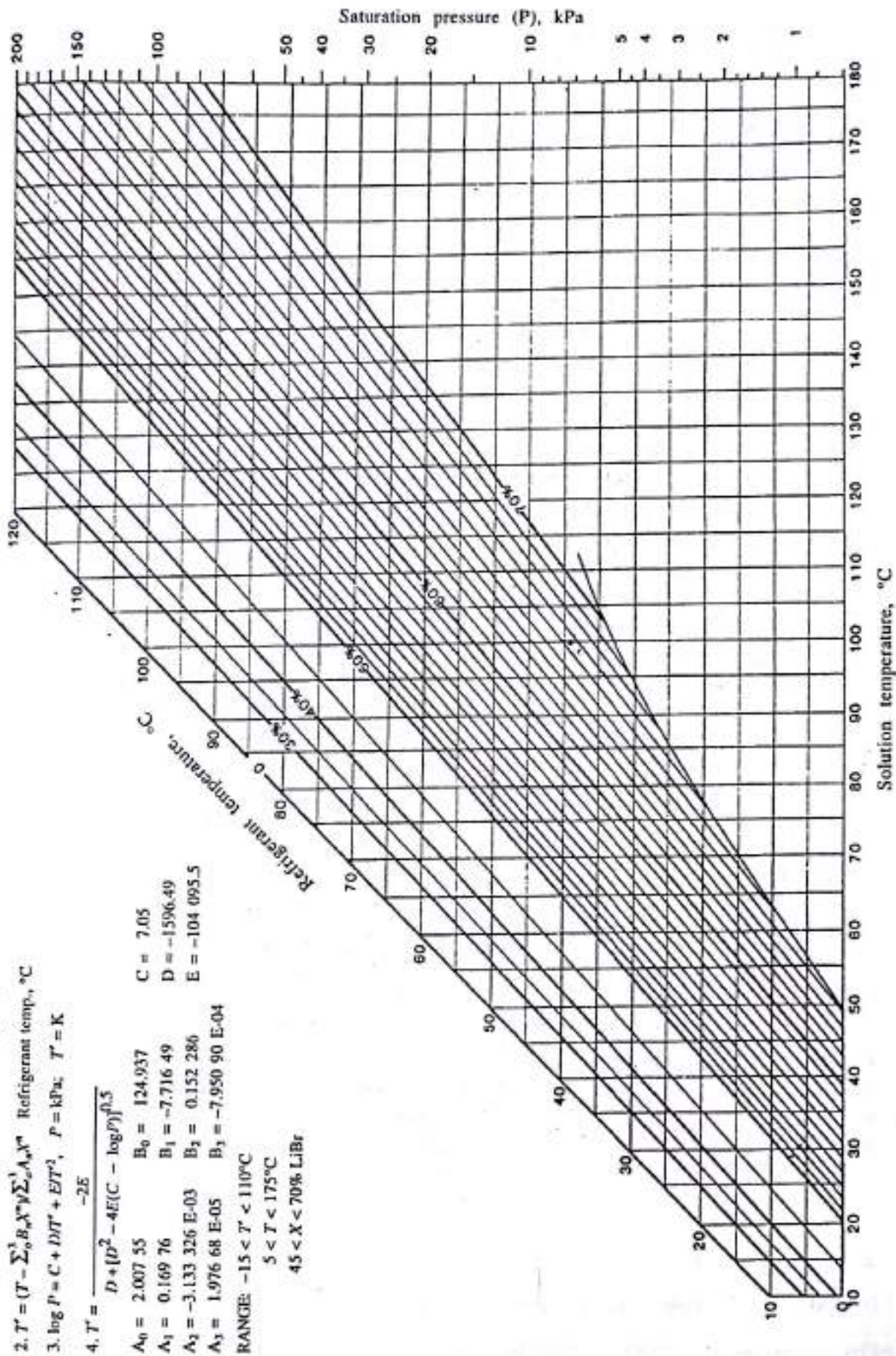
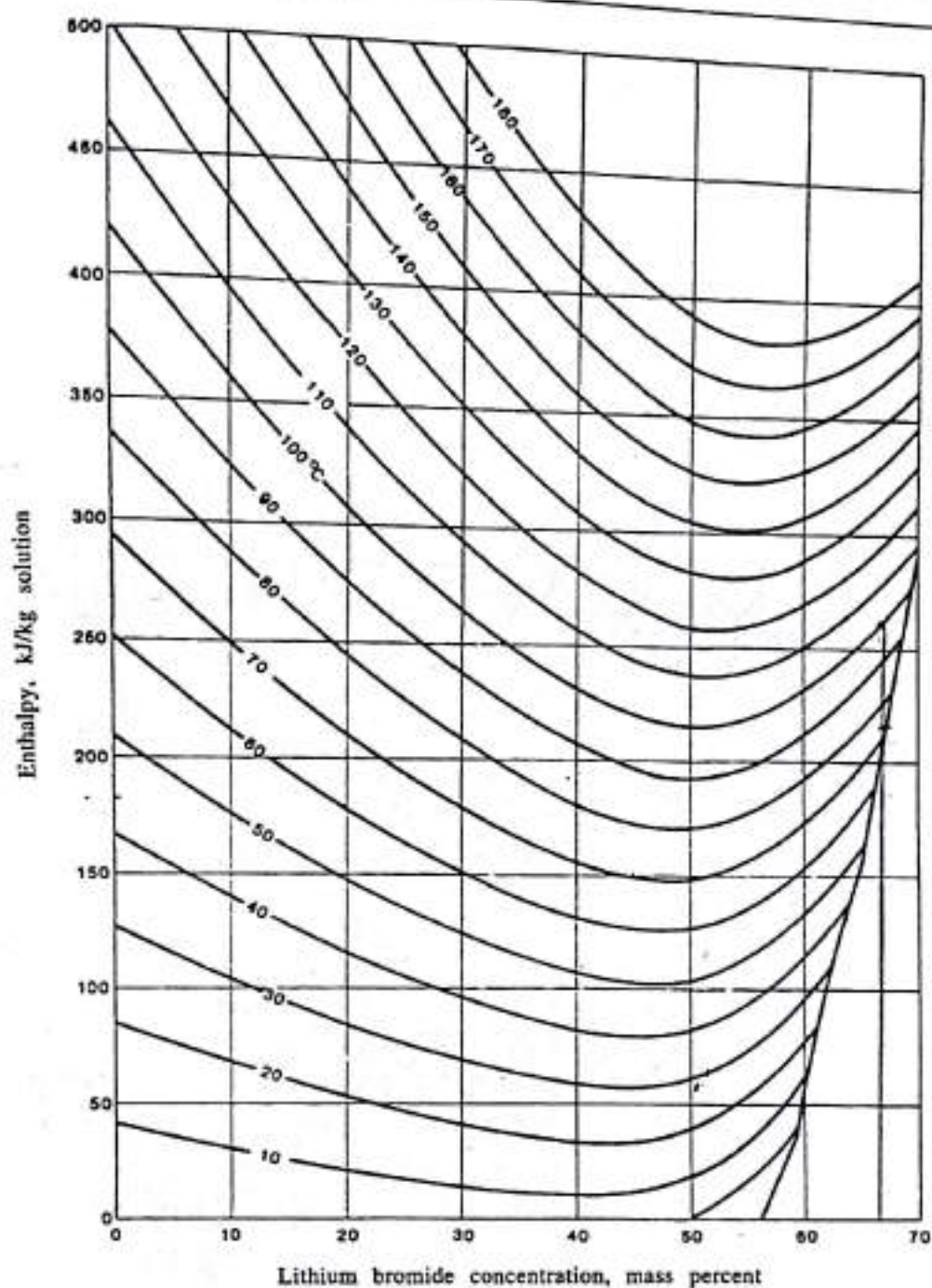


FIGURE B3 Equilibrium chart for aqueous lithium bromide solutions

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Equations

Concentration range
40 < X < 70% LiBrTemperature range
15 < T < 165°C
$$h = \sum_0^4 A_n X^n + T \sum_0^4 B_n X^n + T^2 \sum_0^4 C_n X^n$$
 in kJ/kg, where $T = ^\circ\text{C}$ and $X = \%\text{LiBr}$

$$A_0 = -2024.33$$

$$B_0 = 18.2829$$

$$C_0 = -3.7008214\text{E-}2$$

$$A_1 = 163.309$$

$$B_1 = -1.1691757$$

$$C_1 = 2.8877666\text{E-}3$$

$$A_2 = -4.88161$$

$$B_2 = 3.248041\text{E-}2$$

$$C_2 = -8.1313015\text{E-}5$$

$$A_3 = 6.302948\text{E-}2$$

$$B_3 = -4.034184\text{E-}4$$

$$C_3 = 9.9116628\text{E-}7$$

$$A_4 = -2.913705\text{E-}4$$

$$B_4 = 1.8520569\text{E-}6$$

$$C_4 = -4.4441207\text{E-}9$$

FIGURE B4 Enthalpy-concentration diagram for water/lithium bromide solutions.

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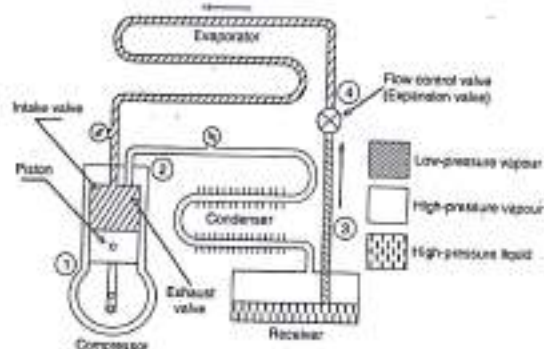


Figure 2.12 Vapour compression system with its components and the condition of refrigerant in the flow circuit.

EXERCISES

1. Define the terms refrigeration, refrigerant and refrigerating equipment.
2. Write a short note on the systems of refrigeration.
3. Briefly explain the applications of a refrigeration system.
4. Explain the working of ice refrigeration with the help of a simple schematic diagram. What are its drawbacks?
5. What do you understand by evaporative refrigeration? Explain with the help of examples.
6. What is dry ice? How can it be used for the refrigeration purpose? List its drawbacks.
7. What do you understand by refrigeration by expansion of air? In which cycle is this concept used?
8. Explain the working of a vapour compression refrigeration system with the help of a neat sketch. Name some of the common refrigerants used in this system.
9. Draw a neat diagram of a simple steam jet refrigeration system and explain its working.
10. Explain the concept of liquid gas refrigeration with the help of a diagram. Where is such refrigeration used in practice?
11. Describe the various methods of refrigeration.

Chapter 3

Air Refrigeration Systems

3.1 INTRODUCTION

Refrigeration is an application of thermodynamics. Refrigeration means transfer of heat from a lower temperature region to a higher temperature region with the help of an external aid. Devices that produce refrigeration effect are called *refrigerators* (or heat pumps) and the cycles on which they operate are called *refrigeration cycles*. The various refrigeration cycles are—reversed Carnot cycle, Bell-Coleman air refrigeration cycle, vapour compression refrigeration cycle, etc. Air refrigeration cycles are dealt with in this chapter.

3.2 DEFINITIONS

Refrigeration is a process of removing heat from a confined space so that its temperature is first lowered and then maintained at that low temperature compared to that of the surroundings, i.e. it is a phenomenon by virtue of which one reduces the temperature of a confined space compared with that of the surroundings.

The device used to produce cold or refrigeration effect is called a *refrigeration system* or *refrigerator*. The basic components of a refrigeration system are evaporator, compressor, condenser and expansion valve. In addition to these, there may be a refrigerant accumulator, temperature controller, etc. The refrigerant is a working substance, which circulates through the refrigeration system during its operation.

The R134a is the refrigerant used in domestic refrigerators and low-temperature refrigeration systems. The R22 is the refrigerant used in window air-conditioning units while ammonia is preferred in large air-conditioning systems. Air is the refrigerant in air-refrigeration cycles.

Unit of refrigeration or rating for refrigeration

The definition of refrigeration indicates that refrigeration is nothing but the rate of removal of heat. The SI unit of heat is joule, the time rate of which is watt. The unit of refrigeration effect is watt (W) or kilowatt (kW).

The standard unit of refrigeration is ton of refrigeration or simply ton denoted by the symbol TR. One TR is equivalent to the production of cold at the rate at which heat is to be removed from one US ton of water at 0°C in 24 hours. One US ton = 2000 lb, and latent heat of fusion of ice at 0°C = 144 Btu/lb.

$$\begin{aligned}\text{Thus, } 1 \text{ TR} &= \frac{1 \times 2000 \text{ lb} \times 144 \text{ Btu/lb}}{24 \text{ h}} \\ &= 12,000 \text{ Btu/h} = 200 \text{ Btu/min} \\ \text{But } 1 \text{ Btu} &= 1.055 \text{ kJ} \\ \therefore 1 \text{ TR} &= 211 \text{ kJ/min} \\ &= 3.516 \text{ kW}\end{aligned}$$

In the above definition of 1 TR, one ton equals 2000 lb instead of 2240 lb as per the conversion factor. In US, one ton is equal to 2000 lb.

3.3 REFRIGERATION LOAD

The refrigeration effect or cooling effect is produced in a refrigeration cycle by the refrigerating equipment. The average rate at which heat is removed from the cold space by the equipment is known as the cooling load. It is expressed in kW or TR as stated above.

The cooling load on refrigerating equipment results from several different sources. Some of the common sources of heat that contribute to the cooling load on the refrigerating equipment are as follows:

1. Ingress of heat into the refrigerated space from outside by conduction through the insulated walls.
2. Solar radiations that enter the refrigerated space through transparent glass or other transparent materials.
3. Heat on account of warm outside air entering the refrigerated space through open doors or through cracks around windows and doors.
4. Heat emitted by warm products whose temperature is to be lowered to the refrigerated space temperature.
5. Heat emitted by people occupying the refrigerated space. For example, people present in an air-conditioned space or the people working in the cold storages during loading and unloading the goods.
6. Heat emitted by any heat generating equipment installed in the refrigerated space such as lamps, motors, electronic devices, etc.

It is to be noted here that all these sources of heat are not present in every application. The significance of any one heat source with relation to the total cooling load will vary considerably with each application.

This topic is discussed in more detail in this chapter.

3.4 HEATING LOAD

In those regions where the atmospheric temperature falls considerably (below 10°C), especially during winters, heating is needed to keep the rooms warm. The rate of heat to be supplied to such a conditioned space is known as heating load. In Western countries, the houses are facilitated with solar heating systems.

3.5 CONCEPT OF HEAT ENGINE, REFRIGERATOR AND HEAT PUMP

It is a well-known fact that heat flows in the direction of decreasing temperature, i.e. from a high temperature body to a low temperature body. Such heat transfer occurs in nature without any external aid or device.

Heat engine

It is a prime mover that generates heat from the fossil fuel. It works according to the second law of thermodynamics stated by Kelvin and Planck—*It is impossible to construct a device that operates continuously in a cycle and produces no effect other than the withdrawal of heat energy from a single reservoir and converts all the heat into useful work.* This means that the heat engine (HE) rejects part of the heat available from the heat source while converting it to useful work. This is shown in Figure 3.1. It takes heat at the rate of Q_1 from the heat source and generates the work at the rate of W while rejecting heat at the rate of Q_2 to the heat sink. The performance of a heat engine is known as 'thermal efficiency' or 'Carnot efficiency' and is mathematically represented as

$$\eta_{\text{Carnot}} = \frac{Q_1 - Q_2}{Q_1} = \frac{T_1 - T_2}{T_1}$$

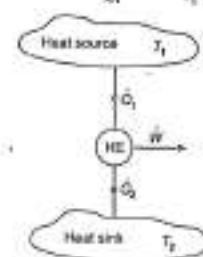


Figure 3.1 A heat engine.

Reverse heat transfer, i.e. from a body at low temperature to a body at high temperature, is possible only with a special device called refrigerator.

Refrigerator

It works according to the second law of thermodynamics stated by Clausius—*It is impossible to construct a device that operates in a cycle and produces no effect other than the transfer of heat from a body at low temperature to a body at high temperature without any external aid.*

Refrigerators are cyclic devices having working substances called refrigerants used in refrigeration cycles. The working principle of the refrigerator (R) is shown in Figure 3.2.

Here Q_2 is the amount of heat removed from the cold space at temperature T_2 . Q_1 is the amount of heat rejected to the environment at temperature T_1 and W is the net work input to the refrigerator.

We know that the term 'efficiency' is used to indicate the performance of a heat engine. The performance of a device means its ability to carry out the assigned task.

Efficiency is the ratio of the output (desired quantity) to the input (reference quantity). The output is always less than the input. The output and input quantities are taken in the same units.

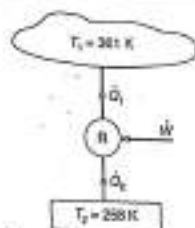


Figure 3.2 A refrigerator.

Coefficient of Performance (COP)

The objective of the refrigerator is to remove heat (Q_c) from a closed space. To accomplish this, it needs W_{in} as work input. Therefore, COP of a refrigerator is:

$$COP_R = \frac{\text{Desired effect}}{\text{Required input}} = \frac{Q_c}{W_{in}} \quad (3.1)$$

But

$$W_{in} = Q_h - Q_c$$

$$\therefore COP_R = \frac{Q_c}{Q_h - Q_c} = \frac{1}{(Q_h / Q_c) - 1} \quad (3.2)$$

The COP may be greater than one in many cases.

Heat pump

A heat pump (HP) transfers heat from a low temperature space to a higher temperature space. The objective of a heat pump is to supply heat Q_h to warm a space as shown in Figure 3.3.

The COP of a heat pump is expressed as the ratio of heat supplied (Q_h) to the work input (W_{in}). Mathematically,

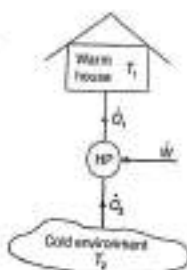


Figure 3.3 A heat pump.

$$COP_{HP} = \frac{\text{Desired effect}}{\text{Work input}} = \frac{\text{Heating effect}}{\text{Work input}} = \frac{Q_h}{W_{in}} \quad (3.3)$$

Comparing Eqs. (3.2) and (3.3),

$$COP_{HP} = COP_R + 1 \quad (3.4)$$

For fixed rates of Q_h and Q_c , Eq. (3.4) shows that the COP_{HP} is always greater than one since the COP_R is a positive value. In reality, however, part of Q_h is lost to the outside air through piping and other device, and COP_{HP} may drop below one when the outside air temperature is too low.

3.6 AIR REFRIGERATION SYSTEMS

Air is used as a refrigerant (working media) in air refrigeration systems. Air absorbs heat from the low temperature space and rejects heat to the high temperature surroundings while undergoing an air refrigeration cycle. Air does not change its phase while undergoing a cyclic process. Therefore, the heat carrying capacity per unit mass of air is very small in comparison to that of a refrigerant of a vapour compression refrigeration cycle. To obtain a required refrigeration effect, a large quantity of air needs to be handled, requiring a bigger-sized compressor, heat exchanger and expansion device. In an aircraft, rammed air is available and hence used for its air conditioning. Therefore, such systems are also discussed in this chapter.

3.6.1 Carnot Refrigerator

The Carnot refrigeration system works on reversed Carnot cycle. It is only a theoretical system in its conception but serves as an ideal cycle ever to be achieved in reality. The $p-v$ and $T-s$ diagrams of reversed Carnot cycle using air as a working medium are shown in Figure 3.4.

The following processes are imagined to take place in a reciprocating compressor:

Process (1-2) reversible adiabatic compression process: Air is compressed from initial pressure p_1 to pressure p_2 till the temperature rises from T_1 to T_2 in a reversible adiabatic compression process. The piston is assumed to move very fast.

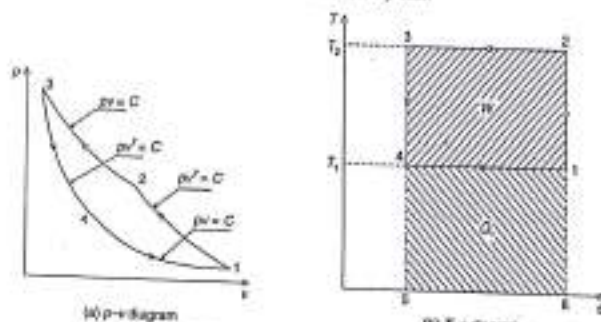


Figure 3.4 Reversed Carnot cycle.

Process (2-3) isothermal compression process: Air is compressed isothermally till its pressure rises from p_2 to p_3 . It is assumed that at point 2, a body at temperature T_2 is brought in contact with cylinder. Heat Q_2 is rejected at constant temperature T_2 during isothermal compression. The piston is assumed to move at dead slow speed.

Process (3-4) reversible adiabatic expansion process: Reversible adiabatic expansion of air in the clearance volume takes place till its pressure and volume change from p_3 and v_3 to p_4 and v_4 during which the temperature falls to T_4 . The piston is assumed to move at a great speed.

Process (4-1) isothermal expansion process: Air from p_4 and v_4 expands isothermally till the pressure and volume reach p_1 and v_1 . It is assumed that when a cold body at temperature T_1 is brought in contact with the cylinder, heat is absorbed by the air. To satisfy the isothermal heat absorption at constant temperature, the piston has to move at a dead slow speed. The heat Q_1 is absorbed during the process. The air attains the original state on completing the cycle.

All the processes involved in the cycle are reversible and therefore the cycle is called a **reversible cycle**.

The heat absorbed from cold body at temperature T_1 (refrigeration effect)

$$Q_1 = \text{area (1-4-5-6) on T-s diagram}$$

$$= T_1(s_1 - s_4), \text{ in kJ/kg}$$

Work done during the cycle

$$W = \text{area (1-2-3-4) on T-s diagram}$$

$$= (T_2 - T_1)(s_1 - s_4), \text{ in kJ/kg}$$

Coefficient of performance of Carnot refrigerator,

$$\text{COP}_R = \frac{\text{Refrigeration effect } Q_1}{\text{Net work done } W}$$

$$= \frac{T_1(s_1 - s_4)}{(T_2 - T_1)(s_1 - s_4)}$$

$$= \frac{T_1}{T_2 - T_1}$$

$$(3.5)$$

$$= \frac{1}{(T_2/T_1) - 1}$$

$$(3.6)$$

The COP of heat pump working on reversed Carnot cycle,

$$\text{COP}_H = \frac{\text{Useful effect}}{\text{Work input}}$$

$$= \frac{\text{Heat rejected } Q_2}{\text{Work input } W}$$

$$Q_2 = \text{Area (6-2-3-5)}$$

$$\text{COP}_H = \frac{T_1(s_1 - s_4)}{(T_2 - T_1)(s_1 - s_4)} \quad (3.7)$$

$$= \frac{T_1}{T_2 - T_1} \quad (3.8)$$

$$= \frac{1}{1 - (T_1/T_2)}$$

It follows that the Carnot COP depends on temperatures T_1 and T_2 only. In the sense, it does not depend upon the working substance (refrigerant) used.

The reversed Carnot cycle may be employed for cooling or heating purpose. For cooling, let T_1 be the refrigeration temperature and T_2 be the surrounding temperature. If we substitute $T_1 = 0$ (absolute zero), the minimum possible refrigeration temperature, in Eq. (3.6) then $\text{COP}_R = 0$. The upper limit to refrigeration temperature is $T_1 = T_2$, i.e. when the refrigeration temperature is equal to the temperature of the surroundings (ambient) at which $\text{COP}_R = \infty$. Thus **Carnot COP for cooling varies between 0 and ∞** .

From Eq. (3.8), it is clear that the Carnot COP for heating varies between 1 and ∞ . Also,

- (i) the refrigeration temperature T_1 should be as high as possible, and
- (ii) the surrounding temperature T_2 should be as low as possible.

These two points are applicable to all refrigerating machines, whether theoretical or practical. A sub-consideration reveals that the refrigeration and surrounding temperature cannot be varied at our will.

The refrigeration temperature is fixed by the particular application, i.e. this must be less than the temperature of the substance in the refrigerated space to be cooled. For example, if the temperature of the refrigerated space would be say -5°C then the actual refrigerant temperature would be less than -10°C . This difference in temperature will be the base to decide the evaporator size for the required rate of cooling.

The effect of raising the refrigeration temperature on the Carnot COP for the fixed surroundings temperature 47°C is shown in Figure 3.5.

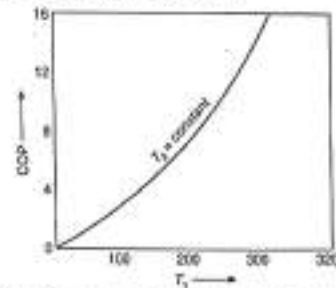


Figure 3.5 Carnot COP variation with respect to T_1 when T_2 is held constant (47°C)

Similarly, the temperature of the refrigerant must be higher than the surrounding temperature (cooling media in condenser) so that heat can be rejected. Thus, it can be said that if the temperature of the available cooling medium like water or air for heat rejection is lower, then Carnot COP for refrigeration will improve. For this reason, the COP of Carnot refrigerator will be higher in winter than in summer. This effect is shown in Figure 3.6.

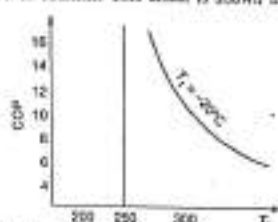


Figure 3.6 Carnot COP variation with respect to T_2 when T_1 is kept constant (-20°C).

EXAMPLE 3.1 A refrigeration system operates on reversed Carnot cycle between the temperature limits -23°C and 45°C . The refrigerating capacity is 10 TR. Determine (a) the COP, (b) the heat rejected from the system per hour, and (c) the power required.

Solution:

$$\begin{aligned} \text{(a)} \quad \text{COP}_R &= \frac{T_1}{T_2 - T_1} \\ &= \frac{273 - 23}{(273 + 45) - (273 - 23)} = \frac{250}{318 - 250} = 3.67 \quad \text{Ans.} \end{aligned}$$

$$\begin{aligned} \text{(b) Also,} \quad \text{COP}_R &= \frac{\text{Refrigerating effect}}{\text{Work input}} \\ \therefore \text{Work input} &= \frac{10 \times 211 \times 60}{3.67} = 34,332 \text{ kJ/h} \\ \text{Heat rejected} &= \text{refrigerating effect per hour} + \text{work input per hour} \\ &= (10 \times 211 \times 60) + 34,332 \\ &= 160,932 \text{ kJ/h} \quad \text{Ans.} \end{aligned}$$

$$\text{(c)} \quad \text{Power in kW} = \frac{34,332}{3600} = 9.536 \text{ kW} \quad \text{Ans.}$$

EXAMPLE 3.2 A Carnot refrigerator works on a reversed Carnot cycle. This unit requires 1.5 kW power for every one TR of refrigeration at -23°C . Determine (a) the COP, (b) the higher temperature of cycle and (c) the heat rejected in kJ/min. Also calculate the heat removal rate and COP when this device is used as a heat pump.

