Appendix A

Mechanical Refrigeration

Contents

Mechanical refrigeration

INTRODUCTION

Definition

The dictionary defines refrigeration as a process of making something cold, and 'cold' can be defined as an absence of heat. Hence, refrigeration becomes a process of taking away heat.

Heat is a form of energy and can be transferred from one body to another by virtue of a temperature difference between them. A fundamental law states that heat may be transferred only from a higher temperature substance to that at a lower temperature. Therefore, refrigeration becomes a process of providing a substance at a lower temperature to which heat can flow from a substance at a higher temperature.

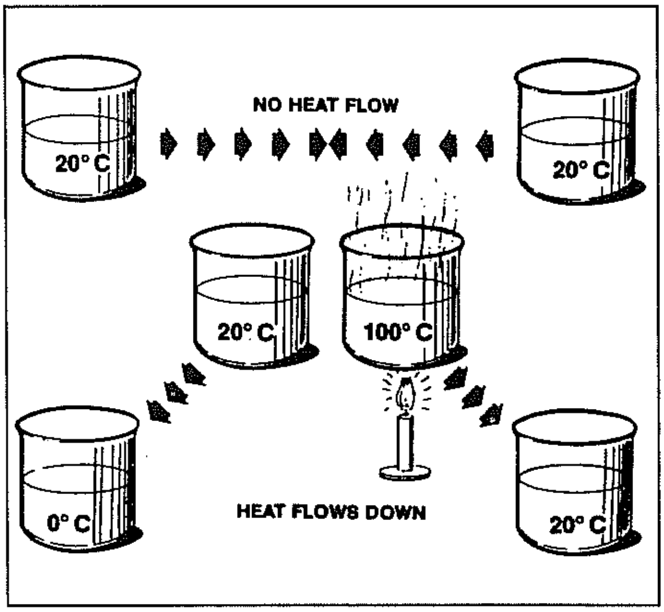


Fig. 1

History - ice and snow

The earliest and most common of the cold substances used for taking away heat was ice or snow. The Chinese were among the first to learn that ice made drinks cooler and more tasty. In early Greek and Roman days, slaves were used to carry snow from mountain tops which was stored in pits, to be used later for creating frozen delicacies. These practices moved on into Europe with the spread of civilization; and in France during the 16th century, ice and snow were used to cool beverages, and frozen dishes became popular.

Francis Bacon, in 1626, was the first to think of refrigeration for preserving foods. He experimented with a chicken stuffed with snow to see if it would preserve it; but it was not until the discovery of the microscope in 1683 that tangible results were obtained. With the microscope, scientists learned about bacteria, enzymes and mould. They discovered that these microscopic organisms multiply with heat, but that they seem to hibernate at temperatures of 10°C and below. Lower temperatures do not kill these organisms, but do control their growth. For the first time, then, food could be maintained in its fresh state by the use of cold instead of preserving it through drying, smoking or salting. The use of cold or refrigeration for this purpose then grew to the worldwide business which exists today.

History - mechanical refrigeration

For nearly a hundred years, all refrigeration for foods was provided by nature through the use of ice or snow. It was not until 1775 that experiments were made to create lower temperatures artificially, but these first experiments never got beyond the laboratory. It was not until 1834 that the first patent was granted on a mechanical refrigeration machine. This was a British patent, and a quotation from it is of interest because in principle it describes the mechanical refrigeration system as it is used today:

'What I claim is an arrangement whereby I am enabled to use volatile fluids for the purpose of producing the cooling or freezing of fluids, and yet at the same time, constantly condensing such volatile fluids and bringing them again and again into operation without waste.'

In 1855, the first mechanical refrigeration machine was put to use in the United States, in Cleveland, Ohio; and it was used for the artificial making of ice. About 1900, electricity began to enter many homes, and with the development of the small electric motor, the 'ice plant' was brought into the home. Now, refrigeration machines are made in sizes up to many hundreds of kilowatts, and multiples of such machines are used to form systems totalling several thousands of kilowatts in a single installation - all with the same principle as described in that original patent over one hundred and fifty years ago.

GENERAL PRINCIPLES

The unit of measure for heat is the joule or kilojoule. It can be defined as the quantity of heat necessary to raise the temperature of one kg of water one degree Celsius.

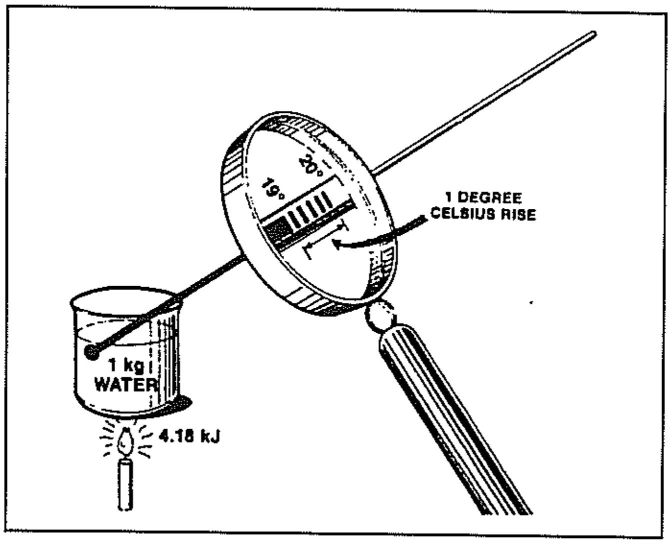


Fig. 2

Change of state

The 'ice boxes' of the past were periodically charged with a cake of ice. The ice melted and the pan of water below the box had to be 'emptied' on schedule to avoid overflow.

In melting from a solid to a liquid, the ice absorbed its latent heat of fusion. This amounts to 335 kJ/kg of ice, and the heat to accomplish this was removed from the foodstuffs within the box.

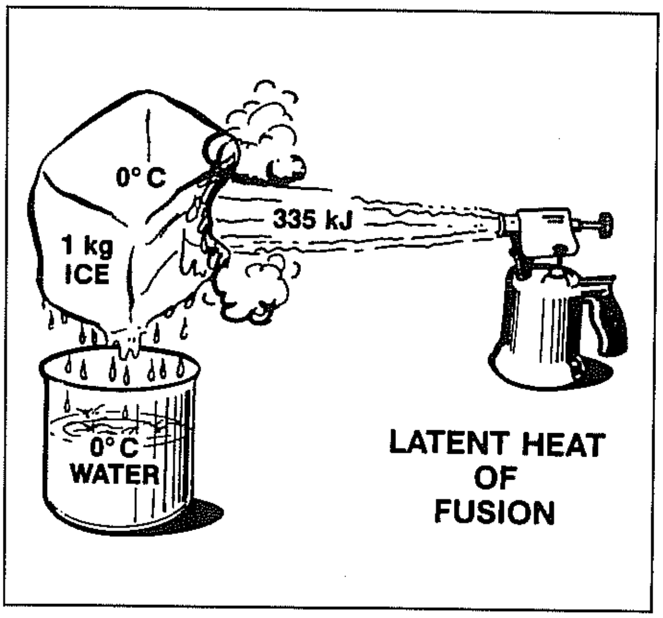


Fig. 3

The water was discarded even though it was cold, because it had very limited refrigeration capacity. This is true because 4.19 kJ raises the temperature of 1 kg of water one degree Celsius. Therefore, when 1 kg of water starting at 0°C absorbs 41.9 kJ, its temperature rises ten degrees to 10°C. This defeats or diminishes its cooling effect.

The change of state is important to the mechanical refrigeration cycle for two reasons. First, the change absorbs a relatively large amount of heat per kilogram of substance; and second, this change of state takes place at constant temperature.

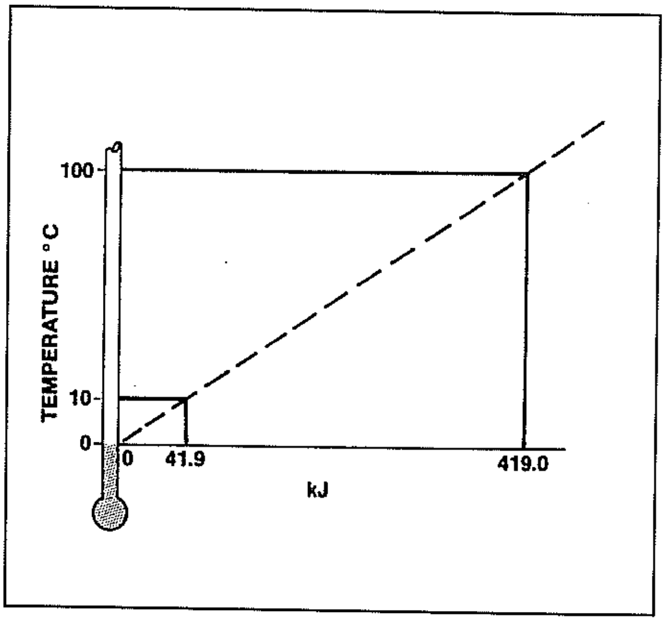


Fig. 4

The boiling process

Since the properties of water are easily observed and since its behaviour is similar to commonly used refrigerants, water is used here to demonstrate the boiling process and to establish terminology.

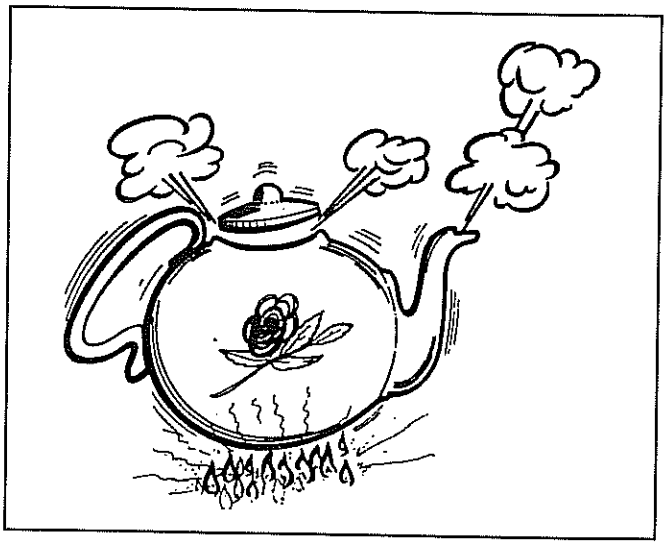


Fig. 5

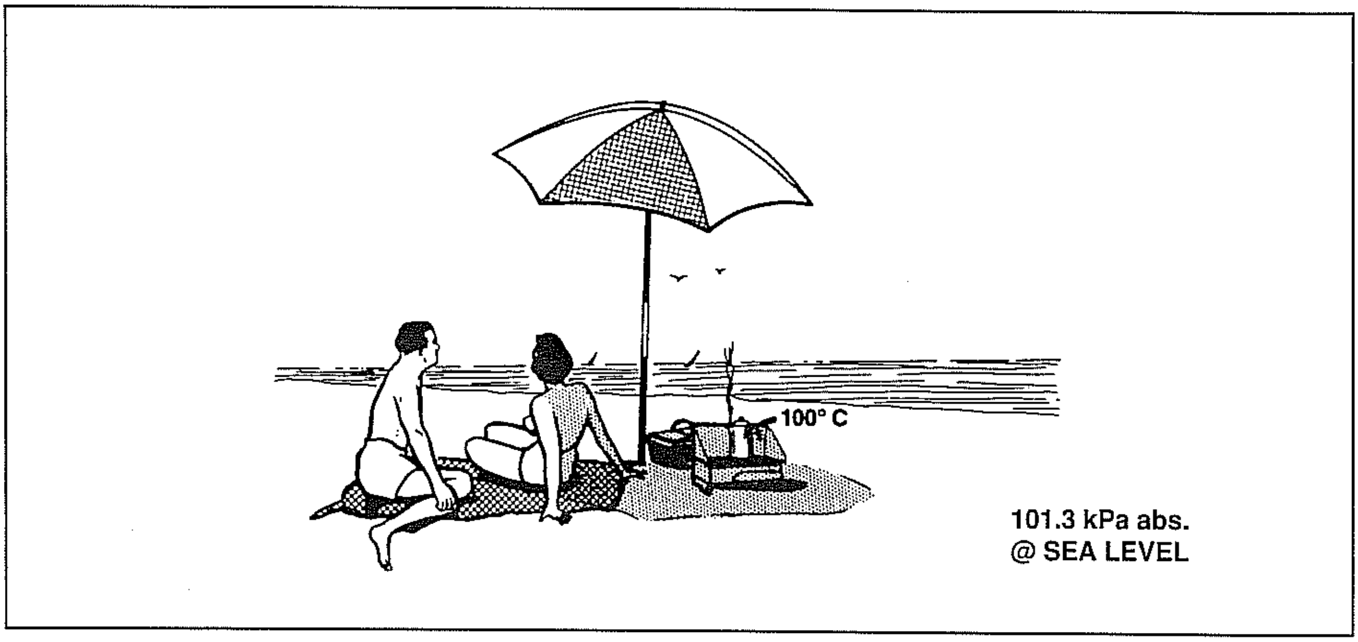


Fig. 6

If 1 kg of water at 0°C is heated, its temperature increases one degree for each 4.19 kJ added. This process continues until the water reaches its boiling point, or boiling temperature. This boiling temperature is determined by the pressure over the water. In an open container, the pressure over the water is atmospheric pressure. In a closed container, the pressure in the container controls the boiling temperature.

