Refrigeration and Air-Conditioning

Refrigeration: The process of removing heat.

Air-conditioning: A form of air treatment whereby temperature, humidity, ventilation, and air cleanliness are all controlled within limits determined by the requirements of the air conditioned enclosure.

BS 5643: 1984

Refrigeration and Air-Conditioning

Third edition

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BUTTERWORTH HEINEMANN

OXFORD AUCKLAND BOSTON JOHANNESBURG MELBOURNE NEW DELHI

Butterworth-Heinemann
Linacre House, Jordan Hill, Oxford OX2 8DP
225 Wildwood Avenue, Woburn, MA 01801-2041
A division of Reed Educational and Professional Publishing Ltd

A member of the Reed Elsevier plc group

First published by McGraw-Hill Book Company (UK) Ltd 1981
Second edition by Butterworths 1989
Third edition by Butterworth-Heinemann 2000

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British Library Cataloguing in Publication Data
A catalogue record for this book is available from the British Library

Library of Congress Cataloguing in Publication Data
A catalogue record for this book is available from the Library of Congress

ISBN 0 7506 4219 X

Typeset in India at Replika Press Pvt Ltd, Delhi 110 040, India
Printed and bound in Great Britain

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Preface

Refrigeration and its application is met in almost every branch of industry, so that practitioners in other fields find that they have to become aware of its principles, uses and limitations. This book aims to introduce students and professionals in other disciplines to the fundamentals of the subject, without involving the reader too deeply in theory. The subject matter is laid out in logical order and covers the main uses and types of equipment. In the ten years since the last edition there have been major changes in the choice of refrigerants due to environmental factors and an additional chapter is introduced to reflect this. This issue is on-going and new developments will appear over the next ten years. This issue has also affected servicing and maintenance of refrigeration equipment and there is an increased pressure to improve efficiency in the reduction of energy use. This edition reflects these issues, whilst maintaining links with the past for users of existing plant and systems. There have also been changes in packaged air-conditioning equipment and this has been introduced to the relevant sections. The book gives worked examples of many practical applications and shows options that are available for the solution of problems in mechanical cooling systems. It is not possible for these pages to contain enough information to design a complete refrigeration system. The design principles are outlined. Finally, the author wishes to acknowledge help and guidance from colleagues in the industry, in particular to Bitzer for the information on new refrigerants.

T.C. Welch
October 1999

1 Fundamentals

1.1 Basic physics – temperature

The general temperature scale now in use is the *Celsius scale*, based nominally on the melting point of ice at 0°C and the boiling point of water at atmospheric pressure at 100°C. (By strict definition, the triple point of ice is 0.01°C at a pressure of 6.1 mbar.) On the Celsius scale, absolute zero is – 273.15°C.

In the study of refrigeration, the *Kelvin* or *absolute temperature scale* is also used. This starts at absolute zero and has the same degree intervals as the Celsius scale, so that ice melts at + 273.16 K and water at atmospheric pressure boils at + 373.15 K.

1.2 Heat

Refrigeration is the process of removing heat, and the practical application is to produce or maintain temperatures below the ambient. The basic principles are those of thermodynamics, and these principles as relevant to the general uses of refrigeration are outlined in this opening chapter.

Heat is one of the many forms of energy and mainly arises from chemical sources. The heat of a body is its thermal or internal energy, and a change in this energy may show as a change of temperature or a change between the solid, liquid and gaseous states.

Matter may also have other forms of energy, potential or kinetic, depending on pressure, position and movement. Enthalpy is the sum of its internal energy and flow work and is given by:

$$H = u + Pv$$

In the process where there is steady flow, the factor Pv will not

change appreciably and the difference in enthalpy will be the quantity of heat gained or lost.

Enthalpy may be expressed as a total above absolute zero, or any other base which is convenient. Tabulated enthalpies found in reference works are often shown above a base temperature of – 40°C, since this is also – 40° on the old Fahrenheit scale. In any calculation, this base condition should always be checked to avoid the errors which will arise if two different bases are used.

If a change of enthalpy can be sensed as a change of temperature, it is called *sensible heat*. This is expressed as specific heat capacity, i.e. the change in enthalpy per degree of temperature change, in kJ/(kg K). If there is no change of temperature but a change of state (solid to liquid, liquid to gas, or vice versa) it is called *latent heat*. This is expressed as kJ/kg but it varies with the boiling temperature, and so is usually qualified by this condition. The resulting total changes can be shown on a temperature–enthalpy diagram (Figure 1.1).

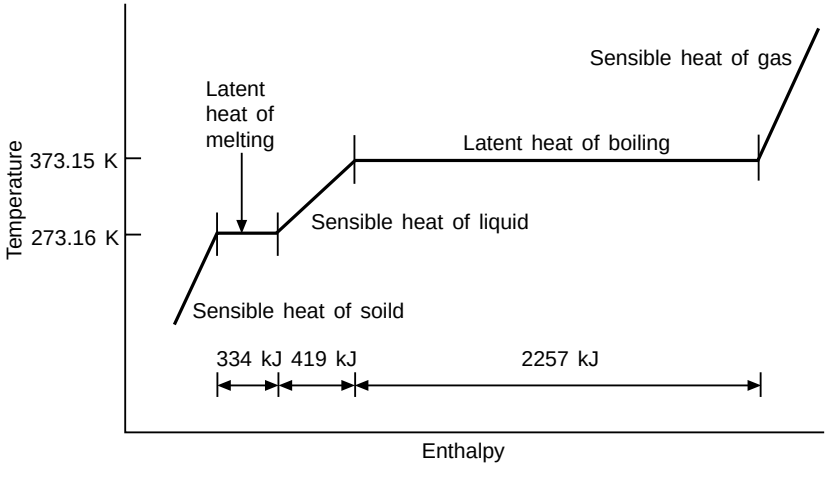


Figure 1.1 *Change of temperature (K) and state of water with enthalpy*

Example 1.1 For water, the latent heat of freezing is 334 kJ/kg and the specific heat capacity averages 4.19 kJ/(kg K). The quantity of heat to be removed from 1 kg of water at 30°C in order to turn it into ice at 0°C is:

$$4.19(30 - 0) + 334 = 459.7 \text{ kJ}$$

Example 1.2 If the latent heat of boiling water at 1.013 bar is 2257 kJ/kg, the quantity of heat which must be added to 1 kg of water at 30°C in order to boil it is:

$4.19(100 - 30) + 2257 = 2550.3$ kJ

Example 1.3 The specific enthalpy of water at 80°C, taken from 0°C base, is 334.91 kJ/kg. What is the average specific heat capacity through the range 0–80°C?

$334.91/(80 - 0) = 4.186$ kJ/(kg K)

1.3 Boiling point

The temperature at which a liquid boils is not constant, but varies with the pressure. Thus, while the boiling point of water is commonly taken as 100°C, this is only true at a pressure of one standard atmosphere (1.013 bar) and, by varying the pressure, the boiling point can be changed (Table 1.1). This pressure–temperature property can be shown graphically (see Figure 1.2).

Table 1.1

Pressure (bar)	*Boiling point* (°C)
0.006	0
0.04	29
0.08	41.5
0.2	60.1
0.5	81.4
1.013	100.0

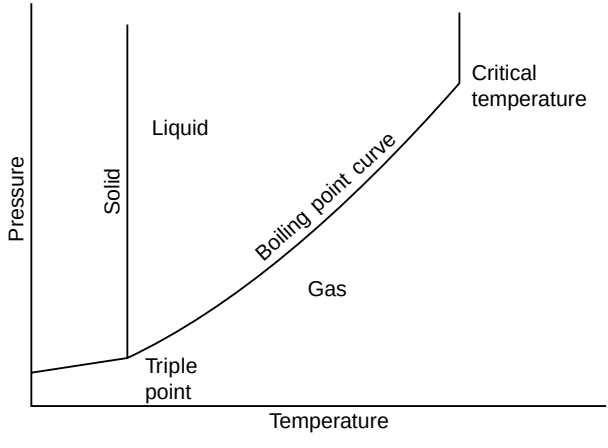


Figure 1.2 *Change of state with pressure and temperature*

The boiling point is limited by the *critical temperature* at the upper end, beyond which it cannot exist as a liquid, and by the *triple point* at the lower end, which is at the freezing temperature. Between these two limits, if the liquid is at a pressure higher than its boiling pressure, it will remain a liquid and will be subcooled below the saturation condition, while if the temperature is higher than saturation, it will be a gas and superheated. If both liquid and vapour are at rest in the same enclosure, and no other volatile substance is present, the condition must lie on the saturation line.