Solution: $T_1 = 273 - 23 = 250 \text{ K}$

(a) For refrigerator:

$$\text{COP}_R = \frac{\text{Refrigerating effect}}{\text{Work done}}$$

$$= \frac{1 \text{ TR}}{1.5 \text{ kW}} = \frac{3.516 \text{ kW}}{1.5 \text{ kW}} = 2.34 \quad \text{Ans.}$$

$$\text{(b) Also,} \quad \text{COP}_R = \frac{T_1}{T_2 - T_1} = \frac{250}{T_2 - 250} \quad \text{Ans.}$$

$$\therefore T_2 = 356.6 \text{ K} \quad \text{Ans.}$$

(c) For heat pump:

Heat rejected or supplied to a space at T_2

$$= \text{Heat absorbed} + \text{Work done}$$

$$= 3.516 + 1.5 = 5.01 \text{ kW} \quad \text{Ans.}$$

$$\therefore \text{COP}_{HP} = \frac{\text{Heat rejected}}{\text{Work done}} = \frac{5.01}{1.5} = 3.3 \quad \text{Ans.}$$

3.6.2 Limitations of Reversed Carnot Cycle

We have seen in Figure 3.4 that the Carnot cycle consists of four processes, namely reversible adiabatic compression (1–2), isothermal compression (2–3), reversible adiabatic expansion (3–4), and isothermal expansion process (4–1).

During the reversible adiabatic compression process (1–2), the piston is supposed to move very fast and for the isothermal compression process, the piston has to move at a very slow speed. It means that the piston has to move at very high speed in the first part of its compression stroke and for the remaining part of the stroke, it has to move at very slow speed, which are both practically impossible situations.

During the isothermal process, it is assumed that a hot or cold imaginary body is brought in contact with cylinder walls so that heat exchange between the system and the surrounding takes place. This is also not practical.

The internal and external friction is assumed negligible in cycle processes, which is not a correct assumption. The working substance is the ideal gas. In practice there is no working substance that behaves as an ideal gas. Therefore, the reversed Carnot cycle is an ideal cycle which gives the upper limit of the COP for a particular refrigeration system.

In a practical refrigeration cycle working on air as a refrigerant, the constant pressure heat absorption/rejection process replaces the isothermal heat absorption/rejection process of the reversed Carnot cycle.

3.6.3 Modified Reversed Carnot Cycle

The reversed Carnot cycle with perfect gas as working fluid is the most efficient cycle. The T - s diagram of such a cycle is a rectangular one as shown Figure 3.7.

If air was used as a refrigerant, this cycle would appear like the one shown in Figure 3.7. The isentropic compression and expansion are the processes 1–2 and 3–4 respectively. The processes 2–3 and 4–1 are constant pressure cooling and heating processes respectively. This cycle differs from the Carnot cycle (Figure 3.4) in that it operates between the same two temperatures but with additional two areas X and Y .

At point 4 the temperature of air must be lower than that of cold space so that air absorbs

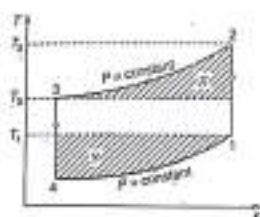


Figure 3.7 Reversed Carnot cycle with air as refrigerant

heat from the cold space and experiences increase in temperature up to T_2 . The addition of area Y decreases not only the COP but also the refrigerating effect. Similarly, at point 2, the temperature of air must be higher than that of the surroundings so that heat can be rejected. This adds area X to the cycle. Therefore, this addition of area X increases not only the power but also decreases the COP.

3.6.4 Reversed Carnot Cycle with Vapour as Refrigerant

Instead of air, a refrigerant can be used that condenses during the heat rejection process and evaporates during the heat absorption (in evaporator) process. With such a refrigerant, the reversed Carnot cycle becomes as if it has been fitted with the saturated liquid and saturated vapour lines as shown in Figure 3.8.

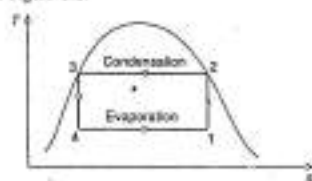


Figure 3.8 Reversed Carnot cycle when a condensable fluid is the refrigerant

Process 2-3: It is a constant temperature condensation process.

Process 4-1: It is also a constant temperature process but an evaporation process. The isothermal and isochoric lines are one and the same in the two-phase region.

At point 1, the refrigerant is a mixture of liquid and vapour. When this mixture is compressed (called wet compression) in a reciprocating compressor, then liquid refrigerant may be trapped in the cylinder head by the rising piston, which may damage the valves and the cylinder head.

Ideally, at point 2 the refrigerant should be in a saturated vapour state but practically the case is different as it contains liquid droplets. This so happens because in high-speed compression, the liquid droplets that are to be vaporized by the internal heat transfer do not get much time to change the phase. Therefore, these droplets remain up to state point 2. It means

that state 2 does not reflect the true condition of the refrigerant. Hence, it is practically impossible to terminate the compression process exactly at state point 2.

Another drawback of wet compression is that the droplets of liquid may wash the lubricating oil from the cylinder walls, which leads to many undesirable effects like increased friction, wear, and blow-by losses, etc.

On account of these disadvantages, the dry compression is preferred. For this, the state point 1 should be on the saturation curve. The refrigerant must be in saturated vapour state at the suction of the compressor. This is as indicated in Figure 3.9.

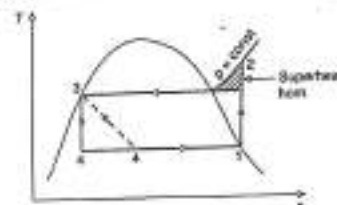


Figure 3.9 Modified reversed Carnot cycle using dry compression and throttling process.

Compression of dry vapour results in a high temperature at the end of compression, say, at point 2. This temperature is much higher than the condensing temperature. The refrigerant leaves the compressor in a superheated condition. The area of that part of the cycle, which is above the condensing temperature, is called the superheat horn, and represents the additional work required by dry compression.

In the reversed Carnot cycle, it is assumed that expansion occurs isentropically and the resulting work would be used to drive the compressor. But it is impractical because the work that is derived by the expansion device is very small. Another difficulty is that the expansion takes place in two phases and poses the lubrication problem. Therefore, the necessity is to reduce the pressure of the liquid from condenser to evaporator, which is certainly possible with a throttling device (a valve or a capillary tube). This throttling process 3-4 is isenthalpic and irreversible in nature. Therefore, this is one more modification introduced in the original reversed Carnot cycle as indicated by the dashed line 3-4 in Figure 3.9. Now the cycle 1-2-3-4 is the standard vapour compression refrigeration cycle which would be discussed in detail in the next chapter.

3.6.5 Bell-Coleman or Reversed Brayton or Joule Cycle

The Bell-Coleman refrigerator using air as a refrigerant is shown with the help of a block diagram in Figure 3.10. It consists of a compressor, a cooler, an expander and a refrigerator.

The Bell-Coleman air refrigeration cycle is the modification of the reversed Carnot cycle with air as a working medium. It can be operated as an open cycle shown in Figure 3.10(a), in which the cold air available at the outlet of the expander is used for refrigeration and is let out in the atmosphere. In the closed cycle [Figure 3.10(b)] the same air is circulated repeatedly. The cold air available at the outlet of the expander is employed to cool the other fluid in the heat

exchanger. The other fluid acts as the secondary refrigerant in this cycle.

The cycle is represented on $p-v$ and $T-s$ diagrams as shown in Figure 3.11 and Figure 3.12 respectively.

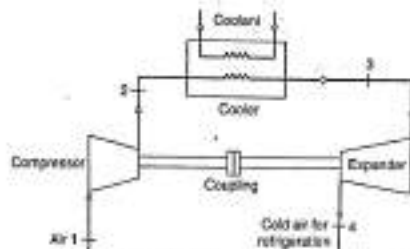


Figure 3.10(a) Bell-Coleman air refrigeration open cycle.

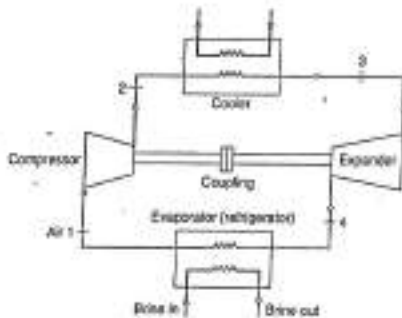


Figure 3.10(b) Bell-Coleman air refrigeration closed cycle (dense air cycle).

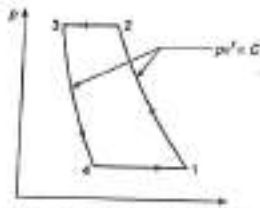


Figure 3.11 Bell-Coleman air cycle on $p-v$ diagram.

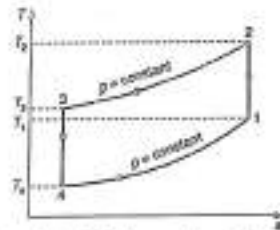


Figure 3.12 Bell-Coleman cycle on $T-s$ diagram.

The four processes of the closed cycle are as follows:

Isentropic compression process (1-2): The air from the refrigerator is drawn into the compressor and compressed isentropically to state 2. During compression, both pressure and temperature increase while the specific volume decreases from v_1 to v_2 . During the isentropic process, no heat is absorbed or rejected by the air.

Constant pressure cooling process (2-3): The high temperature air is cooled from temperature T_2 to T_3 in the cooler at constant pressure. Here the frictional pressure loss on account of the friction between the air and the heat exchanger is neglected. The specific volume decreases from v_2 to v_3 . The heat rejected by the air during this process is

$$Q_{2-3} = c_p(T_2 - T_3)$$

Isentropic expansion process (3-4): The air is expanded isentropically from pressure p_3 ($p_3 = p_2$) to p_4 ($p_4 = p_1 = \text{atm pressure}$), while the temperature of air also decreases from T_3 to T_4 . The specific volume of air increases from v_3 to v_4 . During this no heat exchange takes place.

Constant pressure heat absorption process (4-1): The cold air from the expander is circulated through the refrigerator where it absorbs heat from brine and thus cools the brine. During this, the temperature of the air increases from T_4 to T_1 and the specific volume increases from v_4 to v_1 . The heat absorbed from the refrigerator during constant pressure process per kg of air is

$$Q_{4-1} = c_p(T_1 - T_4) \quad (3.9)$$

Work done during the cycle per kg of air

$$\begin{aligned} &= \text{Heat rejected} - \text{heat absorbed} \\ &= c_p(T_2 - T_3) - c_p(T_1 - T_4) \end{aligned} \quad (3.10)$$

Coefficient of performance, COP = $\frac{\text{Heat absorbed}}{\text{Work input}}$

$$= \frac{c_p(T_1 - T_4)}{c_p(T_2 - T_3) - (T_1 - T_4)} = \frac{T_1 - T_4}{(T_2 - T_3) - (T_1 - T_4)}$$

$$= \frac{T_4 (T_1/T_4 - 1)}{T_3 (T_2/T_3 - 1) - T_4 (T_1/T_4 - 1)} \quad (3.11)$$

For the isentropic compression process 1-2,

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}}$$

Similarly, for the isentropic expansion process 3-4

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4} \right)^{\frac{\gamma-1}{\gamma}} \quad \leftarrow \quad \frac{T_4}{T_3} = \left(\frac{P_4}{P_3} \right)^{\frac{\gamma-1}{\gamma}}$$

Since $P_2 = P_3$ and $P_1 = P_4$, the equations would be:

$$\frac{T_2}{T_1} = \frac{T_3}{T_4}; \quad \frac{T_3}{T_2} = \frac{T_1}{T_4}$$

Putting these values in Eq. (3.11), we get

$$\begin{aligned} \text{COP} &= \frac{T_4}{T_3 - T_4} = \frac{1}{(T_3/T_4) - 1} \\ &= \frac{1}{\left(\frac{P_3}{P_4} \right)^{\frac{\gamma-1}{\gamma}} - 1} = \frac{1}{\left(\frac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}} - 1} \\ &= \frac{1}{r^{\frac{\gamma-1}{\gamma}} - 1} \end{aligned} \quad (3.12)$$

where r = compression or expansion ratio = $\frac{P_2}{P_1} = \frac{P_3}{P_4}$

Practically, the reversible adiabatic process is not possible. The reversible adiabatic or isentropic compression and expansion processes in the above analysis are replaced by the polytropic process $PV^n = \text{constant}$. The COP of the cycle can be obtained as follows:

$$\begin{aligned} \text{Compression work per kg air, } W_C &= \frac{n}{n-1} (P_2 V_2 - P_1 V_1) \text{ kJ/kg} \\ &= \frac{n}{n-1} R(T_2 - T_1) \end{aligned}$$

$$\text{Expansion work per kg air, } W_E = \frac{n}{n-1} R(T_3 - T_4)$$

$$\begin{aligned} \text{The net work done, } W &= W_C - W_E \\ &= \frac{n}{n-1} R[(T_2 - T_1) - (T_3 - T_4)] \end{aligned}$$

$$\begin{aligned} PV^n &= C \\ V &= \frac{C}{P^{1/n}} \\ W_C &= \int_{V_1}^{V_2} P dV \\ &= \int_{V_1}^{V_2} \frac{C}{V^{n+1}} dV \\ &= \frac{n}{n-1} C \left[\frac{V^{n-1}}{n-1} \right]_{V_1}^{V_2} \\ &= \frac{n}{n-1} [P_2 V_2 - P_1 V_1] \end{aligned}$$

$$= \frac{n}{n-1} \left(\frac{\gamma-1}{\gamma} \right) c_p [(T_2 - T_1) - (T_3 - T_4)]$$

since the characteristic gas constant, $R = \left(\frac{\gamma-1}{\gamma} \right) c_p$

From our previous analysis,

$$\begin{aligned} \frac{T_4}{T_1} &= \frac{T_3}{T_2} \\ \text{COP} &= \frac{\text{Refrigerating effect}}{\text{Work done}} \\ &= \frac{c_p (T_1 - T_4)}{\frac{n}{n-1} \left(\frac{\gamma-1}{\gamma} \right) c_p [(T_2 - T_1) - (T_3 - T_4)]} \\ &= \frac{c_p (T_1 - T_4)}{\frac{n}{n-1} \left(\frac{\gamma-1}{\gamma} \right) c_p [(T_2 - T_1) - (T_1 - T_4)]} \\ &= \frac{n-1}{n} \times \frac{\gamma}{\gamma-1} \times \frac{T_4}{T_1 - T_4} \end{aligned} \quad (3.13)$$

3.5.5 Actual Bell-Coleman Cycle

The actual Bell-Coleman air refrigeration cycle shown in Figure 3.13 deviates in four ways from the cycle shown in Figure 3.12.

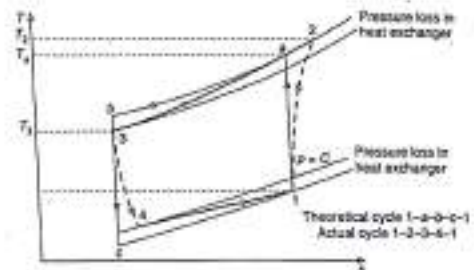


Figure 3.13 Actual Bell-Coleman cycle.

Process (1-2): The compression process in the compressor is non-isentropic due to internal and external friction between the air and the surface temperature T_1 at the end of compression is more than T_1 .

Process (2-3): The cooling of air from temperature T_2 to T_3 takes place in the heat exchanger. Heat exchanger is a coil of tubes in which the air flows. Therefore, due to friction between the air and the metal tube surface a pressure loss occurs. Therefore, the cooling process is not truly at constant pressure.

Process (3-4): The expansion of air from pressure p_3 to p_4 instead of from b to c is on account of friction between the air and the expander surface.

Process (4-1): This process takes place in the heat exchanger, hence a pressure drop.

Practically, the actual cycle indicates that the actual compression work is higher compared with that of isentropic compression. The work obtained in the expander is less than that available from isentropic expansion.

3.6.7 Application of Aircraft Refrigeration

The air refrigeration cycle is exclusively used for air conditioning of all types of aircraft except cargo aircraft. Air is the working substance in such air-conditioning systems. The coefficient of performance of the air refrigeration system for aircraft is very poor compared to the vapour compression refrigeration cycle employed for the same purpose due to its many advantages.

One may ask that since at high altitudes both atmospheric temperature and pressure decrease, then why are aircraft air conditioned? The reasons are attributable to external and internal heat leakages. Also, temperature of the order of -5°C is needed to preserve food and cold drinks.

Following are the external heat sources which add the heat in the compartments of occupants.

External heat sources

1. Solar radiations enter the compartments through windows.
2. Due to solar radiations, the outer surface of the aircraft gets heated. Therefore, heat is conducted into the compartments.
3. The speed of aircraft is very high—of the order of 1500 km/h. Therefore, there is a skin friction between the outer aircraft surface and the air due to which the whole aircraft body gets heated.

Internal heat sources

1. A normal healthy person dissipates heat at the rate of 120 W (100 J/s). For the capacity of 100 persons in an aircraft, the total heat dissipated would be 12 kW. To compensate for such a huge heat load, one requires air conditioning.
2. Electrical and electronic components generate heat throughout their use.
3. The engine parts of aircraft generate heat which is conducted into the compartments.
4. Food products, cold drinks, etc. need to be stored at a low temperature.

3.7 METHODS OF AIR REFRIGERATION SYSTEMS

The various methods of air refrigeration systems used for aircraft are as follows:

- Simple air-cooling system
- Simple air-evaporative cooling system
- Boot-strap air cooling system
- Boot-strap air evaporative cooling system
- Reduced ambient air cooling system
- Regenerative air cooling system

3.7.1 Simple Air-cooling System

Simple air-cooling system of an aircraft is shown with the help of a block diagram in Figure 3.14.

and on T-s diagram in Figure 3.15. The components of the system are a diffuser, a compressor coupled to a turbine, a heat exchanger, a cooling turbine, and an air-cooling fan.

A part of the ram air bled off at state point 3 is cooled in a heat exchanger with the help of returned air to state point 4. It is further cooled due to expansion to cabin pressure in the cooling turbine and is then supplied to the cabin. The work of expansion of cooling turbine is used to drive the air cooling fan which draws the ram air through the heat exchanger.

The remaining air at a high pressure (p_2) and a high temperature (T_2) and the compressed air is supplied to gas turbine through the combustion chamber. Turbine power is used to drive the main compressor and other equipment of the aircraft.

Simple air-cooling system is useful for ground surface air cooling and for aircraft at low speeds.

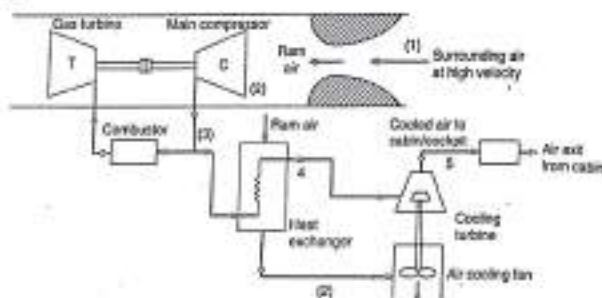


Figure 3.14 Simple air-cooling system.

Ramming process: The ambient air at p_1 and T_1 enters through a diffuser and due to ramming effect its pressure and temperature are raised to p_2 and temperature T_2 . This ideal ramming action is shown by the vertical line 1-2 in Figure 3.15. In practice, because of internal friction and inherent irreversibilities, the temperature of the rammed air is more than T_1 . Thus the actual ramming process is shown by the curve 1-2'. The pressure and temperature of the rammed air are now p_2 and T_2 respectively.

The diffuser efficiency η_d for ram compression is defined as:

$$\eta_d = \frac{\text{Actual pressure rise } (p_2 - p_1)}{\text{Ideal pressure rise } (p_2 - p_1)}$$

Let us apply the steady flow energy equation to the ramming process (diffuser). Let v_1 and v_2 be the relative velocities of air in m/s before and after diffuser respectively.

$$h_1 + \frac{v_1^2}{2} = h_2 + \frac{v_2^2}{2}$$

But

$$v_2 = 0 \text{ after ram compression.}$$

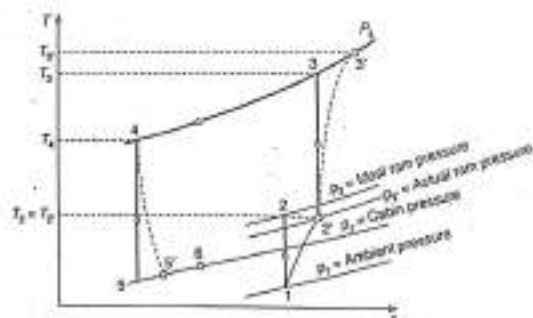


Figure 3.15 T-s diagram for simple air cooling system.

$$h_2 - h_1 = \frac{v^2}{2}$$

$$c_p(T_2 - T_1) = \frac{v^2}{2} \text{ for unit mass of air.}$$

$$T_2 = T_1 + \frac{v^2}{2c_p}$$

$$\frac{T_2}{T_1} = 1 + \frac{v^2}{2c_p T_1} \quad (3.14)$$

We know that $c_p - c_v = R$

$$c_p \left[1 - \frac{c_v}{c_p} \right] = R \text{ or } c_p \left[1 - \frac{1}{\gamma} \right] = R \quad (\because c_p/c_v = \gamma)$$

$$c_p = \frac{\gamma R}{\gamma - 1}$$

$$T_2 = T_1 + \frac{1}{2} \left(\frac{\gamma - 1}{\gamma} \right) \frac{v_1^2}{R}$$

$$\frac{T_2}{T_1} = 1 + \frac{(\gamma - 1)}{2} \frac{v_1^2}{\gamma R T_1} \quad (3.15)$$

$$T_2 = T_1$$

But sonic velocity of ambient air, $c_1 = \sqrt{\gamma R T_1}$

$$\frac{T_2}{T_1} = 1 + \left(\frac{\gamma - 1}{2} \right) \frac{v_1^2}{c_1^2} = 1 + \left(\frac{\gamma - 1}{2} \right) M^2 \quad (3.16)$$

where Mach number,

$$M = \frac{\text{Actual velocity, } v_1}{\text{Sound velocity, } c_1}$$

and

$$\frac{p_2}{p_1} = \left(\frac{T_2}{T_1} \right)^{\frac{\gamma}{\gamma - 1}} \quad (3.17)$$

where M is the mach number of the flight. It is defined as the ratio of aircraft velocity (v) to the local sound velocity (c).

The temperature $T_1 = T_2$ is called the *stagnation temperature* of the ambient air entering the main compressor. The velocity of air at the outlet of the diffuser is assumed to be zero which is why the 'stagnation' is used. The properties corresponding to the point are stagnation properties. The stagnation pressure after isentropic compression (p_2) is determined with the help of diffuser efficiency.

Compression process: The isentropic compression of air in the main compressor is represented by the line 2'-3. In practice, because of friction, the compression process does not remain isentropic. So the actual compression is represented by the curve 2'-3' on T-s diagram. The work done for compression process is given by

$$W_c = m_c c_p (T_3' - T_2') \quad (3.18)$$

where m_c = mass of air bled from the main compressor for refrigeration purposes since we are analyzing the refrigeration system.

Cooling process: The ram air is used to cool the compressed air in the heat exchanger. This process is a constant pressure cooling shown by the curve 3-4 in Figure 3.15. In practice, there is a pressure drop in the heat exchanger, which is not shown in the figure. The temperature of air decreases from T_3 to T_4 . The heat rejected in the heat exchanger is given by

$$Q_R = m_c c_p (T_3 - T_4) \quad (3.19)$$

Expansion process: The cooled air is now expanded isentropically in the cooling turbine as shown by the curve 4-5. The actual expansion in the cooling turbine is shown by the curve 4-5'. The work obtained in cooling turbine due to expansion process is given by

$$W_R = m_c c_p (T_4 - T_5') \quad (3.20)$$

The work of this turbine is used to drive the cooling air fan, which draws cooling air from the heat exchanger.

Refrigeration process: Cool air from the outlet of cooling turbine (i.e. after expansion) is supplied to the cabin and cockpit where it gets heated by the heat of equipment and occupancy.

$$\frac{Q_{\text{cool}}}{\text{kJ}} = \frac{Q_{\text{cool}}}{\text{kJ}}$$

This process is shown by the curve 5'-6 in Figure 3.15. The refrigerating effect produced or heat absorbed is given by

$$= m_a c_p (T_6 - T_5) = m_a (h_6 - h_5) \quad (3.21)$$

where T_5 = inside temperature of the cabin.

COP of the air cycle,

$$\begin{aligned} &= \frac{\text{Refrigerating effect produced}}{\text{Work done}} \\ &= \frac{m_a c_p (T_6 - T_5)}{m_a c_p (T_2 - T_1)} = \frac{T_6 - T_5}{T_2 - T_1} \end{aligned} \quad (3.22)$$

If Q tons of refrigeration is the cooling load in the cabin, then the air required for the refrigeration purpose,

$$m_a = \frac{211 Q}{c_p (T_6 - T_5)} \text{ kg/min} \quad (3.23)$$

Power required for the refrigeration system,

$$P = \frac{m_a c_p (T_2 - T_1)}{60} \text{ kW} \quad (3.24)$$

And COP of the refrigerating system

$$= \frac{211 Q}{m_a c_p (T_2 - T_1)} = \frac{211 Q}{P \times 60} \quad (3.25)$$

EXAMPLE 3.3 An aircraft air conditioning needs refrigeration capacity of 10 TR. At the altitude of aircraft the atmospheric pressure and temperature are 0.9 bar and 10°C respectively. The pressure of air after ramming effect in a diffuser increases to 1.013 bar. The temperature of the air is reduced by 50°C in the heat exchanger. The pressure in the cabin is 1.01 bar and the temperature of air leaving the cabin is 25°C. The pressure of the compressed air is 3.5 bar. Assume that all the expansions and compressions are isentropic. Also carry out the calculation if the compression and expansion efficiencies are 90%.

Determine the following:

(a) Power required to take the load of cooling in the cabin.

(b) COP of the system.

If the efficiencies of the expansion and compression processes are 90%, analyse the problem for power requirement in both the cases.

Solution: Refer to Figure 3.16.

Given: $Q = 10$ TR; $p_1 = 0.9$ bar; $T_1 = 10^\circ\text{C} = 10 + 273 = 283$ K;

$p_2 = 1.013$ bar; $p_3 = p_4 = 1.01$ bar; $T_6 = 25^\circ\text{C} = 25 + 273 = 298$ K; $p_1 = 3.5$ bar.