At sea level or standard barometric pressure of 101.3 kPa, water boils at 100°C. If the pressure is greater than 101.3 kPa, the temperature at which water boils also increases. For example, the boiling temperature for water in a pressure cooker operating at an absolute pressure of 143.3 kPa above atmospheric pressure is 110°C. At an absolute pressure of 198.5 kPa, this temperature increases to 120°C.

Conversely, if the pressure is less than 101.3 kPa absolute, as in a vacuum, the temperature at which water boils is lower. For instance, at a pressure corresponding to 12.3 kPa absolute or 89.0 kPa vacuum below standard atmospheric pressure, the boiling temperature of water is 50°C; and at 7.4 kPa absolute or 93.9 kPa vacuum below atmospheric, it is 40°C.

If the pressure is lowered still further, water can be made to boil at temperatures sufficiently low to provide cooling for air conditioning purposes

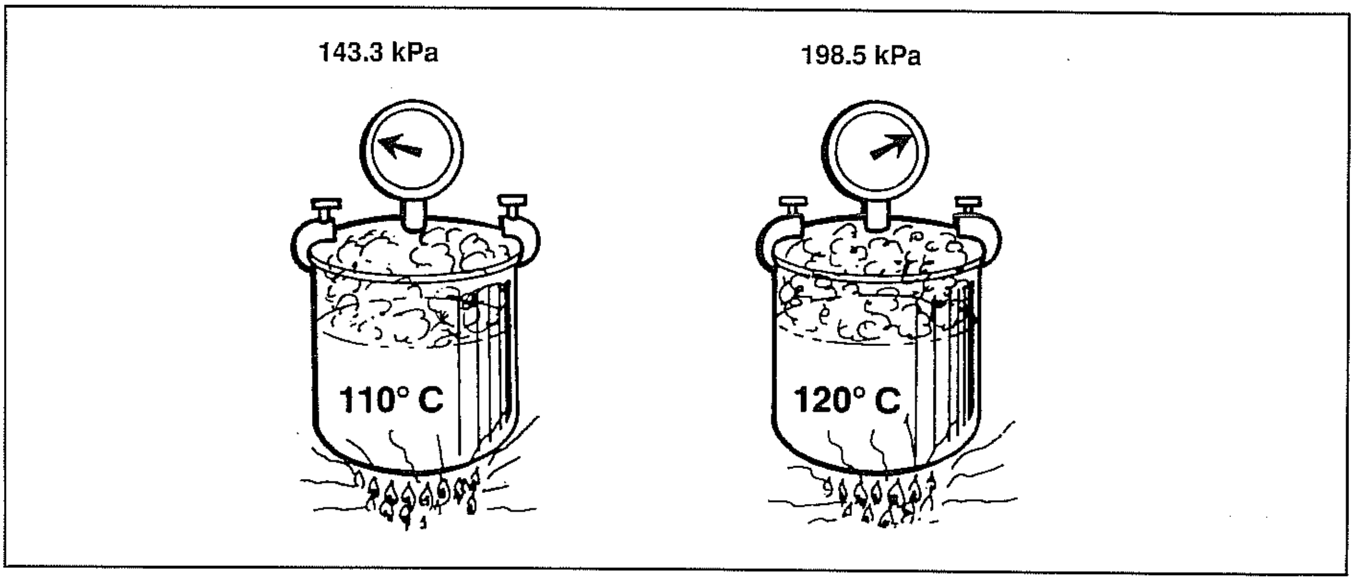


Fig. 7

At the boiling point, either the pressure or the temperature establishes the other. Water at the boiling temperature for a given pressure is identified as saturated liquid, i.e. saturated with all the heat it can contain at the pressure and still remain a liquid.

Boiling temp. °C	kPa absolute	± standard atmosphere gauge pressure - kPa
120	198.5	97.22 gauge
110	143.3	41.90 gauge
100	101.3	0 gauge
90	70.1	32.21 vacuum
70	31.2	70.16 vacuum
10	1.2	100.10 vacuum
5	0.9	100.45 vacuum

Fig. 9

Heat of vaporization

After a fluid has been heated to the boiling temperature, further addition of heat at that pressure results in evaporation of the fluid. The heat required to change a fluid from liquid to vapour is called the heat of vaporization. At standard barometric pressure, it takes 2257 kJ/kg to completely evaporate 1 kg of water from water at 100°C to steam at 100°C.

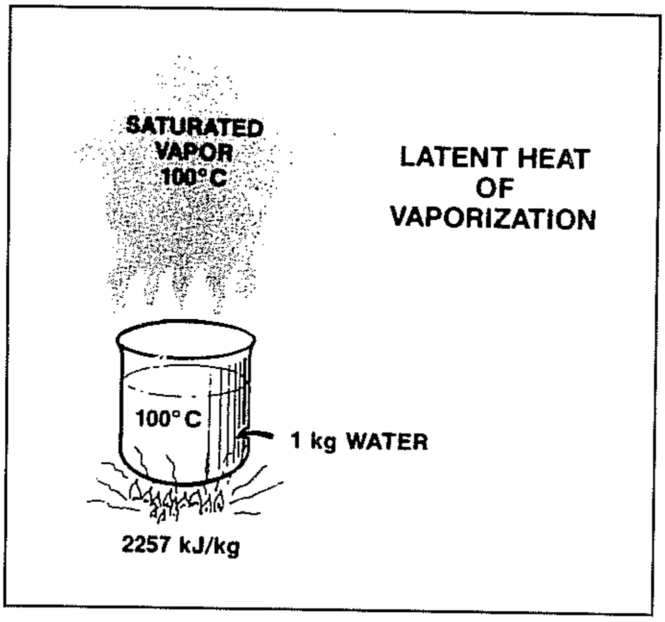


Fig. 10

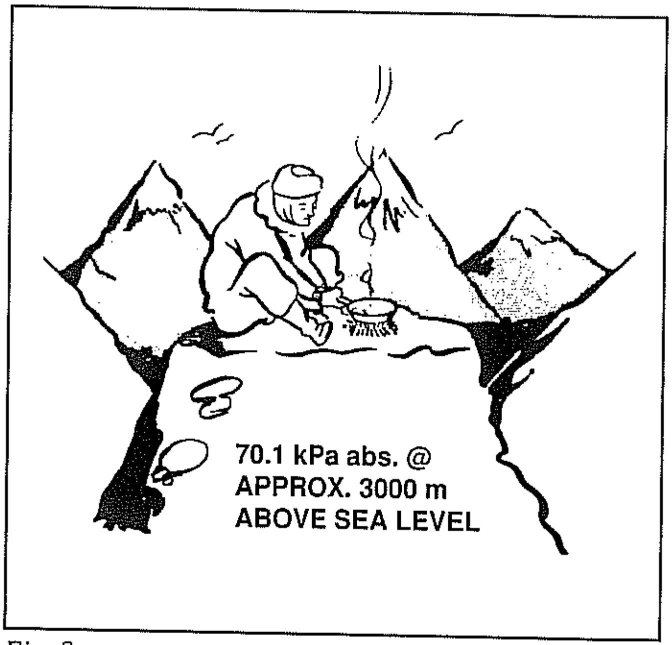


Fig. 8

Superheat

When heat is added to a fluid so it has all evaporated, it reaches a point identified as saturated vapour; i.e., saturated with all the heat it can contain at that pressure and still remain at the saturation temperature.

The change of state has been completed, and any additional heat at constant pressure results in a rise in temperature of vapour. This additional heat is called superheat.

In the superheat region, the vapour expands slightly in volume as its temperature is raised. Moreover, the specific heat of the vapour is different from that of the same fluid in liquid form. For example, it takes only 1.89 kJ to raise 1 kg of water vapour by 1 K. If a pound of steam is superheated by 20°C, then 20°C x 1.89 or 37.8 kJ is required.

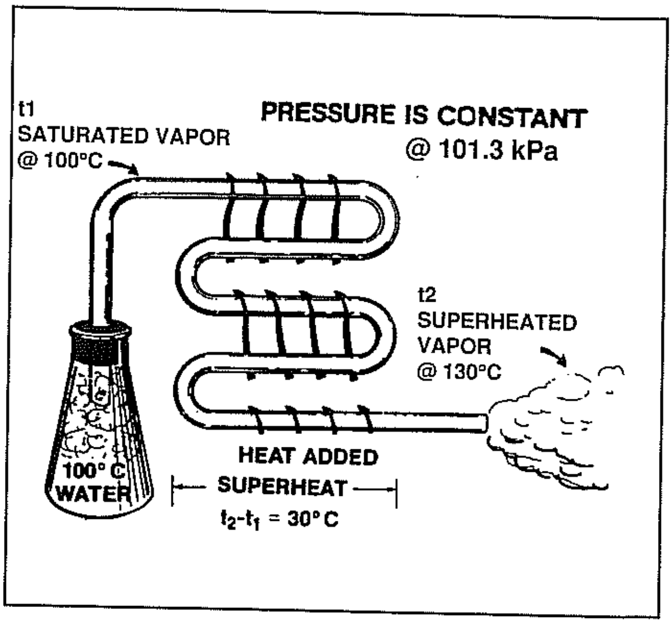


Fig. 11

Enthalpy or heat content

Actually, all matter contains heat or energy down to a temperature at absolute zero (273.2°C); but, for convenience purposes, other datum lines are selected arbitrarily at which to assign a 'zero' enthalpy value.

Because water freezes at 0°C, the datum line for water is taken at 0°C. Any values of heat content below 0°C are thus negative.

Total energy in a fluid involves not only the internal energy represented by heat, but also the external energy or work required to reach that state. The sum of these energies is called enthalpy. Since it is the more accurate term, enthalpy is used in refrigerant tables and is used in this text to replace the term total heat content.

Temperature-enthalpy diagram

The properties discussed thus far can be illustrated on a temperature enthalpy diagram. Values of enthalpy are given for 1 kg of water and at standard atmospheric pressure of 101.3 kPa.

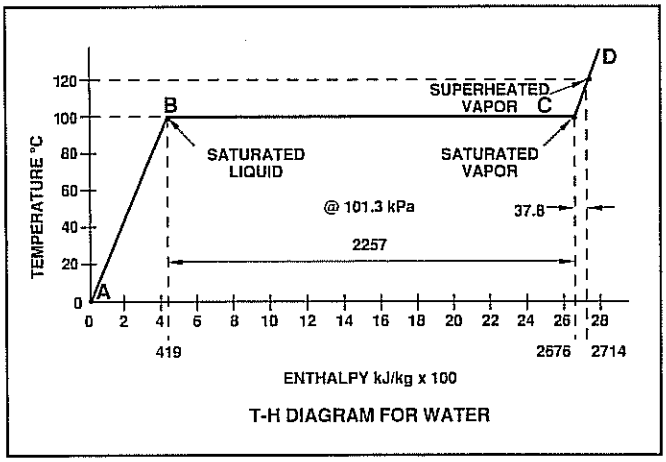


Fig. 12

Figure 12, starting at Point 'A' shows that 1 kg of water at 0°C has zero heat content or enthalpy (kJ/kg). Line A-B represents the sensible heat needed to raise the water from 0°C to 100°C which is its boiling temperature or saturated liquid temperature at standard atmospheric pressure. The difference between 0°C and 100°C represents a rise of 100 K. As stated previously, for every degree rise, 1 kJ must be absorbed by the liquid. Therefore, the heat content or enthalpy at 100°C is 100 x 4.191 kJ/kg or 419 kJ.

Line B-C represents the latent heat of vaporization needed to completely change 1 kg of saturated liquid at Point 'B' to dry saturated vapour at Point 'C'. The heat of vaporization for water at standard atmospheric pressure is 2257 kJ/kg. Hence, the enthalpy of the saturated vapour at Point 'C' is 419 + 2257 or 2676 kJ. As shown in the diagram, the change of state from Point 'B' to 'C' represents no change in temperature; therefore, heat absorbed during this process is called latent heat.