At a pressure below the triple point pressure, the solid can change directly to a gas (sublimation) and the gas can change directly to a solid, as in the formation of carbon dioxide snow from the released gas.

The liquid zone to the left of the boiling point line is subcooled liquid. The gas under this line is superheated gas.

1.4 General gas laws

Many gases at low pressure, i.e. atmospheric pressure and below for water vapour and up to several bar for gases such as nitrogen, oxygen and argon, obey simple relations between their pressure, volume and temperature, with sufficient accuracy for engineering purposes. Such gases are called 'ideal'.

Boyle's Law states that, for an ideal gas, the product of pressure and volume at constant temperature is a constant:

$$pV = \text{constant}$$

Example 1.4 A volume of an ideal gas in a cylinder and at atmospheric pressure is compressed to half the volume at constant temperature. What is the new pressure?

$$p_1V_1 = \text{constant}$$
$$= p_2V_2$$
$$\frac{V_1}{V_2} = 2$$

so $p_2 = 2 \times p_1$

$= 2 \times 1.013\ 25$ bar (101 325 Pa)

$= 2.026\ 5$ bar (abs.)

Charles' Law states that, for an ideal gas, the volume at constant pressure is proportional to the absolute temperature:

$$\frac{V}{T} = \text{constant}$$

Example 1.5 A mass of an ideal gas occupies 0.75 m^3 at 20°C and is heated at constant pressure to 90°C. What is the final volume?

$$V_2 = V_1 \times \frac{T_2}{T_1}$$

$$= 0.75 \times \frac{273.15 + 90}{273.15 + 20}$$

$$= 0.93 \text{ m}^3$$

Boyle's and Charles' laws can be combined into the ideal gas equation:

$pV = (\text{a constant}) \times T$

The constant is mass $\times R$, where R is the specific gas constant, so:

$pV = mRT$

Example 1.6 What is the volume of 5 kg of an ideal gas, having a specific gas constant of 287 J/(kg K), at a pressure of one standard atmosphere and at 25°C?

$pV = mRT$

$$V = \frac{mRT}{p}$$

$$= \frac{5 \times 287(273.15 + 25)}{101\,325}$$

$$= 4.22 \text{ m}^3$$

1.5 Dalton's law

Dalton's Law of partial pressures considers a mixture of two or more gases, and states that the total pressure of the mixture is equal to the sum of the individual pressures, if each gas separately occupied the space.

Example 1.7 A cubic metre of air contains 0.906 kg of nitrogen of specific gas constant 297 J/(kg K), 0.278 kg of oxygen of specific gas constant 260 J/(kg K) and 0.015 kg of argon of specific gas constant 208 J/(kg K). What will be the total pressure at 20°C?

$pV = mRT$

$V = 1\ \text{m}^3$

so $p = mRT$

For the nitrogen $p_N = 0.906 \times 297 \times 293.15 =$ 78 881 Pa
For the oxygen $p_O = 0.278 \times 260 \times 293.15 =$ 21 189 Pa
For the argon $p_A = 0.015 \times 208 \times 293.15 =$ 915 Pa

Total pressure = 100 985 Pa
(1.009 85 bar)

1.6 Heat transfer

Heat will move from a hot body to a colder one, and can do so by the following methods:

1. *Conduction*. Direct from one body touching the other, or through a continuous mass
2. *Convection*. By means of a heat-carrying fluid moving between one and the other
3. *Radiation*. Mainly by infrared waves (but also in the visible band, e.g. solar radiation), which are independent of contact or an intermediate fluid.

Conduction through a homogeneous material is expressed directly by its area, thickness and a conduction coefficient. For a large plane surface, ignoring heat transfer near the edges:

$$\text{Conductance} = \frac{\text{area} \times \text{thermal conductivity}}{\text{thickness}}$$

$$= \frac{A \times k}{L}$$

and the heat conducted is

$$Q_f = \text{conductance} \times (T_1 - T_2)$$

Example 1.8 A brick wall, 225 mm thick and having a thermal conductivity of 0.60 W/(m K), measures 10 m long by 3 m high, and has a temperature difference between the inside and outside faces of 25 K. What is the rate of heat conduction?

$$Q_f = \frac{10 \times 3 \times 0.60 \times 25}{0.225}$$

$= 2000$ W (or 2 kW)

Thermal conductivities, in watts per metre kelvin, for various common materials are as in Table 1.2. Conductivities for other materials can be found from standard reference works [1, 2, 3].

Table 1.2

Material	*Thermal conductivity* (W/(m K))
Copper	200
Mild steel	50
Concrete	1.5
Water	0.62
Cork	0.040
Expanded polystyrene	0.034
Polyurethane foam	0.026
Still air	0.026

Convection requires a fluid, either liquid or gaseous, which is free to move between the hot and cold bodies. This mode of heat transfer is very complex and depends firstly on whether the flow of fluid is 'natural', i.e. caused by thermal currents set up in the fluid as it expands, or 'forced' by fans or pumps. Other parameters are the density, specific heat capacity and viscosity of the fluid and the shape of the interacting surface.

With so many variables, expressions for convective heat flow cannot be as simple as those for conduction. The interpretation of observed data has been made possible by the use of a number of groups which combine the variables and which can then be used to estimate convective heat flow.

The main groups used in such estimates are as shown in Table 1.3.

A typical combination of these numbers is that for turbulent flow in pipes:

$$(\mathrm{Nu}) = 0.023\ (\mathrm{Re})^{0.8}\ (\mathrm{Pr})^{0.4}$$

The calculation of every heat transfer coefficient for a refrigeration or air-conditioning system would be a very time-consuming process, even with modern methods of calculation. Formulas based on these factors will be found in standard reference works, expressed in terms of heat transfer coefficients under different conditions of fluid flow [1, 4–8].

Example 1.9 A formula for the heat transfer coefficient between forced draught air and a vertical plane surface ([1], Chapter 3, Table 6) gives:

$$h' = 5.6 + 18.6V$$

Table 1.3

Number	*Sign*	*Parameters*
Reynolds	Re	Velocity of fluid Density of fluid Viscosity of fluid Dimension of surface
Grashof	Gr	Coefficient of expansion of fluid Density of fluid Viscosity of fluid Force of gravity Temperature difference Dimension of surface
Nusselt	Nu	Thermal conductivity of fluid Dimension of surface Heat transfer coefficient
Prandtl	Pr	Specific heat capacity of fluid Viscosity of fluid Thermal conductivity of fluid

What is the thermal conductance for an air velocity of 3 m/s?

$h' = 5.6 + 18.6 \times 3$

$= 61.4 \text{ W}/(\text{m}^2 \text{ K})$

Where heat is conducted through a plane solid which is between two fluids, there will be the convective resistances at the surfaces. The overall heat transfer must take all of these resistances into account, and the unit transmittance, or 'U' factor, is given by:

$R_t = R_i + R_c + R_o$

$U = 1/R_t$

where R_t = total thermal resistance
R_i = inside convective resistance
R_c = conductive resistance
R_o = outside convective resistance

Example 1.10 A brick wall, plastered on one face, has a thermal conductance of 2.8 $\text{W}/(\text{m}^2 \text{ K})$, an inside surface resistance of 0.3 $(\text{m}^2 \text{ K})/\text{W}$, and an outside surface resistance of 0.05 $(\text{m}^2 \text{ K})/\text{W}$. What is the overall transmittance?