Case 1: With 100% efficiency of compression and expansion processes:

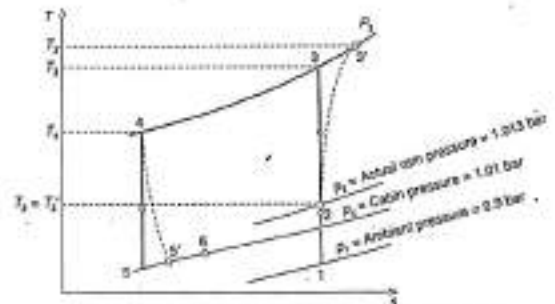


Figure 3.16 T-s diagram for simple air-cooling system—Example 3.3.

Temperature after ramming effect

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1.013}{0.9} \right)^{\frac{1.4-1}{1.4}} = (1.125)^{0.286} = 1.034$$

$$\therefore T_2 = T_1 \times 1.034 = 283 \times 1.034 = 292.6 \text{ K}$$

$$\begin{aligned} \text{Similarly, } \frac{T_1}{T_2} &= \left(\frac{p_1}{p_2} \right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{0.9}{1.013} \right)^{\frac{1.4-1}{1.4}} \\ &= (0.888)^{0.286} = 0.974 \end{aligned}$$

$$\therefore T_1 = T_2 \times 0.974 = 292.6 \times 0.974 = 285 \text{ K} = 12^\circ\text{C}$$

The temperature of air is reduced by 50°C in the heat exchanger, therefore, the temperature of air leaving the heat exchanger,

$$T_4 = 12 - 50 = -38^\circ\text{C} = 235 \text{ K}$$

$$\text{We know that } \frac{T_3}{T_4} = \left(\frac{p_3}{p_4} \right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1.01}{-38} \right)^{\frac{1.4-1}{1.4}} = (0.284)^{0.286} = 0.7$$

$$\therefore T_3 = T_4 \times 0.7 = 235 \times 0.7 = 164.5 \text{ K}$$

We know that the mass of air required for the refrigeration purpose is

$$m_a = \frac{211 Q}{c_p (T_6 - T_5)} = \frac{211 \times 10}{1(298 - 164.5)} = 130.7 \text{ kg/min}$$

(a) Power required to take the load of cooling in the cabin,

$$P = \frac{m_a c_p (T_1 - T_2)}{60} = \frac{51.2 \times 1.004 (417 - 292.6)}{60} = 106 \text{ kW} \quad \text{Ans.}$$

(b) COP of the system:

We know that COP of the system is

$$= \frac{211 Q}{P \times 60} = \frac{211 \times 10}{106 \times 60} = 0.33 \quad \text{Ans.}$$

Case 2: When the efficiencies of compression and expansion are 90%:
Compression (2-3):

$$\eta_c = \frac{T_3 - T_2}{T_3' - T_2} = 0.9 = \frac{417 - 292.6}{T_3' - 292.6}$$

$$T_3' = 430.8 \text{ K} = 157.8^\circ\text{C}$$

The temperature of air is reduced by 50°C in the heat exchanger, therefore, the temperature of air leaving the heat exchanger is

$$T_4 = 157.8^\circ\text{C} - 50^\circ\text{C} = 107.8^\circ\text{C} = 380.8 \text{ K}$$

We know that

$$\frac{T_3}{T_4} = \left(\frac{p_3}{p_4} \right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1.01}{3.5} \right)^{\frac{1.4-1}{1.4}} = (0.288)^{0.286} = 0.7$$

$$T_3 = T_4 \times 0.7 = 380.8 \times 0.7 = 266.5 \text{ K}$$

Turbine efficiency:

$$\eta_t = \frac{T_4 - T_5}{T_4 - T_5'} = 0.9 = \frac{380.8 - T_5}{380.8 - 266.5}$$

$$T_5' = 277.93 \text{ K}$$

Mass of air required for the refrigeration purposes,

$$m_a = \frac{211 Q}{c_p (T_4 - T_5')} = \frac{211 \times 10}{1.004 (380.8 - 277.93)} = 104.6 \text{ kg/min}$$

(a) Power required to take the load of cooling in the cabin,

$$P = \frac{m_a c_p (T_1 - T_2)}{60} = \frac{104.6 \times 1.004 (430.8 - 292.6)}{60} = 240.9 \text{ kW} \quad \text{Ans.}$$

(b) COP of the system:

We know that COP of the system is

$$= \frac{211 Q}{P \times 60} = \frac{211 \times 10}{240.9 \times 60} = 0.145 \quad \text{Ans.}$$

From the above calculation it is evident that a 10% reduction in the efficiency of both compression and expansion processes would increase the power requirement from 106 kW to 240.9 kW. In percentage, it is an increase of 127%. It is clear that utmost care should be taken to maintain the condition of both the compressor and turbine to the maximum possible.

EXAMPLE 3.4 An aircraft refrigeration plant has a capacity of 30 TR. The ambient temperature is 17°C . The atmospheric air is compressed to 0.95 bar and 30°C due to ram action. The air is then further compressed in a compressor to 4.75 bar and is then cooled in a heat exchanger to 67°C . It then expands in a turbine to 1 bar before it is supplied to the cabin. Air leaves the cabin at 27°C . The isentropic efficiencies of the compressor and the turbine are 0.9.

Determine the following:

(a) Mass flow rate of air circulated/second

(b) COP.

(c) Specific power required. Take $c_p = 1.004 \text{ kJ/kg}\cdot\text{K}$ and $\gamma = 1.4$ for air. Sketch the cycle on $T-s$ diagram.

Solution: Refer to Figures 3.17 and 3.18.

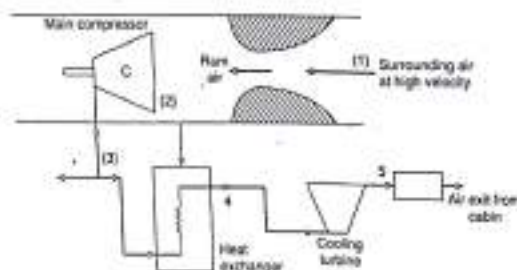


Figure 3.17 Simple air-cooling system—Example 3.4.

Given: Capacity = 30 TR = $30 \times 3.517 = 105.51 \text{ kW}$

$T_1 = 17^\circ\text{C} = 290 \text{ K}$

$p_2 = 0.95 \text{ bar}$

$T_2 = 30^\circ\text{C} = 303 \text{ K}$

$p_3 = 4.75 \text{ bar}$

$T_4 = 67^\circ\text{C} = 340 \text{ K}$

$p_5 = 1 \text{ bar}$

Cabin exit temperature = $27^\circ\text{C} = 300 \text{ K}$; Also, $\eta_c = \eta_T = 0.9$

In compressor,

$$T_3 = T_2 \left(\frac{p_3}{p_2} \right)^{\frac{\gamma-1}{\gamma}} = 303 \left(\frac{4.75}{0.95} \right)^{\frac{1.4-1}{1.4}}$$

$$T_3 = 479.89 \text{ K}$$

$$\eta_c = \frac{T_3 - T_2}{T_3' - T_2} = 0.9 = \frac{479.89 - 303}{T_3' - 303}$$

$$T_3' = 499.55 \text{ K}$$

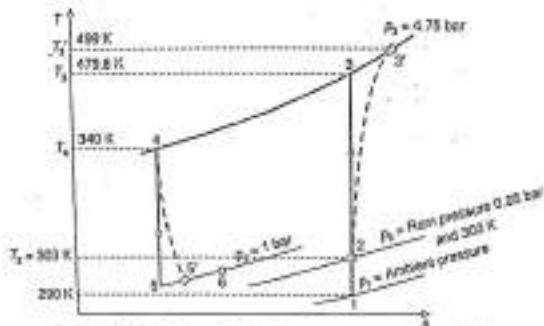


Figure 3.13 T-s diagram for simple air-cooling system—Example 3.4

For turbine,

$$T_3 = T_4 \left(\frac{p_3}{p_4} \right)^{\frac{\gamma-1}{\gamma}} = 340 \left(\frac{1}{4.75} \right)^{\frac{1.4-1}{1.4}}$$

$$T_3 = 217.86 \text{ K}$$

$$\eta_T = \frac{T_4 - T_3}{T_4 - T_2} = 0.9 = \frac{340 - T_3}{340 - 217.86}$$

$$T_2 = 230.0 \text{ K}$$

$$\begin{aligned} \text{Refrigerating effect/kg} &= c_p (\text{cabin temperature} - T_3) \\ &= 1.005(300 - 230.0) = 70.29 \text{ kJ/kg} \\ \text{Capacity} &= 105.51 \text{ kW} \end{aligned}$$

$$\text{Mass flow rate in kg/s} = \frac{\text{Capacity}}{\text{Refrigerating effect}} = \frac{105.51}{70.29}$$

$$m = 1.5 \text{ kg/s}$$

$$\text{Work done } W = W_C - W_T$$

$$\begin{aligned} W &= c_p (T_2 - T_3) - c_p (T_4 - T_3) \\ &= 1.005(230.0 - 217.86) - 1.005(340 - 217.86) \\ &= 87.04 \text{ kJ/kg} \\ W &= 87.04 \times 1.5 = 130.56 \text{ kW} \end{aligned}$$

$$\text{COP} = \frac{\text{Capacity}}{W} = \frac{105.51}{130.56} = 0.81$$

$$\text{Specific power required (kW/TR)} = \frac{\text{Work in kW}}{\text{Capacity in TR}} = 4.352 \text{ kW/TR}$$

EXAMPLE 3.5 An aircraft is moving at a speed of 1000 km/h at an altitude of 8000 m, where the ambient pressure and temperature are 0.35 bar and -15°C respectively. The cabin of the plane is maintained at 25°C by using a simple air refrigeration system. The pressure ratio of compressor is 3. The air is passed through heat exchanger after compression and cooled to its original condition entering into the plane. A pressure loss of 0.1 bar takes place in the heat exchanger. The pressure of the air leaving the cooling turbine is 1.06 bar and the air pressure in the cabin is 1.013 bar. Considering the total cooling load of plane to be 70 kW, determine the following:

- Stagnation temperature and pressure.
- Mass flow rate of air circulated through the cabin.
- Volume handled by the compressor and expander.
- Net power delivered to the refrigeration system and COP of the system.

Solution: Refer to Figures 3.19(a) and (b).

Given:

$$\text{Plane speed, } C = 1000 \text{ km/h} = 277.78 \text{ m/s;}$$

$$p_1 = 0.35 \text{ bar; } T_1 = -15^\circ\text{C} = 258 \text{ K;}$$

$$\frac{p_2}{p_1} = 3 \quad \text{or} \quad p_2 = 3p_1;$$

$$p_4 = (p_3 - 0.1) \text{ bar; cabin temperature} = 25^\circ\text{C} = 298 \text{ K}$$

$$\text{pressure at exit of turbine} = 1.06 \text{ bar; pressure in the cabin} = 1.013 \text{ bar;}$$

$$\text{cooling load} = 70 \text{ kW}$$

$$(i) \text{ Stagnation temperature, } T_2 = T_1 + \frac{C^2}{2000 c_p}$$

$$= 258 + \frac{(277.78)^2}{2000 \times 1.005}$$

$$T_2 = 296.6 \text{ K}$$

Stagnation pressure,

$$\frac{p_2}{p_1} = \left(\frac{T_2}{T_1} \right)^{\frac{\gamma}{\gamma-1}}$$

$$p_2 = 0.35 \left(\frac{296.6}{258} \right)^{\frac{1.4}{0.4}} = 0.57 \text{ bar}$$

Stagnation pressure,

$$p_2 = 0.57 \text{ bar}$$

$$p_3 = 3p_2 = 1.71 \text{ bar}$$

$$T_3 = T_2 \left(\frac{p_3}{p_2} \right)^{\frac{\gamma-1}{\gamma}} = 405.97 \text{ K}$$

$$\text{Pressure at turbine inlet, } p_4 = p_3 - 0.1 = 1.61 \text{ bar}$$

Ans.

Ans.

Ans.

Ans.

Ans.

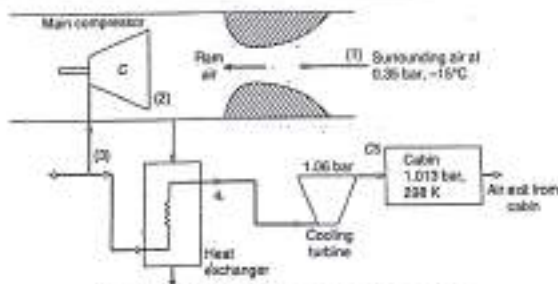


Figure 1.19(a) Simple air-cooling system—Example 3.5.

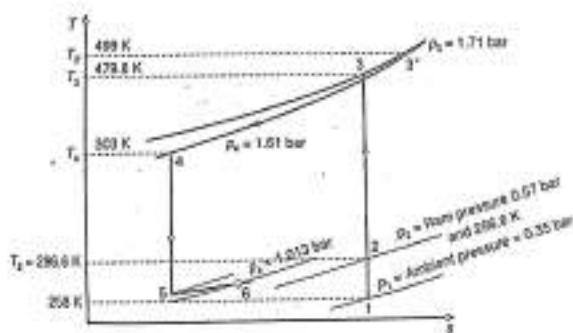


Figure 1.19(b) T-s diagram for simple air-cooling system—Example 3.5.

Turbine inlet temperature = original temperature of the air entering the plane
i.e. $T_4 = 258 \text{ K}$

$$\text{Turbine exit temp., } T_3 = T_4 \left(\frac{p_3}{p_4} \right)^{\frac{\gamma-1}{\gamma}}$$

$$T_3 = 258 \left(\frac{1.06}{1.61} \right)^{\frac{1.4-1}{1.4}} = 264.25 \text{ K}$$

$$\text{Refrigerating effect in cabin} = c_p(T_6 - T_5) = 1.005(298 - 264.25) = 33.92 \text{ kJ/kg}$$

$$\begin{aligned} \text{(b) Mass flow rate} &= \frac{\text{Cooling capacity in kJ/s}}{\text{Refrigerating effect}} \\ &= \frac{70}{33.92} = 2.06 \text{ kg/s} \quad \text{Ans.} \end{aligned}$$

$$\text{(c) Volume through compressor} = \frac{mRT_1}{p_1} = \frac{2.06 \times 0.287 \times 258}{0.35 \times 10^5} = 1.66 \text{ m}^3/\text{s} \quad \text{Ans.}$$

$$\text{Volume through expander} = \frac{mRT_3}{p_3} = \frac{2.06 \times 0.287 \times 229.86}{1.06 \times 10^5} = 1.282 \text{ m}^3/\text{s} \quad \text{Ans.}$$

$$\begin{aligned} \text{(d) Net work} &= W_c - W_T = m c_p(T_2 - T_3) - m c_p(T_4 - T_5) \\ &= m c_p(T_2 - T_3 - T_4 + T_5) \\ \text{or } W &= 2.06 \times 1.005(499.97 - 296.6 - 258 + 229.86) \\ &= 167.33 \text{ kW} \quad \text{Ans.} \end{aligned}$$

$$\text{and } \text{COP} = \frac{Q}{W} = \frac{70}{167.33} = 0.42 \quad \text{Ans.}$$

EXAMPLE 3.6 An air-cooling system for a jet plane cockpit operates on the simple cycle. The cockpit is to be maintained at 25°C . The ambient air pressure and temperature are 0.35 bar and -15°C respectively. The pressure ratio of the jet compressor is 3. The speed of the plane is 1000 km/h. The pressure drop through the cooler coil is 0.1 bar. The pressure of the air leaving the cooling turbine is 1.06 bar and that in the cockpit is 1.01325 bar. The cockpit cooling load is 58.05 kW.

Calculate the following:

- Stagnation temperature and pressure of the air entering the compressor.
- Mass flow rate of the air circulated.
- Volume handled by the compressor and expander.
- Net power delivered by the engine to the refrigeration unit.
- COP of the system.

Solution:

$$\begin{aligned} \text{Given: } \text{Cockpit temperature} &= 25^\circ\text{C} = 298 \text{ K} \\ p_1 &= 0.35 \text{ bar; } T_1 = -15^\circ\text{C} = 258 \text{ K} \\ \frac{p_2}{p_1} &= 3 \quad \text{or } p_2 = 3p_1, \\ p_3 &= 1.06 \text{ bar} \end{aligned}$$

Plane speed, $C = 1000 \text{ km/h} = 277.78 \text{ m/s}$
Pressure drop through the cooler coil = 0.1 bar, Pressure of air leaving the turbine = 1.06 bar.
Pressure of air in the cockpit = 1.01325 bar, Cooling load = 58.05 kW

$$\text{(a) Stagnation temperature, } T_2 = T_1 + \frac{C^2}{2c_p}$$

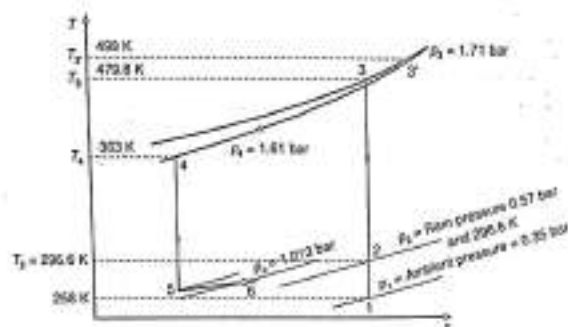


Figure 3.28 T-s diagram for simple air-cooling system—Example 3.6.

$$T_2 = 258 + \frac{(277.78)^2}{2 \times 1000} = 296.58 \text{ K} \quad \text{Ans.}$$

$$\text{Stagnation pressure, } p_2 = p_1 \left(\frac{T_2}{T_1} \right)^{\frac{\gamma}{\gamma-1}} = 0.35 \left(\frac{296.58}{258} \right)^{1.4(1.4-1)} = 0.57 \text{ bar} \quad \text{Ans.}$$

(b) Compressor outlet pressure, $p_2 = 3p_1 = 1.71 \text{ bar}$
= cooler inlet pressure

Cooler exit pressure = cooler inlet - pressure drop
= $1.71 - 0.1 = 1.61 \text{ bar}$
= turbine inlet pressure

Assuming cooling turbine inlet temperature is $30^\circ\text{C} = 303 \text{ K}$

$$\frac{\text{Turbine exit temp.}}{\text{Turbine inlet temp.}} = \left(\frac{\text{Turbine exit pressure}}{\text{Turbine inlet pressure}} \right)^{\frac{\gamma-1}{\gamma}}$$

$$\text{Turbine exit temperature} = 303 \left(\frac{1.06}{1.61} \right)^{\frac{1.4-1}{1.4}} = 294.1 \text{ K}$$

$$\text{Refrigerating effect/kg} = c_p (\text{cockpit temperature} - \text{turbine exit temperature}) = 1.005(298 - 294.1) = 3.92 \text{ kJ/kg}$$

$$\text{Mass flow rate} = \frac{\text{Cooling required in kJ/s}}{\text{Refrigerating effect kJ/kg}} = \frac{58.05}{2.5} = 23.2 \text{ kg/s} \quad \text{Ans.}$$

$$(c) \quad \text{Compressor volume} = \frac{mRT_2}{p_2} = \frac{23.22 \times 0.287 \times 296.58}{0.57 \times 10^5} = 34.67 \text{ m}^3/\text{s} \quad \text{Ans.}$$

$$\text{Expander volume} = \frac{mRT_3}{p_3} = \frac{23.22 \times 0.287 \times 295.51}{1.06 \times 10^5} = 18.13 \text{ m}^3/\text{s} \quad \text{Ans.}$$

(d) Assuming isentropic compression in compressor,

$$T_3 = T_2 \left(\frac{p_3}{p_2} \right)^{\frac{\gamma-1}{\gamma}} = 296.58 \left(\frac{1.71}{0.57} \right)^{\frac{1.4-1}{1.4}}$$

$$T_3 = 405.9 \text{ K}$$

$$\text{Compressor work/kg} = c_p(T_3 - T_2) = 1.005(405.9 - 296.58) = 109.9 \text{ kJ/kg}$$

$$\text{Compressor work/s} = \text{Mass flow} \times W = 23.22 \times 109.91 = 2552.0 \text{ kJ/s}$$

$$\text{Turbine work/s} = m c_p \Delta T = 23.22 \times 1.005(333 - 295.51) = 874.87 \text{ kJ/s}$$

$$\text{Net work} = W_C - W_T = 2552 - 874.87 = 1677.13 \text{ kJ/s} \quad \text{Ans.}$$

$$(e) \quad \text{COP} = \frac{\text{Refrigerating effect in kJ/s}}{\text{Net work in kJ/s}} = \frac{58.05}{1677.13} = 0.035 \quad \text{Ans.}$$

EXAMPLE 3.7 A simple air refrigeration system is used for an aircraft to take a load of 20 TR. The ambient pressure and temperature are 0.9 bar and 22°C respectively. The pressure of air is increased to 1 bar due to isentropic ramming action. The air is further compressed in a compressor to 3.5 bar and then cooled in a heat exchanger to 72°C . Finally the air is passed through the cooling turbine and then supplied to the cabin at 1.03 bar. The air leaves the cabin at 25°C . Assuming the isentropic efficiency of compressor and turbine as 80% and 75% respectively, find

- the power required to take the cooling load in the cabin.
- the COP of the system.

Take $c_p = 1.005 \text{ kJ/kg}\cdot\text{K}$; $\gamma = 1.4$

Solution: The cycle of operations is represented on the T-s diagram shown in Figure 3.21.

Given:

$$Q = 20 \text{ TR} = 20 \times 3.517 = 70.34 \text{ kJ/s}; \quad p_1 = 0.9 \text{ bar}; \quad T_1 = 22^\circ\text{C} = 22 + 273 = 295 \text{ K};$$

$$p_2 = 1 \text{ bar}; \quad p_3 = p_4 = 3.5 \text{ bar}; \quad T_4 = 72^\circ\text{C} = 72 + 273 = 345 \text{ K};$$

$$T_6 = 25^\circ\text{C} = 25 + 273 = 298 \text{ K}; \quad p_5 = p_7 = p_8 = 1.03 \text{ bar};$$

$$\eta_c = 80\%; \quad \eta_T = 75\%$$

Ram compression (1-2): For isentropic process,

$$T_2 = T_1 \left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} = 295 \left(\frac{1}{0.9} \right)^{\frac{1.4-1}{1.4}} = 304 \text{ K}$$

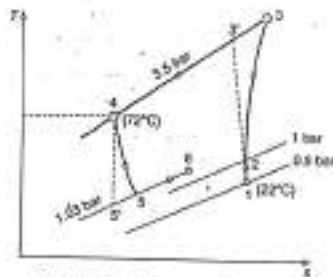


Figure 3.21 T-s diagram—Example 27.

Compressor: Consider the isentropic process (2–3'),

$$T_{3'} = T_2 \left(\frac{p_3}{p_2} \right)^{\gamma-1/\gamma} = 304 \left(\frac{3.5}{1} \right)^{(1.4-1)/1.4} = 434.8 \text{ K}$$

$$\eta_C = \frac{T_2 - T_3}{T_3 - T_{3'}} \text{ or } 0.8 = \frac{434.8 - 304}{T_3 - 304} \text{ or } T_3 = 467.5 \text{ K}$$

Cooling turbine: Consider the isentropic process (4–5'),

$$T_{5'} = T_4 \left(\frac{p_5}{p_4} \right)^{\gamma-1/\gamma} = 345 \left(\frac{1.03}{3.5} \right)^{(1.4-1)/1.4} = 243.2 \text{ K}$$

$$\eta_T = \frac{T_4 - T_5}{T_4 - T_{5'}} \text{ or } 0.75 = \frac{345 - T_5}{345 - 243.2} \text{ or } T_5 = 268.7 \text{ K}$$

(a) Power required to take the cooling load in cabin

Let \dot{m}_a be the mass flow rate circulated to cabin.

$$Q = \dot{m}_a c_p (T_6 - T_5)$$

$$70.34 = \dot{m}_a \times 1.005 (298 - 268.7)$$

$$\dot{m}_a = 2.3887 \text{ kg/s}$$

$$\text{Net work (Power)} = \dot{m}_a c_p (T_3 - T_2) = 2.3887 \times 1.005 \times (467.5 - 304) = 392.5 \text{ kJ/s or kW} \quad \text{Ans.}$$

(b) COP of the system

$$\text{COP} = \frac{\text{Refrigerating effect in kW}}{\text{Net work in kW}} = \frac{70.34}{392.5} = 0.179 \quad \text{Ans.}$$

EXAMPLE 3.5 The following data refers to a simple air refrigeration cycle of 20 TR capacity.

Ambient air temperature and pressure	= 20°C and 0.8 bar
Ram air pressure	= 0.9 bar
Compressor outlet pressure	= 3.6 bar
Temperature of air leaving H.E.	= 60°C
Pressure of air leaving the turbine	= 1 bar
Temperature of air leaving the cabin	= 22°C
Compressor efficiency	= 80%
Turbine efficiency	= 75%

Assume no pressure drop in H.E. and isentropic ramming process.

Sketch the cycle on T-s and p-v diagrams.

Calculate the net power required and the COP of the system.

Solution: Refer to Figure 3.22.

Given: $R_E = 20 \text{ TR} = 20 \times 3.517 = 70.34 \text{ kJ/s}$

Ambient pressure, $p_1 = 0.8 \text{ bar}$; Ambient temperature, $T_1 = 20^\circ\text{C} = 293 \text{ K}$;

Ram pressure, $p_2 = 0.9 \text{ bar}$; Compressor pressure, $p_3 = 3.6 \text{ bar}$; $\eta_C = 80\%$;

Temperature at exit of heat exchanger, $T_4 = 60^\circ\text{C} = 60 + 273 = 333 \text{ K}$;

Turbine exit pressure = cabin pressure = $p_5 = 1 \text{ bar}$;

Turbine efficiency, $\eta_T = 75\%$;

$T_6 = 22^\circ\text{C} = 22 + 273 = 295 \text{ K}$.