Line C-D illustrates the effect of adding sensible heat to the vapour. This process is called superheating. For every degree of superheat, 1.89 kJ must be added. For example, if 30° of superheat are added, the temperature at Point 'D' is 100°C + 30 = 120°C and requires the addition 20 x 1.89 or 37.8 kJ/kg. The enthalpy of the superheated vapour at Point 'D' then becomes 2676 + 37.8 or 2714 kJ/kg.

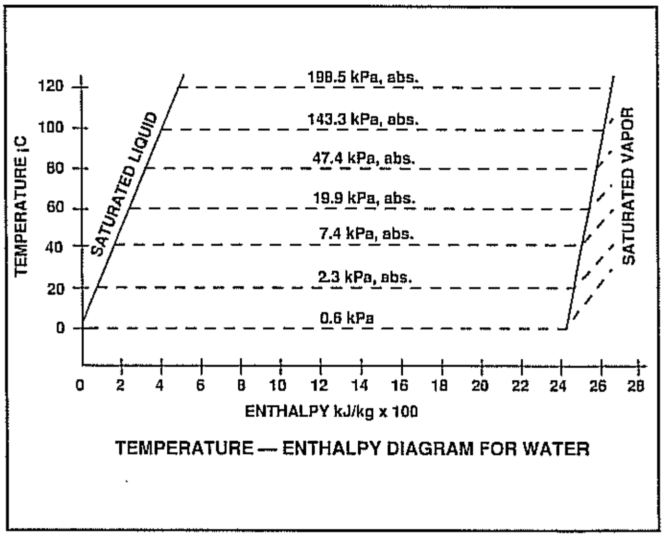


Fig. 13

Charts would be plotted for water at other pressures, or lines superimposed on the saturation points and the vaporization process at other pressures, as shown in Figure 13.

Pressure-enthalpy diagram

This produces a usable chart for discussion of the refrigeration cycle. However, the temperature scale is uniform or linear, but the corresponding pressure values are not. Since the flow of refrigerant through a cycle involves pressures, it is more convenient to replot the diagram so that the pressure scale is uniform or linear, and then superimpose the corresponding saturation temperature. This creates a pressure-enthalpy diagram as shown in Figure 14 for water.

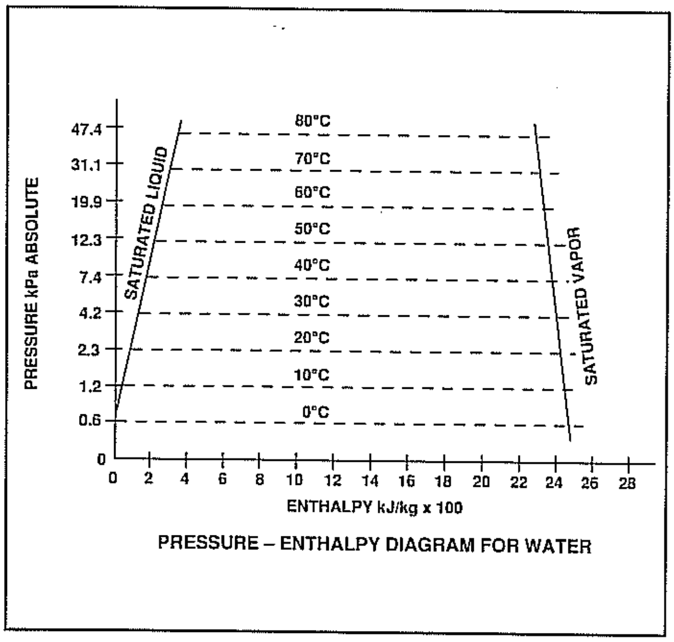


Fig. 14

This chart serves to illustrate and emphasize that for each pressure, there is a corresponding saturation temperature. This establishes the values for saturated liquid, saturated vapour and the heat of vaporization between them. Connecting the points for liquid and for vapour creates the saturated liquid line and the saturated vapour line.

The condensing process

The change of state from liquid to vapour is reversible; that is, the fluid can be changed from a vapour back to a liquid; and this process is called condensing. Just as heat must be added to effect evaporation, heat must be taken away to change a vapour back to a liquid - and in the same amount. Just as pressure establishes the temperature at which a fluid boils, pressure also fixes the temperature at which condensation takes place.

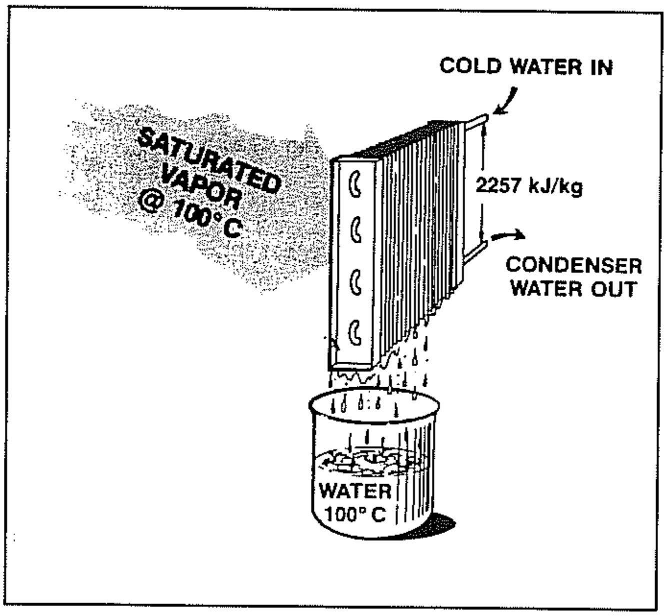


Fig. 15

REFRIGERANTS

Water as a refrigerant

The general principles of refrigeration have been introduced using water as the volatile fluid or refrigerant. Actually, water has several desirable properties as a 'refrigerant'. It is low in cost and readily available; is entirely safe, being non-toxic and non-flammable; it also has a large heat of vaporization.

Although water is used as the refrigerant in steam-jet and in absorption refrigeration machines, it is not a practical refrigerant for a mechanical refrigeration cycle.

It is limited in temperature to a minimum of 0°C, for below this, the water freezes. To obtain water cool enough for most air conditioning purposes would require operating at extremely low absolute pressures or high vacuums, which are difficult to obtain with reciprocating equipment. Moreover, the specific volume of water vapour is large at the low pressure required, and such volumes are impractical with reciprocating machines. and centrifugal pumps are unable to maintain the required vacuum.

Therefore, it is desirable to use other volatile fluids whose overall characteristics and properties make them more practical for this particular purpose.

Desirable characteristics

Commercially used refrigerants should have certain desirable characteristics. The following are among the more important ones, and their order of importance varies, depending on use and on the specific requirements of the application.

1. The refrigerant should be non-flammable and non-toxic.
2. The pressure corresponding to the temperatures available with normal condensing media should not be excessive, so as to eliminate the need for extremely heavy construction.
3. The pressures corresponding to the temperatures required for most air conditioning and refrigeration processes should be above atmospheric pressure so as to avoid inward leakage of air and water vapour.
4. A relatively large heat of vaporization is desirable so that the required capacities can be obtained with the smallest weight flow of refrigerant.
5. The vapour should have a relatively low specific volume, because it is this volume that establishes the required size or displacement of the

compressor. This property is more important to the reciprocating compressor than it is to the centrifugal machine which is inherently a large volume, low pressure pump.

6. It is desirable that the refrigerant have a low specific heat in the liquid state in order that less heat will be required to cool the liquid from the condensing temperature down to the temperature at which cooling is to be accomplished. The heat required for this cooling results in 'flash gas', and subtracts from the refrigeration effect or cooling capacity of the refrigerant circulated.
7. The refrigerant should be easily detected by suitable indicators to locate leaks in the system.
8. The refrigerant should be compatible with the usual lubricants and oils, and should not detract from their effectiveness as lubricants.
9. The coefficients of heat transfer and the viscosity should be conducive to good heat transfer rates.
10. The refrigerant should be readily available, low in cost and easily handled.
11. The refrigerant should be non-corrosive to the metals usually employed in a refrigeration system and should be chemically stable.
12. The refrigerant should not be damaging to the environment.

Common refrigerants

The following paragraphs give a brief discussion and history on a few of the more commonly known refrigerants. Each is identified with its chemical formula and also its saturation temperature at standard atmospheric pressure.

1. Ammonia (NH_3) (-33.3°C)
Ammonia is one of the early refrigerants. It is still used today in the industrial field, in breweries, packing plants and similar applications because of its high cycle efficiency. It has a low specific volume, a relatively high latent heat of vaporization and low cost. However, in spite of these desirable properties, ammonia is limited to industrial applications and excluded from comfort air conditioning because of the first requirement listed under desirable characteristics. It is highly toxic and flammable, requiring special handling and possesses a sharp, penetrating odour.

2. Carbon dioxide (CO_2) (-78.3°C)
Carbon dioxide is an inert gas which is colourless and odourless. It is non-toxic and non-flammable, but the primary objection to its use is the heavy equipment required by its high operating pressures. This and its relatively high kilowatt requirement have limited its use as a refrigerant.

3. Methyl chloride (CH_3CI) (-23.3°C)
Methyl chloride is a colourless refrigerant with a faint sweet, non-irritating odour. It was used in the earlier models of household refrigerators and replaced ammonia and carbon dioxide for many new installations. It was used widely during World War II as a substitute for Freon which was then available only through government priorities; but at present, methyl chloride is seldom used.

4. Fluorinated hydrocarbons
Halide refrigerants are the most widely used of all refrigerants. They are odourless in concentrations up to 20% by volume in air. At concentrations above that, a mild, ether-like odour results. Vapours from these refrigerants are substantially odourless and non-irritating. However, recent research has identified these refrigerants as contributing to the depletion of the polar ozone layer and they are being phased out.

The halide refrigerants in this series are essentially non-toxic. Their vapours and liquids are non-flammable and non-combustible, for they contain no elements which support combustion. They are non-corrosive to the metals commonly used in refrigeration equipment as long as the refrigerants are dry or free from water vapour. In the presence of water vapour, these refrigerants can be quite corrosive. They have a high solvent action on natural rubber, but synthetic rubber materials can be used satisfactorily. In either liquid or vapour form, they have no effect on the odour, taste, colour or structure of refrigerated materials such as dairy products, meats, vegetables, plant life or furs and fabrics.

Figure 16 on the following page lists the refrigerant numbers, chemical formulas and boiling temperature at standard atmospheric pressure of the more common halide refrigerants of this type in use today.

Fluorocarbon refrigerants

Refrigerant	Chemical formula	Boiling temp. °C at standard atmospheric pressure	Ozone depletion factor
R-11	CCL_3F	23.7	1.00
R-12	CCL_2F_2	29.8	1.00
R-22	$CHCLF_2$	40.8	0.05
R-123	$CHCL_2CF_3$	27.6	0.02
R-134a	CH_2FCF_3	26.5	0.00
R-500	$(CCL_2F_2)\ (CH_3CHF_2)$	33.3	
R-502	$(CCLF_2)\ (CHCLF_2)$	45.6	

Fig. 16

R-12 was introduced about 1930 and was the common refrigerant for reciprocating compressors during World War II. At air conditioning levels (5°C saturated suction and 40°C saturated condensing temperature), R-12 produces a refrigeration effect of about 115 kJ/kg. R-12 is due to be phased out by 1st January 1995.

R-22 was developed initially for single stage reciprocating compressor applications below -30°C. Later, R-22 was applied to air conditioning systems. Its increased refrigeration effect and its lower specific volume results in a 60% increase in compressor capacity as compared to R-12. Use of R-22 results in a considerable reduction in the size, weight and cost of the compressor required to handle a given load. A compressor having a given capacity with R-12 will produce 1.6 times that when using R-22 at the same operating conditions, but with a proportionate increase in compressor kilowatts. Due to be phased out by 1st January 2030.

R-123 is a new refrigerant recently introduced in an effort to minimise ozone depletion. It is a hydrochlorofluorocarbon (HCFC) with an ozone depletion factor of 0.02 and is thus better than R-22. However, recent test results seem to show carcinogenic and toxic side effects in rats and most manufacturers are now recommending that this refrigerant is avoided. Additionally, recent recommendations from ANSI/ASHRAE that all HCFCs be treated as a group, means that R-123 could be phased out at the same time as other HCFCs, including R-22. This obviously does not make it a good long-term investment.

R-134a is also a new introduction aimed at overcoming ozone depletion. It is a hydrofluorocarbon (HFC), with no chlorine molecules and a zero ozone depletion factor. Unfortunately its global warming factor is higher than that of R-123. Testing is ongoing, but no problems are foreseen, and therefore this refrigerant is seen as the current front runner. R-134a results in an increase in compressor size of 30% over R-22.