$R_t = R_i + R_c + R_o$

$= 0.3 + \dfrac{1}{2.8} + 0.05$

$= 0.707$

$U = 1.414$ W/(m^2 K)

Typical overall thermal transmittances are:

Insulated cavity brick wall, 260 mm thick, sheltered exposure on outside	0.69 W/(m^{2}K)
Chilled water inside copper tube, forced draught air flow outside	15–28 W/(m^{2}K)
Condensing ammonia gas inside steel tube, thin film of water outside	450–470 W/(m^{2}K)

Special note should be taken of the influence of geometrical shape, where other than plain surfaces are involved.

The overall thermal transmittance, U, is used to calculate the total heat flow. For a plane surface of area A and a steady temperature difference ΔT, it is

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

If a non-volatile fluid is being heated or cooled, the sensible heat will change and therefore the temperature, so that the ΔT across the heat exchanger wall will not be constant. Since the rate of temperature change (heat flow) will be proportional to the ΔT at any one point, the space–temperature curve will be exponential. In a case where the cooling medium is an evaporating liquid, the temperature of this liquid will remain substantially constant throughout the process, since it is absorbing latent heat, and the cooling curve will be as shown in Figure 1.3.

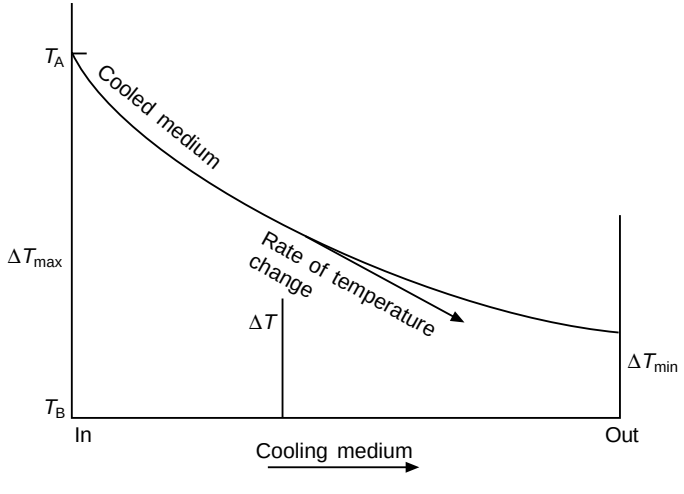


Figure 1.3 *Changing temperature difference of a cooled fluid*

Providing that the flow rates are steady, the heat transfer coefficients do not vary and the specific heat capacities are constant throughout the working range, the average temperature difference over the length of the curve is given by:

$$\Delta T = \frac{\Delta T_{\max} - \Delta T_{\min}}{\ln(\Delta T_{\max} / \Delta T_{\min})}$$

This is applicable to any heat transfer where either or both the media change in temperature (see Figure 1.4). This derived term is the *logarithmic mean temperature difference* (ln MTD) and can be used as ΔT in the general equation, providing U is constant throughout the cooling range, or an average figure is known, giving

$$Q_f = A \times U \times \ln \text{ MTD}$$

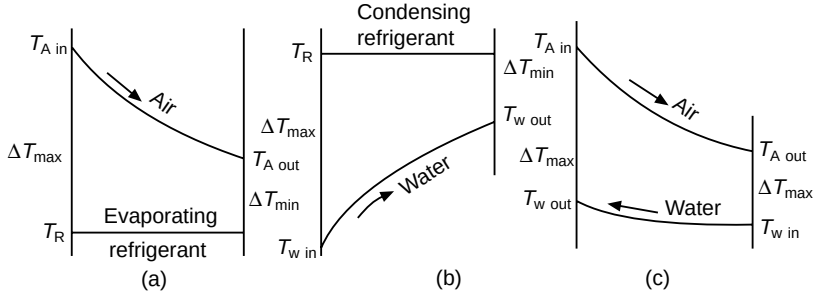


Figure 1.4 *Temperature change. (a) Refrigerant cooling fluid. (b) Fluid cooling refrigerant. (c) Two fluids*

Example 1.11 A fluid evaporates at 3°C and cools water from 11.5°C to 6.4°C. What is the logarithmic mean temperature difference and what is the heat transfer if it has a surface area of 420 m^2 and the thermal transmittance is 110 W/(m^2 K)?

$$\Delta T_{\max} = 11.5 - 3 = 8.5 \text{ K}$$

$$\Delta T_{\min} = 6.4 - 3 = 3.4 \text{ K}$$

$$\ln \text{ MTD} = \frac{8.5 - 3.4}{\ln(8.5/3.4)}$$

$$= 5.566 \text{ K}$$

$$Q_f = 420 \times 110 \times 5.566$$

$$= 257\,000 \text{ W or } 257 \text{ kW}$$

In practice, many of these values will vary. A pressure drop along a pipe carrying boiling or condensing fluid will cause a change in the

saturation temperature. With some liquids, the heat transfer values will change with temperature. For these reasons, the ln MTD formula does not apply accurately to all heat transfer applications.

If the heat exchanger was of infinite size, the space–temperature curves would eventually meet and no further heat could be transferred. The fluid in Example 1.11 would cool the water down to 3°C. The *effectiveness* of a heat exchanger can be expressed as the ratio of heat actually transferred to the ideal maximum:

$$\Sigma = \frac{T_{A\,\text{in}} - T_{A\,\text{out}}}{T_{A\,\text{in}} - T_{B\,\text{in}}}$$

Taking the heat exchanger in Example 1.11:

$$\Sigma = \frac{11.5 - 6.4}{11.5 - 3.0}$$
$$= 0.6 \text{ or } 60\%$$

Radiation of heat was shown by Boltzman and Stefan to be proportional to the fourth power of the absolute temperature and to depend on the colour, material and texture of the surface:

$$Q_f = \sigma \varepsilon T^4$$

where σ is Stefan's constant ($= 5.67 \times 10^{-8}\,\text{W}/(\text{m}^2\,\text{K}^4)$) and ε is the surface *emissivity*.

Emissivity figures for common materials have been determined, and are expressed as the ratio to the radiation by a perfectly black body, viz.

Rough surfaces such as brick, concrete, or tile, regardless of colour	0.85–0.95
Metallic paints	0.40–0.60
Unpolished metals	0.20–0.30
Polished metals	0.02–0.28

The metals used in refrigeration and air-conditioning systems, such as steel, copper and aluminium, quickly oxidize or tarnish in air, and the emissivity figure will increase to a value nearer 0.50.

Surfaces will absorb radiant heat and this factor is expressed also as the ratio to the *absorptivity* of a perfectly black body. Within the range of temperatures in refrigeration systems, i.e. –70°C to +50°C (203–323 K), the effect of radiation is small compared with the conductive and convective heat transfer, and the overall heat transfer factors in use include the radiation component. Within this temperature range, the emissivity and absorptivity factors are about equal.

The exception to this is the effect of solar radiation when considered as a cooling load, such as the air-conditioning of a building which is subject to the sun's rays. At the wavelength of sunlight the absorptivity figures change and calculations for such loads use tabulated factors for the heating effect of sunlight. Glass, glazed tiles and clean white-painted surfaces have a lower absorptivity, while the metals are higher.

1.7 Transient heat flow

A special case of heat flow arises when the temperatures through the thickness of a solid body are changing as heat is added or removed. This *non-steady* or *transient* heat flow will occur, for example, when a thick slab of meat is to be cooled, or when sunlight strikes on a roof and heats the surface. When this happens, some of the heat changes the temperature of the first layer of the solid, and the remaining heat passes on to the next layer, and so on. Calculations for heating or cooling times of thick solids consider the slab as a number of finite layers, each of which is both conducting and absorbing heat over successive periods of time. Original methods of solving transient heat flow were graphical [1, 5], but could not easily take into account any change in the conductivity or specific heat capacity or any latent heat of the solid as the temperature changed.

Complicated problems of transient heat flow can be resolved by computer. Typical time–temperature curves for non-steady cooling are shown in Figures 16.1 and 16.2, and the subject is met again in Section 26.2.