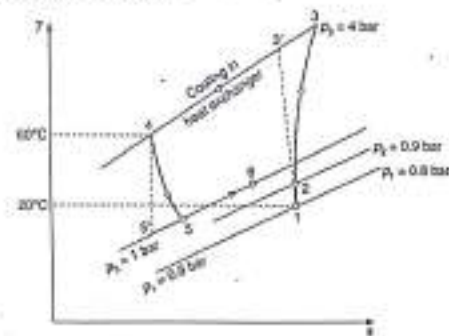


Figure 3.22 T-s diagram—Example 3.5.

$$\text{Ram compression (1–2): } T_2 = T_1 \left(\frac{p_2}{p_1} \right)^{\gamma-1/\gamma} = 293 \left(\frac{0.9}{0.8} \right)^{(1.4-1)/1.4} = 303 \text{ K}$$

Main compressor: For the isentropic compression process (2-3'),

$$T_{3'} = T_2 \left(\frac{P_3}{P_2} \right)^{\frac{\gamma-1}{\gamma}} = 303 \left(\frac{3.6}{0.9} \right)^{\frac{1.4-1}{1.4}} = 450.2 \text{ K}$$

$$\eta_c = \frac{T_{3'} - T_2}{T_{3'} - T_2} \text{ or } 0.8 = \frac{450.2 - 303}{T_{3'} - 303}$$

$$T_{3'} = 487.0 \text{ K}$$

Gas turbine: For the isentropic expansion process (4'-5'),

$$T_{5'} = T_{4'} \left(\frac{P_5}{P_{4'}} \right)^{\frac{\gamma-1}{\gamma}} = 333 \left(\frac{1}{3.6} \right)^{\frac{1.4-1}{1.4}} = 230.9 \text{ K}$$

$$\eta_T = \frac{T_{4'} - T_{5'}}{T_{4'} - T_{5'}} \text{ or } 0.75 = \frac{333 - T_{5'}}{333 - 230.9}$$

$$T_{5'} = 256.4 \text{ K}$$

Mass flow rate of air, \dot{m}_a

$$R_c = \dot{m}_a \times c_p (T_{4'} - T_{5'})$$

$$70.34 = \dot{m}_a \times 1.005 (295 - 256.4)$$

$$\dot{m}_a = 1.813 \text{ kg/s}$$

(a) Net compressor power required, P

$$P = \dot{m}_a c_p (T_{3'} - T_2)$$

$$= 1.813 \times 1.005 (487.0 - 303.0)$$

$$= 335.12 \text{ kJ/s or kW}$$

Ans.

(b) COP of the system

$$\text{COP} = \frac{R_c}{P} = \frac{70.34}{335.1} = 0.2098$$

Ans.

3.7.2 Simple Air Evaporative Cooling System

The simple air evaporative cooling system is similar to the simple air cooling system with just one modification. The system has an additional evaporative type heat exchanger to cool the air to a large extent before it is expanded in the cooling turbine. A simple air evaporative cooling system is shown in Figure 3.23.

In the evaporative type heat exchanger, water evaporates and cools the air to a greater extent (i.e. from point 4 to point 4'). At high altitudes, the evaporative cooling may be obtained by using alcohol or ammonia. Water, alcohol and ammonia all have different refrigerating effects at different altitudes. At the altitude of the aircraft, atmospheric pressure is of the order of 0.8 to 0.9 bar. Water boils and provides a cooling effect to the air.

The various processes involved in the simple air evaporative refrigeration cycle are:

- Process (1-2) → Ideal ram compression
- Process (1-2') → Actual ram compression
- Process (2'-3) → Isentropic compression of air in main compressor
- Process (2'-3') → Actual compression of air in main compressor
- Process (3'-4) → Constant pressure air cooling by ram air in heat exchanger
- Process (4-4') → Constant pressure air cooling by refrigerant in evaporator
- Process (4'-5) → Isentropic expansion of air in cooling turbine
- Process (4'-5') → Actual expansion of air in cooling turbine
- Process (5'-6) → Constant pressure heating of air to cabin temperature T_6

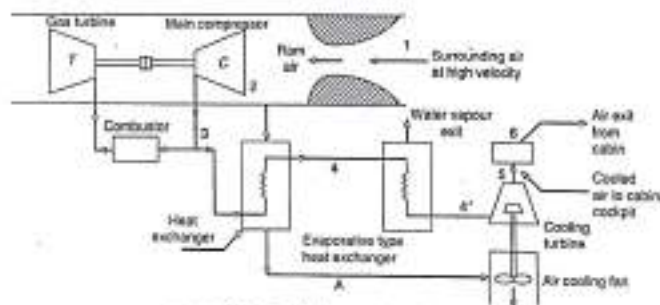


Figure 3.23 Simple air evaporative cooling system.

The T - s diagram for a simple evaporative cooling system is shown in Figure 3.24.

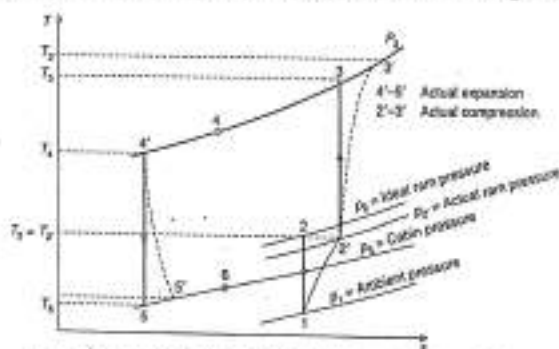


Figure 3.24 T - s diagram for a simple air evaporative cooling system of aircraft.

The cooling effect achieved in the aircraft refrigeration system

$$= \dot{m}_a \times c_p \times (T_6 - T_7)$$

where \dot{m}_a is the mass flow rate of air in kg/min, T_6 and T_7 are the cabin temperature and the air temperature at the outlet of the cooling turbine. If Q is the cooling load requirement of the cabin, then the air required for the refrigeration purpose,

$$\dot{m}_a = \frac{211 Q}{c_p (T_6 - T_7)} \text{ kg/min}$$

Power required for the refrigerating system,

$$P = \frac{\dot{m}_a c_p (T_7 - T_2)}{60} \text{ kW}$$

and COP of the refrigerating system

$$= \frac{211 Q}{\dot{m}_a c_p (T_7 - T_2)} = \frac{211 Q}{P \times 60}$$

EXAMPLE 3.9 A simple evaporative air refrigeration system is used for an aeroplane to take 20 TR of refrigeration load to maintain the cabin temperature at 25°C. The ambient air conditions are 20°C and 0.9 bar. The ambient air is rammed isentropically to a pressure of 1 bar. The air leaving the main compressor at pressure 4 bar is first cooled in the heat exchanger to a temperature of 100°C and then in the evaporator where its temperature is reduced by 10°C. The air from the evaporator is passed through the cooling turbine and then it is supplied to the cabin at a pressure of 1.05 bar. If the compression efficiency of the compressor is 80% and the expansion efficiency of cooling turbine is 75%, determine the following:

- Mass of air bled off the main compressor.
- Power required for the refrigerating system.
- COP of the refrigerating system.

Solution:

Given:	$Q = 20 \text{ TR};$	$T_1 = 20^\circ\text{C} = 20 + 273 = 293 \text{ K};$
	$p_1 = 0.9 \text{ bar};$	$p_2 = 1 \text{ bar}; p_3 = p_7 = 4 \text{ bar};$
	$\eta_c = 75\%$	$\eta_e = 80\%$
	$T_6 = 25^\circ\text{C} = 25 + 273 = 298 \text{ K};$	$p_6 = 1.05 \text{ bar}$

The T - s diagram for the simple evaporative air refrigeration system with the given conditions is shown in Figure 3.25.

- Suppose: T_1 = temperature of air entering the main compressor,
 T_2 = temperature of air after isentropic compression in the main compressor,
 T_3 = actual temperature of air leaving the main compressor, and
 T_4 = temperature of air entering the evaporative type heat exchanger.

We know that for an isentropic ramming process 1-2, taking ($\gamma = 1.4$)

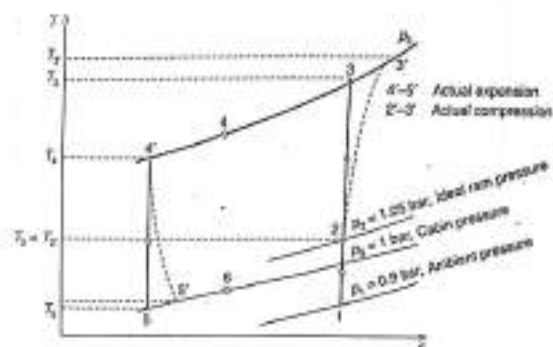


Figure 3.25 T - s diagram for a simple evaporative cooling system—Example 3.9

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1}{0.9} \right)^{\frac{1.4-1}{1.4}} = (1.11)^{0.286} = 1.03$$

$$\therefore T_2 = T_1 \times 1.03 = 293 \times 1.03 = 301.8 \text{ K}$$

Now for the isentropic compression process 2-3,

$$\frac{T_3}{T_2} = \left(\frac{p_3}{p_2} \right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{4}{1} \right)^{\frac{1.4-1}{1.4}} = (3.5)^{0.286} = 1.485$$

$$\therefore T_3 = T_2 \times 1.485 = 301.8 \times 1.485 = 448.5 \text{ K}$$

We know that the efficiency of the compressor,

$$\eta_c = \frac{\text{Isentropic increase in temperature}}{\text{Actual increase in temperature}} = \frac{T_3 - T_2}{T_3 - T_2}$$

$$\text{or } 0.8 = \frac{448.5 - 301.8}{T_3 - 301.8} = \frac{146.6}{T_3 - 301.8}$$

$$\therefore T_3 = 301.8 + \frac{146.6}{0.8} = 485.1 \text{ K}$$

The temperature of air leaving the first heat exchanger, given in the example, $T_4 = 100^\circ\text{C}$. The temperature of air in the evaporative type HE is reduced by 10°C , therefore the temperature of air leaving the evaporator and entering the cooling turbine,

$$T_5 = T_4 - 10 = 100 - 10 = 90^\circ\text{C} = 363 \text{ K}$$

Now for the isentropic expansion process 4-5,

$$\frac{T_5}{T_4} = \left(\frac{p_5}{p_4} \right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{4}{1.05} \right)^{\frac{1.4-1}{1.4}} = (3.81)^{0.286} = 1.46$$

$$T_5 = T_4 / 1.46 = 363 / 1.46 = 247.6 \text{ K}$$

Efficiency of the cooling turbine,

$$\eta_T = \frac{\text{Actual decrease in temperature}}{\text{Isentropic decrease in temperature}} = \frac{T_4 - T_5}{T_4 - T_5}$$

$$0.75 = \frac{363 - T_5}{363 - 247.6} = \frac{363 - T_5}{115.4}$$

$$T_5 = 363 - 0.75 \times 115.4 = 276.45$$

(a) The mass of air bled off the main compressor,

$$\dot{m}_a = \frac{211 \dot{Q}}{c_p (T_4 - T_5)} = \frac{211 \times 20}{1(298 - 276.4)} = 195 \text{ kg/min}$$

Ans.

(b) Power required for the refrigerating system,

$$P = \frac{\dot{m}_a c_p (T_2 - T_1)}{60} = \frac{195 \times 1(485.1 - 301.8)}{60} = 595.7 \text{ kW}$$

Ans.

(c) The COP of the refrigeration system

$$= \frac{210 \dot{Q}}{P \times 60} = \frac{210 \times 20}{595.7 \times 60} = 0.011$$

Ans.

3.7.3 Boot-strap Air Cooling System

A boot-strap air cooling system is shown in Figure 3.26. The main advantage of this system is that the compression of air is carried out in two stages with intercooling. Cold atmospheric air is used as a coolant in both the heat exchangers. The air bled from the main compressor is first cooled in the first heat exchanger using cold ram air. This cooled air, after compression in the secondary compressor, leads to the second heat exchanger, where it is again cooled by the ram air before passing to the cooling turbine. This type of cooling system is mostly used in transport type aircraft.

The T-s diagram for a boot-strap air cycle cooling system is shown in Figure 3.27. The various processes are as follows:

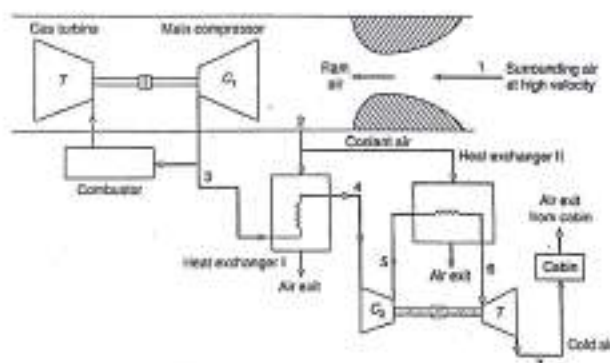


Figure 3.26 Boot-strap air refrigeration cycle for aircraft.

Process 1-2: It represents the isentropic ramming of ambient air from pressure p_1 and temperature T_1 to pressure p_2 and temperature T_2 . The process 1-2' represents the actual ramming process.

Process 2'-3: It represents the isentropic compression of air in the main compressor and the process 2'-3' represents the actual compression of air.

Process 3'-4: The compressed air from the main compressor is cooled in heat exchanger I using ram air. The pressure drop in the heat exchanger is neglected.

Process 4-5: The isentropic compression of cooled air is from the first heat exchanger, in the second compressor. The actual compression is represented by the process 4-5'.

Process 5'-6: The compressed air from the secondary compressor is cooled in heat exchanger II using ram air. The pressure drop in the heat exchanger is neglected.

Process 6-7: It is isentropic expansion of cooled air in the cooling turbine up to the cabin pressure. The process 6-7' represents the actual expansion of the cooled air in the cooling turbine.

Process 7'-8: It represents the heating of air up to the cabin temperature T_8 .

The quantity of air required for the refrigeration purpose to meet the \dot{Q} TR of refrigeration load in the cabin is

$$\dot{m}_a = \frac{211 \dot{Q}}{c_p (T_8 - T_7)} \text{ kg/min}$$

Power needed for the refrigerating system,

$$P = \frac{\dot{m}_a c_p (T_2 - T_1)}{60} \text{ kW}$$

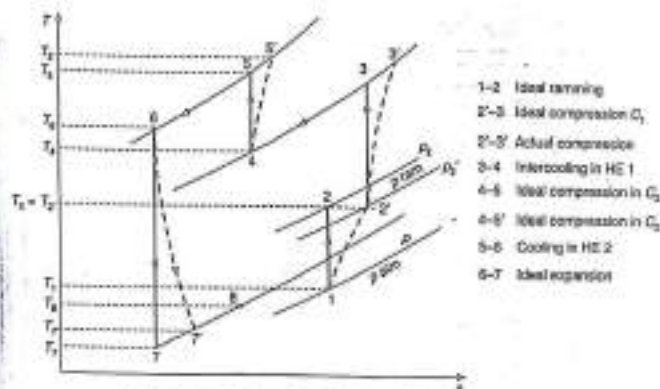


Figure 3.27 T-s diagram for boot-strap air cycle cooling system.

and COP of the refrigerating system

$$= \frac{211Q}{m_2 c_p (T_3 - T_2)} = \frac{211}{P \times 60}$$

EXAMPLE 3.10 A boot-strap cooling system is used in an aeroplane for 15 TR capacity. The ambient air temperature and pressure are 20°C and 0.85 bar respectively. The pressure of air rises from 0.85 bar to 1 bar due to ramming action. The pressure of air at the outlet of the main compressor is 3 bar. The discharge pressure of air from the secondary compressor is 4.5 bar. The efficiency of each of the compressors is 85% and that of the turbine is 90% . 75% of the heat content of the air discharged from the main compressor is removed in the heat exchanger I and 30% of the heat of air discharged from the auxiliary compressor is removed in the heat exchanger II using rammed air. Assuming ramming action to be isentropic, the required cabin pressure to be 0.9 bar and the temperature of the air leaving the cabin to be not more than 20°C , find the following:

- The mass of air bled off the main compressor.
- The power required to operate the system.
- The COP of the system.

Take $\gamma = 1.4$ and $c_p = 1$ kJ/kg-K.

Solution:

Given: $Q = 15$ TR; $T_1 = 20^\circ\text{C} = 20 + 273 = 293$ K
 $p_1 = 0.85$ bar; $p_2 = 1$ bar
 $p_3 = p_7 = p_4 = 3$ bar; $p_5 = p_7 = p_6 = 4.5$ bar
 $\eta_{C1} = \eta_{C2} = 85\%$; $\eta_T = 90\%$
 $p_7 = p_T = p_8 = 0.9$ bar; $T_8 = 20^\circ\text{C} = 20 + 273$ K
 $\gamma = 1.4$; $c_p = 1$ kJ/kg-K;

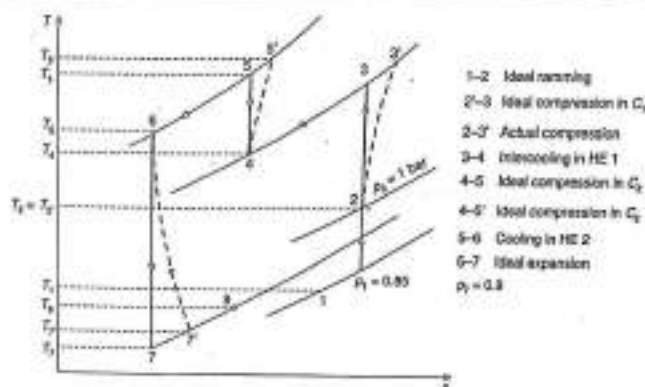


Figure 3.28 T-s diagram for boot-strap air cycle cooling system—Example 3.10.

The temperature entropy ($T-s$) diagram with the given conditions is shown in Figure 3.28. We know that for isentropic ramming process 1-2,

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1} \right)^{\gamma-1/\gamma} = \left(\frac{1}{0.85} \right)^{1.4-1/1.4} = (1.176)^{0.286} = 1.047$$

$$\therefore T_2 = T_1 \times 1.047 = 293 \times 1.047 = 306.8 \text{ K}$$

Now for the isentropic compression process 2-3,

$$\frac{T_3}{T_2} = \left(\frac{p_3}{p_2} \right)^{\gamma-1/\gamma} = \left(\frac{3}{1} \right)^{1.4-1/1.4} = (3)^{0.286} = 1.37$$

$$\therefore T_3 = T_2 \times 1.37 = 306.8 \times 1.37 = 420.3 \text{ K}$$

Isentropic efficiency of the compressor,

$$\eta_{C1} = \frac{\text{Isentropic increase in temperature}}{\text{Actual increase in temperature}} = \frac{T_3 - T_2}{T_3' - T_2}$$

$$\text{or } 0.85 = \frac{420.3 - 306.8}{T_3' - 306.8} = \frac{113.5}{T_3' - 306.8}$$

$$\therefore T_3' = 306.8 + 113.5/0.85 = 448.7 \text{ K} = 175.7^\circ\text{C}$$

For perfect intercooling, the temperature of air at the outlet of heat exchanger I is equal to the temperature of air entering the main compressor. But in intercooling only 75% of the

compression heat is removed and so the temperature of air at point 4 is

$$T_4 = T_2 + 0.25(T_3' - T_2) = 306.8 + 0.25(420.3 - 306.8) = 335 \text{ K}$$

The isentropic process 4-5,

$$\frac{T_4}{T_5} = \left(\frac{p_1}{p_6}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{4.5}{3}\right)^{\frac{1.4-1}{1.4}} = (1.5)^{0.286} = 1.122$$

$$\therefore T_5 = T_4 \times 1.122 = 335 \times 1.122 = 376 \text{ K} = 118.5^\circ\text{C}$$

Isentropic efficiency of the auxiliary (secondary) compressor,

$$\eta_{is} = \frac{T_5 - T_4}{T_5 - T_6}$$

$$\text{or } 0.85 = \frac{376 - 335}{T_5 - 335} = \frac{41}{T_5 - 335}$$

$$\therefore T_5 = 383.2 \text{ K} = 110^\circ\text{C}$$

For perfect after-cooling, the temperature of air at the outlet of heat exchanger II is equal to the temperature of air entering the auxiliary (secondary) compressor. But in after-cooling, only 30% of the compression heat is removed. Therefore, the temperature of air at point 6 is

$$T_6 = T_4 + 0.7(T_5 - T_4) = 335 + 0.7(383.2 - 335) = 368.7 \text{ K}$$

For the isentropic process 6-7,

$$\frac{T_6}{T_7} = \left(\frac{p_1}{p_6}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{0.9}{4.5}\right)^{\frac{1.4-1}{1.4}} = (0.2)^{0.286} = 0.631$$

$$= (0.225)^{0.286} = 0.631$$

$$\therefore T_7 = T_6 \times 0.631 = 368.7 \times 0.631 = 232.6 \text{ K} = -40.3^\circ\text{C}$$

The turbine efficiency,

$$\eta_T = \frac{\text{Actual decrease in temperature}}{\text{Isentropic decrease in temperature}} = \frac{T_6 - T_7}{T_6 - T_7}$$

$$\text{or } 0.9 = \frac{368.7 - T_7}{368.7 - 232.6} = \frac{368.7 - T_7}{136.1}$$

$$\therefore T_7 = 246.2 \text{ K} = -26.8^\circ\text{C}$$

(ii) Flow rate of air:

The mass of air bled off the main compressor

$$\dot{m}_a = \frac{211Q}{c_p(T_4 - T_7)} = \frac{211 \times 15}{1 \times (335 - 246.2)} = 67.3 \text{ kg/min}$$

Ans.

(b) Power required to operate the system:

$$P = \frac{\dot{m}_a c_p (T_2 - T_7)}{60} = \frac{67.3 \times 1 (448.7 - 306.8)}{60} = 159 \text{ kW}$$

Ans.

(c) COP of the system:

We know the COP of the system to be

$$= \frac{211Q}{\dot{m}_a c_p (T_2 - T_7)} = \frac{211 \times 15}{67.3 \times 1 (448.7 - 306.8)} = 0.33$$

Ans.

EXAMPLE 3.11 An aeroplane uses the boot-strap air cooling system. It requires 16 TR of refrigeration. Ambient temperature and pressure are -13°C and 0.6 bar respectively. The speed of the plane is 800 kmph. The ram air is compressed in the main compressor up to 3.6 bar pressure and this is further compressed in the secondary compressor up to 4.4 bar. The isentropic efficiency for each compressor is 85%. The air in the heat exchanger I is cooled up to 113°C by ram air while the air from the secondary compressor is cooled by ram air up to 87°C . This cooled air is then expanded in cooling turbine up to the cabin pressure of 1 bar with an isentropic efficiency of 90%. The cabin temperature is required to be maintained at 20°C . Find the following:

(a) Mass flow rate of air in kg/min.

(b) Power required to operate the system.

(c) COP of the system.

Assume, $\gamma = 1.4$ and $c_p = 1.01 \text{ kJ/kg}\cdot\text{K}$.

Solution:

Given:

$Q = 16 \text{ TR}$	$C = 800 \text{ kmph}$
$p_1 = 0.6 \text{ bar}$	$T_1 = -13^\circ\text{C} = -13 + 273 = 260 \text{ K}$
$p_3 = p_2 = p_4 = 3.6 \text{ bar}$	$p_2 = p_1 = p_6 = 4.4 \text{ bar}$
$\eta_c = \eta_{c2} = 85\%$	$\eta_T = 90\%$
$T_6 = 87^\circ\text{C} = 87 + 273 = 360 \text{ K}$	$p_7 = p_T = p_8 = 1 \text{ bar}$
$T_8 = 20^\circ\text{C} = 20 + 273 = 293 \text{ K}$	$T_4 = 113^\circ\text{C} = 113 + 273 = 386 \text{ K}$

The $T-s$ diagram is shown in Figure 3.29 and system diagram in Figure 3.26.

Ram compression (1-2):

$$\text{Velocity of air, } C = 800 \text{ kmph} = \frac{800 \times 1000}{3600} = 222.2 \text{ m/s}$$

$$T_2 = T_1 + \frac{C^2}{2c_p} = 260 + \frac{(222.2)^2}{2 \times (1.01 \times 1000)} = 284.5 \text{ K}$$

$$p_2 = p_1 \left(\frac{T_2}{T_1}\right)^{\frac{\gamma}{\gamma-1}} = 0.6 \left(\frac{284.5}{260}\right)^{\frac{1.4}{1.4-1}} = 0.822 \text{ bar}$$

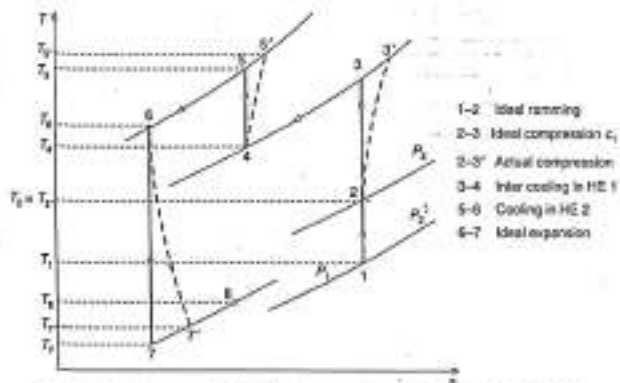


Figure 3.29 T-s diagram for bootstrap air evaporative cooling system—Example 3.11.

Process (2-3) in main compressor:

$$T_3 = T_2 \left(\frac{P_3}{P_2} \right)^{\frac{\gamma-1}{\gamma}} = 284.5 \left(\frac{3.6}{0.822} \right)^{\frac{1.4-1}{1.4}}$$

or $T_3 = 433.7 \text{ K}$

$$\eta_{C1} = \frac{T_3 - T_2}{T_3 - T_2'} \quad \text{i.e.} \quad T_3 = T_2 + \frac{T_3 - T_2}{\eta_{C1}}$$

$$\therefore T_3 = 284.5 + \frac{(433.7 - 284.5)}{0.85} = 460.0 \text{ K}$$

Secondary compression process (4-5):

$$T_5 = T_4 \left(\frac{P_5}{P_4} \right)^{\frac{\gamma-1}{\gamma}} = 386 \left(\frac{4.4}{3.6} \right)^{\frac{1.4-1}{1.4}} = 413.0 \text{ K}$$

But $\eta_{C2} = \frac{T_5 - T_4}{T_5 - T_4'}$

$$\text{or} \quad T_5 = T_4 + \frac{T_5 - T_4}{\eta_{C2}} = 386 + \frac{(413.0 - 386)}{0.85} = 452.5 \text{ K}$$

Expansion in cooling turbine (6-7):

$$T_7 = T_6 \left(\frac{P_7}{P_6} \right)^{\frac{\gamma-1}{\gamma}} = 360 \left(\frac{1}{4.4} \right)^{\frac{1.4-1}{1.4}}$$

$$= 235.8 \text{ K}$$

$$\eta_T = 0.9 = \frac{360 - T_7}{360 - 235.8}$$

or

$$T_7 = 248.2 \text{ K}$$

(a) Refrigerating effect, i.e. heating of air in cabin in process (7-8)

$$Q = \dot{m}_a c_p (T_8 - T_7)$$

$$\text{or} \quad 16 \times 211 = \dot{m}_a = 1.01(293 - 248.2)$$

\therefore

$$\dot{m}_a = 74.61 \text{ kg/min}$$

Ans.

(b) Total compressor power required is to run the main compressor.

$$P_C = \dot{m}_a [c_p (T_3 - T_2)]$$

$$= \frac{74.61}{60} \times 1.01(460.0 - 284.5) = 220.60 \text{ kN/s or kW}$$

Ans.