R-500 is an azeotropic mixture of R-12 and ethylidene fluoride (R-100). It has a refrigeration capacity and kilowatt requirement approximately 18% greater than that of R-12 with a given compressor. The revolution per second (r/s) of a hermetic compressor operating on 50 Hertz current is about 17% less than when it operates on 60 Hertz current. A 50 Hertz current compressor using R-500 will produce about the same capacity as one operating on 60 Hertz current and charged with R-12. R-500 is due to be phased out by 1st January 1995.

R-502 is also a fluorocarbon refrigerant. It is an azeotropic mixture of R-22 and R-115. It was developed initially for use in single stage, low temperature refrigerant applications such as frozen food processing and storage. It has the advantage of higher capacities than R-22, while compressor discharge temperatures are considerably lower than those of R-22. In refrigerant-cooled hermetic compressors, motor cooling is better than that obtained with R-22. R-502 is also due to be phased out by 1st January 1995.

Many other refrigerants and blends of refrigerants aimed at reducing ozone depletion are currently being investigated. The majority of manufacturers of small water chillers suitable for the hydronic market are still using R-22, but have plans to switch to R-134a in the short term.

THE REFRIGERATION CYCLE

Evaporation or cooling effect

If a volatile refrigerant is available in liquid form, it will absorb heat in evaporating or changing to a vapour. It is this change of state that accomplishes the cooling effect in a refrigeration cycle.

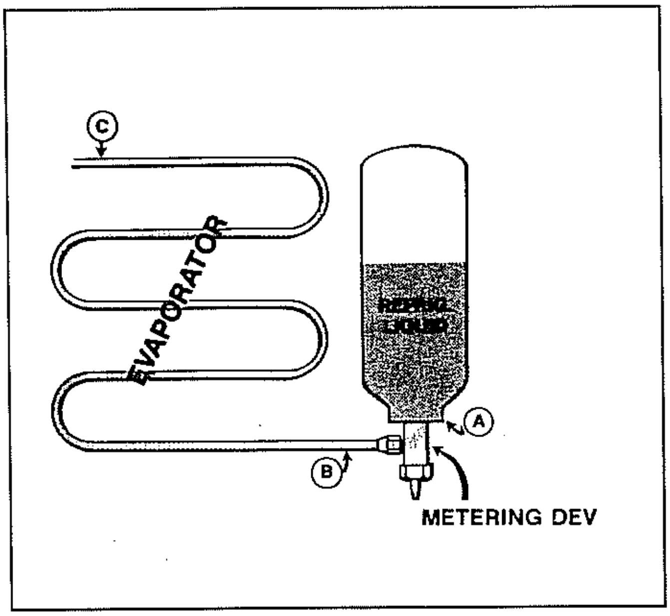


Fig. 17

If refrigerant is allowed to expand through a cooling coil which is open to the atmosphere, the refrigerant absorbs heat from the surrounding air and evaporates at a temperature corresponding to atmospheric pressure. With refrigerant R-22, this would be -40.8°C (see Figure 17).

Since it is evaporation which accomplishes the cooling, that part of the equipment in which the cooling is done and the evaporation takes place is called the 'evaporator'.

The expansion process

When liquid refrigerant flows from the higher pressure of the refrigerant drum to the low pressure of the evaporator, its rate of flow must be controlled by some type of metering device. This might be a thermal expansion valve, a hand valve, a float valve or a capillary tube. In any of these, the refrigerant expands as it enters the area of lower pressure. The process is often referred to as throttling.

The expansion takes place quickly and in a relatively short distance through a valve. Hence, it is practical in an analysis of the refrigerant cycle to assume that there is neither a heat gain nor a heat loss from the refrigerant in this process. Therefore, the refrigerant enters the evaporator with a heat content or enthalpy equal to that as it entered the metering device or 'expansion valve'.

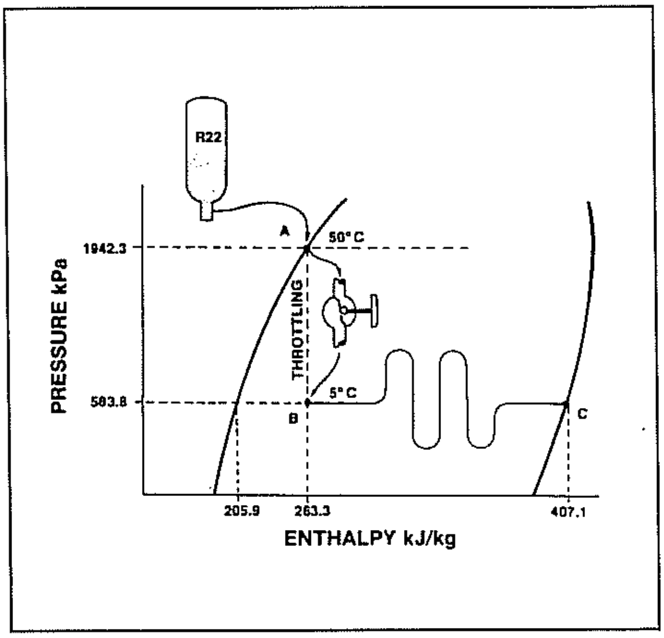


Fig. 18

Flash gas or cooling the liquid

Liquid refrigerant supplied in a drum would be at a temperature (and hence saturation pressure) in equilibrium with the ambient temperature. Liquid refrigerant from an air-cooled condenser will be available at the elevated pressure and temperature at which condensing takes place. As the refrigerant flows through the metering device to the lower pressure of the evaporator, the liquid cools to the saturation temperature corresponding to the lower pressure.

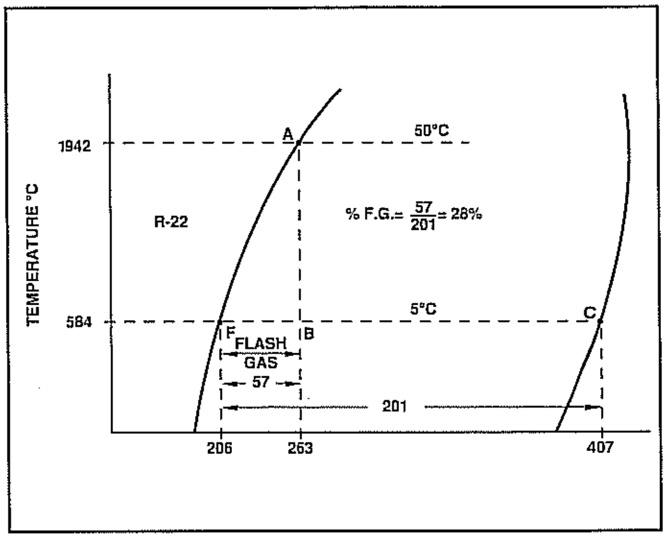


Fig. 19

The vapour resulting from this evaporation in cooling the liquid is called flash gas, and its amount is referred to as per cent flash gas.

Per cent flash gas at air conditioning levels of temperature is in the range of 20 to 30%. It is an inherent part of a refrigerant cycle; and since it detracts from the useful capacity of the cycle, it is desirable that the refrigerant liquid have a low specific heat so the flash gas will be minimum.

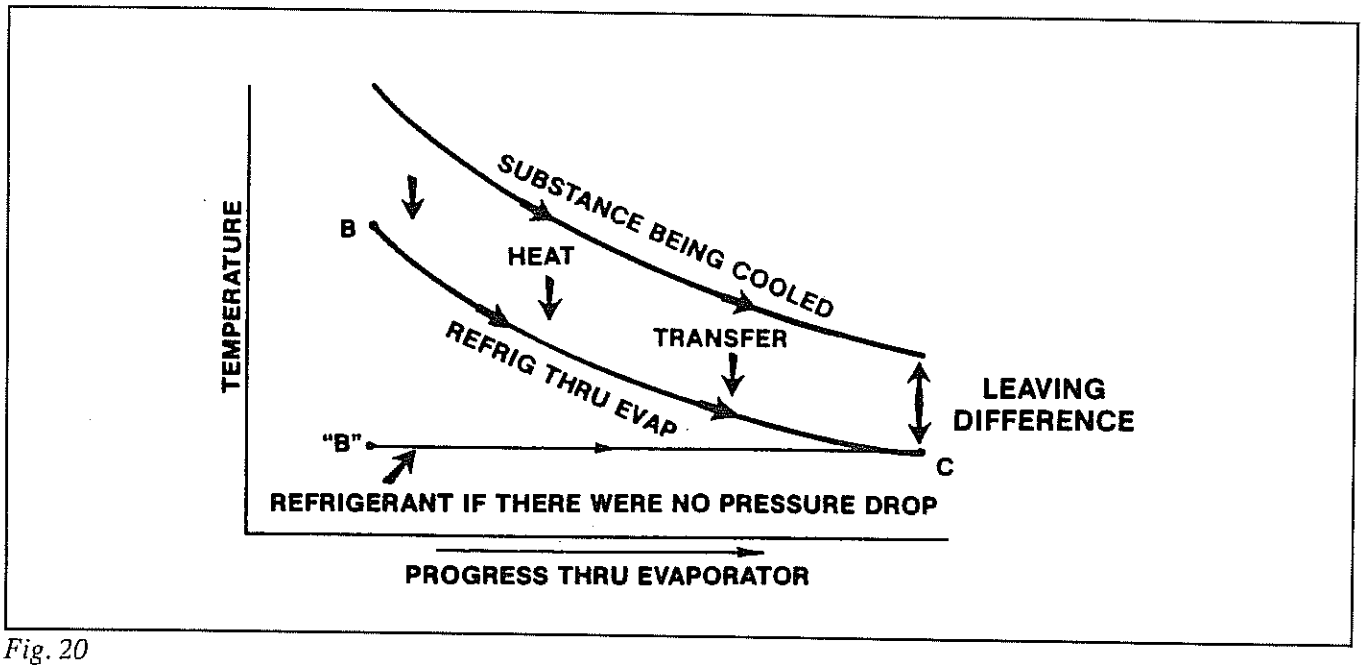


Fig. 20

Refrigerant temperature

If heat is to flow from a substance to a refrigerant, the temperature of the refrigerant must be below that of the substance. The difference between the two temperatures is established by the relative size and performance of the heat transfer surface in the evaporator.

The heat flow through the metal which makes up the evaporator must be in balance with the following equation:

$W = U \times A \times \Delta T$

Where:

W = Rate of heat flow in Watts

U = Overall heat transfer coefficient in $\text{W/m}^2 \times {}^\circ\text{C}$ including the air or water side film, the metal resistance and the refrigerant film

A = Square metres (m^2) of heat transfer surface, accounting for both prime surface at the tubes and any additional fin surface

ΔT = The temperature difference between the fluids on either side of the metal in the coil or evaporator, K

For any one rate of heat flow, if the surface (A) is large, then the temperature difference (ΔT) can be smaller; and vice versa. Also for any one piece of equipment in an operating cycle, as the load decreases and W becomes smaller, with little or no change in U, the ΔT will decrease. This is important in the analysis of performance at part loads.

There must be a pressure difference between the inlet and outlet of an evaporator to affect a flow of refrigerant through it. As the pressure is lowered, the corresponding saturation temperature goes down; and the balance of the liquid refrigerant must be cooled by further evaporation of refrigerant. The per cent flash gas increases and refrigeration effect decreases as the refrigerant progresses into lower pressures in its path through the evaporator. However, the total flash gas and net refrigeration effect calculated on the premise that there is no pressure drop through the evaporator are the same as those resulting with the actual pressure drop, and this premise simplifies the analysis of a cycle.

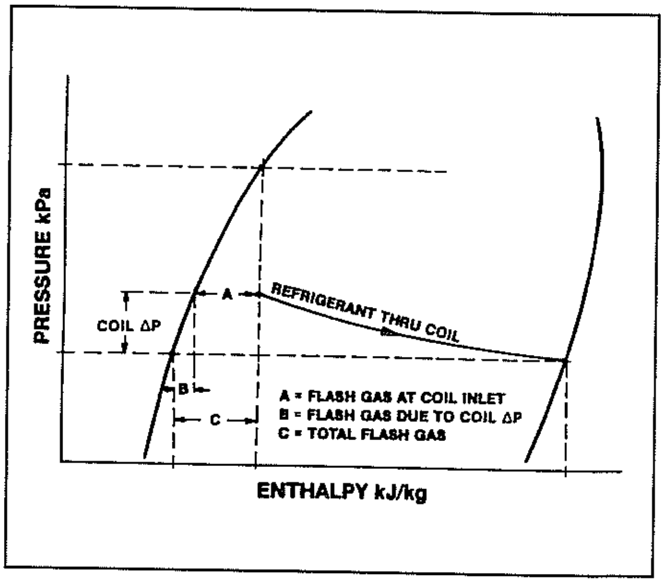


Fig. 21

Most evaporators are rated in terms of the refrigerant saturation temperature corresponding to the pressure at the leaving end or outlet of the evaporator. This is called the 'refrigerant temperature'.