1.8 Two-phase heat transfer

Where heat transfer is taking place at the saturation temperature of a fluid, evaporation or condensation (mass transfer) will occur at the interface, depending on the direction of heat flow. In such cases, the convective heat transfer of the fluid is accompanied by conduction at the surface to or from a thin layer in the liquid state. Since the latent heat and density of fluids are much greater than the sensible heat and density of the vapour, the rates of heat transfer are considerably higher. The process can be improved by shaping the heat exchanger face (where this is a solid) to improve the drainage of condensate or the escape of bubbles of vapour. The total heat transfer will be the sum of the two components.

Rates of two-phase heat transfer depend on properties of the volatile fluid, dimensions of the interface, velocities of flow and the

extent to which the transfer interface is blanketed by fluid. The driving force for evaporation or condensation is the difference of vapour pressures at the saturation and interface temperatures. Equations for specific fluids are based on the interpretation of experimental data, as with convective heat transfer.

Mass transfer may take place from a mixture of gases, such as the condensation of water from moist air. In this instance, the water vapour has to diffuse through the air, and the rate of mass transfer will depend also on the concentration of vapour in the air. In the air–water vapour mixture, the rate of mass transfer is roughly proportional to the rate of heat transfer at the interface and this simplifies predictions of the performance of air-conditioning coils [1, 5, 9].

2 The refrigeration cycle

2.1 Basic vapour compression cycle

A liquid boils and condenses – the change between the liquid and gaseous states – at a temperature which depends on its pressure, within the limits of its freezing point and critical temperature. In boiling it must obtain the latent heat of evaporation and in condensing the latent heat must be given up again.

The basic refrigeration cycle (Figure 2.1) makes use of the boiling and condensing of a working fluid at different temperatures and, therefore, at different pressures.

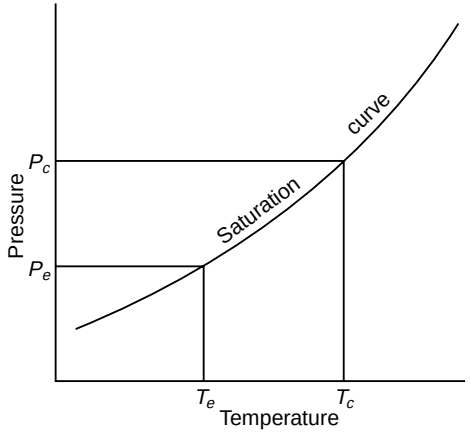


Figure 2.1 *Evaporation and condensation of a fluid*

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

The total cooling effect will be the heat transferred to the working fluid in the boiling or evaporating vessel, i.e. the change in enthalpies between the fluid entering and the vapour leaving the evaporator. For a typical circuit, using the working fluid Refrigerant 22, evaporating at – 5°C and condensing at 35°C, the pressures and enthalpies will be as shown in Figure 2.2.

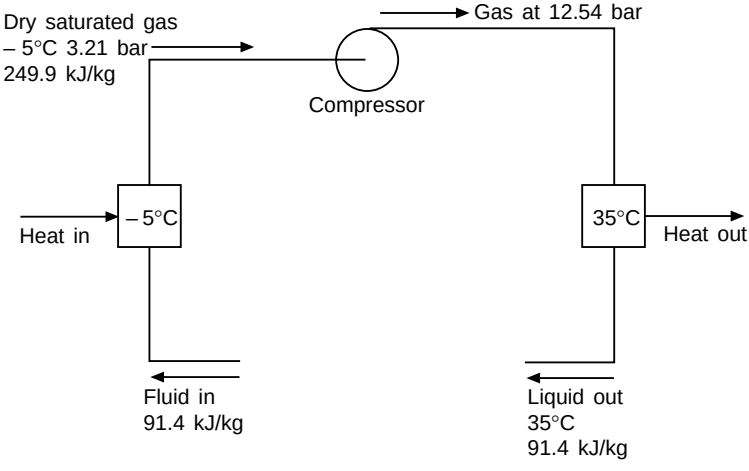


Figure 2.2 *Basic refrigeration cycle*

Enthalpy of fluid entering evaporator = 91.4 kJ/kg

Enthalpy of saturated gas leaving evaporator = 249.9 kJ/kg

Cooling effect = 249.9 – 91.4 = 158.5 kJ/kg

A working system will require a connection between the *condenser* and the inlet to the *evaporator* to complete the circuit. Since these are at different pressures this connection will require a pressure-reducing and metering valve. Since the reduction in pressure at this valve must cause a corresponding drop in temperature, some of the fluid will flash off into vapour to remove the energy for this cooling. The volume of the working fluid therefore increases at the valve by this amount of flash gas, and gives rise to its name, the *expansion valve*. (Figure 2.3.)

2.2 Coefficient of performance

Since the vapour compression cycle uses energy to move energy, the ratio of these two quantities can be used directly as a measure of the performance of the system. This ratio, the coefficient of performance, was first expressed by Sadi Carnot in 1824 for an

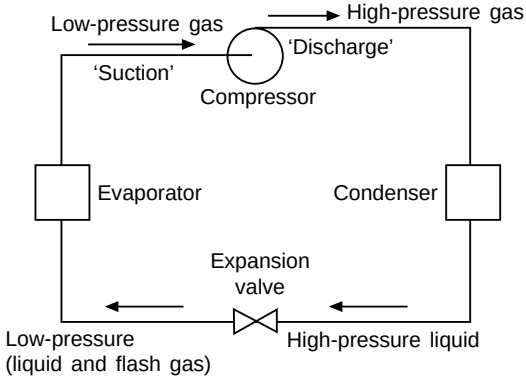


Figure 2.3 *Complete basic cycle*

ideal reversible cycle, and based on the two temperatures of the system, assuming that all heat is transferred at constant temperature (see Figure 2.4). Since there are mechanical and thermal losses in a real circuit, the coefficient of performance (COP) will always be less than the ideal Carnot figure. For practical purposes in working

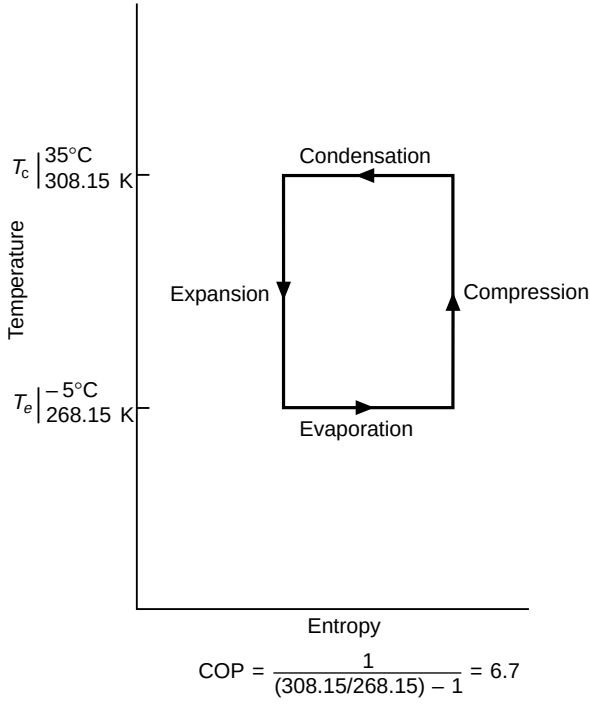


Figure 2.4 *Ideal reversed Carnot cycle*

systems, it is the ratio of the cooling effect to the input compressor power.

At the conditions shown in Figure 2.2, evaporating at – 5°C and condensing at 35°C (268.15 K and 308.15 K), the Carnot coefficient of performance is 6.7.

Transfer of heat through the walls of the evaporator and condenser requires a temperature difference. This is shown on the modified reversed Carnot cycle (Figure 2.5). For temperature differences of 5 K on both the evaporator and condenser, the fluid operating temperatures would be 263.15 K and 313.15 K, and the coefficient of performance falls to 5.26.