(c)

$$\text{Since } 1 \text{ TR} = 3.517 \text{ kW}$$

$$\text{COP} = \frac{Q}{P_C} = \frac{16 \times 3.517}{220.6} = 0.255$$

Ans.

EXAMPLE 3.12 A bootstrap air refrigeration system is used in an aeroplane for the 10 TR refrigeration load. The ambient air conditions are 15°C and 0.9 bar. This air is rammed isentropically to a pressure of 1.1 bar. The pressure of air bled off the main compressor is 3.5 bar and this is further compressed in the secondary compressor to a pressure of 4.5 bar. The isentropic efficiency of both the compressors is 90% and that of cooling turbine is 85%. The effectiveness of both the heat exchangers is 0.6. If the cabin is to be maintained at 23°C and the pressure in the cabin is 1 bar, find the following:

- Mass of air passing through the cabin.
- Power used for the refrigeration system.
- COP of the system.

Draw the schematic and T-s diagram for the system, take $\gamma = 1.4$ and $c_p = 1 \text{ kJ/kg}\cdot\text{K}$.

Solution: Figure 3.30 shows the T-s diagram.

Given:

$$\text{Refrigeration effect} = 10 \text{ TR} = 10 \times 3.517 = 35.17 \text{ kW};$$

$$P_1 = 0.9 \text{ bar};$$

$$T_1 = 15^\circ\text{C} = 15 + 273 = 288 \text{ K};$$

$$P_2 = 1.1 \text{ bar};$$

$$P_3 = P_2 = P_6 = 3.5 \text{ bar};$$

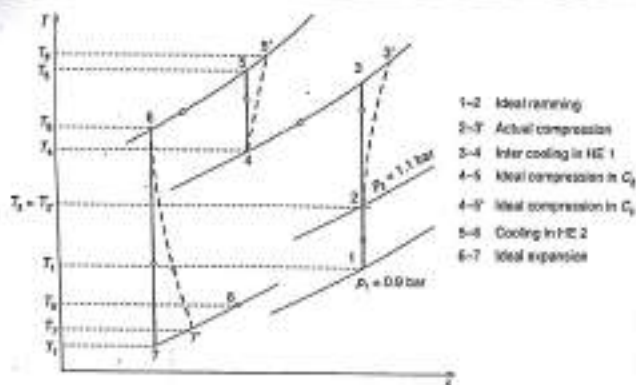


Figure 1.30 T-s diagram for boot-strap air evaporative cooling system—Example 1.12.

$$p_1 = p_2 = p_3 = 4.5 \text{ bar};$$

$$\eta_C = \eta_{C1} = \eta_{C2} = 90\%;$$

$$\eta_T = 85\%;$$

$$E_1 = E_2 = E_3 = 0.6$$

$$\text{Cabin temperature, } T_4 = 25^\circ\text{C} = 25 + 273 = 298 \text{ K, } p_T = p_2 = p_3 = 1 \text{ bar}$$

Isentropic ram compression (1-2):

$$T_2 = T_1 \left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} = 288 \left(\frac{1.1}{0.9} \right)^{\frac{1.4-1}{1.4}} = 305 \text{ K}$$

Main compressor: Isentropic compression process (2-3):

$$T_3 = T_2 \left(\frac{p_3}{p_2} \right)^{\frac{\gamma-1}{\gamma}} = 305 \left(\frac{3.5}{1.1} \right)^{\frac{1.4-1}{1.4}} = 424.5 \text{ K}$$

$$\eta_{C1} = \frac{T_3 - T_2}{T_Y - T_2}; 0.9 = \frac{424.5 - 305}{T_Y - 305}; \therefore T_Y = 437.8 \text{ K}$$

Heat exchanger (HE-1): Its effectiveness is

$$E_1 = \frac{T_Y - T_4}{T_Y - T_2}; 0.6 = \frac{437.8 - T_4}{437.8 - 305}; \therefore T_4 = 358.1 \text{ K}$$

Secondary compressor: Consider the isentropic process (4-5),

$$T_5 = T_4 \left(\frac{p_5}{p_4} \right)^{\frac{\gamma-1}{\gamma}} = 358.1 \left(\frac{4.5}{3.5} \right)^{\frac{1.4-1}{1.4}} = 384.8 \text{ K}$$

$$\eta_{C2} = 0.9 = \frac{384.8 - 358.1}{T_Y - 358.1}; \therefore T_Y = 387.8 \text{ K}$$

Heat exchanger II: Its effectiveness is

$$E_2 = \frac{T_Y - T_6}{T_Y - T_3}$$

$$\text{Hence, } 0.6 = \frac{387.8 - T_6}{387.8 - 305}; \therefore T_6 = 338.1 \text{ K}$$

Cooling turbine: Consider the isentropic process (6-7),

$$T_7 = T_6 \left(\frac{p_7}{p_6} \right)^{\frac{\gamma-1}{\gamma}} = 338.1 \left(\frac{1}{4.5} \right)^{\frac{1.4-1}{1.4}} = 220 \text{ K}$$

$$\eta_T = \frac{T_6 - T_7}{T_6 - T_Y}$$

$$\text{or } 0.85 = \frac{338.1 - T_7}{338.1 - 220}; \therefore T_7 = 237.7 \text{ K}$$

(a) Mass of air passing through the cabin, \dot{m}_a

$$\text{Refrigerating effect} = \dot{m}_a \times c_p (T_4 - T_7)$$

$$35.17 = \dot{m}_a \times 1 \times (298 - 237.7)$$

$$\therefore \dot{m}_a = 0.58325 \text{ kg/s} = 35 \text{ kg/min} \quad \text{Ans.}$$

(b) Power required for the refrigeration system, P_c

$$P_c = \dot{m}_a \times c_p (T_Y - T_2) = 0.58325 \times 1 \times (437.8 - 305)$$

$$= 77.46 \text{ kW} \quad \text{Ans.}$$

(c) COP of the system

$$\text{COP} = \frac{\text{Refrigerating effect}}{\text{Power supplied}} = \frac{35.17}{77.46} = 0.454 \quad \text{Ans.}$$

EXAMPLE 1.13 A boot-strap refrigeration system is used in an airplane. The following observations are made. The ambient air temperature is 15°C and the pressure is 0.85 bar. Due to ramming action the pressure increases to 1 bar. This ram air is used for heat exchangers. This air is then compressed in primary compressor to 3.25 bar. The discharge pressure of air from the secondary compressor is 4.25 bar. Assume compression efficiency of 0.9 and turbine efficiency of 0.85. Effectiveness for both the heat exchangers is 0.7. The cabin pressure is 0.9

bar and the temperature of air leaving the cabin is 22°C. Assume the remaining action to be isentropic. Calculate the COP and the power required per ton of refrigeration.

Solution: Refer to Figure 3.31 for the cycle and the T-s diagram.

Given : $T_1 = 15^\circ\text{C} = 15 + 273 = 288\text{ K}$ $p_1 = 0.85\text{ bar}, p_2 = 1\text{ bar}$
 $p_3 = p_4 = 3.25\text{ bar}$ $p_5 = p_6 = 4.25\text{ bar}$
 $\eta_{C1} = \eta_{C2} = 0.9$ $\eta_T = 0.8, E_r = E_{r1} = E_{r2} = 0.7$
 $p_7 = p_8 = p_9 = 0.9\text{ bar}$ $T_8 = 22^\circ\text{C} = 22 + 273 = 295\text{ K}$

Isentropic ram compression (1-2):

$$T_2 = T_1 \left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} = 288 \left(\frac{1}{0.85} \right)^{0.4-0.1/1.4} = 301.7\text{ K}$$

Main compressor: Considering isentropic compression,

$$T_3 = T_2 \left(\frac{p_3}{p_2} \right)^{\frac{\gamma-1}{\gamma}} = 301.7 \left(\frac{3.25}{1} \right)^{0.4-0.1/1.4} = 422.5\text{ K}$$

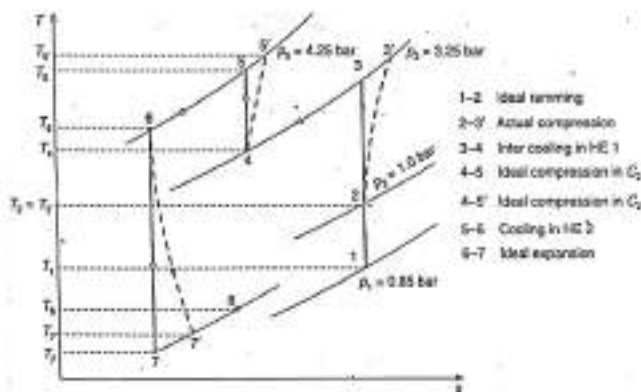


Figure 3.31 T-s diagram for boot-strap air evaporative cooling system—Example 3.11.

But the compression is not isentropic:

$$\eta_{C1} = 0.9 = \frac{T_3 - T_2}{T_3' - T_2}; \quad T_3' = 435.9\text{ K}$$

Heat exchanger I: Ram air is employed in the heat exchangers for cooling the compressed air. Its effectiveness E_r is

$$E_r = \frac{T_3' - T_4}{T_3' - T_2}; \quad 0.7 = \frac{435.9 - T_4}{435.9 - 301.7}$$

$$T_4 = 362.0\text{ K}$$

Compression in secondary compressor: Considering isentropic compression (4-5),

$$T_5 = T_4 \left(\frac{p_5}{p_4} \right)^{\frac{\gamma-1}{\gamma}} = 362 \left(\frac{4.25}{3.25} \right)^{0.4-0.1/1.4} = 369.3\text{ K}$$

$$\eta_{C2} = 0.9 = \frac{369.3 - 342}{T_6 - 362}; \quad \therefore T_6 = 372.2\text{ K}$$

Heat exchanger II: Cold ram air is used for cooling the compressed air in HE2. Therefore,

$$E_r = \frac{T_6 - T_7}{T_6 - T_5}; \quad 0.7 = \frac{372.2 - T_7}{372.2 - 301.7}$$

$$T_7 = 322.6\text{ K}$$

Cooling turbine:

Considering isentropic expansion (6-7),

$$T_7 = T_6 \left(\frac{p_7}{p_6} \right)^{\frac{\gamma-1}{\gamma}} = 322.6 \left(\frac{0.9}{4.25} \right)^{0.4-0.1/1.4} = 207.2\text{ K}$$

$$\eta_T = \frac{T_6 - T_7}{T_6 - T_7'}; \quad 0.80 = \frac{322.6 - T_7'}{322.6 - 207.2}$$

$$T_7' = 230.3\text{ K}$$

(a) Consider 1 kg of air:

$$\begin{aligned} \text{Compressor work} &= c_p(T_3' - T_2) + c_p(T_5 - T_4) \\ &= 1.005(435.9 - 301.7) + (372.2 - 362.0) \\ &= 134.9 + 10.35 = 165.2\text{ kJ/kg} \end{aligned}$$

$$\begin{aligned} \text{Specific refrigerating effect, } Q_1 &= c_p(T_8 - T_7') = 1.005(295.0 - 230.3) \\ &= 65\text{ kJ/kg} \end{aligned}$$

$$\text{COP} = \frac{\text{Refrigerating effect}}{\text{Work input}} = \frac{65}{165.2} = 0.393 \quad \text{Ans.}$$

(b) Power required per ton of refrigeration, P

$$1\text{ TR} = 3.517\text{ kW}$$

$$Q = \dot{m}_a \times q$$

$$\dot{m}_a = \frac{1 \text{ TR}}{\text{Specific refrigerating effect}} = \frac{3.517}{65} \text{ kg/s}$$

and

$$P = \dot{m}_a \times \text{compressor work} = \frac{3.517}{65} \times 165.2$$

$$= 8.94 \text{ kW}$$

Ans.

3.7.4 Boot-strap Air Evaporative Cooling System

A boot-strap air evaporative cooling system is shown in Figure 3.32. The simple boot-strap air cooling system is modified introducing an evaporator between the second heat exchanger and the cooling turbine. Air is cooled to a low temperature in the evaporator before it is expanded in the cooling turbine. This improves the COP of the cycle or requires less compressor work for the same refrigeration effect.

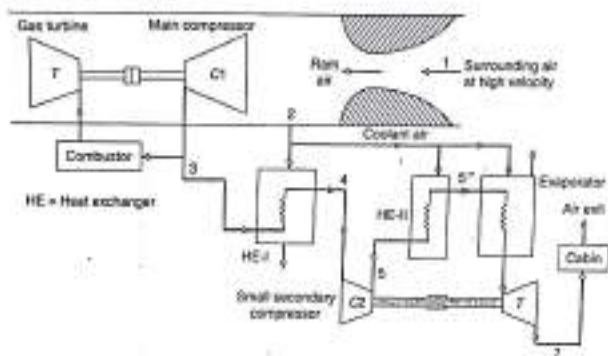


Figure 3.32 Boot-strap air evaporative cooling cycle for aircraft.

The $T-s$ diagram for a boot-strap air evaporative cooling system is shown in Figure 3.33. The various processes of this cycle are same as those a simple boot-strap system and are indicated on the $T-s$ diagram itself. Since the temperature of the air leaving the cooling turbine in the boot-strap evaporative system is lower than that of the simple boot-strap system, the mass of air (\dot{m}_a) per TR of refrigeration is less in the boot-strap evaporative system.

The quantity of air required for the refrigeration load of Q TR in the cabin will be

$$\dot{m}_a = \frac{211Q}{c_p(T_4 - T_7)} \text{ kg/min}$$

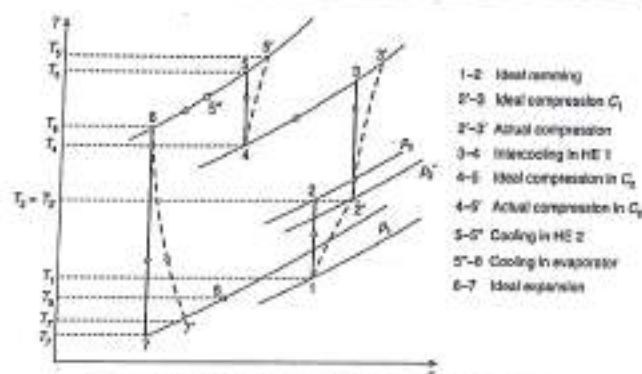


Figure 3.33 $T-s$ diagram for a boot-strap air evaporative cooling system.

Power required for the refrigeration system is given by

$$P = \frac{\dot{m}_a c_p (T_7 - T_7')}{60} \text{ kW}$$

and COP of the refrigeration system

$$= \frac{211Q}{\dot{m}_a c_p (T_7 - T_7')} = \frac{211Q}{P \times 60}$$

3.7.5 Reduced Ambient Air Cooling System

The reduced ambient air cooling system is shown in Figure 3.34. It is used for air cooling of very high speed aircraft. Due to high speed of the aircraft, the temperature of the ram air is relatively high compared with other cooling systems seen earlier. Therefore, ram air cannot be employed directly as a cooling media but it is first expanded in the turbine T_1 and then used for cooling the compressed air in the heat exchanger.

This cooling system includes two cooling turbines and one heat exchanger. This high pressure and high temperature air bled off from the main compressor is cooled initially in the heat exchanger. The cooled air from the heat exchanger is passed through the second cooling turbine from where the air is supplied to the cabin. The work of the cooling turbine is used to drive the cooling fan (through reduction gears), which draws cooling air from the heat exchanger.

The $T-s$ diagram for the reduced ambient air cooling system is shown in Figure 3.35. The various processes are as follows:

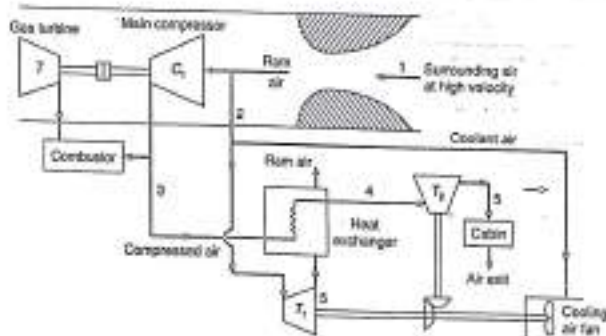


Figure 1.34 Reduced ambient air cooling cycle for aircraft.

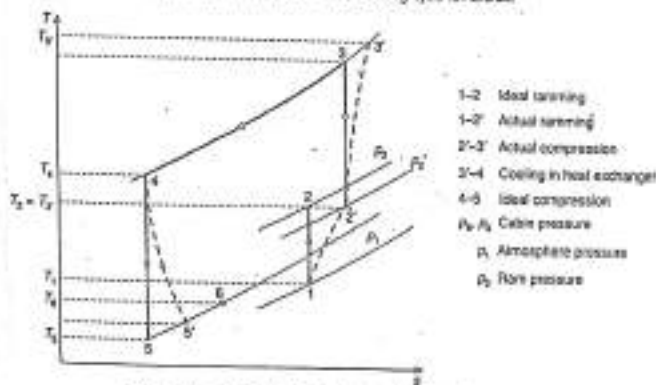


Figure 1.35 T-s diagram for reduced ambient air cooling system.

- Process (1-2): Isentropic compression of ram air, actual compression shown by 1-2' line.
 Process (2-3): Isentropic compression in main compressor, actual process by 2'-3'.
 Process (3'-4): Constant pressure cooling of compressed air drawn from main compressor in heat exchanger.
 Process (4-5): Isentropic expansion of cooled air in the second cooling turbine (T₂), and the actual expansion is represented by the process (4-5').
 Process (5-6): Constant pressure heating of air up to temperature T₆ in cabin to produce refrigerating effect.

1. The quantity of air required to produce refrigeration effect of Q TR will be

$$m_a = \frac{211 Q}{c_p (T_6 - T_2)} \text{ kg/min}$$

2. Power required for the refrigeration system is

$$P = \frac{m_a c_p (T_1 - T_2)}{60} \text{ kW}$$

3. COP of the system

$$= \frac{211 Q}{m_a c_p (T_1 - T_2)} = \frac{211 Q}{P \times 60}$$

where, P = power in kW, Q = refrigeration effect in TR.

EXAMPLE 1.14 The reduced ambient air refrigeration system used for an aircraft consists of two cooling turbines, one heat exchanger and one air cooling fan. The speed of the aircraft is 1500 km/h. The ambient air conditions are 0.8 bar and 10°C. The rammed air used for cooling is expanded in the first cooling turbine and leaves it at a pressure of 0.8 bar. The air bled from the main compressor at 6 bar is cooled in the heat exchanger and leaves it at 100°C. The cabin is to be maintained at 20°C and 1 bar. If the isentropic efficiency for the main compressor is 85% and both of the cooling turbines have 80% efficiency, find the following.

- (a) Mass flow rate of air supplied to cabin to take a cabin load of 10 TR of refrigeration.
 (b) Quantity of air passing through the heat exchanger if the temperature rise of ram air is limited to 30 K.
 (c) Power used to drive the cooling fan.
 (d) COP of the system.

Solution:

Given: $V = 1500 \text{ km/h} = 417 \text{ m/s}$; $p_1 = 0.8 \text{ bar}$; $T_1 = 10^\circ\text{C} = 10 + 273 = 283 \text{ K}$;
 $p_3 = p_4 = 6 \text{ bar}$; $T_4 = 100^\circ\text{C} = 100 + 273 = 373 \text{ K}$; $T_6 = 20^\circ\text{C} = 20 + 273 = 293 \text{ K}$; $p_6 = 1 \text{ bar}$;
 $\eta_c = 85\%$; $\eta_{t1} = \eta_{t2} = 80\%$; Refrigeration effect = $Q = 10 \text{ TR}$

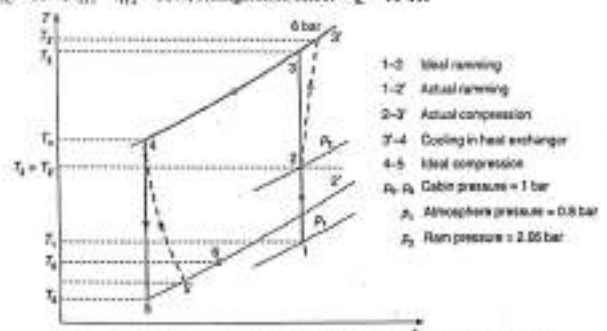


Figure 1.36 T-s diagram for reduced ambient cooling system—Example 1.14.

The T - s diagram for the reduced ambient air refrigeration system with the given conditions is shown in Figure 3.36.

Suppose T_1 = Stagnation temperature of ambient air entering the main compressor
 p_1 = Pressure of air at the end of isentropic ramming, and

we know that $T_2 = T_1 + \frac{V^2}{2000 c_p} = 283 + \frac{(417)^2}{2000 \times 1} = 370 \text{ K}$

Isentropic ramming process 1-2:

$$p_1 = p_2 \left(\frac{T_1}{T_2} \right)^{\frac{\gamma}{\gamma-1}} = 0.8 \left(\frac{370}{283} \right)^{1.4/(1.4-1)} = 2.06 \text{ bar}$$

The expansion of ram air in the first cooling turbine is shown in Figure 3.37. The vertical line 2-1 represents the isentropic expansion process and the curve 2-1 represents the actual expansion process.

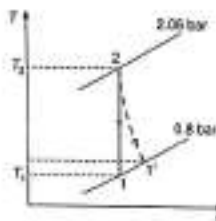


Figure 3.37 Expansion of ram air in first turbine on T - s diagram—Example 3.14.

Isentropic expansion of ram air in the first turbine,

$$T_2 = T_1 \left(\frac{p_1}{p_2} \right)^{\frac{\gamma-1}{\gamma}} = 370 / \left(\frac{2.06}{0.8} \right)^{0.4/(1.4)} = 282 \text{ K}$$

Isentropic efficiency of the first cooling turbine,

$$\eta_{T1} = \frac{\text{Actual decrease in temperature}}{\text{Isentropic decrease in temperature}} = \frac{T_2 - T_1}{T_2 - T_1}$$

$$0.8 = \frac{370 - T_1}{370 - 282}$$

$$T_1 = 299.6 \text{ K}$$

Main compressor: Considering isentropic compression (2-3),

$$T_3 = T_2 \left(\frac{p_1}{p_2} \right)^{\frac{\gamma-1}{\gamma}} = 370 \left(\frac{6}{2.06} \right)^{0.4/(1.4)} = 502.6 \text{ K}$$

We know that isentropic efficiency of the compressor,

$$\eta_c = \frac{\text{Isentropic increase in temperature}}{\text{Actual increase in temperature}} = \frac{T_3 - T_2}{T_3 - T_2}$$

$$0.85 = \frac{502.6 - 370}{T_3 - 370}; \quad \therefore T_3 = 525.5 \text{ K}$$

$$p_3 = p_2 = p_4 = 1.0 \text{ bar}$$

Now for the isentropic expansion of air in the second cooling turbine (process 4-5),

$$\frac{T_4}{T_5} = \left(\frac{p_4}{p_5} \right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{6}{1.0} \right)^{0.4/(1.4)} = 1.669$$

$$T_5 = \frac{T_4}{1.669} = \frac{373}{1.669} = 223 \text{ K}$$

$$\eta_{T2} = \frac{\text{Actual decrease in temperature}}{\text{Isentropic decrease in temperature}} = 0.8 = \frac{373 - T_5}{373 - 223} = \frac{373 - T_5}{149}$$

$$\therefore T_5 = 253.3 \text{ K}$$

(a) Mass flow rate of air supplied to cabin:

For mass flow rate of air supplied to cabin, apply the energy balance equation:

Total refrigeration load = heat carried away by the air

$$211 \times Q = m_a \times c_p \times (T_6 - T_5)$$

$$m_a = \frac{211 Q}{c_p (T_6 - T_5)} = \frac{211 \times 10}{1 (293 - 253.3)} = 53.14 \text{ kg/min}$$

Ans.

(b) Quantity of ram air passing through the heat exchanger:

Let m_a be the quantity of ram air passing through the heat exchanger.

Consider the heat balance for the heat exchanger (Refer Figure 3.36).

Heat given out by compressed air = Heat gained by ram air

$$m_a c_p (T_3 - T_4) = m_a \times c_p (\text{Increase in temperature of ram air})$$

$$53.14 \times 1.0 \times (525.6 - 373) = m_a \times 1.0 \quad (80)$$

$$m_a = 101.2 \text{ kg/s}$$

Ans.

(c) Power to drive the cooling fan:

Work output from the first cooling turbine,

$$W_{T1} = m_a \times c_p (T_2 - T_1)$$

$$= 101.2 \times 1(370 - 299.6) = 7124.5 \text{ kJ/min}$$

and work output from the second cooling turbine,

$$W_{T2} = m_a \times c_p (T_4 - T_5)$$

$$= 53.14 \times 1(373 - 253.3) = 6360.8 \text{ kJ/min}$$

Combined work output from both the cooling turbines,

$$W_T = W_{T1} + W_{T2} = 7124.5 + 6360.8 = 13484 \text{ kJ/min}$$

The power generated by both the turbines is combined and supplied to the cooling fan.
Power used to drive the cooling fan = 13484 kJ/min = 224.7 kW

Ans.

(d) COP of the system:

We know that COP of the system

$$= \frac{211 Q}{m_a c_p (T_5 - T_2)} = \frac{211 Q}{53.14 \times (525.60 - 370)} = 0.255$$

Ans.

3.7.6 Regenerative Air Cooling System

The regenerative air cooling system is shown in Figure 3.38 and its corresponding T-s diagram in Figure 3.39. The speciality of this system is the regenerative heat exchanger. The high pressure and high temperature air from the main compressor is first cooled by the ram air in the heat exchanger. This air is further cooled in the regenerative heat exchanger with a portion of the air bled after expansion in the cooling turbine. This type of cooling system is used for supersonic aircraft and rockets.

The various processes are as follows:

Process 1-2: Isentropic ramming of air and the curve 1-2' represents the actual ramming process of air.

Process 2-3: Isentropic compression of air in the main compressor and the process 2'-3' represents the actual compression of air.

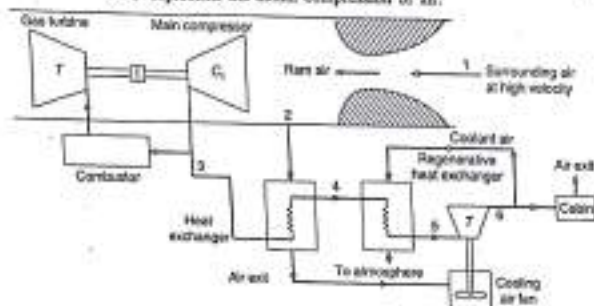


Figure 3.38 Regenerative air cooling system.

- Process 3'-4: Cooling of compressed air by ram air in the heat exchanger.
Process 4-5: Cooling of air in the regenerative heat exchanger.
Process 5-6: Isentropic expansion of air in the cooling turbine up to the cabin pressure and 5-6' represents the actual expansion of air.
Process 6'-7: Increase in air temperature while passing through the cabin.