Refrigeration effect

Refrigerant enters an evaporator as a mixture of cold liquid and the vapour which flashed to cool the liquid. As the refrigerant takes on heat from the media to be cooled, the balance of the liquid evaporates. In order to make full use of the refrigerant circulated, it is desirable to evaporate all the refrigerant before it leaves the evaporator. In coolers, such as those controlled by thermal expansion valves, the refrigerant takes on additional heat and leaves as a superheated vapour.

The difference between the heat content of the refrigerant as it entered the evaporator and the heat content as it leaves the evaporator is called the refrigeration effect (see Figure 22).

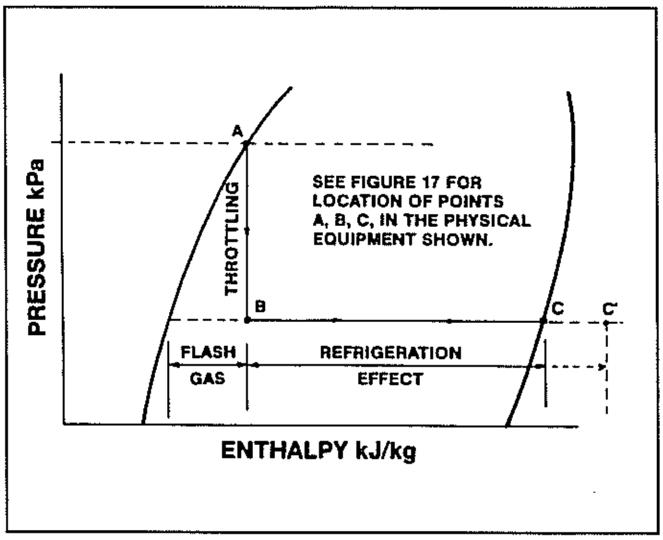


Fig. 22

The compressor

The elements of the cycle shown in Figure 17 would provide refrigeration or cooling. However, such an arrangement would be very expensive to operate because the refrigerant would be wasted, also the escaping refrigerant would be dangerous to the environment. Therefore, additional equipment is needed and used to recover the refrigerant, so that 'the volatile fluids are brought again and again into operation without waste'.

If the refrigerant vapour from the evaporator were merely passed into a chamber or tank, the pressure in that chamber would rise until the pressure in the chamber and that of the refrigerant supply reached equilibrium at a pressure corresponding to ambient temperature, and the cooling effect would stop. Therefore, it is necessary to continually remove the refrigerant vapour as rapidly as it is formed to maintain proper pressure, and hence, saturation or boiling temperature in the evaporator. This is one function of the compressor (see Figure 23).

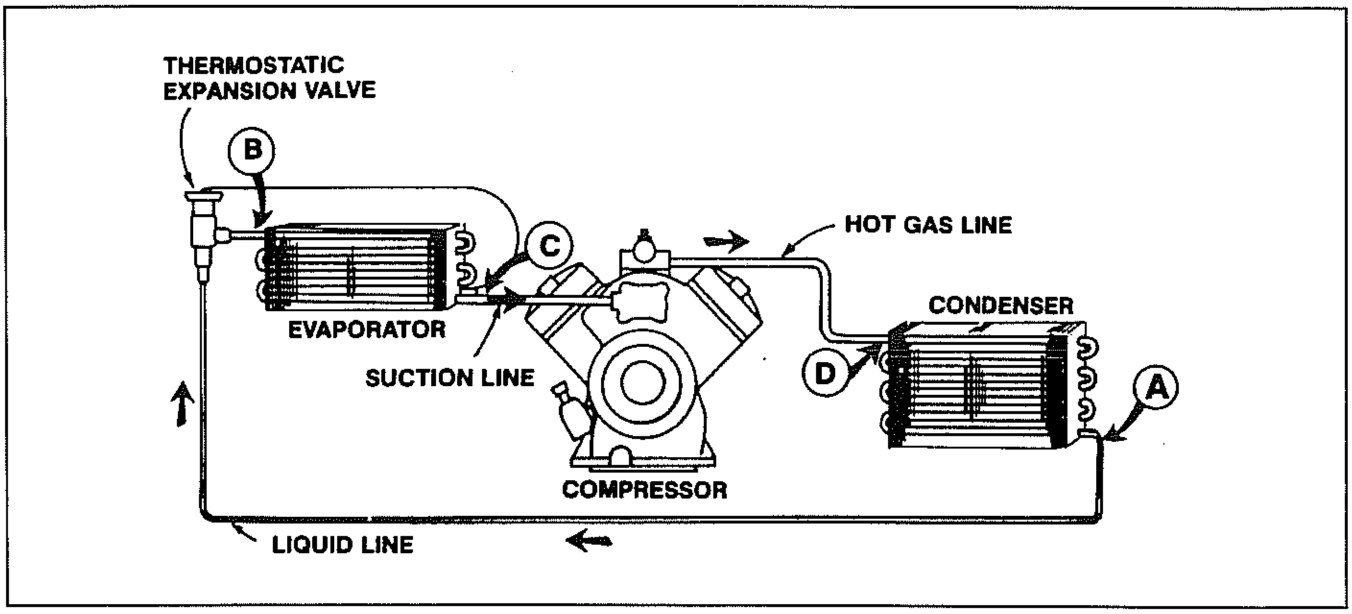


Fig. 23

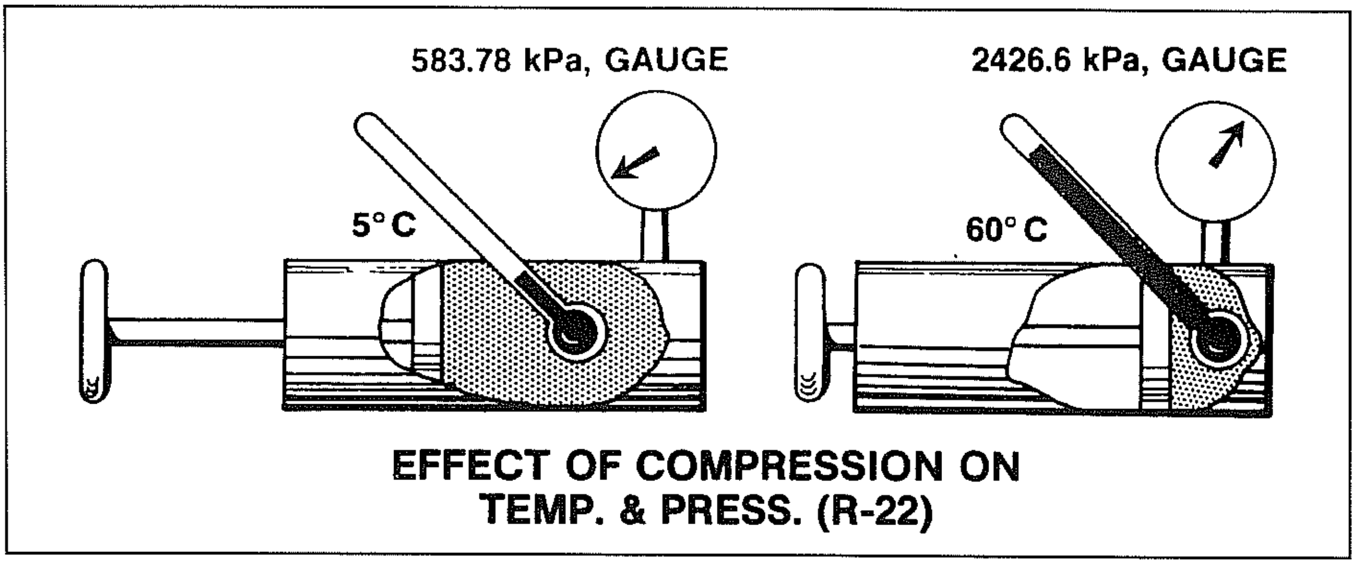


Fig. 24

If the compressor withdraws vapour faster than vapour is created by the load on the evaporator, the pressure and hence temperature in the evaporator goes down. If the load increases and boils off refrigerant faster than the compressor withdraws it, the pressure and temperature rise. Thus, a refrigeration cycle 'rides with the load', until compressor capacity and evaporator load reach a balance.

To convert the refrigerant to a liquid so that it may be used again, it must be condensed by removing heat from it. To do this, some medium to which the heat can flow must be used. To affect heat flow, the temperature of the refrigerant must be higher than the temperature of the medium. A second function of the compressor is to compress the refrigerant, which raises its temperature, and permits the condensing medium to cool the refrigerant and condense it to a liquid state.

Work of compression

Refrigerant leaves the evaporator and enters the compressor as saturated vapour. The refrigerant takes on heat during compression, and leaves the compressor in a superheated state with more energy, and hence more heat than it had when it entered the compressor. This increase in heat is provided by the compressor motor and is referred to as the 'heat of compression', and is equivalent to the work done on the vapour as it is being compressed.

The heat of compression varies only slightly with the different refrigerants, but varies greatly with different levels of temperature and pressure between which the cycle is operating. For example, at temperature levels of comfort air conditioning, namely 5°C in the evaporator and the 40°C commonly associated with water-cooled condensers, this work of compression amounts to approximately 1.0 kW input per 5 kW of refrigeration capacity. For a frozen food cabinet, on the other hand, with -23°C in the evaporator and 49°C in an air-cooled condenser, the compressor would require approximately 1.0 kW input for 1.5 kW of capacity.

The condenser

In the condenser, heat is removed from the refrigerant by providing a medium a lower temperature to which the heat can flow and be dissipated. the condenser removes heat taken on by the refrigerant in the evaporator, and it also removes the heat pumped into the refrigerant during the compression process.

The first step in the heat removal process lowers the temperature of the superheated refrigerant until it reaches the saturation point at the pressure existing in the condenser, as from Point D to E in Figure 25. Further removal of heat then effects a change of state and condenses the refrigerant to a liquid, as from Point E to A, so that it can be used over again in the cycle.

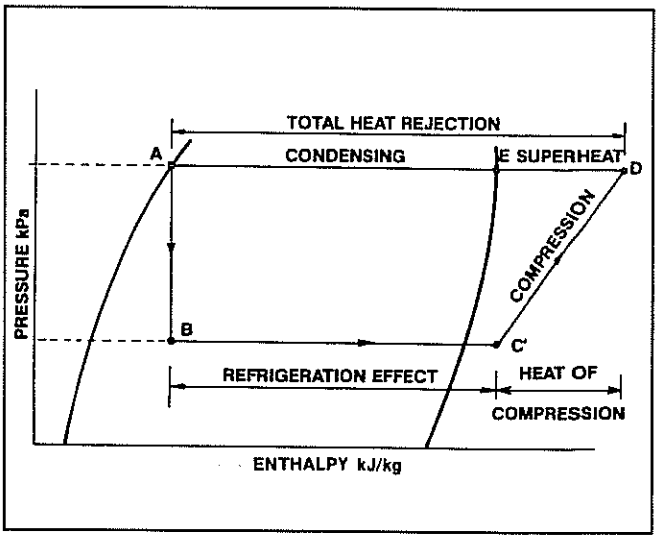


Fig. 25

Condensing temperature

The temperature of the medium (air or water) to which heat flows in the condenser establishes the temperature at which condensation takes place. As it takes on heat from the refrigerant, the temperature of the condensing medium will rise. To maintain the flow of heat, the temperature of the refrigerant in the condenser must always be above the temperature of the medium.

With water-cooled condensers, the refrigerant is in the heat exchanger shell, and there is little variation in pressure and saturated condensing temperature throughout the shell. The difference between the saturated condensing temperature and the temperature of the condensing medium as it leaves the condenser is referred to as the leaving difference (see Figure 26 for illustration of these temperature relationships).

In an air-cooled condenser, the refrigerant passes back and forth through the coil several times; and to accomplish flow, there must be a pressure difference, or pressure drop. In most condensers, this will be approximately 70 kPa or less (see Figure 27 for illustration of the pressure drop on a pressure-enthalpy diagram).