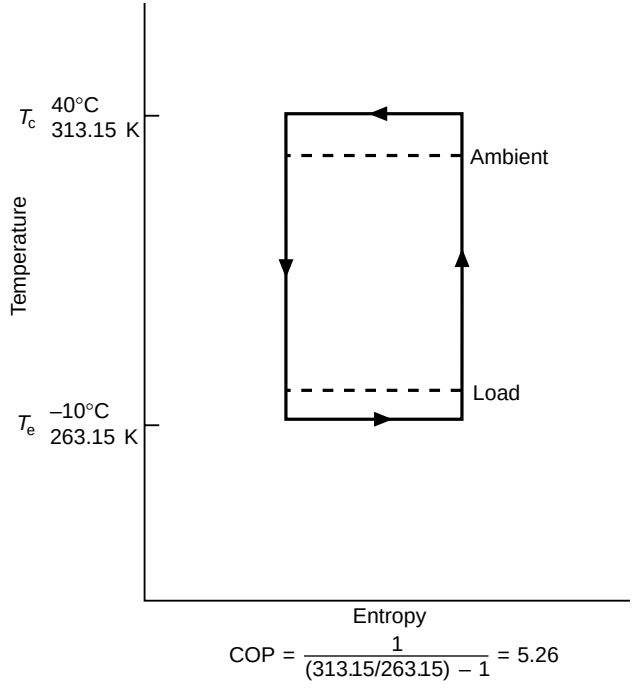


Figure 2.5 *Modified reversed Carnot cycle*

A more informative diagram is the pressure–enthalpy chart which shows the liquid and vapour states of the fluid (Figure 2.6). In this diagram, a fluid being heated passes from the subcooled state (a), reaches boiling point (b) and is finally completely evaporated (c) and then superheated (d). The distance along the sector b–c shows the proportion which has been evaporated at any enthalpy value.

The refrigeration cycle is shown by the process lines *ABCD* (Figure 2.7). Compression is assumed to be adiabatic, but this will alter

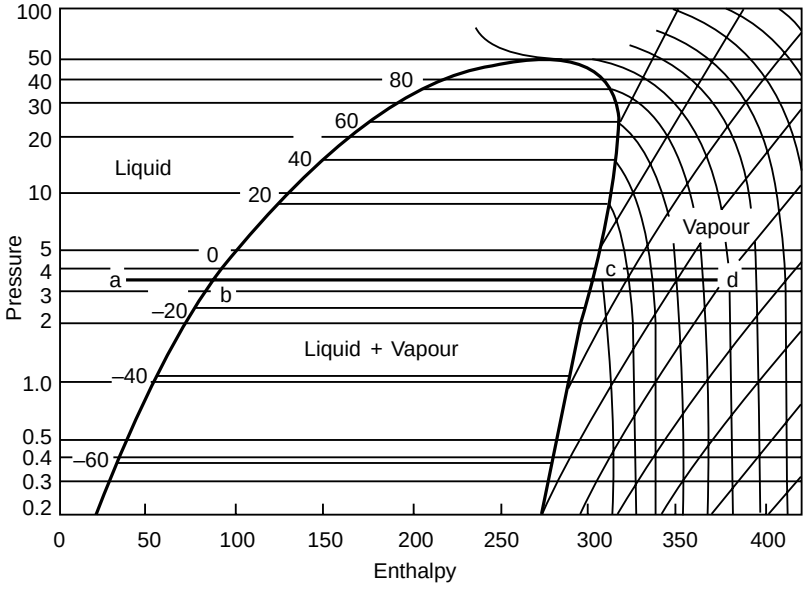


Figure 2.6 *Pressure–enthalpy diagram*

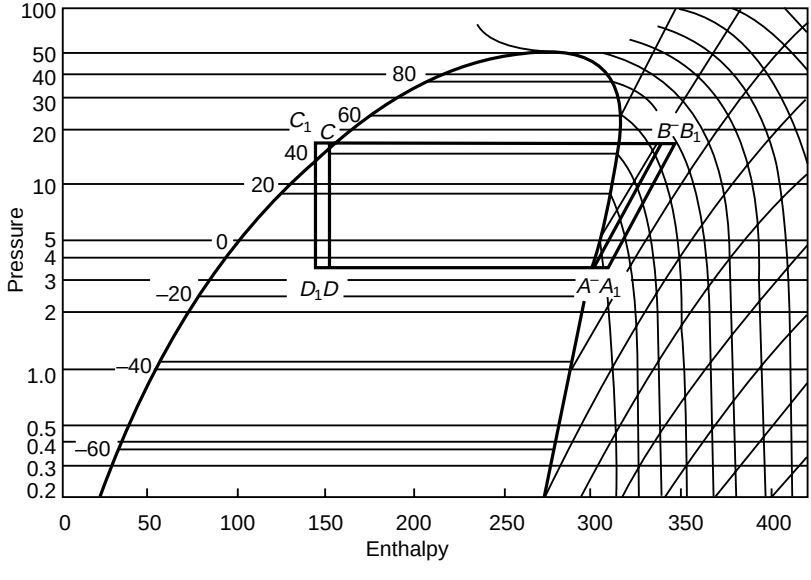


Figure 2.7 *Pressure–enthalpy or Mollier diagram (From [10], Courtesy of the Chartered Institution of Building Services Engineers)*

according to the type of compressor. Since there is no energy input or loss within the expansion valve, these two points lie on a line of equal enthalpy. The pressure–enthalpy chart can give a direct measure of the energy transferred in the process.

In a working circuit, the vapour leaving the evaporator will probably be slightly superheated and the liquid leaving the condenser subcooled. The gas leaving the evaporator is superheated to point A_1 and the liquid subcooled to C_1. Also, pressure losses will occur across the gas inlet and outlet, and there will be pressure drops through the heat exchangers and piping. The final temperature at the end of compression will depend on the working limits and the refrigerant. Taking these many factors into account, the refrigerating effect ($A_1 - D_1$) and the compressor energy ($B_1 - A_1$) may be read off directly in terms of enthalpy of the fluid.

The distance of D_1 between the two parts of the curve indicates the proportion of flash gas at that point. The condenser receives the high-pressure superheated gas, cools it down to saturation temperature, condenses it to liquid, and finally subcools it slightly. The energy removed in the condenser is seen to be the refrigerating effect plus the heat of compression.

2.3 Heat exchanger size

Transfer of heat through the walls of the evaporator and condenser requires a temperature difference, and the larger these heat exchangers are, the lower will be the temperature differences and so the closer the fluid temperatures will be to those of the load and condensing medium. The closer this approach, the nearer the cycle will be to the ideal reversed Carnot cycle. (See Table 2.1.)

These effects can be summarized as follows.

Larger evaporator 1. Higher suction pressure to give denser gas entering the compressor and therefore a greater mass of gas for a given swept volume, and so a higher refrigerating duty; 2. Higher suction pressure, so a lower compression ratio and less power for a given duty.

Larger condenser 1. Lower condensing temperature and colder liquid entering the expansion valve, giving more cooling effect; 2. Lower discharge pressure, so a lower compression ratio and less power.

2.4 Volumetric efficiency

In a reciprocating compressor, there will be a small amount of

Table 2.1

	Evaporator		*Condenser*		*Compression ratio*	*Reversed Carnot COP*
	Temperature	*Pressure*	*Temperature*	*Pressure*		
Ideal reversed Carnot	–5°C	4.24	35°C	13.68	3.23	6.70
Modified reversed Carnot, ΔT = 5 K	–10°C	3.54	40°C	15.34	4.33	5.26
Modified reversed Carnot, ΔT = 10 K	–15°C	2.96	45°C	17.3	5.85	4.30

Pressures are bar absolute for an R.22 circuit.

clearance space at the top of the stroke, arising from gas ports, manufacturing tolerances, and an allowance for thermal expansion and contraction of the components in operation. High-pressure gas left in this space at the end of the discharge stroke must re-expand to the suction inlet pressure before a fresh charge of gas can be drawn in. This clearance space is usually of the order of 4–7% of the swept volume, but it is possible to design compressors with less clearance.

This loss of useful working stroke will increase with the ratio of the suction and discharge absolute pressures, and the compressor efficiency will fall off. This effect is termed the volumetric efficiency [11]. Typical figures are shown in Figure 2.8.