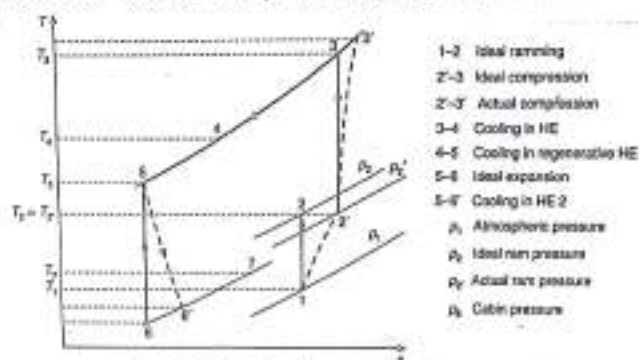


Figure 3.39 T-s diagram for regenerative air cooling system.

1. Mass flow rate of air supplied to cabin

Let Q_{TR} be the cooling load in the cabin.

Mass flow rate of air supplied to cabin, apply energy balance equation.

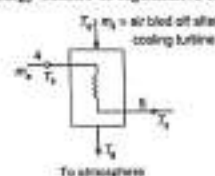
Total refrigeration load = heat carried away by the air

$$211 = Q = m_a \times c_p \times (T_1 - T_6)$$

$$m_a \text{ in kg/min} = \frac{211 Q}{c_p (T_1 - T_6)}$$

where, m_a = total mass of air bled from the main compressor.

2. Mass flow rate of air bled off after cooling turbine and used for regenerative cooling. From the energy balance of regenerative heat exchanger, we have



$$m_1 c_p (T_4 - T_0) = m_2 c_p (T_4 - T_2) \quad \therefore m_2 = \frac{m_1 (T_4 - T_3)}{(T_4 - T_0)}$$

T_4 = temperature of air escaping to atmosphere from the regenerative heat exchanger,
 m_2 = mass of cold air bled from the cooling turbine for regenerative heat exchanger.

3. Power required for the refrigeration system

$$P = \frac{m_2 c_p (T_2 - T_0)}{60} \text{ kW}$$

4. COP of the refrigerating system

$$= \frac{211 Q}{m_2 c_p (T_2 - T_0)} = \frac{211 Q}{P \times 60}$$

where, P = power in kW, Q = refrigeration effect in TR.

EXAMPLE 3.15 A regenerative air cooling system is used in an aircraft to take 20 TR load. The ambient air at pressure 0.8 bar and temperature 10°C is rammed isentropically to a pressure of 1.2 bar. The air is further compressed in the main compressor to 4.75 bar and is cooled by the ram air in the heat exchanger whose effectiveness is 80%. The air from the heat exchanger is further cooled to 60°C in the regenerative heat exchanger with a portion of the air bled after expansion in the cooling turbine. The cabin is to be maintained at a temperature of 25°C and a pressure of 1 bar. If the isentropic efficiency for the compressor as well as that for the turbine is 90%, and the temperature of air escaping to atmosphere from the regenerative heat exchanger is 50°C , find the following:

- Mass of the air bled from cooling turbine to be used for regenerative cooling.
- Power required for maintaining the cabin at the required condition.
- COP of the system.

Solution:

Given: $Q = 20 \text{ TR}$; $p_1 = 0.8 \text{ bar}$ $T_1 = 10^\circ\text{C} = 10 + 273 = 283 \text{ K}$
 $p_2 = 1.2 \text{ bar}$ $p_3 = p_4 = p_5 = 4.75 \text{ bar}$
 effectiveness of HE = 0.8 $T_3 = 60^\circ\text{C} = 60 + 273 = 333 \text{ K}$
 $T_2 = 25^\circ\text{C} = 25 + 273 = 298 \text{ K}$ $p_7 = p_8 = p_0 = 1 \text{ bar}$
 $\eta_c = \eta_T = 90\%$ $T_8 = 100^\circ\text{C} = 100 + 273 = 373 \text{ K}$

The T - s diagram for the regenerative air-cooling system is shown in Figure 3.40. For the isentropic ramming of air (Process 1-2),

$$T_2 = T_1 \left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} = 283 \left(\frac{1.2}{0.8} \right)^{0.4-1/1.4} = 317.8 \text{ K}$$

Main compressor: Considering isentropic compression,

$$T_3 = T_2 \left(\frac{p_3}{p_2} \right)^{\frac{\gamma-1}{\gamma}} = 317.8 \left(\frac{4.75}{1.2} \right)^{0.4-1/1.4} = 471 \text{ K}$$

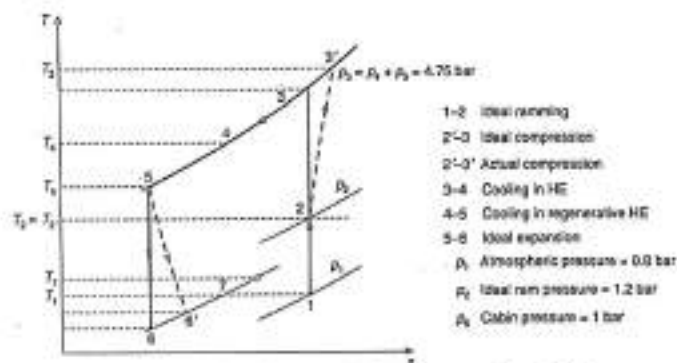


Figure 3.40 T - s diagram for regenerative air cooling system—Example 3.15.

Isentropic efficiency of the compressor,

$$\eta_c = \frac{\text{Isentropic increase in temperature}}{\text{Actual increase in temperature}} = \frac{T_3 - T_2}{T_3' - T_2}$$

$$\therefore 0.9 = \frac{471 - 317.8}{T_3' - 317.8} = \frac{153.2}{T_3' - 317.8}$$

$$\therefore T_3' = 488 \text{ K}$$

Effectiveness of the heat exchanger (ϵ_0),

$$0.8 = \frac{T_3' - T_4}{T_3' - T_2} = \frac{488 - T_4}{488 - 317.8} = \frac{488 - T_4}{170.25}$$

$$\therefore T_4 = 351.8 \text{ K}$$

Isentropic expansion in the cooling turbine (process 5-6),

$$\frac{T_5}{T_4} = \left(\frac{p_5}{p_4} \right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{4.75}{1.0} \right)^{0.4-1/1.4} = 1.561$$

$$\therefore T_5 = \frac{T_4}{1.561} = \frac{333}{1.561} = 213.3 \text{ K}$$

Isentropic efficiency of the cooling turbine,

$$\eta_T = \frac{\text{Actual decrease in temperature}}{\text{Isentropic decrease in temperature}} = \frac{T_5 - T_6}{T_5 - T_4}$$

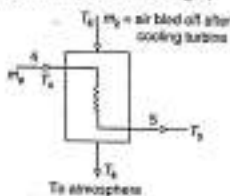
$$0.8 = \frac{333 - T'_a}{333 - 213} = \frac{333 - T'_a}{119.7}$$

$$T'_a = 225.2 \text{ K}$$

(a) Mass of air bled from the cooling turbine to be used for regenerative cooling:

Let m_2 = mass of air bled from the cooling turbine to be used for regenerative cooling,
 m_a = total mass of air bled from the main compressor, and
 m_1 = mass of cold air supplied to the cabin.

For the energy balance of regenerative heat exchanger,



$$m_2 c_p (T_3 - T_4) = m_1 c_p (T_2 - T_1); \quad \therefore m_2 = \frac{m_1 (T_2 - T_1)}{(T_3 - T_4)}$$

$$m_2 = \frac{m_a (351.3 - 333)}{(323 - 225.2)} = 0.1923 m_a \quad (i)$$

We know that the mass of air supplied to the cabin,

$$m_a = m_1 + m_2$$

$$= \frac{211 Q}{c_p (T_1 - T'_a)} = \frac{211 \times 20}{1 (298 - 225.2)} = 58.0 \text{ kg/min} \quad (ii)$$

From Eq. (i), we find that

$$m_a - m_1 = 58.0 \text{ or } m_a - 0.1923 m_a = 58.0$$

$$\therefore m_a = \frac{58.0}{1 - 0.1923} = 71.8 \text{ kg/min}$$

$$\text{and } m_2 = 0.1923 m_a = 0.1923 \times 71.8 = 13.8 \text{ kg/min} \quad \text{Ans.}$$

(b) Power required for the compressor:

Power required for maintaining the cabin at the required condition is given as:

$$P = \frac{m_a c_p (T_2 - T_1)}{60} = \frac{71.8 \times 1 (488 - 317.8)}{60} = 203.6 \text{ kW} \quad \text{Ans.}$$

(c) COP of the system:

$$\text{COP of the system} = \frac{211 Q}{m_a c_p (T_2 - T_1)} = \frac{211 \times 20}{71.8 \times 1 (488 - 317.8)} = 0.345 \quad \text{Ans.}$$

3.8 COMPARISON OF VARIOUS AIR COOLING SYSTEMS USED FOR AIRCRAFT

The performance curves for the various air cooling systems used for aircraft are shown in Figure 3.41. These curves are plotted for various Mach numbers vs temperature of air at the outlet of cooling turbine.

- One can observe that the simple air cooling system gives maximum cooling effect on the ground surface and decreases as the speed of the aircraft increases. It would be useful for the Mach numbers ranging between 0.4 and 1.5. This simple system can be employed with the evaporative system at high speeds.
- The boot-strap system is better at low Mach number (1.4 and above) since it needs ram air for cooling in heat exchangers.
- Boot-strap system is modified using evaporative system in addition to heat exchangers. At high Mach numbers, the temperature of ram air rises due to high speeds. Such high temperature ram air would be less effective in cooling the compressed air of the main compressor in the primary heat exchanger to the desired low temperature.
- Regenerative air cooling systems are useful for low Mach numbers as can be seen from the graph.
- Reduced ambient air cooling system is suitable to render low temperatures at high Mach numbers, hence it is only suitable for supersonic aircraft.

The turbine discharge temperature of the air is variable. Therefore, in order to maintain the constant temperature of supply air to the cabin, it requires some control system.

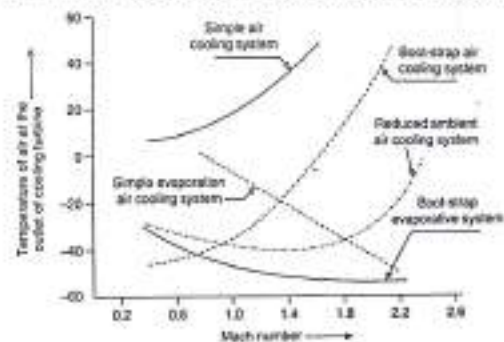


Figure 3.41 Performance curves of various air cooling systems.

EXERCISES

1. The COP of the heat pump is greater by unity compared to the reversed Carnot refrigerator system. Explain.
2. Discuss the working of a Carnot refrigerator with working substance as air as well as vapour. Derive an expression for its COP.
3. Differentiate between an engine, a refrigerator and a heat pump.
4. Define the term coefficient of performance as applied to refrigerators and heat pumps.
5. Discuss the Bell-Colemann air refrigeration cycle with the help of schematic, $p-v$ and $T-s$ diagrams. Deduce an expression for its COP. For what purpose is this cycle used in aircraft?
6. What are the advantages and disadvantages of Bell-Colemann air cycle?
7. Discuss the closed Bell-Colemann air cycle. Why are these solutions used in these cycles?
8. State and explain the various air refrigeration cycles used for aircraft.
9. What are the advantages or benefits of using the air cycles for aircraft cooling?
10. Draw a neat diagram and explain the working of the simple air cooling cycle for aircraft. Also represent the cycle on $T-s$ diagram and explain the various processes involved. Explain the procedure of obtaining the COP of the cycle.
11. Explain the working of boot-strap air refrigeration system.
12. Describe the working of the simple air evaporative cycle with the help of schematic and $T-s$ diagrams.
13. Differentiate and enumerate the advantages of the boot-strap air evaporative system from the boot-strap air cooling system.
14. When can one prefer to have the reduced ambient air cooling system in aircraft? Explain its working with the help of a neat sketch.
15. Explain the working of the regenerative air cooling system with the help of schematic and $T-s$ diagrams. Explain the various processes involved and write the expression for its COP.
16. Write short notes on the following:
 - Limitations of Carnot cycle with gas as refrigerant.
 - Refrigeration effect and its units.
 - Aircraft air conditioning.
17. Compare the different types of air cooling systems in terms of aircraft speed.

NUMERICALS

1. In a Bell-Colemann refrigeration plant, the air is drawn from cold chamber at 1 bar and 12°C , and compressed to 5 bar. The same is cooled to 25°C in the cooler before expanding in the expansion cylinder to cold chamber pressure of 1 bar.

- (a) Determine the theoretical COP of the plant and the theoretical net refrigeration effect/kg of air. The compression and expansion be assumed isentropic. Assume $\gamma = 1.41$ and $c_p = 1.009 \text{ kJ/kg}\cdot\text{K}$.
- (b) If the compression and expansion laws followed are $pv^{1.31} = C$ and $pv^{1.3} = C$ respectively, how will the result be modified?

2. A refrigeration unit working on Bell-Colemann cycle takes air from cold chamber at -10°C and compresses it from 1 bar with index of compression being 1.2. The compressed air is cooled to a temperature 10°C above the ambient temperature of 25°C before it is expanded in the expander where the index of expansion is 1.35.

Determine the following:

- (i) COP.
 - (ii) Quantity of air circulated per minute for the production of 2000 kg ice per day at 0°C from water at 20°C .
 - (iii) Capacity of the plant in ton-refrigeration.
- Assume $c_p = 1 \text{ kJ/kg}\cdot\text{K}$ for air.

3. An air refrigeration plant 25 TR capacity comprises a centrifugal compressor, a cooler heat exchanger and an air turbine. The compressor is coupled directly to the air turbine. The compressor also receives power from another prime mover. The processes in the compressor and the turbines are adiabatic but not isentropic. Air at temperature 21°C and 0.85 bar enters the compressor. It leaves the compressor at 90°C . The same air enters the turbine at 38°C and 1.5 bar. The turbine exit is at 0°C . Assuming no pressure drop in the cooler and the refrigerator (evaporator) section, and constant specific heats as $c_p = 1.006 \text{ kJ/kg}\cdot\text{K}$ and $c_v = 0.712 \text{ kJ/kg}\cdot\text{K}$, determine the following:

- (i) The compressor efficiency.
- (ii) The turbine efficiency.
- (iii) The flow rate of air.
- (iv) The power input to the plant.
- (v) The coefficient of performance.

4. A high altitude flight aircraft is flying at an altitude of 1.5 km with a speed of 1.2 mach. The ambient atmospheric pressure and temperature are 0.2 bar and -40°C . The cabin is pressurized to 0.7 bar and has to be maintained at 25°C . The main compressor pressure ratio is 5 and the air enters the cooling turbine at 40°C . The exit from the cooling turbine is at 0.75 bar. The cockpit cooling load is 10 tonnes. Assume internal efficiency of compressor as 85% and that of cooling turbine as 75%. Ram efficiency is 90%. Assume $\gamma = 1.4$ and $c_p = 1 \text{ kJ/kg}\cdot\text{K}$. Determine the following:

- (i) Stagnation temperature and pressure of air entering and leaving the main compressor.
- (ii) Mass flow rate of air.
- (iii) Ram air heat exchanger effectiveness.
- (iv) Volume handled by compressor and cooling turbine.
- (v) Net power delivered by engine to the refrigerating unit (Pressurisation + Refrigeration).
- (vi) COP of the system based on compressor work.
- (vii) Additional power only for refrigeration.

5. A boot-strap air refrigeration system is used for an airplane to take 20 TR of cooling load. The ambient conditions are 5°C and 0.85 bar. The air pressure increases to 1.1 bar due to ramming action which is considered ideal (isentropic). The pressure of air bled off the main compressor is 3.5 bar and this air is further compressed in the secondary compressor to 4.8 bar. The isentropic or internal efficiency of the main compressor as well as that of the secondary compressor is 90% and that of the turbine is 80%. Heat exchanger effectiveness of the primary heat exchanger is 0.6 and that of secondary heat exchanger is 0.6. Assuming $c_p = 1.0 \text{ kJ/kg}\cdot\text{K}$, determine the following:

(a) The power required to take the cabin load and (b) the COP of the system.

The cabin may be maintained at 1 bar and 25°C . The cooling turbine runs the secondary compressor and uses its surplus power to run a fan for blowing in the ram air to waste.

6. A regenerative air refrigeration system of an aircraft with flight speed of 1500 km/h has 30 TR cooling load while the ambient conditions are 0.1 bar and -63°C . The ram efficiency is 90%. The pressure ratio of the main compressor is 5 with internal efficiency of 0.9. The air bled off the main compressor is cooled by ram air in a heat exchanger which is 60% effective. The air from the heat exchanger passes on to the cooling air turbine whose internal efficiency is 0.8. Some portion of air from the cooling turbine is led to the regenerative heat exchanger reducing the temperature to 30°C of the bled off compressed air. The cooling air gets heated to 92°C before discharging to atmosphere. The cabin is pressurized to 0.8 bar and maintained at 25°C . Determine the following:

(a) The percentage of air extracted for regenerative cooling.

(b) Power required to maintain the cabin at the required condition.

(c) COP.

Assume the cooling turbine power to be used for air exhaust fan.

7. The following data refer to the aircraft refrigeration system with evaporative cooling.

Ram air pressure = 1.04 bar; ram air temperature = 25°C ; compressor delivery pressure = 4 bar; pressure drop in ram air heat exchanger = 0.2 bar; ram air exchanger effectiveness = 0.8; cabin pressure = 1 bar; mass flow rate of air through cabin bleed from main compressor = 30 kg/min; temperature of air leaving the cabin = 25°C and pressure = 1 bar; evaporative cooling effect = 30 kJ/kg of air flow. Determine the following:

(a) Refrigeration capacity, (b) Power for refrigeration, and (c) COP.

Chapter 4

Simple Vapour Compression Refrigeration Systems

4.1 INTRODUCTION

The vapour compression system is the basis of operation for many of the refrigeration units. Therefore, one must understand the operation of the compression cycle accurately so as to be able to diagnose (identify) faults. Before studying the development of vapour compression cycle, let us review the advantages and disadvantages (limitations) of air refrigeration.

4.2 ADVANTAGES AND LIMITATIONS OF AIR REFRIGERATION

The advantages and limitations of air refrigeration are given in the following subsections:

4.2.1 Advantages of Air Refrigeration

- Air refrigeration systems use air as refrigerant which is available abundantly free of cost.
- Air is an inert gas. It is inflammable, non-toxic and hence safe as a refrigerant.
- Air refrigeration systems, which work on Bell-Coleman cycle, are suitable for aircraft on account of availability of ram air.

4.2.2 Limitations of Air Refrigeration

Various air refrigeration systems employed for cooling the cabins of aircraft were discussed in Chapter 1. In general, there are some limitations of using air as a refrigerant, which are as follows:

- The heat carried away by air in air refrigeration systems is $\dot{m}_a \times c_p (\bar{T}_{\text{out}} - T_a)$. It depends on the mass flow rate of air (\dot{m}_a), specific heat c_p and temperature difference. The value of c_p is 1.00 kJ/kg which is very low. Therefore, for a fixed value of

6

Other Refrigeration Cycles

The vapour compression refrigeration cycle is a firmly established system for refrigeration and air-conditioning applications all over the world. This is because of its high COP and moderate cost. Similarly, the vapour absorption cycles are widely used when a cheap source of energy like waste steam is readily available. There are also other cycles such as (a) thermoelectric system and (b) vortex tube, which are used for specific applications despite the fact that their COPs are not as high as the vapour compression or vapour absorption systems. The steam jet water refrigeration system was very popular in the early days of refrigeration which, however, has been superseded by the newer and more efficient systems.

6.1 THERMOELECTRIC REFRIGERATION

Thermoelectric refrigeration is based on the discovery of Seebeck who demonstrated in 1821 that in a circuit of two conductors made of different metals, a current flowed when the two junctions were held at different temperatures. Following this discovery, Peltier demonstrated in 1934 that when an electric current passes around a circuit composed of two different metals, one junction becomes cool and the other becomes warm (Figure 6.1).

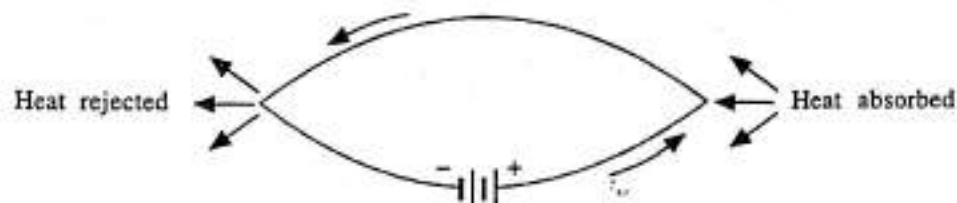


FIGURE 6.1 Peltier effect.

With pure metals the Peltier effect is comparatively small; however, with the advent of semiconductors in the 1950s, it became possible in Peltier junctions (Figure 6.1) to produce an effect much larger than that observed with pure metals. Semiconductors are solid crystalline substances, for example germanium or silicon, that have electrical conductivity greater than insulators but less than good conductors. Compounds of the metals lead and bismuth with semimetals tellurium and selenium are the most common semiconductors used for cooling, for example, bismuth telluride. The lowest temperature obtained by this method is -145°C .

Thermoelectric refrigerator

The coefficient of performance of a thermoelectric refrigerator is much less than that of a conventional vapour compression system when the required cooling capacity is high. Figure 6.2 shows a simple thermoelectric refrigerator. When current flows through the circuit, the cold

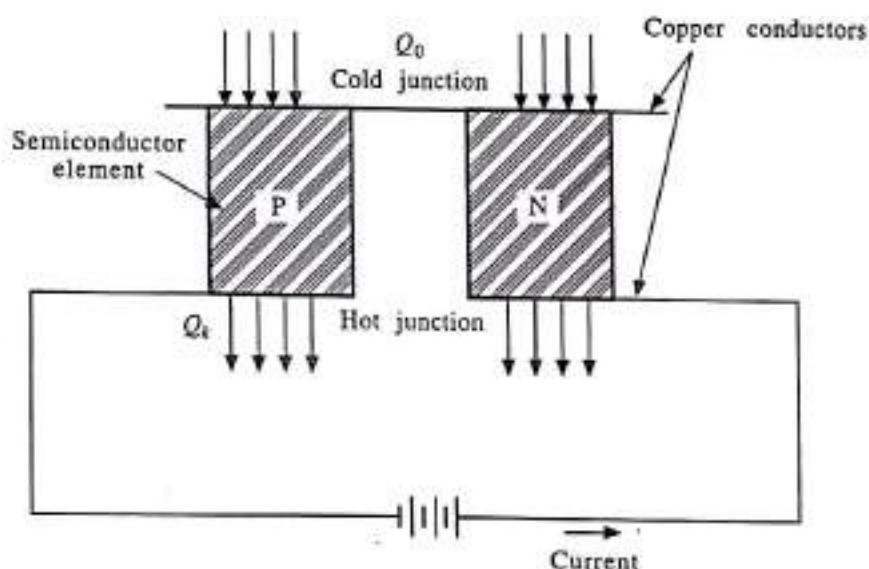
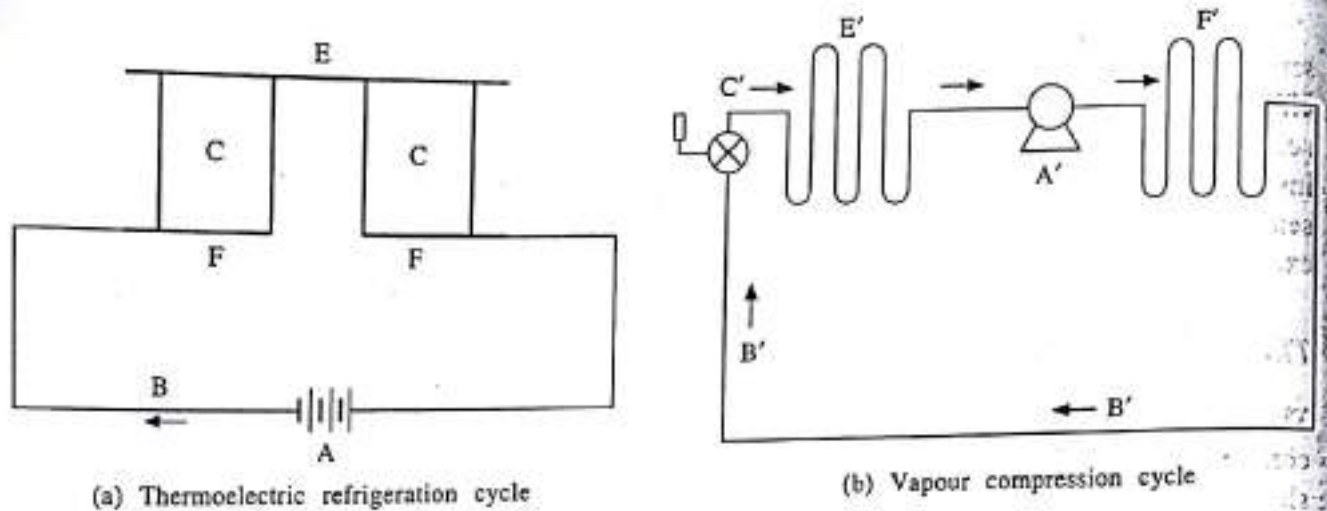


FIGURE 6.2 A simple thermoelectric refrigerator.

junction becomes cold thus serving as a cooler, while the hot junction becomes hot and acts like a condenser in a vapour compression system. The components of this cycle are comparable to those of the vapour compression cycle as shown in Figure 6.3. However, whereas the COP of the conventional unit falls off rapidly as the cooling capacity is decreased, that of the thermoelectric cycle remains almost constant, as shown in Figure 6.4. The thermoelectric method is preferred for low cooling rates. Apart from the low COP, the cost of the thermoelectric units makes them unsuitable for applications where a large cooling capacity must be provided unless other considerations are more important than the economic ones. There are certain advantages offered by the thermoelectric refrigerator, making it suitable for some applications in spite of its obvious handicaps. The cooling units are of compact size, silent, have no moving parts, and contain no liquid or gases. They have a very long life as well. Moreover, it is very simple to control the rate of cooling by adjustment of the current, the response to changes in the supply being very rapid. To obtain larger cooling capacities, many couples may be combined into modules. By reversing the direction of current, a refrigeration unit is transformed into a heat pump.



Thermoelectric cycle	Vapour compression
Electron flow (B)	Refrigerant flow (B')
Cold junction (E)	Evaporator (E')
Battery (A)	Compressor (A')
Hot junction (F)	Condenser (F')
Semiconductor (C)	Expansion valve (C')

FIGURE 6.3 Comparison of the components of thermoelectric and vapour compression cycles.