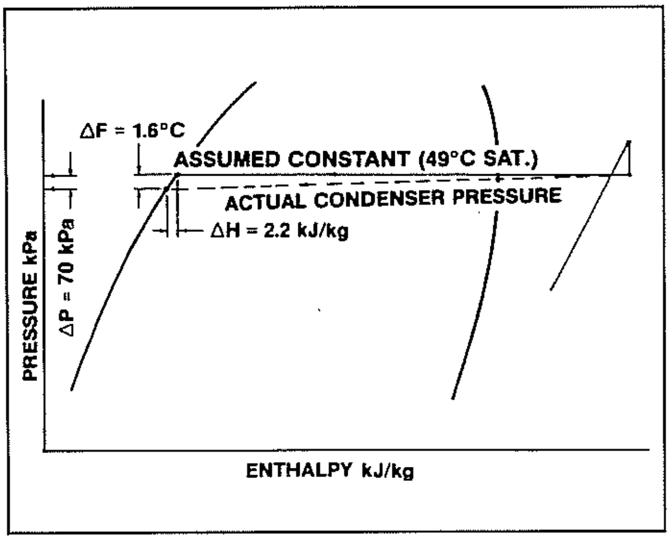


Fig. 27

Desuperheating represents between 10 and 20% of the total heat removal in the condenser; and since the density of the vapour is light, the pressure drop in this section will be small. In a strict mathematical analysis of a cycle in which the refrigerant leaves as saturated liquid, it would be necessary to establish the amount of pressure drop and the pressure at the leaving end of the condenser.

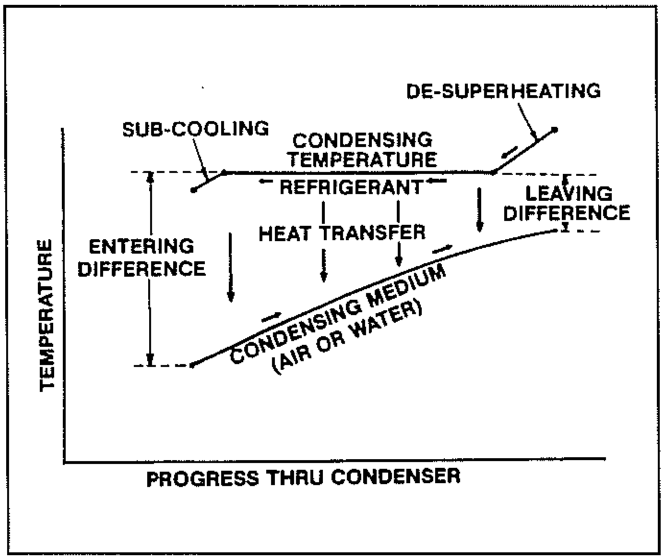


Fig. 26

However, a drop of 70 kPa when using R-22 entering at a saturation temperature of 50°C represents a reduction in saturation temperature of only 1.6°C, and a change in the enthalpy of saturated liquid of only 2.2 kJ/kg. Therefore, it is customary to refer to the condensing temperature as the saturation temperature at the pressure entering the condenser, and to neglect the pressure drop in a cycle analysis.

Subcooling the liquid

Most condensers are arranged so as to provide some subcooling of the liquid refrigerant as it leaves the condenser. If this subcooling is retained until the liquid reaches the metering device (such as the expansion valve), then it reduces the per cent flash gas and increases the refrigeration effect of each kilogram of refrigerant circulated.

Fig. 28 - Properties of refrigerant 22, liquid and saturated vapour

Temp. °C	Pressure kPa	Volume $m^3/kg\ 10^3$		Density $1/m^3/kg\ 10^3$		Enthalpy kJ/kg			Temp. °C
		Liquid v_f	Vapour v_g	Liquid $1/v_f$	Vapour $1/v_g$	Liquid h_f	Latent h_{fg}	Vapour h_g	
0	497.59	0.77834	47.1354	1.28479	0.02122	200.000	205.361	405.361	0
1	514.01	0.78041	45.6757	1.28139	0.02189	201.174	204.550	405.724	1
2	530.83	0.78249	44.2702	1.27797	0.02259	202.351	203.733	406.084	2
3	548.06	0.78460	42.9166	1.27453	0.02330	203.530	202.910	406.440	3
4	565.71	0.78673	41.6124	1.27108	0.02403	204.713	202.080	406.793	4
5	583.78	0.78889	40.3556	1.26760	0.02478	205.899	201.243	407.143	5
6	602.28	0.79107	39.1441	1.26412	0.02555	207.089	200.400	407.489	6
7	621.22	0.79327	37.9759	1.26061	0.02633	208.281	199.550	407.831	7
8	640.59	0.79549	36.8493	1.25708	0.02714	209.477	198.693	408.169	8
9	660.42	0.79775	35.7624	1.25353	0.02796	210.675	197.829	408.504	9
10	680.70	0.80002	34.7136	1.24997	0.02881	211.877	196.958	408.835	10
11	701.44	0.80232	33.7013	1.24638	0.02967	213.083	196.079	409.162	11
12	722.65	0.80465	32.7239	1.24277	0.03056	214.291	195.194	409.485	12
13	744.33	0.80701	31.7801	1.23915	0.03147	215.503	194.301	409.804	13
14	766.50	0.80939	30.8683	1.23550	0.03240	216.719	193.400	410.119	14
15	789.15	0.81180	29.9874	1.23183	0.03335	217.937	192.492	410.430	15
16	812.29	0.81424	29.1361	1.22813	0.03432	219.160	191.577	410.736	16
17	835.93	0.81671	28.3131	1.22442	0.03532	220.385	190.653	411.038	17
18	860.08	0.81922	27.5173	1.22068	0.03634	221.615	189.721	411.336	18
19	884.75	0.82175	26.7477	1.21692	0.03739	222.848	188.782	411.629	19
20	909.93	0.82431	26.0032	1.21313	0.03846	224.084	187.834	411.918	20
21	935.64	0.82691	25.2829	1.20932	0.03955	225.324	186.877	412.202	21
22	961.89	0.82954	24.5857	1.20548	0.04067	226.568	185.913	412.481	22
23	988.67	0.83221	23.9107	1.20162	0.04182	227.816	184.939	412.755	23
24	1016.0	0.83491	23.2572	1.19773	0.04300	229.068	183.957	413.025	24
25	1043.9	0.83765	22.6242	1.19382	0.04420	230.324	182.965	413.289	25
26	1072.3	0.84043	22.0111	1.18987	0.04543	231.583	181.965	413.548	26
27	1101.4	0.84324	21.4169	1.18590	0.04669	232.847	180.955	413.802	27
28	1130.9	0.84610	20.8411	1.18190	0.04798	234.115	179.935	414.050	28
29	1161.1	0.84899	20.2829	1.17787	0.04930	235.387	178.906	414.293	29
30	1191.9	0.85193	19.7417	1.17381	0.05065	236.664	177.867	414.530	30
31	1223.2	0.85491	19.2168	1.16971	0.05204	237.944	176.817	414.762	31
32	1255.2	0.85793	18.7076	1.16559	0.05345	239.230	175.758	414.987	32
33	1287.8	0.86101	18.2135	1.16143	0.05490	240.520	174.687	415.207	33
34	1321.0	0.86412	17.7341	1.15724	0.05639	241.814	173.606	415.420	34
35	1354.8	0.86729	17.2686	1.15301	0.05791	243.114	172.514	415.627	35
36	1389.2	0.87051	16.8168	1.14875	0.05946	244.418	171.410	415.828	36
37	1424.3	0.87378	16.3779	1.14445	0.06106	245.727	170.294	416.021	37
38	1460.1	0.87710	15.9517	1.14012	0.06269	247.041	169.167	416.208	38
39	1496.5	0.88048	15.5375	1.13574	0.06436	248.361	168.027	416.388	39
40	1533.5	0.88392	15.1351	1.13133	0.06607	249.686	166.875	416.561	40
41	1571.2	0.88741	14.7439	1.12687	0.06782	251.016	165.710	416.726	41
42	1609.6	0.89097	14.3636	1.12237	0.06962	252.352	164.531	416.883	42
43	1648.7	0.89459	13.9938	1.11783	0.07146	253.694	163.339	417.033	43
44	1688.5	0.89823	13.6341	1.11324	0.07335	255.042	162.133	417.174	44
45	1729.0	0.90203	13.2841	1.10861	0.07528	256.396	160.912	417.308	45
46	1770.2	0.90586	12.9436	1.10392	0.07726	257.756	159.676	417.432	46
47	1812.1	0.90976	12.6122	1.09919	0.07929	259.123	158.425	417.548	47
48	1854.8	0.91374	12.2895	1.09441	0.08137	260.497	157.158	417.655	48
49	1898.2	0.91779	11.9753	1.08957	0.08351	261.877	155.875	417.752	49
50	1942.3	0.92193	11.6693	1.08468	0.08570	263.264	154.575	417.839	50
51	1987.2	0.92616	11.3711	1.07973	0.08794	264.659	153.257	417.916	51
52	2032.8	0.93047	11.0806	1.07472	0.09025	266.062	151.921	417.983	52
53	2079.3	0.93488	10.7975	1.06965	0.09261	267.472	150.566	418.039	53
54	2126.5	0.93939	10.5214	1.06452	0.09504	268.891	149.192	418.083	54
55	2174.4	0.94400	10.2521	2.05932	0.09754	270.318	147.798	418.116	55
56	2223.2	0.94872	9.98952	1.05405	0.10010	271.754	146.383	418.137	56
57	2272.8	0.95355	9.73328	1.04871	0.10274	273.199	144.946	418.145	57
58	2323.2	0.95850	9.48319	1.04330	0.10545	274.654	143.486	418.141	58
59	2374.5	0.96357	9.23904	1.03781	0.10824	276.119	142.003	418.122	59
60	2426.6	0.96878	9.00062	1.03223	0.11110	277.594	140.495	418.089	60
61	2479.5	0.97412	8.76773	1.02657	0.11405	279.080	138.962	418.041	61
62	2533.3	0.97960	8.54016	1.02082	0.11709	280.577	137.401	417.978	62
63	2587.9	0.98524	8.31772	1.01498	0.12023	282.086	135.813	417.898	63
64	2643.5	0.99104	8.10023	1.00904	0.12345	283.607	134.194	417.802	64

REFRIGERANT DATA

Refrigerant tables

For a particular refrigerant, either a saturated refrigerant temperature or a pressure in kPa absolute will establish a point on the saturated liquid line and/or the saturated vapour line. Therefore, corresponding data for saturated liquid or vapour can be presented in tabular form.

Figure 28 shows such a table for R-22. The first column is the saturation temperature, the second is the corresponding absolute pressure.

The two columns titled 'volume' are values of specific volume in $m^3/kg \times 10^3$.

'Density' is usually applied to refrigerant liquid, and is given in kg/m^3. It would be useful in determining the weight of refrigerant in a liquid line, or the pressure represented by a column of liquid.

'Enthalpy' (or heat content) in kilojoules per kilogram is given for saturated liquid (h_f) and for saturated vapour (h_g), as well as the difference between them which is the heat of vaporization and/or condensation (h_{fg}).

Refrigerant charts

A single value of either saturation temperature or absolute pressure will locate values on the saturation lines, and hence can be conveniently presented in tables.

Data in the superheat region, however, must be established by two values and this can be done on a standard pressure-enthalpy chart or in tables of superheated vapour properties.

Figure 29 shows a pressure-enthalpy diagram for R-22. Similar charts are available for other refrigerants.

The horizontal line or pressure level can be located either by use of the absolute pressure scale on the left, or by the corresponding saturation temperature on either the saturated liquid or saturated vapour lines. The horizontal lines or pressure levels carry straight across the entire chart, through the values at saturation, on across the superheat region.

Enthalpy of saturated liquid or vapour can be read on the enthalpy scale along the bottom, but such values at saturation are more readily obtained from saturation tables.

Liquid to the left of the saturated liquid line is subcooled liquid, and there are no temperature lines in this region. However, within a limited range of pressures, the specific heat of liquid is constant, and lines of constant temperature are vertical.

For example, liquid at 35°C would have the same enthalpy whether it is saturated at 35°C, or whether it were liquid at a pressure corresponding to 40°C but subcooled to 35°C. Properties of subcooled liquid can be found in the tables by reference to data at saturation for that liquid temperature.