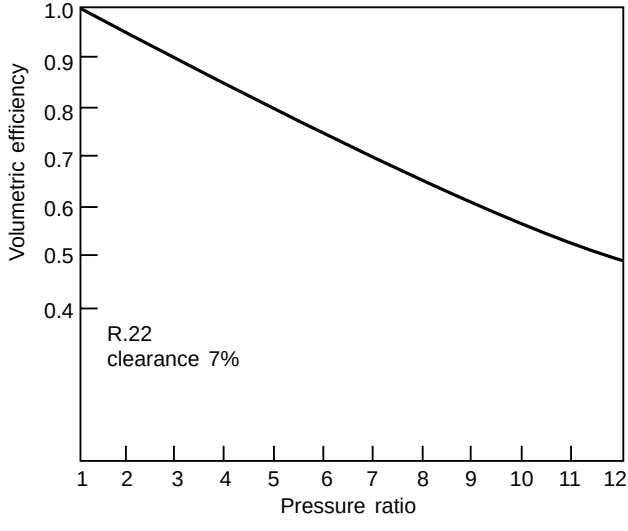


Figure 2.8 *Volumetric efficiency*

2.5 Multistage cycles

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

Compound systems use the same refrigerant throughout a common circuit, compressing in two or more stages (Figure 2.9). Discharge gas from the first compression stage will be too hot to pass directly to the high-stage compressor, so it is cooled in an intercooler, using some of the available refrigerant from the condenser. The opportunity is also taken to subcool liquid passing to the evaporator. Small compound systems may cool the interstage gas by direct injection of liquid refrigerant into the pipe.

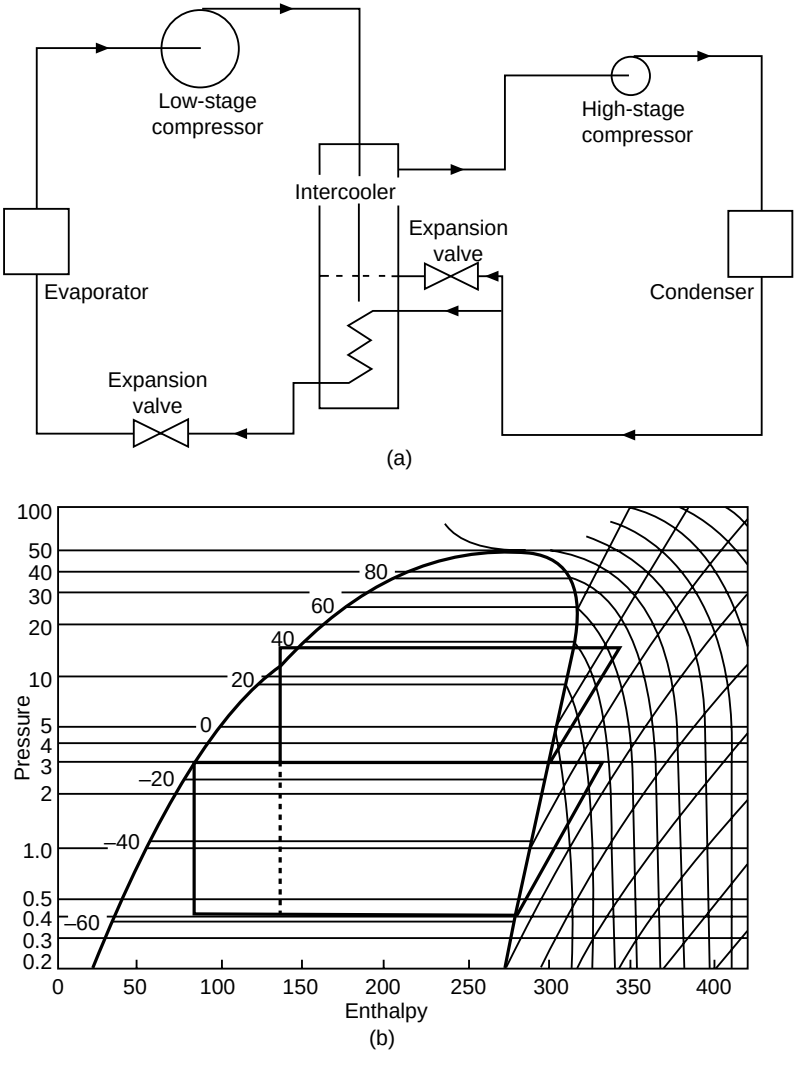


Figure 2.9 *Compound cycle. (a) Circuit. (b) Mollier diagram (compound)*

The *cascade cycle* has two separate refrigeration systems, one acting as a condenser to the other (see Figure 2.10). This arrangement permits the use of different refrigerants in the two systems, and high-pressure refrigerants such as R.13 are common in the lower stage.

The Mollier diagrams for compound and cascade systems (Figures 2.9 and 2.10) indicate the enthalpy change per kilogram of circulated refrigerant, but it should be borne in mind that the mass flows are different for the low and high stages.

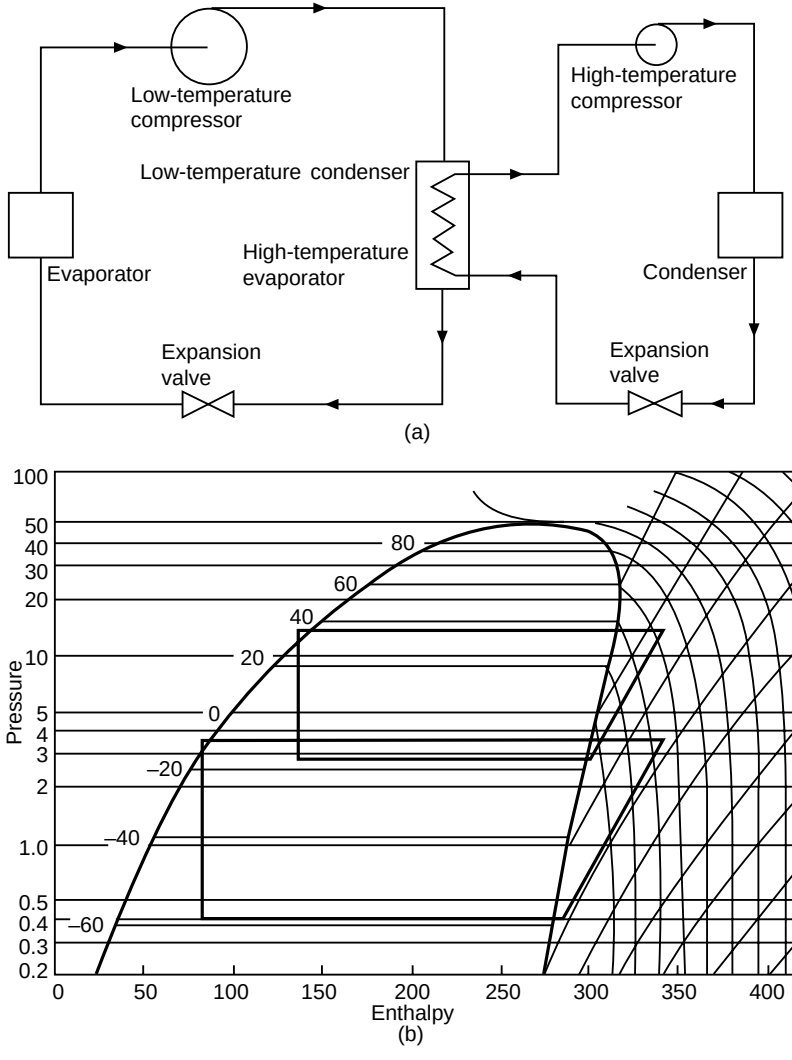


Figure 2.10 *Cascade cycle. (a) Circuits. (b) Mollier diagram (cascade)*

2.6 Refrigerants for vapour compression cycles

The requirements for the working fluid are as follows:

1. A high latent heat of vaporization
2. High density of suction gas
3. Non-corrosive, non-toxic and non-flammable
4. Critical temperature and triple point outside the working range

5. Compatibility with materials of construction, with lubricating oils, and with other materials present in the system
6. Convenient working pressures, i.e. not too high and preferably not below atmospheric pressure
7. High dielectric strength (for compressors having integral electric motors)
8. Low cost
9. Ease of leak detection
10. Environmentally friendly

No single working fluid has all these properties and a great many different chemicals have been used over the years. The present situation has been dominated by the need for fluids which are environmentally friendly. This is dealt with in Chapter 3.