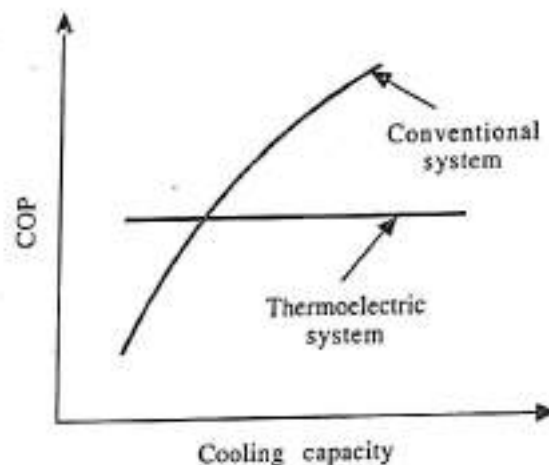


FIGURE 6.4 Variation of COP with cooling capacity.

Currently, the thermoelectric refrigerators are used for cooling instruments, medicines and other electronic devices which generate heat. If at certain times, heating instead of cooling is required to maintain a constant temperature, this heating can be accomplished by reversing the flow of current. Both refrigerators and air conditioners cooled with thermoelectric modules have been built for nuclear submarines, to enable submarines to avoid detection by staying underwater for extended periods.

Much experimental work has been done to find materials that would give the best thermoelectric results. Compounds of the metals lead and bismuth with the semimetals tellurium

and selenium have given the best results. The most common semiconductor used today for cooling purposes is bismuth telluride, Bi_2Te_3 . To obtain a larger cooling capacity than that produced by a single couple, many couples are combined in modules.

Figure 6.5 shows the schematic diagram of a practical application of a thermoelectric refrigeration cycle.

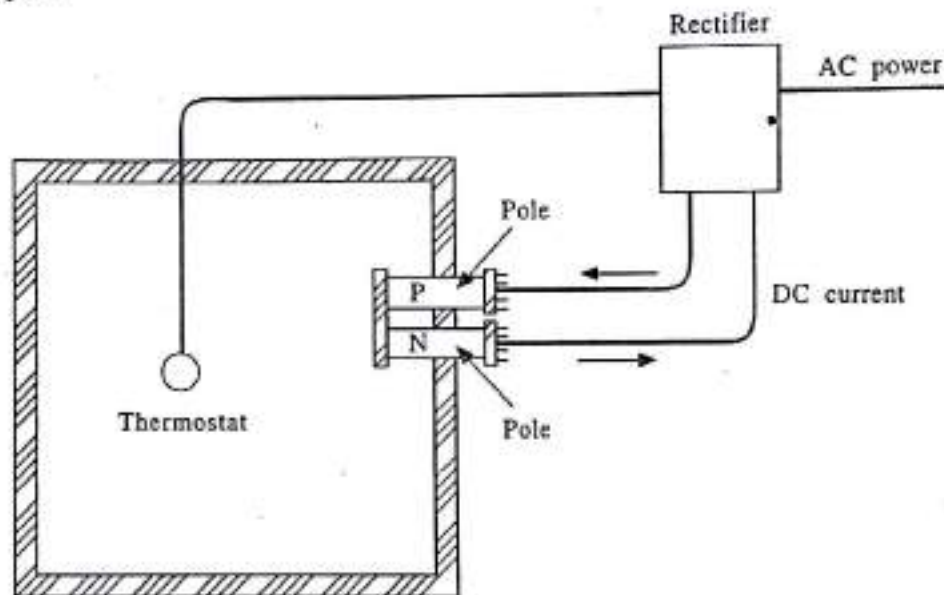


FIGURE 6.5 Schematic diagram of a thermoelectric couple used in a refrigerator.

Thermodynamic analysis of thermoelectric refrigeration system

In the cooling element shown in Figure 6.2, the Peltier heat absorbed at the cold end and the Peltier heat rejected at the hot end are given by

$$Q_c = \pi_c I \quad (6.1)$$

$$Q_h = \pi_h I \quad (6.2)$$

where π_c and π_h are the Peltier coefficients which are functions of temperatures T_c and T_h of the cold and hot ends respectively. It may be seen that if T_h is maintained at ambient temperature, T_c will be lower than the ambient temperature. The direction of the flow of current will decide as to which of the junctions or ends will become cold or hot.

The energy balance of the system is

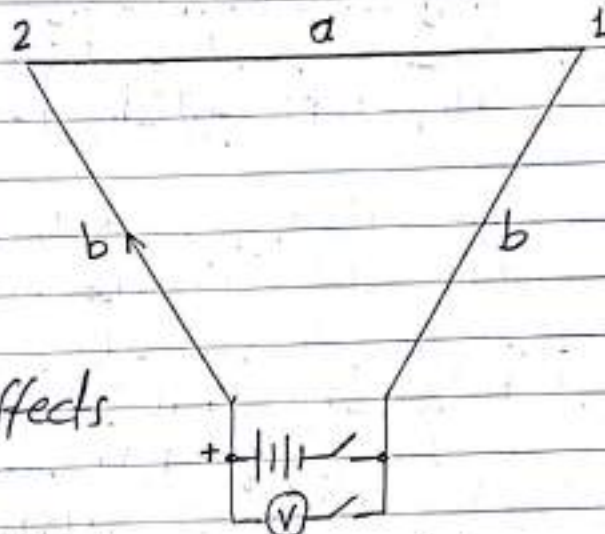
$$Q_h - Q_c = EI \quad (6.3)$$

where, I is the current, E is the emf supplied, Q_c and Q_h are the actual amounts of heat flows at the cold and hot ends respectively.

Thermoelectric phenomena

When two dissimilar metals or semiconductors are connected and the junctions held at different temperatures, five phenomena take place simultaneously:

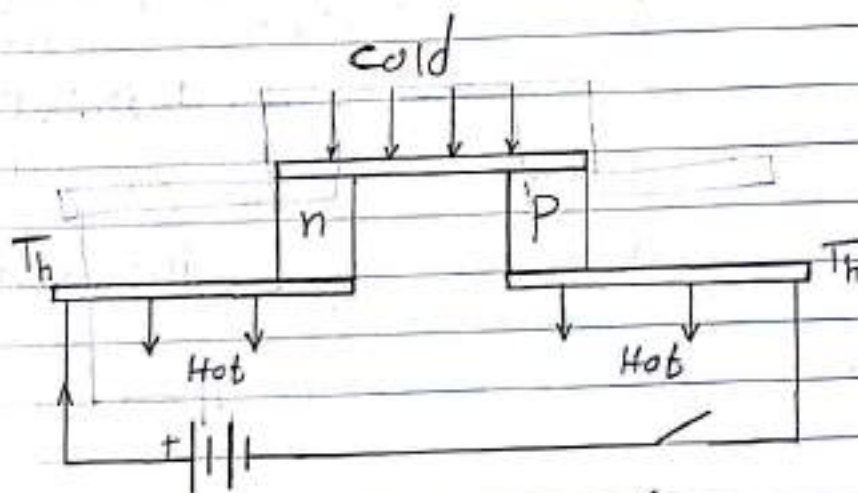
Thermoelectric Refrigeration



Seebeck & Peltier effects

لقد استغل تأثير سيبيك في صنع المبردات الحرارية متدخلة تحويل ولكن تأثير بلتيير لم يلعب كثيراً من الاهتمام الا مؤخراً (المستحدثات من الفهم الجاهلي) وبالمخصوص بعد اكتشاف أشباه الموصلات (Semiconductors)

هناك نوعان من أشباه الموصلات الأول يسمى نوع n وله معادل سيبيك سالبة ويمثل خاضعاً من الإلكترونات ولثاني نوع p وله معادل سيبيك موجب وهو مشحون بالإلكترونات وتجمع المواد كبيرة خضراً لصنع ما يسمى وحدة كهروحرارية ويوضع بعضها مادة عازلة.



وحدة كهروحرارية

هناك أربع تقاطعات ولكن هناك قطبان كهربائيان فقط القطب العلوي هو بارد في هذا الزئبق والقطب السفلي الساخن وتوضع هبات من هذا النوع (أي أزواج أقطاب) مع بعضها وتربط على التوالي لتشكل صففاً واحداً ثم يتم تجميع صفوف متعددة إلى جانب بعضها البعض لتشكل صففاً متعدداً على ارتفاع الحرارة في جانبها البارد وفيه بذلك نعمل التلبيح بينا يقوم الجانب الآخر بطرد الحرارة.

إن معامل أداء التلاهيبة الكهربائية صغير جداً مقارنة مع حورة التلبيح الانضغاطية فاعلة عند ازدياد الفرق بين درجات الحرارة الجانبين البارد والساخن. كما أن كلفة استهلاك الرصولات وصناعة وحدة كهربائية مرتفعة جداً. مناسب للاستخدام في مركبات الفضاء وفي تبريد بعض المعدات الإلكترونية مما يستدعي في هافنات الرصوات التي يولد قبل ازديادهم وفي تطبيقات محدودة أخرى تتطلب قدوم حركة الأجزاء ونظافة التلاهيبة في المواد الخارجية وهناك بعض التطبيقات العسكرية لتبريد معدات الإلكترونية بدون وجود ضوضاء أجزاء متحركة.

إن إحدى الخصائص التي تحكم من عمل الوحدات الكهربائية حاسية وتسمى الاستحقاق (figure of merit) الذي يشار إليه بالرمز Z

$$Z = \frac{\alpha^2}{(\sqrt{k_p p} + \sqrt{k_n n})^2}$$

α : الفرق المعلوم بين معاملي سبيل للمادتين p و n
 k : معامل انتقال الحرارة
 m : المعادلة الكهربائية
 p, n : الحاريتين

إن معامل الأداء للتلاهيبة الكهربائية يتحكم مع ارتفاع قيمة Z

- Seebeck effect
- Joule effect
- Fourier effect
- Peltier effect
- Thomson effect

Seebeck effect: If ΔE and ΔT are the emf output and the temperature difference of the junctions, then

$$\Delta E \propto \Delta T \quad (6.4)$$

The proportionality constant is denoted by

$$\text{Seebeck coefficient, } \alpha_{ab} = \frac{\Delta E}{\Delta T} \quad (6.5)$$

where

$$\alpha_{ab} = \alpha_a - \alpha_b \quad (6.6)$$

and α_a , α_b are the coefficients for metals A and B or P and N.

Peltier effect: If direct current is passed through a pair of dissimilar metals, there is heating at one junction, cooling at the other depending upon the material combinations. Peltier varied the current I and observed the heating and cooling rate Q for different sets of elements. It is found that:

$$Q \propto I \quad (6.7)$$

i.e.

$$Q = \pi_{ab} I \quad (6.8)$$

where

$$\pi_{ab} = \pi_a - \pi_b \quad (6.9)$$

Thomson effect: When a current passes through a single conductor having a temperature gradient, heat transfer is given by:

$$\frac{dQ}{dx} = \tau I \left(\frac{dT}{dx} \right) \quad (6.10)$$

where τ is the Thomson coefficient (volt/K) and dQ/dx is the Thomson heat transfer per unit length. Using the first and second laws of thermodynamics, Zemansky obtained the relation between Seebeck and Peltier coefficients as

$$\pi_{ab} = \alpha_{ab} T \quad (6.11)$$

or

$$\frac{\pi_a - \pi_b}{T} = \frac{d\alpha_{ab}}{dT} \quad (6.12)$$

Substituting Eq. (6.11) into Eq. (6.8), we find that

$$Q = \alpha_{ab} IT \quad (6.13)$$

Joule effect: When electrical current I flows through a conductor of resistance R , there is dissipation of electrical energy. This is the well known joule effect. The energy dissipated is given by

$$q_I = I^2 R \quad (6.14)$$

Fourier (conduction) effect: If the ends of any element are maintained at different temperatures, the heat transfer from the hot end to the cold end is related by

$$Q_{\text{cond}} = U(T_h - T_c) \quad (6.15)$$

where

$$U = \frac{kA}{L} \quad (6.16)$$

The cooling and heating effects due to thermoelectric effect are given by

$$Q_c = \alpha_{ab} IT_c \quad (6.17)$$

$$Q_h = \alpha_{ab} IT_h \quad (6.18)$$

6.2 VORTEX TUBE REFRIGERATION

G. Ranque, a French engineer discovered this phenomenon in 1931 which is called the "vortex effect" and the device is known as *vortex tube* or *Ranque tube*. Subsequently, R. Hilsch carried out extensive investigations on vortex tubes and recommended optimum dimensions and measured the performance. Since then, sporadic research has been carried out, mainly in the USSR, in order to improve its efficiency. Though the general principle of operation is fairly clear, the quantitative theory of its performance has not yet been established.

The vortex tube is an instrument capable of converting an ordinary supply of compressed air (7–10 bar) into two streams, one hot and one cold. This device is essentially a straight length of cylindrical tube into which compressed air is admitted tangentially at the outer radius, creating cyclone spinning at very high rpm. While spiralling inwards (Figure 6.6) and so

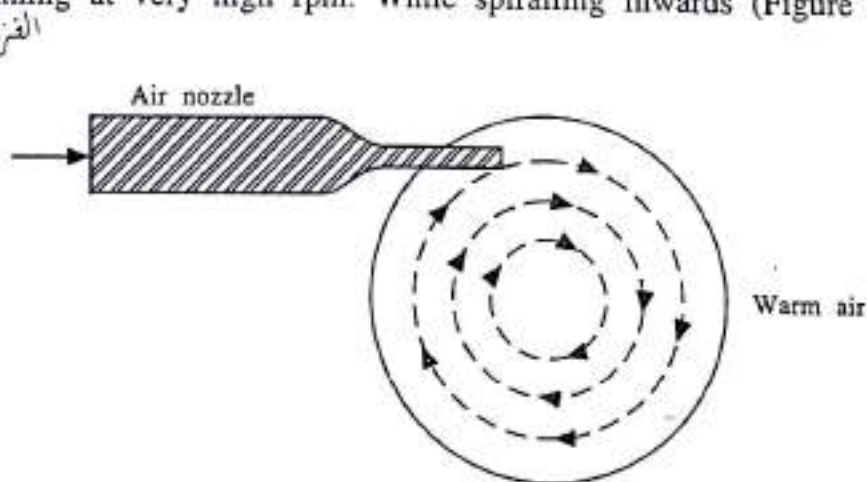


FIGURE 6.6 Cyclone spinning of air admitted into the vortex tube.

throttled that the central core of the resulting air stream can be separated from the peripheral flow. With this device, cold air temperatures 50°C below the nozzle temperature are readily obtainable with moderate inlet pressures. There are two versions of the tube—counterflow (Figure 6.7) or uniflow (Figure 6.8). In the counterflow version, hot air and cold air come out through the opposite ends of the tube. In the uniflow version, the cold stream comes out of the same end as the warm stream, as a central core of air which is separated by a special arrangement of orifice and valve. The uniflow tube is less efficient compared to the counterflow type.

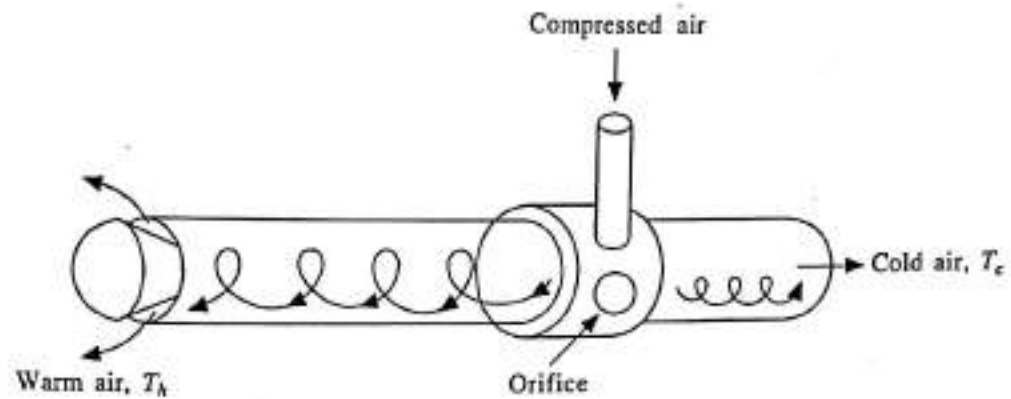


FIGURE 6.7 Counterflow type vortex tube.

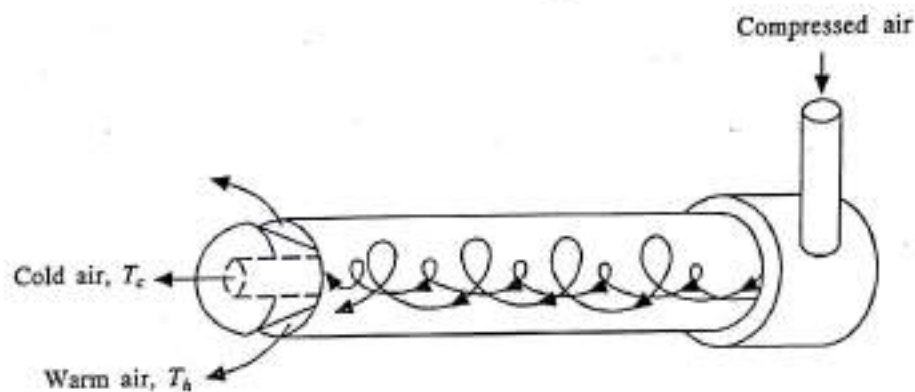


FIGURE 6.8 Uniflow type vortex tube.

The counterflow type of vortex tube at one end is partially closed by a diaphragm with a central orifice approximately half the tube diameter. At the other end of the tube a valve restricts the exit. When the valve is partly open, a stream of cold air leaves through the orifice and a stream of warm air leaves through the valve. The temperature of the cold stream and its flow rate depend on the setting of the valve. When the valve is fully closed, all the air comes out through the orifice and no reduction in temperature is found. As the valve is opened, the flow rate of the cold stream rises and its temperature falls to a minimum value when the flow rate is about one-quarter or one-third of the total.

The geometry of the vortex tube (Figure 6.9) can be described by the diameter of the vortex tube D , the diameter of the orifice d , the length of the vortex tube L , the geometry and the

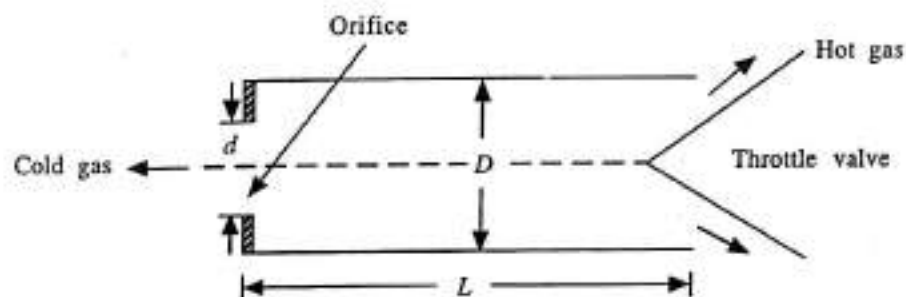


FIGURE 6.9 The geometry of vortex tube.

number of nozzles, and the design of the valve. The setting of the valve determines the ratio of the hot and cold mass flow fractions respectively. The optimum diameter of the orifice is found to be half the tube diameter, i.e.

$$d_{\text{optimum}} = \frac{D}{2}$$

Figure 6.10 shows a practical setup of a vortex tube application, where the compressor supplies compressed air to storage tank. The compressed air enters through the nozzle into the vortex tube, where the air is split into two streams. The cold air produced at E flows into the space F that is to be cooled. Warm air is exhausted through the tube D. The system has the advantage of great mechanical simplicity and instantaneous operation but its efficiency is very low.

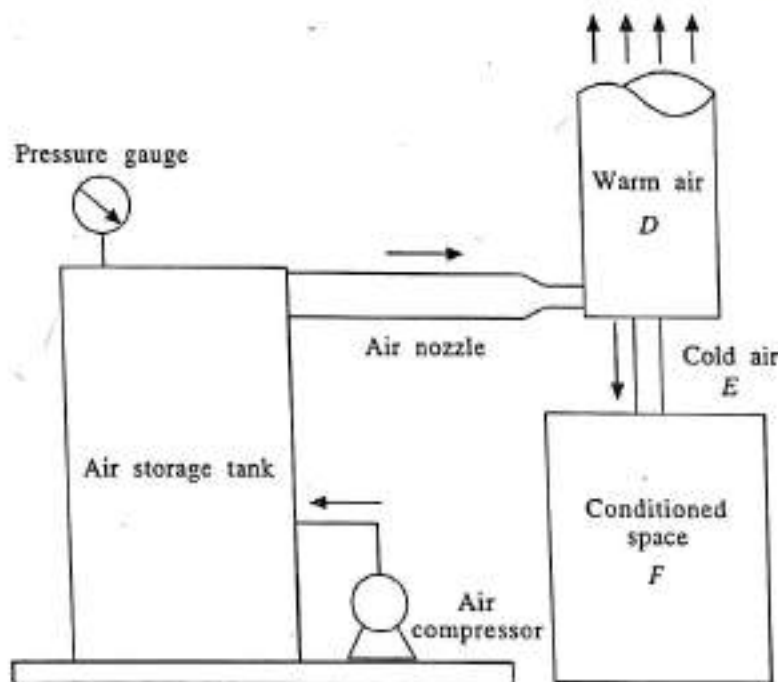


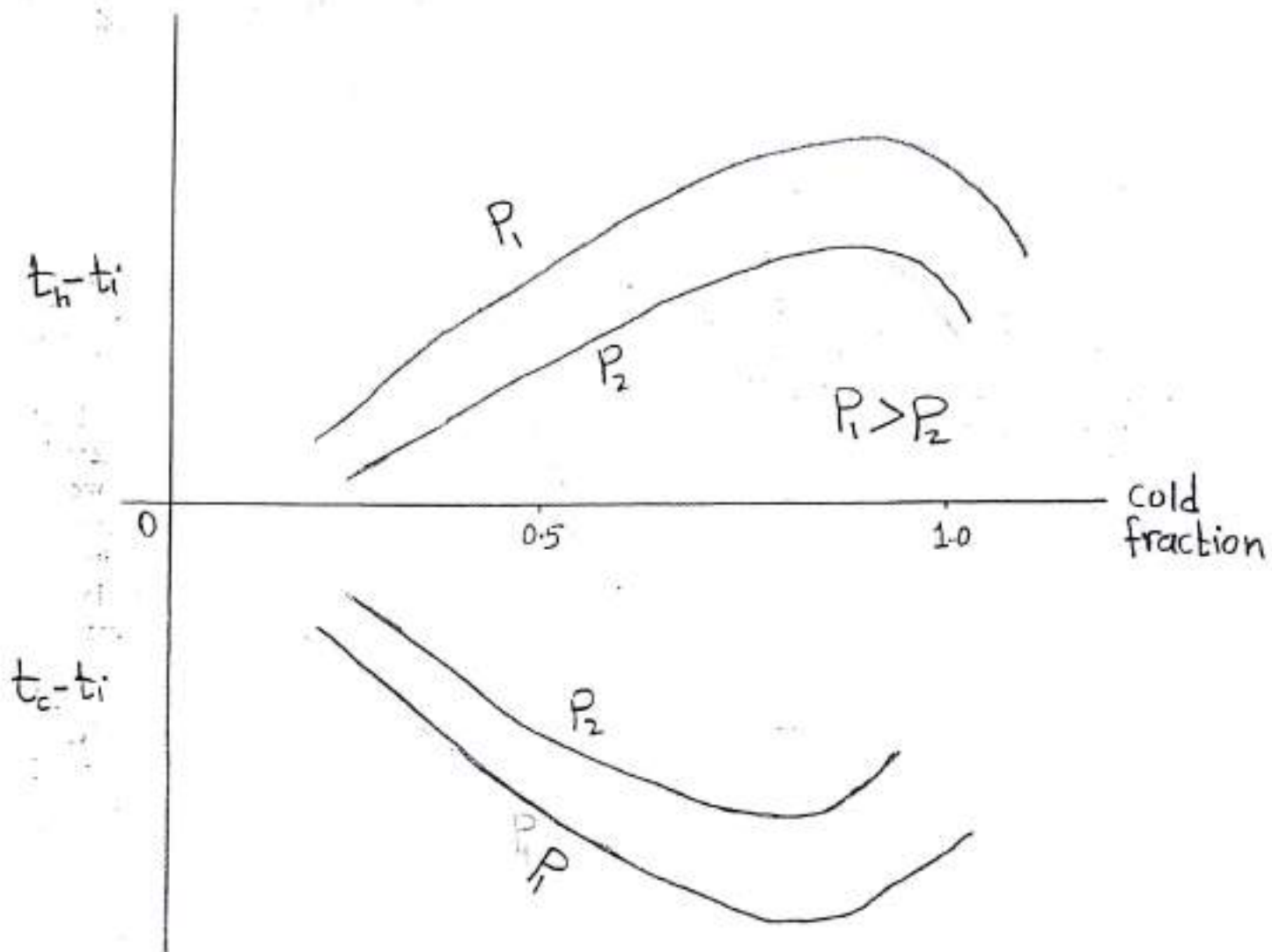
FIGURE 6.10 A practical vortex tube refrigeration system.

The use of vortex tube is convenient where a supply of compressed air is available, to obtain rapidly and momentarily a fall in temperature. Its applications are still relatively limited to laboratory trials, to control of temperature of small electronic boxes, to aviation or to obtain a rapid fall in temperature in certain manufactures, such as chocolate, or waxes, cooling machining operations, ^{الغرف الباردة} deep mine air conditioning, etc. The use of vortex cooling may often prove to be a better proposition than the more efficient conventional vapour compression systems when any of the following requirements apply:

- Light weight, portability
- Low capacity or localized cooling or freezing
- No moving parts—high reliability
- May be used where electricity is not available or prohibited on safety grounds
- Low initial cost
- Instantaneous operation
- No expert attendant required

مميزات
عيوب

لدينا سرعة
مظور



تأثير الضغط الابتدائي ونسبة الهواء البارد على درجة حرارة التيارات الباردة والحارة

في اوطار دقيقة حرارة ممكنة نظرياً مع. جهاز تدر كذا حي غير الوكاه التدر عكوسياً اربابانياً
من الحالة الابتدائية الى الحالة النهائية. ومنه نطبق معادلة الغاز المثالي التالية

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}}$$

The main disadvantage of the vortex tube is its low efficiency, i.e. COP ranging from 0.1–0.2. It is also uneconomical for large systems.