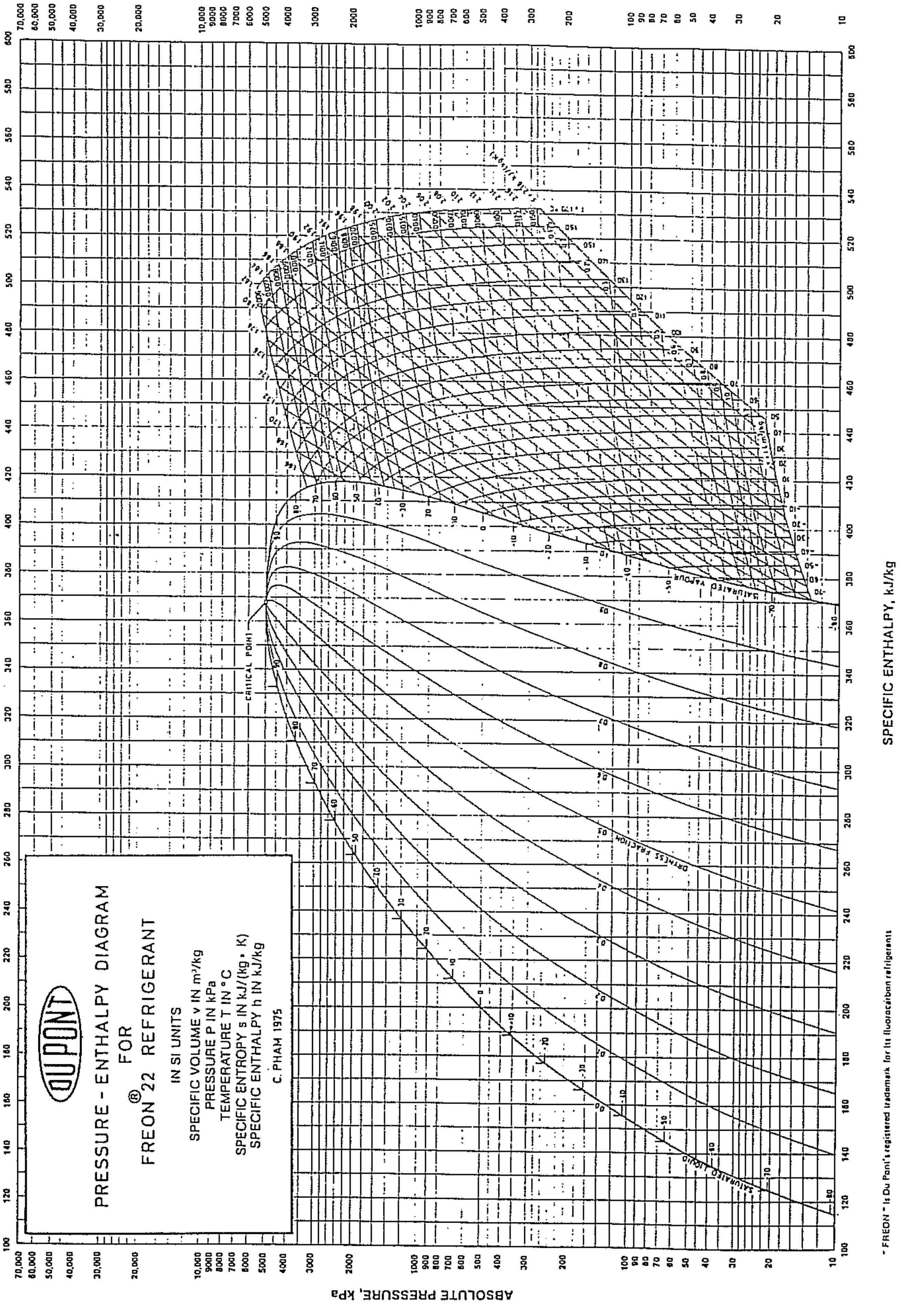
Lines of constant temperature in the superheat region slope to the right and down from each saturation temperature. It should be noted that a 50°C temperature line (for example) starts at 50°C saturated vapour, and also applies for gas at a pressure of 40°C saturated and superheated by 10°C to an actual temperature of 50°C, or at pressure of 20°C saturated and superheated by 30°C to gas temperature of 50°C.

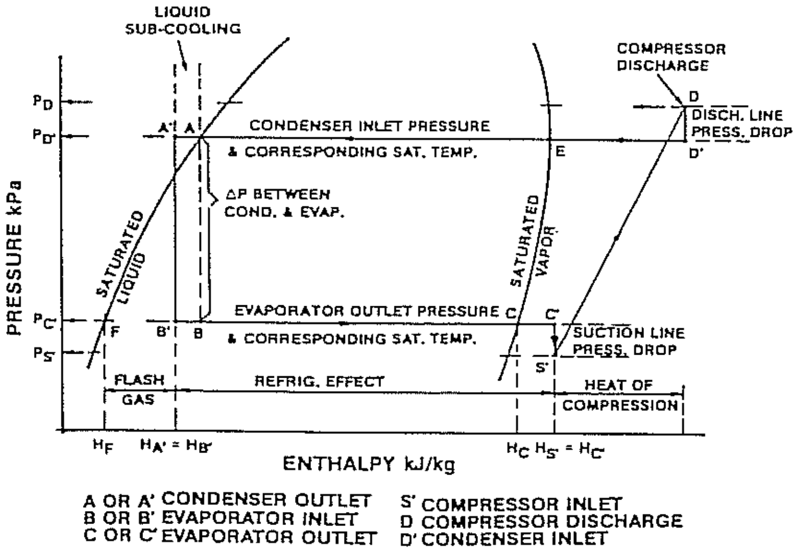
The horizontal line or pressure level can be located either by use of the absolute pressure scale on the left, or by the corresponding saturation temperature on either the saturated liquid or saturated vapour lines. The horizontal lines or pressure levels carry straight across the entire chart, through the values at saturation, on across the superheat region.

Specific volume lines extend off to the right from the saturated vapour line, and their numerical values are over close to the right hand margin. There is only a small change in specific volume with a change in temperature, but values of specific volume vary rapidly as pressure changes.

Lines of constant entropy slope upward to the right, and can be used to follow the theoretical path of compression in a 'perfect' compressor.

Fig. 29


DUPONT
PRESSURE - ENTHALPY DIAGRAM
FOR
FREON® 22 REFRIGERANT
IN SI UNITS
SPECIFIC VOLUME v IN m³/kg
PRESSURE P IN kPa
TEMPERATURE T IN °C
SPECIFIC ENTROPY s IN kJ/(kg • K)
SPECIFIC ENTHALPY h IN kJ/kg
C. PHAM 1975
ABSOLUTE PRESSURE, kPa
SPECIFIC ENTHALPY, kJ/kg
CRITICAL POINT
SATURATED LIQUID
SATURATED VAPOUR
DRYNESS FRACTION
" FREON " is Du Pont's registered trademark for its fluorocarbon refrigerants



IDEALIZED REFRIG. CYCLE ON PRESSURE — ENTHALPY DIAGRAM

Fig. 30

Refrigerant cycle diagram

A plot of the refrigeration cycle on a pressure-enthalpy diagram is the best means of describing and discussing each part or function of the cycle. It also services to identify values with which to make a mathematical analysis of the cycle. The following pages will do both, review and/or discuss each part, and apply data for typical operating conditions to each function.

The cycle is presented in Figure 30 with descriptive identifications. The same point designations will be used in the following analysis. The analysis assumes that the refrigeration load and evaporator ratings require a saturated refrigerant temperature of 5°C, and that the condensing medium and the condenser ratings are such that the saturated condensing temperature will be 50°C. The refrigerant will be R-22.

Corresponding pressures

In the air conditioning and refrigeration industries load requirements (such as chilled water temperature) and the condensing medium (such as outdoor air or cooling tower water) are identified with temperatures. Hence, it is convenient to express the performance of the evaporator and the condenser with required saturation temperatures, rather than in terms of pressure.

The table of refrigerant properties at saturation can be used to convert these saturation temperatures to corresponding pressures.

R-22

50°C saturation temperature	=	1942.3 kPa absolute
5°C saturation temperature	=	583.78 kPa absolute

Line losses

The connecting pipe between the condenser and the evaporator inlet carries liquid refrigerant, and is identified as the liquid line. The pipe from the evaporator outlet to the compressor inlet carries refrigerant vapour, and is called the suction line. The pipe from the compressor discharge to the condenser inlet carries superheated vapour, and is called the hot gas line.

Each of the lines requires a pressure differential, or pressure drop, to maintain flow. This pressure drop is an arbitrary amount used to establish a corresponding size of the line for the required flow.

Because the conditions at the inlet and outlet of these lines are usually expressed as saturation temperatures, it is convenient to express the pressure drop through the line as a corresponding change or drop in saturation temperature. The general practice at air conditioning levels of temperature is to size each line for a pressure drop corresponding to a change in saturation temperature of 1 K.

Liquid line			
Saturated condensing temperature	50°C	=	1942 kPa, absolute
Liquid line drop	1 K	=	44 kPa, absolute
Saturation temperature at inlet to metering device	49°C	=	1898 kPa, absolute
Suction line			
Saturated refrigerant temperature	5°C	=	584 kPa, absolute
Suction line drop	1 K	=	18 kPa, absolute
Saturated suction temperature at compressor inlet	4°C	=	566 kPa, absolute
Hot gas line			
Saturated condensing temperature	50°C	=	1942 kPa, absolute
Hot gas line drop	1 K	=	45 kPa, absolute
Saturation temperature at compressor discharge	51°C	=	1987 kPa, absolute

It should be noted that refrigerant flow is from the compressor to the condenser, and hence the loss or drop in the hot gas line is added to the saturated condensing temperature to determine saturation temperature at compressor discharge.

Also, it is seen that the same loss expressed as 1 K change in saturation temperature does not create equal changes in actual pressure. The smallest actual change is in the suction line, and this is fortunate, in that the cycle capacity is more sensitive to pressure change at the compressor inlet than it is to pressure change at other parts in the cycle.

CYCLE ANALYSIS

Per cent flash gas (Points B, B' and F)

Per cent flash gas can be defined as that part of the refrigerant which must be evaporated at the evaporator (low) pressure to cool the remaining liquid from its temperature at the condensing (high) pressure to the saturation temperature corresponding to the low pressure.

With no subcooling of the liquid:

$$\% \text{ flash gas} = \frac{H_A - H_F}{H_C - H_F}$$

H_A = enthalpy of saturated liquid at 50°C = 263 kPa

H_F = enthalpy of saturated liquid at 5°C = 206 kPa

H_C = enthalpy of saturated vapour at 5°C = 407 kPa

$$\% \text{ flash gas} \quad \frac{263 - 206}{407 - 206} = \frac{57}{201} = 28.3\%$$

Subcooling the liquid (Points A and A')

The saturation temperature at the condenser was given as 50°C, and if the liquid were subcooled by 8°C, it would bring the refrigerant temperature down to 42°C. No data is given for enthalpies in the subcooled region. However, earlier in this text, it was pointed out that the temperature lines in the subcooled area are vertical because the specific heat of the liquid is assumed constant within a limited range of pressure.

Hence, the enthalpy of liquid at a pressure corresponding to 50°C saturated and subcooled to 42°C will be equal to that of saturated liquid at 42°C, and the latter is available from the tables of refrigerant properties. The value of the enthalpy of saturated liquid at 42°C = 252 kJ/kg.

$$\% \text{ flash gas} \quad \frac{252 - 206}{407 - 206} = \frac{46}{201} = 22.9\%$$

The above examples show one of the benefits of subcooling, i.e., a reduction in the percentage of flash gas. This means that a greater percentage of the refrigerant will reach the evaporator as a liquid, where its change of state to a vapour will be accomplished by heat from the refrigeration load.

In a water-cooled condenser, a small amount of subcooling, approximately 2.8 K is obtained by circuiting the water so that the cooler inlet water flows through the tubes at the bottom of the condenser, where they are immersed in the liquid refrigerant. Some have a separate subcooling circuit and provide as much as 8 K of subcooling.

In an air-cooled condenser, subcooling can be obtained by over-sizing the condenser and 'over-charging' the system with refrigerant. This backs-up liquid refrigerant in the condenser tubes at the bottom of the condenser, so that the incoming air can cool the accumulated liquid to a temperature below that at which it condensed.

This decreases the ΔT or leaving difference, and produces a saturated liquid at a lower saturation temperature and pressure. However, the condenser surface required to do this is larger and more expensive than the surface required in a smaller condenser.

Evaporator performance (Points B', B, C and C')

As pointed out earlier in this text, it is assumed that the enthalpy of refrigerant entering the evaporator is the same as that leaving the condenser. This is a reasonable assumption, since passage through the metering device is short and quick, and the loss of heat from the liquid line will be a minor amount.

The enthalpies at B or B' are not known, except that they are equal to the enthalpies at A and A' respectively. Hence, the enthalpy entering the evaporator is found by establishing the enthalpy leaving the condenser.

The heat picked up in the evaporator from the refrigerant load is identified as refrigeration effect and is the difference between the enthalpy entering and the enthalpy leaving.

The most effective evaporator would be one from which the refrigerant leaves as a saturated refrigerant - where the heat picked up from the load was all used in evaporating the balance of the refrigerant, changing its state. It is important to make certain that only refrigerant vapour goes to the compressor, as slugs of liquid refrigerant can cause damage, and even destroy a compressor. To ensure that only gas will reach the compressor the suction gas is superheated as it leaves the evaporator by locating the expansion valve sensing bulb at the evaporator outlet.

Thermal expansion valve, 6°C superheat, no subcooling

(TX valve - Points A and B, superheat - Points C and C')

With 6°C superheat, the enthalpy value of $H_{C'}$ must be read from the refrigerant chart, not the table. The saturation temperature of 5°C establishes the pressure level, and the horizontal line at this pressure is followed to the right until it intersects the 11°C temperature line (5°C saturated plus 6°C superheat). The enthalpy at this point is read from the bottom scale at approximately 411 kJ/kg.