2.7 Total loss refrigerants

Some volatile fluids are used once only, and then escape into the atmosphere. Two of these are in general use, carbon dioxide and nitrogen. Both are stored as liquids under a combination of pressure and low temperature and then released when the cooling effect is required. Carbon dioxide is below its critical point at atmospheric pressure and can only exist as 'snow' or a gas. Since both gases come from the atmosphere, there is no pollution hazard. The temperature of carbon dioxide when released will be – 78.4°C. Nitrogen will be at – 198.8°C. Water ice can also be classified as a total loss refrigerant.

2.8 Absorption cycle

Vapour can be withdrawn from an evaporator by absorption (Figure 2.11) into a liquid. Two combinations are in use, the absorption of ammonia gas into water and the absorption of water vapour into lithium bromide. The latter is non-toxic and so may be used for air-conditioning. The use of water as the refrigerant in this combination restricts it to systems above its freezing point. Refrigerant vapour from the evaporator is drawn into the absorber by the liquid absorbant, which is sprayed into the chamber. The resulting solution (or liquor) is then pumped up to condenser pressure and the vapour is driven off in the generator by direct heating. The high-pressure refrigerant gas given off can then be condensed in the usual way and passed back through the expansion valve into the evaporator. Weak liquor from the generator is passed through another pressure-reducing valve to the absorber. Overall thermal efficiency is improved

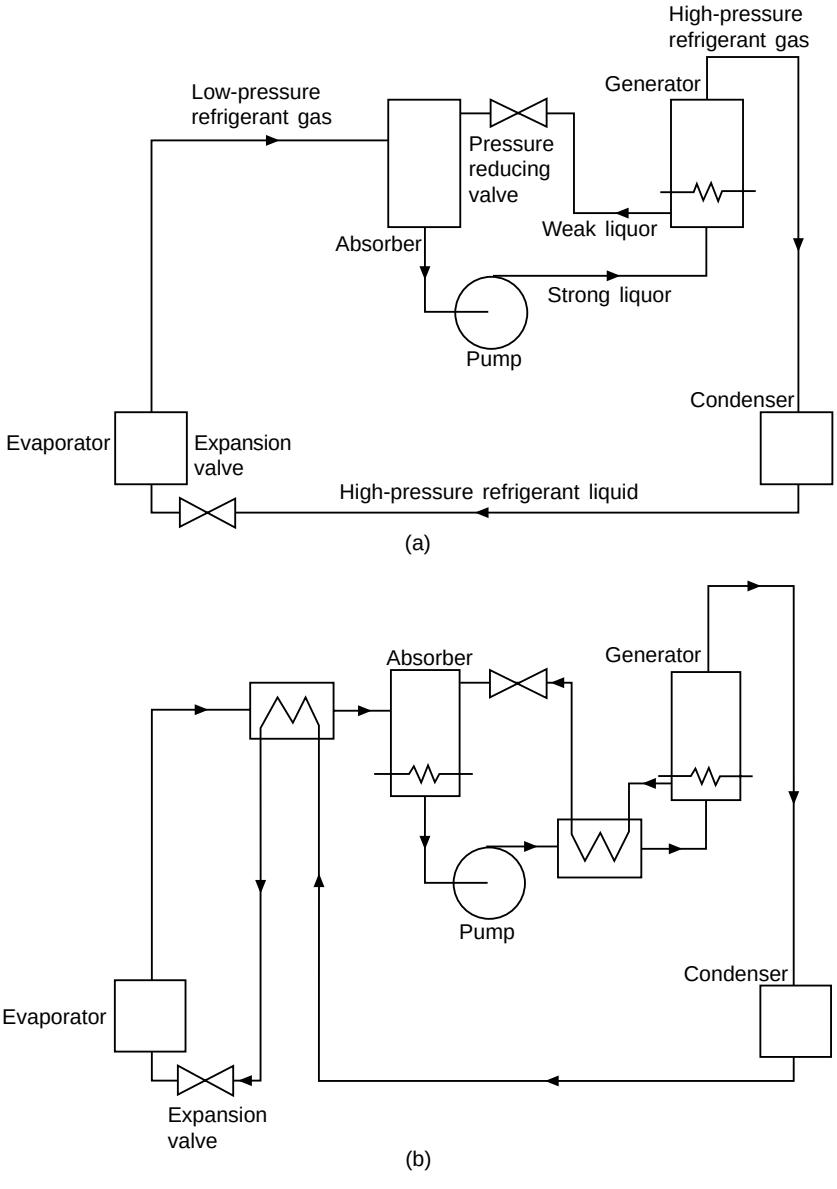


Figure 2.11 *Absorption cycle. (a) Basic circuit. (b) Circuit with heat interchange*

by a heat exchanger between the two liquor paths and a suction-to-liquid heat exchanger for the refrigerant. Power to the liquor pump will usually be electric, but the heat energy to the generator may be any form of low-grade energy such as oil, gas, hot water or steam.

(Solar radiation can also be used.) The overall energy used is greater than with the compression cycle, so the COP (coefficient of performance) is lower. Typical figures are as shown in Table 2.2.

Table 2.2 Energy per 100 kW cooling capacity at 3°C evaporation, 42°C condensation

	Absorption	*Vapour compression*
Load	100.0	100.0
Pump/compressor (electricity)	0.1	30.0
Low-grade heat	165	–
Heat rejected	265.1	130.0

The absorption system can be used to advantage where there is a cheap source of low-grade heat or where there are severe limits to the electrical power available. A modified system of the ammonia–water absorption cycle has been developed for small domestic refrigerators.

2.9 Steam ejector system

The low pressures (8–22 mbar) required to evaporate water as a refrigerant at 4–7°C for air-conditioning duty can be obtained with a steam ejector. High-pressure steam at 10 bar is commonly used. The COP of this cycle is somewhat less than with the absorption system, so its use is restricted to applications where large volumes of steam are available when required (large, steam-driven ships) or where water is to be removed along with cooling, as in freeze-drying and fruit juice concentration.

2.10 Air cycle

Any gas, when compressed, rises in temperature. Conversely, if it is made to do work while expanding, the temperature will drop. Use is made of the sensible heat only (although it is, of course, the basis of the air liquefaction process).

The main application for this cycle is the air-conditioning and pressurization of aircraft. The turbines used for compression and expansion turn at very high speeds to obtain the necessary pressure ratios and, consequently, are noisy. The COP is lower than with other systems [15].

The normal cycle uses the expansion of the air to drive the first stage of compression, so reclaiming some of the input energy (Figure 2.12).

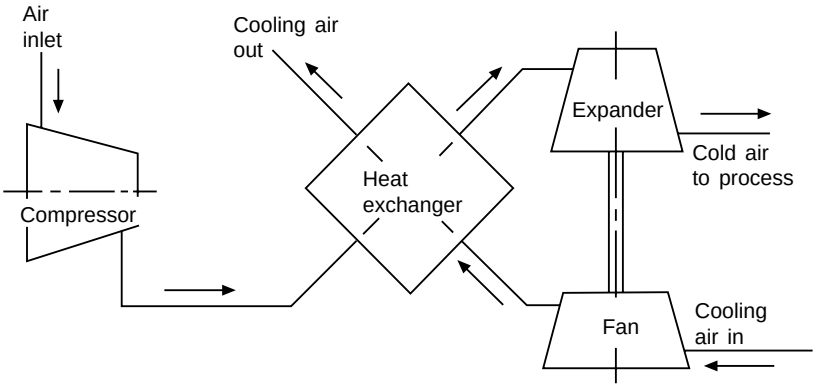


Figure 2.12 *Air cycle cooling*

2.11 Thermoelectric cooling

The passage of an electric current through junctions of dissimilar metals causes a fall in temperature at one junction and a rise at the other, the Peltier effect. Improvements in this method of cooling have been made possible in recent years by the production of suitable semiconductors. Applications are limited in size, owing to the high electric currents required, and practical uses are small cooling systems for military, aerospace and laboratory use (Figure 2.13).