On comparing with the Joule-Thomson refrigeration system (Chapter 9), it may be observed that the vortex tube system is a modification of the same cycle with the expander having been replaced by a vortex tube. In the Joule cycle, where the work of expansion is utilized, a temperature drop is obtained equal to the isentropic temperature drop (Figure 6.11), i.e. $\Delta T_s = T_3 - T_4$, where

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4} \right)^{(k-1)/k} \quad (6.19)$$

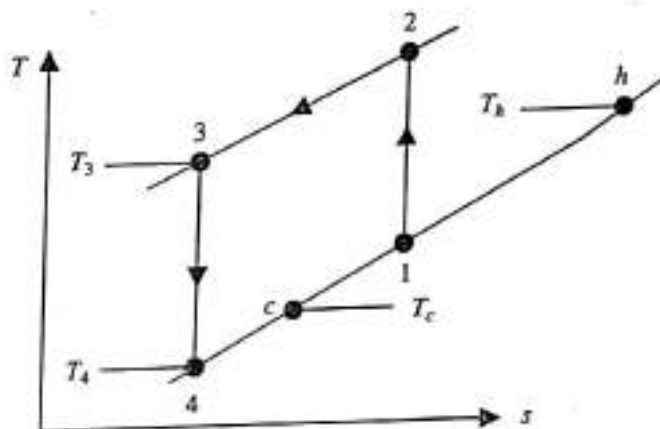


FIGURE 6.11 Temperature in vortex tube.

The temperature drop obtained with the vortex tube is smaller, about one-half that of isentropic drop. The air expands to a temperature T_4 and velocity c which is given by

$$\frac{c^2}{2} = c_p (T_3 - T_4) \quad (6.20)$$

If all this kinetic energy could be removed, cold air would be obtained at temperature T_4 .

From the nozzle, the high velocity gas travels from the periphery of the tube to the axis during which the separation of kinetic energy occurs. The kinetic energy is obtained by the outer layers due to which they are heated and emerge from the hot end of the tube at state h . The central core after having lost the kinetic energy emerges from the cold end at state c , i.e. at a temperature slightly above the static temperature of the expanded gas. The pressure of the cold gas stream is usually lowered further due to expansion in the vortex chamber.

The COP of the system is less than that of the Joule cycle as it can be seen that the temperature drop is less than the isentropic temperature drop. Also, the quantity of cold air is only a fraction of the air supplied. The energy balance of the vortex tube gives

$$T_3 = \mu_c T_c + \mu_h T_h \quad (6.21)$$

where, $\mu_c + \mu_h = 1$ and T_c and T_h are the stagnation temperatures of the two streams.

The energy efficiency η_e of the vortex tube can now be defined as the ratio of the actual cooling to ideal cooling, which is given by

$$\eta_e = \frac{\mu_c (T_3 - T_c)}{T_3 - T_4} \quad (6.22)$$

It is seen that if all the air goes through the throttle valve, there will be no energy separation. Similarly, if all the air leaves through the orifice, there will again be no energy separation. Thus, there will be optimum values of fractions η_c and η_h each for the maximum temperature drop, refrigerating effect, and COP.

6.3 STEAM JET WATER VAPOUR REFRIGERATING SYSTEM

Water was considered as a suitable refrigerating medium in the earlier times. However, water could not be introduced for refrigeration due to its inherent disadvantages as a refrigerant, i.e. the high evaporating vacuum required, the enormous volume of vapour to be compressed, and the relatively high freezing point. With the advent of air conditioning requiring relatively a high evaporator temperature, interest in this system was revived.

In a steam jet water vapour refrigerating system, refrigeration is obtained by direct evaporation and subsequent self-cooling of water. The principle of operation of this system is based on the fact that water under a high vacuum boils at a relatively low temperature which causes evaporation to occur and this reduces the temperature. Since water is the refrigerant used in this type of systems, only temperatures down to about 4°C are practicable and as such its use is limited primarily to air conditioning. If temperatures below 0°C are to be maintained in the evaporator, antifreeze may be added or brine may be used, however, such use is uncommon mainly because of the necessity of maintaining extremely high vacuum.

The use of this system is limited to installations where there is an abundance of steam and condensing water at a low cost and where the desired temperatures are in the range of 4°C to 10°C . Apart from air conditioning, these systems were particularly adapted to the processing of the cold water for such applications as rubber mills, distilleries, chemical plants, paper mills, food processing plants and aboard ships. It has another application in the cooling of lettuce and other leafy vegetables. The system, once very popular has largely been superseded by more efficient systems.

The advantages of this cycle in air conditioning comprise complete safety of water as a refrigerant, freedom from vibration, and ability to adjust quickly to load variations. In factories or plants, additional auxiliary equipment is not necessary. Moreover, water vapour is non-toxic, non-explosive and cheap. The inherent disadvantages of water as a refrigerant are the high evaporating vacuum required, the enormous volume of vapour to be compressed, and the comparatively high freezing point.

A typical steam-jet system is shown diagrammatically in Figure 6.12 and the corresponding temperature-entropy diagram is shown in Figure 6.13. High pressure steam is supplied to the nozzle inlet and is expanded to 2 in the ejector. Here water vapour, originating from the flash chamber at 6, is entrained with the high velocity steam jet and compressed through 4 into the diffuser exit at 5. The velocity of the mixture is then reduced, the kinetic energy is converted into an increase in static head, and the mass is discharged to the condenser. Both the motive vapour and the vapour drawn from the evaporator are condensed, and the condensate is then divided into two flows, one to feed the evaporator as make up water and the other to supply the boiler through a feed pump. The water in the flash chamber is cooled by the absorption from it of the latent heat of that portion of water flashed and removed by the ejector. The amount of water lost in the cooling process is very small and the remaining water is circulated from the chamber as a secondary cooling medium to the point of the load. Upon return it is usually sprayed into the flash chamber to provide exposure of the maximum surface area.

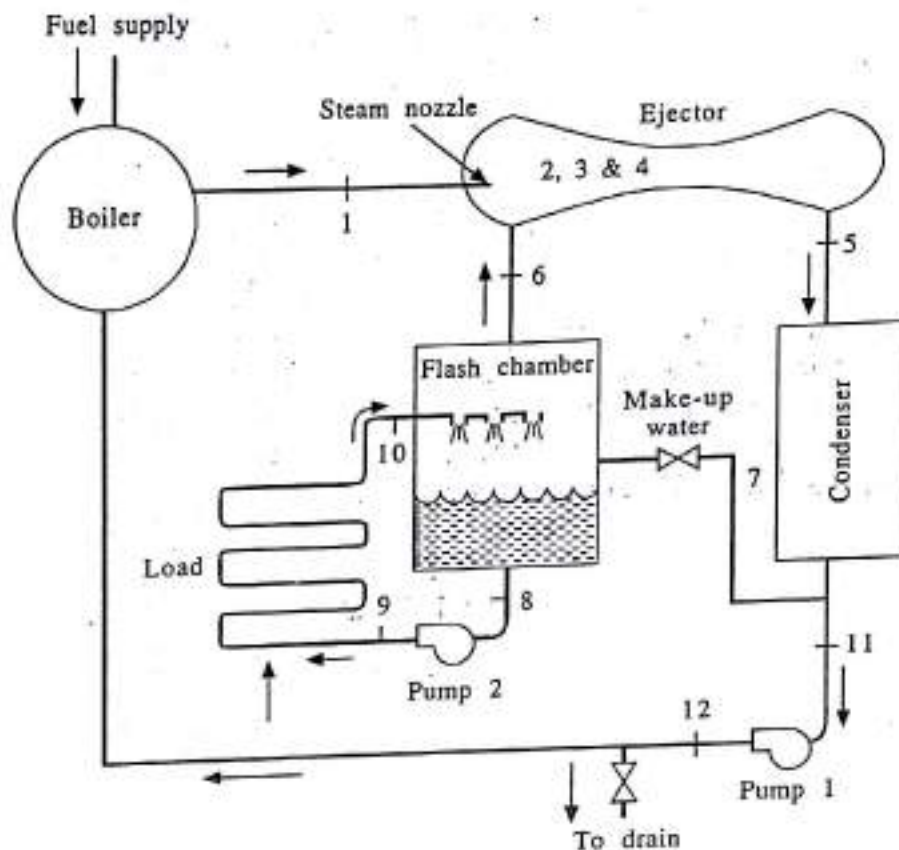


FIGURE 6.12 Water vapour refrigeration system using steam ejector.

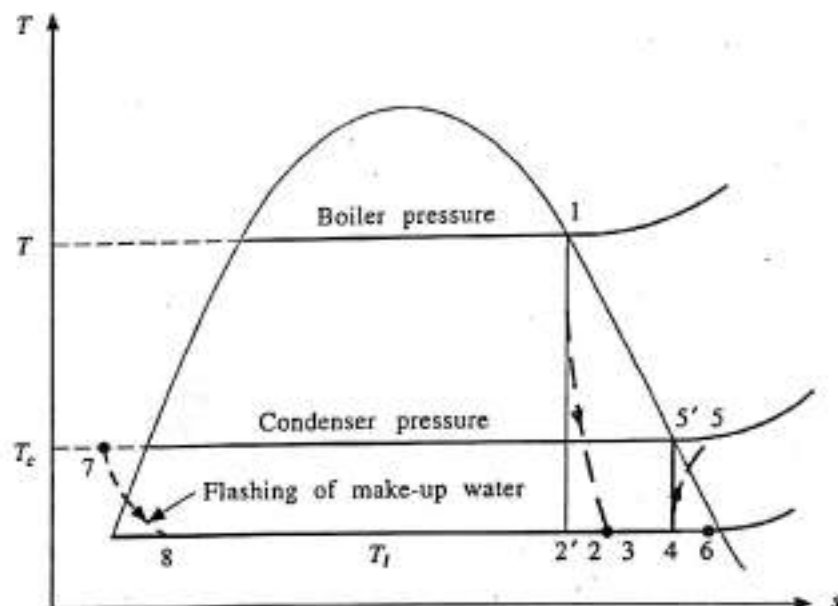


FIGURE 6.13 Steam-jet refrigeration cycle represented on a T - s diagram.

The operating pressures required are extremely low, since an evaporation temperature of 4°C requires an absolute pressure in the evaporator of 6 mm of mercury. If condensation is to be accomplished at approximately 38°C , the condenser pressure must be 50 mm of mercury

absolute. Operation under these conditions results in a compression ratio of 8 to 1, which approaches the limit for efficient operation. The air and water vapour remaining in the condenser are removed by means of a secondary ejector or a vacuum pump and are discharged from the system.

In Figure 6.12, 1 represents the initial condition of the motive steam before passage through the nozzle to 2, thereafter losing kinetic energy the final condition becomes 3. Point 6 represents the initial condition of the water vapour flashed in the evaporator and 4 the resultant condition of the mixture of high-velocity steam and entrained water vapour before compression. If compression were isentropic, the final condition would then be 5' at discharge of the mixture to the condenser. Make up water is available at 7, slightly lower in temperature than the condenser conditions, and is throttled to point 8 in the flash chamber. Because of unavoidable losses encountered in expansion, entrainment, and compression, the actual path of expansion and entrainment of the motive steam is 1-2-4. Compression of the mixture actually occurs along 4-5.

The nozzle efficiency η_n is the ratio of the actual enthalpy drop to the isentropic enthalpy drop of the motive steam passing through the nozzle. This efficiency will ordinarily range from 0.87 to 0.95 in properly designed nozzles.

The entrainment efficiency, η_e , indicates the losses that originate in the transfer of a portion of the kinetic energy of the motive steam to the flashed vapour and should be taken as 65 per cent. The loss of this kinetic energy, $1 - \eta_e$, is dissipated as heat added to the mixture. The compression efficiency η_c indicates the loss encountered in utilizing the kinetic energy of the mixture for compression in the diffuser section of the ejector. The ratio between the isentropic enthalpy increases theoretically required to the actual enthalpy increase needed to accomplish this compression is the compression efficiency, which may be taken as 75 to 80 per cent.

Thermodynamics of cooling

During expansion of steam through the nozzle, the irreversibilities and fluid friction cause the reduction in the speed of steam, i.e. the enthalpy drop is reduced. By definition of the nozzle efficiency,

$$h_1 - h_2 = \eta_n(h_1 - h'_2) \quad (6.23)$$

The quantity $(h_1 - h_2)$ is equivalent to kinetic energy of the motive steam, i.e. $C^2/2$, now available for entrainment of the vapour in the flash chamber. For the well-designed nozzle, the value of η_n ranges from 0.9 to 0.94.

During the entrainment process the motive steam loses its kinetic energy, thereby the state of the motive steam is represented by state 3, and is expressed by the entrainment efficiency, η_e , i.e.

$$h_1 - h_3 = \eta_e(h_1 - h_2) \quad (6.24)$$

The reported values of η_e are around 0.65.

The state of the mixture of motive steam and flash vapour is represented by point 4.

The isentropic compression of vapour is represented by 4-5'. However, the compression is again far from the ideal value due to friction, which necessitates the introduction of η_c , compression efficiency, whose value ranges from 0.75 to 0.8, i.e.

$$h'_5 - h_4 = \eta_c(h_5 - h_4) \quad (6.25)$$

The energy for compression comes from motive steam. If m_1 be the mass of motive steam per unit mass of flash vapour, then

$$m_1(h_1 - h_3) = (m_1 + 1)(h_5 - h_4) \quad (6.26)$$

where the quantity $(m_1 + 1)$ is the total mass of vapour compressed. Using Eqs. (6.23–6.25), Eq. (6.26) yields

$$\begin{aligned} m_1 &= \frac{h'_5 - h_4}{\eta_n \eta_e \eta_c (h_1 - h'_2) - (h'_5 - h_4)} \\ &= \frac{1}{\eta_n \eta_e \eta_c \frac{h'_1 - h'_2}{h'_5 - h_4} - 1} \end{aligned} \quad (6.27)$$

in terms of isentropic enthalpies, h'_2 and h'_5 .

The flash vapour at state 6 is wet and has to be evaluated. It is done using the ejector as the control volume, i.e.

$$m_1 h_1 + h_6 = (m_1 + 1) h_5 \quad (6.28)$$

Substituting the value of m_1 from Eq. (6.27) into Eq. (6.28) and simplifying, we get

$$h_6 = h_5 - \frac{h_1 - h_5}{\eta_n \eta_e \eta_c \frac{h'_1 - h'_2}{h'_5 - h_4} - 1} \quad (6.29)$$

The refrigerating effect may be evaluated by carrying out energy balance across the flash chamber, i.e.

$$Q_r + |W_{p_2}| + m_7 h_7 = m_6 h_6 \quad (6.30)$$

Since $m_6 = m_7 = 1$, we have

$$Q_r = (h_6 - h_7) - |W_{p_2}| \quad (6.31)$$

The energy input to the steam per unit mass of the flash vapour can be expressed as

$$Q_{\text{steam}} = m_1(h_1 - h_{11}) - \frac{(m_1 + 1)(W_{p_1})}{\eta_{p_1}} \quad (6.32)$$

Therefore,

$$\text{COP} = \frac{h_6 - h_7 - |W_{p_2}|}{m_1(h_1 - h_{11}) - \frac{(1 + m_1)(W_{p_1})}{\eta_{p_1}}} \quad (6.33)$$

In practice, the amount of steam (kg/min) needed for given capacity (TR) of refrigeration plant is required and the same is found from

$$m'_1 = \frac{210 \text{ TR } m_1}{h_6 - h_7 - |W_{p_2}|} \quad (6.34)$$

i.e.

$$m_{\text{make-up}} = \frac{210 \text{ TR}}{h_6 - h_7 - |W_{p_2}|} \quad (6.35)$$

BIBLIOGRAPHY

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Sparks, N.R. and C.C. Dilio, *Mechanical Refrigeration*, McGraw-Hill Book Company, Ltd., 1959.

REVIEW EXERCISES

- 6.1 What is Peltier effect? Briefly discuss how the Peltier effect forms the basis of thermoelectric refrigeration.
- 6.2 Draw a block diagram of a thermoelectric refrigerator and indicate how different components of this cycle correspond to those of the vapour compression cycle.
- 6.3 What are the five phenomena that take place simultaneously when two dissimilar metals or semiconductors are connected and the junctions held at different temperatures?
- 6.4 What are the applications of thermoelectric refrigerators?
- 6.5 With the aid of a sketch, briefly describe how does the vortex tube refrigerator work?
- 6.6 What are the advantages and limitations of a vortex tube refrigeration system?
- 6.7 Why is a vortex tube refrigeration system suitable for mining operations?
- 6.8 Briefly describe the principle of operation of a steam jet water vapour refrigerating system. What are its advantages and limitations?
- 6.9 Define the various efficiencies in a steam jet water vapour refrigerating system.
- 6.10 What is the function of an ejector in a steam jet water vapour refrigerating system?

- The dry saturated vapour (at state 2) is drawn by the compressor from evaporator at lower pressure p_1 and then it (vapour) is compressed isentropically to the upper pressure p_2 .
- The isentropic compression is shown by the line 2-3. Since the vapour is dry and saturated at the start of compression it becomes superheated at the end of compression as given by point 3.
- The process of *condensation which takes place at constant pressure* is given by the line 3-4.
- The vapour now reduced to saturated liquid is throttled through the expansion valve and the process is shown by the line 4-1.
- At the point 1 a mixture of vapour and liquid enters the evaporator where it gets dry saturated as shown by the point 2. The cycle is thus completed.

Heat extracted (or net refrigerating effect produced),

$$R_n = h_2 - h_1$$

Work done $W = h_3 - h_2$

$$\therefore \text{C.O.P.} = \frac{R_n}{W} = \frac{h_2 - h_1}{h_3 - h_2}$$

The values of h_1 , h_2 and h_3 can be directly read for p - h chart.

3.7 FACTORS AFFECTING THE PERFORMANCE OF A SIMPLE VAPOUR COMPRESSION SYSTEM

The factors which affect the performance of a vapour compression system are as follows:

1. Effect of suction pressure: The effect of *decrease* in suction pressure is shown in Fig. 3.7.

The C.O.P. of the original cycle,

$$\text{C.O.P.} = \frac{h_2 - h_1}{h_3 - h_2}$$

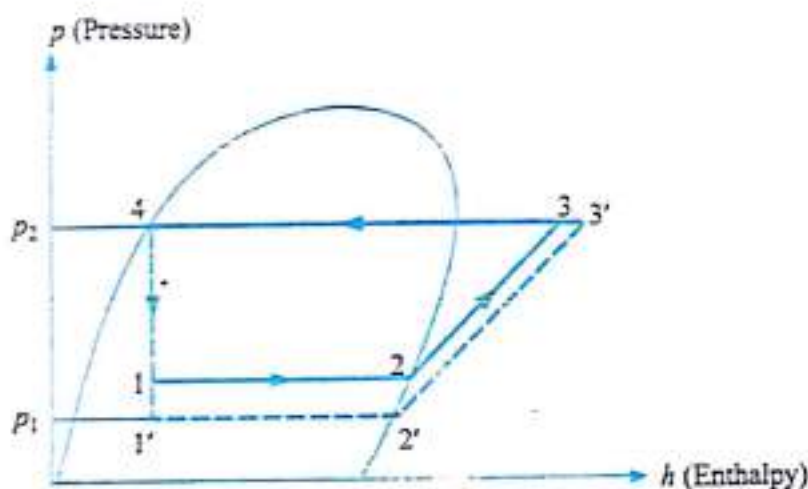


Fig. 3.7. Effect of decrease in suction pressure.

The C.O.P. of the cycle when suction pressure is decreased,

$$\text{C.O.P} = \frac{h_2 - h_1}{h_3 - h_2} = \frac{(h_2 - h_1) - (h_2 - h_2)}{(h_3 - h_2) - (h_2 - h_2) + (h_3 - h_3)} \quad (\because h_1 = h_2)$$

This shows that the *refrigerating effect is decreased and work required is increased. The net effect is to reduce the refrigerating capacity of the system (with the same amount of refrigerant flow) and the C.O.P.*

2. Effect of delivery pressure: Fig. 3.8 shows the effect of increase in delivery pressure.

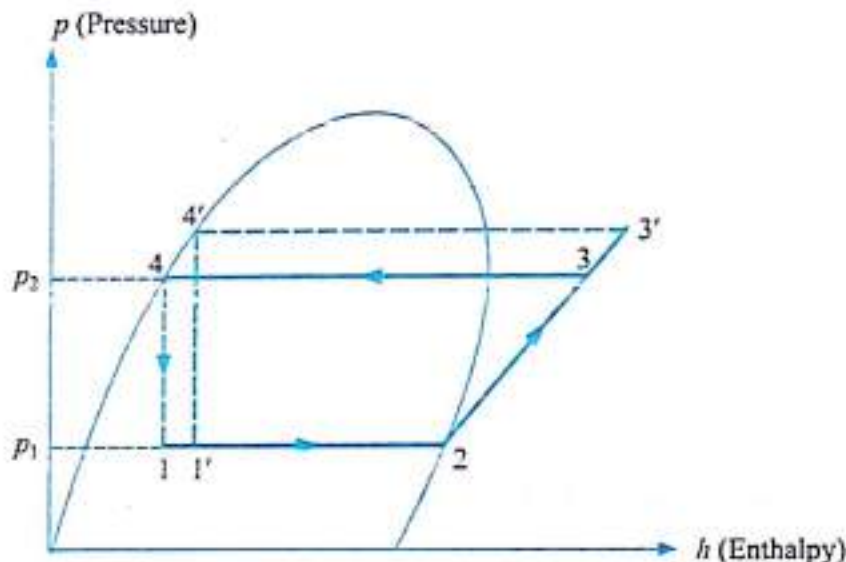


Fig. 3.8. Effect of increase in delivery pressure.

C.O.P. of the original cycle,

$$\text{C.O.P} = \frac{h_2 - h_1}{h_3 - h_2}$$

C.O.P. of the cycle when delivery pressure is increased,

$$\text{C.O.P.} = \frac{h_2 - h_{1'}}{h_3 - h_2} = \frac{(h_2 - h_1) - (h_{1'} - h_1)}{(h_3 - h_2) + (h_3 - h_3)}$$

The effect of increasing the delivery/discharge pressure is just similar to the effect of decreasing the suction pressure. *The only difference is that the effect of decreasing the suction pressure is more predominant than the effect of increasing the discharge pressure.*

The following points may be noted:

- (i) As the discharge temperature required in the summer is more as compared with winter, the same machine will give less refrigerating effect (load capacity decreased) at a higher cost.
- (ii) The increase in discharge pressure is necessary for high condensing temperature and decrease in suction pressure is necessary to maintain low temperature in the evaporator.

3. Effect of superheating. As may be seen from the Fig. 3.9 the effect of superheating is to increase the refrigerating effect but this increase in refrigerating effect is at the cost of increase in amount of work spent to attain the upper pressure limit. Since the increase in work is more as compared to increase in refrigerating effect, therefore *overall effect of the superheating is to give a low value of C.O.P.*

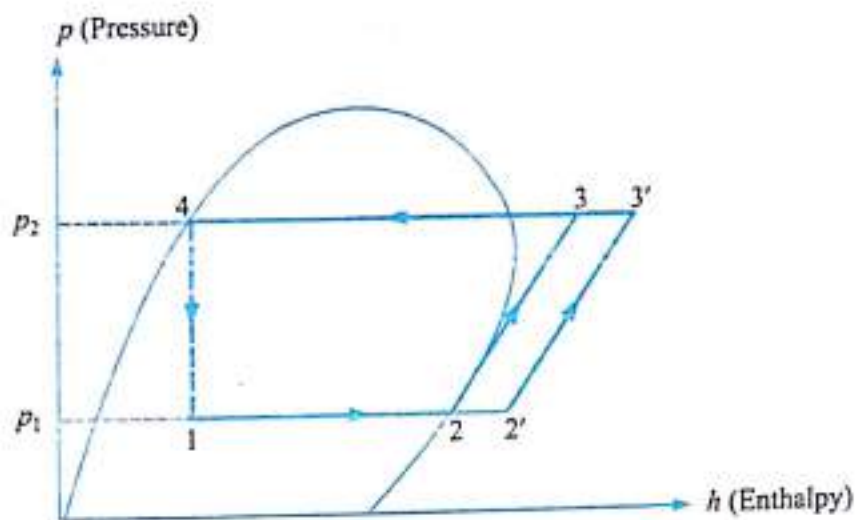


Fig. 3.9. Effect of superheating.

4. Effect of sub-cooling of liquid. 'Sub-cooling' is the process of cooling the liquid refrigerant below the condensing temperature for a given pressure. In the Fig. 3.10 the process of sub-cooling is shown by 4-4'. As is evident from the figure the effect of sub-cooling is to increase the refrigerating effect. Thus sub-cooling results in increase of C.O.P, provided that no further energy has to be spent to obtain the extra cold coolant required.

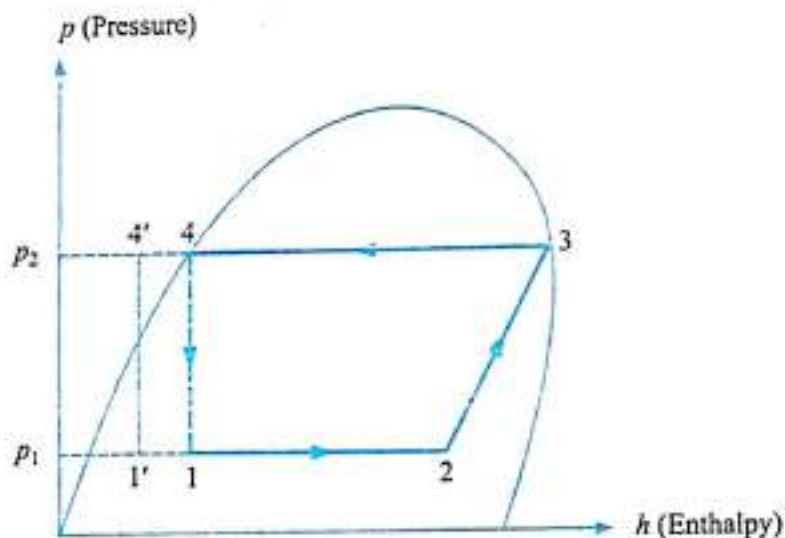


Fig. 3.10. Effect of sub-cooling of liquid.

The sub-cooling or under-cooling may be done by any of the following methods:

- (i) Inserting a special coil between the condenser and the expansion valve.
- (ii) Circulating greater quantity of cooling water through the condenser.
- (iii) Using water cooler than main circulating water.

5. Effect of suction temperature and condenser temperature. The performance of the vapour compression refrigerating cycle varies considerably with both vaporising and condensing temperature. Of the two, the vaporising temperature has far the greater effect. It is seen that the capacity and performance of the refrigerating system improve as the vaporising temperature increases and the condensing temperature decreases. Thus refrigerating system should always be designed to operate at the highest possible vaporising temperature and lowest possible condensing temperature, of course, keeping in view the requirements of the application.