Refrigeration effect = $H_{C'} - H_B$

= 411 - 263 = 148 kJ/kg

Thermal expansion valve, 6°C superheat, 8 K subcooling

(TX valve - Points A' and B', superheat - Points C and C')

Refrigeration effect = HC' - HB

= 411 - 252 = 159 kJ/kg

Suction line to compressor inlet (Points C' and S')

The preceding discussion has established the pressure and temperature, and hence enthalpy of the refrigerant leaving the evaporator (Point C or C'). Conditions at the compressor inlet will depend on the path taken by the line representing the suction line while undergoing its pressure drop, 1 K change in saturation temperature. At air conditioning levels of temperature with refrigerant leaving the evaporator between 5°C and 10°C saturated, plus 6°C superheat provided by an expansion valve, the actual gas temperatures in the suction line will be in the neighbourhood of 11°C to 16°C. Whether the refrigerant takes on some heat from or gives off some heat to the ambient air depends on the ambient temperature, and is normally ignored.

If the gas temperature is below the ambient dewpoint, the suction line will likely be insulated to prevent drippage from condensation. If the line passes through a warm equipment room, it may be insulated to prevent its taking on heat, but the transfer surface is relatively small, and there is no forced air motion to promote heat transfer.

In any case, the amount of heat transfer will be small; and for purpose of cycle analysis, it is reasonable to assume that the enthalpy will remain the same. In other words, $H_S = H_C$ and $H_{S'} = H_{C'}$.

Compressor inlet conditions (Point S')

Data from the preceding paragraphs have now fixed the conditions at the compressor inlet. The saturation temperature (hence, pressure) and its enthalpy have been established. These two values fix a point on the refrigerant chart from which other properties can be read.

A thermal expansion valve is a device which senses the amount of superheat in the refrigerant leaving the evaporator, and uses this superheat to control the supply of liquid refrigerant into the evaporator. The amount of superheat is usually between 4°C and 8°C, generally taken as 6°C.

A thermal expansion valve is a compact, simple and relatively inexpensive metering device. Hence, its broad usage. However, it requires additional evaporator surface through which to obtain the heat exchange to produce the superheat needed to ensure that liquid does not reach the compressor, perhaps 7% of the evaporator surface. This sacrifice is made to enable the use of thermal expansion valves.

Compression ratio

In a refrigeration compressor, there are no cam shafts and valve lifters that open and close the valves at particular points in the stroke and hence establish a fixed compression ratio. Instead, the discharge and suction valves of a refrigeration compressor are spring loaded, so that the discharge valves open when the compression stroke has raised the cylinder pressure enough above the pressure in the discharge line to overcome the spring pressure. Conversely, the suction valves open when the suction stroke has lowered the pressure in the cylinder to something less than the pressure in the suction line.

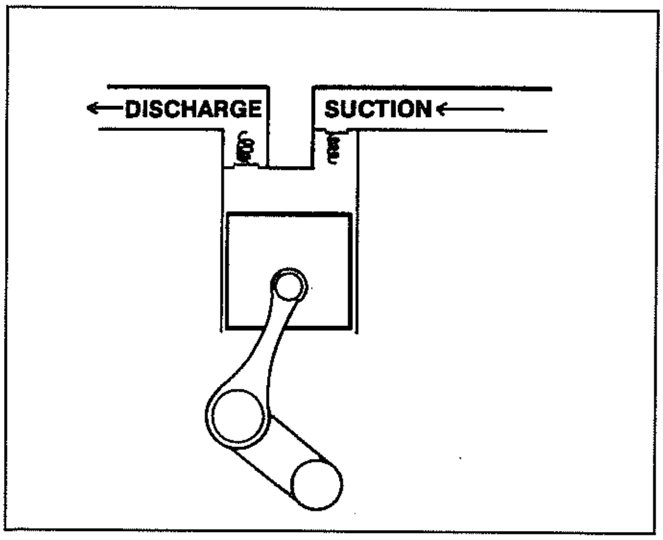


Fig. 32

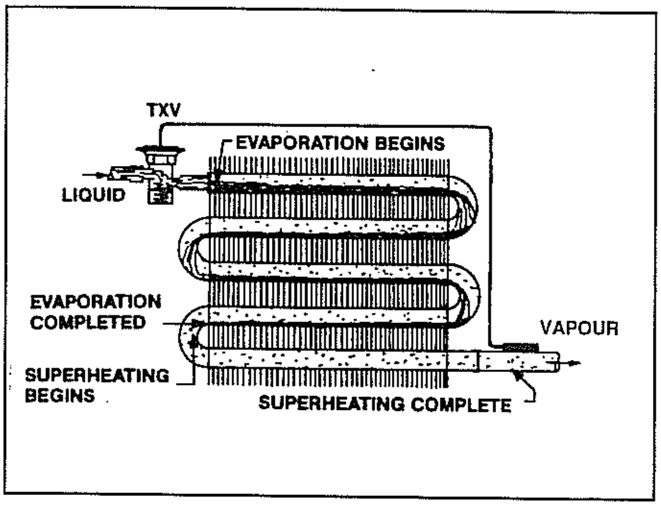


Fig. 31

Therefore, a refrigeration compressor 'rides with load', and its compression ratio depends on the conditions at which it is operating.

$$\text{Compression ratio} = \frac{\text{Pressure at discharge in kPa, absolute}}{\text{Pressure at inlet in kPa, absolute}}$$

$$= \frac{\text{Pressure at D}}{\text{Pressure at S'}} \quad \begin{matrix} = \\ = \end{matrix} \quad \frac{51°\text{C saturated}}{4°\text{C}}$$

$$= \frac{1987}{566} \quad = \quad \frac{3.5}{1}$$

A compressor is tested in the laboratory with the discharge identified at a point downstream of the discharge shut-off valve and the inlet at a point upstream of the inlet shut-off valve. Therefore, the pressure losses through the shut-off valves and through the spring loaded discharge and suction valves are a 'part of the compressor'. The compression ratio calculated above is the 'required' compression ratio, and assumes the compressor selected will be in balance with the load under those conditions.

Compressor discharge temperature (Point D)

When gas is compressed, work is done on it. In addition to a pressure increase, its temperature and enthalpy also increase. The increase in pressure and enthalpy caused by the compressor is shown by the line S' - D of Figure 30.

Discharge temperature can be computed from the following formula:

$$T_d = T_s \times (C.R.)^{\frac{n-1}{n}}$$

Where:

T_d = discharge temperature Kelvin
T_s = suction temperature Kelvin
C.R. = compression ratio
n = compression exponent

The performance of an actual compressor can be established only by tests; and during such tests, the discharge temperatures are read over a range of operating conditions. Rather than publish a table of results showing discharge temperatures, the test results are substituted in the above formula, and a value of 'n' is computed.

Heat of compression

With the saturated discharge temperature (i.e., discharge pressure) having been established and the actual gas temperature at discharge having been calculated as above, the compressor discharge at Point D can be fixed on the pressure-enthalpy diagram. The enthalpy at discharge can be found at the bottom horizontal scale.

The heat of compression, or the work done on the gas, is then $H_D - H_{S'}$ assuming the use of a thermostatic expansion valve and superheated vapour leaving the evaporator.

D = 51°C saturated and 89°C actual temperature

H_D = 455 kJ/kg

$$\text{Heat of compression} = H_D - H_{S'} = 455 - 412 = 43 \text{ kJ/kg}$$

This represents an 'actual' heat of compression since it is established by using an 'n' value obtained by tests on actual compressors.

Actual heats of compression are published in manufacturer's ratings, and are usually given as kw input required at the compressor shaft, or as kW input to the motor on hermetic compressors. The latter accounts for efficiency of the electric motor as well as mechanical friction losses in the compressor.

Coefficient of performance

Efficiency is usually thought of as the ratio of output to input (COP). For the refrigeration cycle presented here, this would appear as follows:

$$\text{COP} = \frac{\text{Output}}{\text{Input}} = \frac{\text{Refrigeration effect}}{\text{Heat of compression}}$$

For conditions of thermal expansion valve, 6° superheat, no subcooling

$$\text{COP} = \frac{411 - 263}{455 - 411} = \frac{148}{44} \quad \text{or} \quad 3.36$$

Hot gas line to condenser inlet (Points D and D')

Refrigerant leaves the compressor as a highly superheated vapour. As it passes through the hot gas line, there will likely be some dissipation of heat to the surrounding air. The extent of this loss depends on the length of the hot gas line and the temperature of the ambient air.

For purposes of cycle analysis, it is assumed that the heat loss is negligible, and the refrigerant enters the condenser with an enthalpy equal to that at the compressor discharge.

$$H_{D'} = H_D$$

There will be a slight temperature reduction as the refrigerant vapour 'expands' through the pressure drop in the hot gas line.

	H_D	$H_{D'}$
Enthalpy	455	455 kJ/kg
Saturation temperature	51°C	50°C
Actual gas temperature	89°C	88°C

Heat rejection (Points D', E, A and A')

The condensing medium (air or water) must remove the total heat or energy input. This is made up of the refrigeration effect or heat taken from the refrigeration load <u>plus</u> the work or heat of compression.

Total heat rejection = refrigeration effect and heat of compression

$$= (H_{C'} - H_B) + (H_D - H_{C'})$$

$$= 411 - 263 + 455 - 411$$

$$= 148 + 44 = 192 \text{ kJ/kg}$$

The function of the condenser is first to desuperheat the refrigeration from the condition at D' to that of a saturated vapour at E. The major part of the condenser capacity is used to condense the refrigerant, changing its state back to that of a liquid, shown as Point A on Figure 30.

This takes the refrigerant back to its original state as a liquid to be recycled through the system 'over and over again'.

COMPRESSOR CAPACITY/SYSTEMS CAPACITY

Capacity balance

As pointed out earlier in the text, a refrigeration compressor 'rides with the load'. Because of the valve action, the operating conditions of the compressor depend on those of the system (i.e., the evaporator and condenser pressures). Therefore, compressor capacity and system capacity are equal and synonymous.

If the load changes with no change in compressor displacement (speed or number of active cylinders), the operating conditions or pressure levels will change until the compressor and load are again in balance.

Mechanically, or as a piece of hardware by itself, the size or capacity of a compressor can be expressed only as its displacement. This is the volume 'swept through' by the pistons. It is a product of piston area and stroke, revolutions per second and number of cylinders. The displacement in litres per second is part of the data published by most manufacturers.

THERMAL EXPANSION VALVES

The purpose of a thermal expansion valve (TXV) is to control the flow rate of liquid refrigerant entering the evaporator in proportion to the load on the evaporator. For maximum utilization of evaporator surface, it would be desirable to spread the load uniformly through the entire evaporator, with the refrigerant leaving the evaporator as a saturated vapour. However, to protect the compressor from liquid refrigerant, and to provide a means of operating the TXV, the TXV measures and controls a certain amount of superheat (usually 6°) in the refrigerant vapour leaving the evaporator.

A bulb is strapped to the outside of the suction line leaving the evaporator, and there it senses the actual temperature of the refrigerant. The bulb and its connecting tubing are charged with both vapour and liquid refrigerant, and this charge will come to the temperature of the superheated vapour. This pressure acts on the top of the diaphragm in the TXV and, operating through push rods, serves to open the valve.

Evaporator pressure is applied to the underside of the diaphragm. This pressure is obtained either through an internal passage on the leaving side of the evaporator at a point near that at which the thermal bulb is attached. The latter is used with larger evaporators through which there is appreciable pressure drop, and thus appreciable difference in pressure between the entering and leaving ends. The difference in pressure above and below the diaphragm of the TXV is that between saturation pressure corresponding to the temperature of the superheated refrigerant and the actual pressure of the refrigerant. This difference is opposed by a spring, and setting the spring tension establishes the amount of superheat to which the valve will control.

Thermostatic expansion valves are available with different types of charges, depending on the application on which they are to be used. The most common one for air conditioning applications is that of a charge using the same refrigerant as that in the system, with a limited amount of liquid. This means that the liquid in the bulb will all have evaporated at some predetermined temperature, and above that temperature there will be little rise in the opening pressure.

On shut-down, the evaporator pressure rises. This combines with the spring pressure to close the valve; and with little or no increase in the bulb or opening pressure, the TXV closes and remains closed during shut-down.

On start-up, the evaporator pressure drops, but before the TXV will open, the evaporator must pull-down to a pressure below that of the predetermined temperature of the limited liquid charge. This keeps the valve closed during initial pull-down, minimizes floodback of liquid refrigerant to the compressor, and limits the compressor motor load during pull-down.