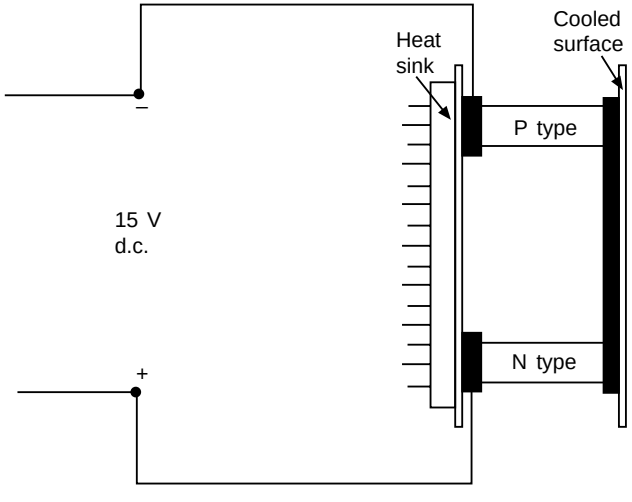


Figure 2.13 *Thermoelectric cooling*

3 Refrigerants[73]

3.1 Background

The last decade has seen radical changes in the selection and use of refrigerants, mainly in response to the environmental issues of 'holes in the ozone layer' and 'global warming or greenhouse effect'. Previously there had not been much discussion about the choice of refrigerant, as the majority of applications could be met by the well-known and well-tested fluids, R11, R12, R22, R502 and ammonia (R717). The only one of these fluids to be considered environmentally friendly today is ammonia, but it is not readily suited to commercial or air-conditioning refrigeration applications because of its toxicity, flammability and attack by copper.

This chapter is about the new refrigerants and the new attitude needed in design, maintenance and servicing of refrigeration equipment.

3.2 Ideal properties for a refrigerant

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

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

No single fluid has all these properties, and meets the new environmental requirements, but this chapter will show the developments that are taking place in influencing the selection and choice of a refrigerant.

3.3 Ozone depletion potential

The ozone layer in our upper atmosphere provides a filter for ultraviolet radiation, which can be harmful to our health. Research has found that the ozone layer is thinning, due to emissions into the atmosphere of chlorofluorocarbons (CFCs), halons and bromides. The Montreal Protocol in 1987 agreed that the production of these chemicals would be phased out by 1995 and alternative fluids developed. From Table 3.1, R11, R12, R114 and R502 are all CFCs used as refrigerants, while R13B1 is a halon. They have all ceased production within those countries which are signatories to the Montreal Protocol. The situation is not so clear-cut, because there are countries like Russia, India, China etc. who are not signatories and who could still be producing these harmful chemicals. Table 3.2 shows a comparison between old and new refrigerants.

Table 3.1 Typical uses of refrigerants before 1987

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

It should be noted that prior to 1987, total CFC emissions were made up from aerosol sprays, solvents and foam insulation, and that refrigerant emissions were about 10% of the total. However, all the different users have replaced CFCs with alternatives.

R22 is an HCFC and now regarded as a transitional refrigerant, in that it will be completely phased out of production by 2030, as agreed under the Montreal Protocol. A separate European Community decision has set the following dates.

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

Table 3.2 Comparison of new refrigerants

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

3.4 Global warming potential (GWP)

Global warming is the increasing of the world's temperatures, which results in melting of the polar ice caps and rising sea levels. It is caused by the release into the atmosphere of so-called 'greenhouse' gases, which form a blanket and reflect heat back to the earth's surface, or hold heat in the atmosphere. The most infamous greenhouse gas is carbon dioxide (CO_2), which once released remains in the atmosphere for 500 years, so there is a constant build-up as time progresses.

The main cause of CO_2 emission is in the generation of electricity at power stations. Each kWh of electricity used in the UK produces

about 0.53 kg of CO_2 and it is estimated that refrigeration compressors in the UK consume 12.5 billion kWh per year.

Table 3.3 shows that the newly developed refrigerant gases also have a global warming potential if released into the atmosphere. For example, R134a has a GWP of 1300, which means that the emission of 1 kg of R134a is equivalent to 1300 kg of CO_2. The choice of refrigerant affects the GWP of the plant, but other factors also contribute to the overall GWP and this has been represented by the term *total equivalent warming impact* (TEWI). This term shows the overall impact on the global warming effect, and includes refrigerant leakage, refrigerant recovery losses and energy consumption. It is a term which should be calculated for each refrigeration plant. Figures 3.1 and 3.2 show the equation used and an example for a medium temperature R134a plant.

Table 3.3 Environmental impact of some of the latest refrigerants

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

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

$$\text{TEWI} = (\text{GWP} \times L \times n) + (\text{GWP} \times m\,[1 - \alpha_{recovery}] + (n \times E_{annual} \times \beta)$$

Leakage — Recovery losses — Energy consumption

direct global warming potential — indirect global warming potential

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

Figure 3.1 *Method for the calculation of TEWI figures*

Example

Medium temperature R134a

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

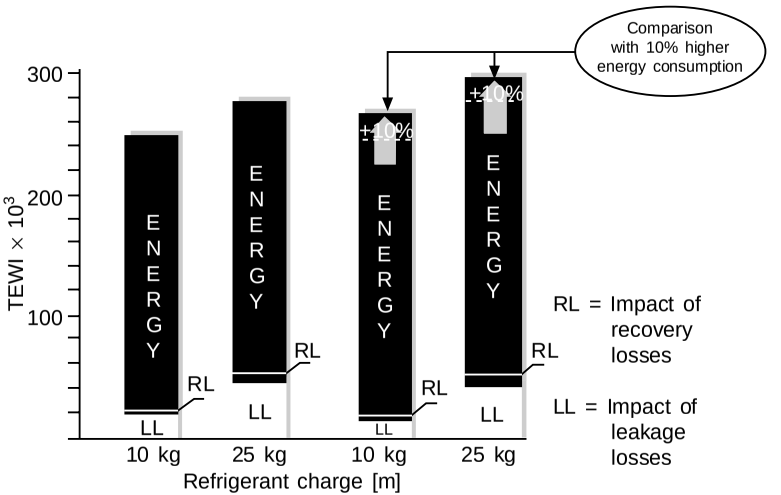


Figure 3.2 *Comparison of TEWI figures (example)*

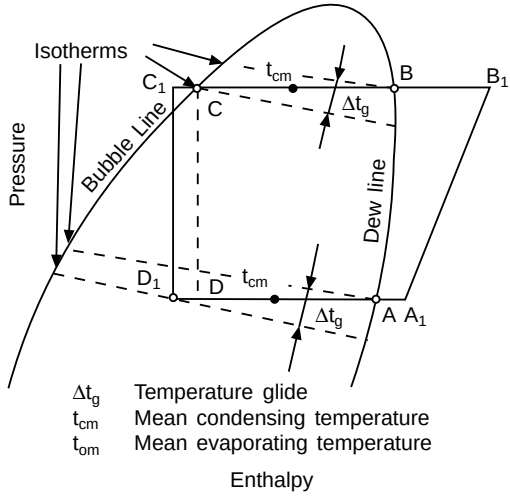


Figure 3.3 *Evaporating and condensing behaviour of zeotropic blends*

One thing that is certain is that the largest element of the TEWI is energy consumption, which contributes CO_2 emission to the atmosphere. The choice of refrigerant is therefore about the efficiency of the refrigerant and the efficiency of the refrigeration system. The less the amount of energy needed to produce each kW of cooling, the less will be the effect on global warming.

3.5 Ammonia and the hydrocarbons

These fluids have virtually zero ODP and zero GWP when released into the atmosphere and therefore present a very friendly environmental picture. Ammonia has long been used as a refrigerant for industrial applications. The engineering and servicing requirements are well established to deal with its high toxicity and flammability. There have been developments to produce packaged liquid chillers with ammonia as the refrigerant for use in air-conditioning in supermarkets, for example. Ammonia cannot be used with copper or copper alloys, so refrigerant piping and components have to be steel or aluminium. This may present difficulties for the air-conditioning market where copper has been the base material for piping and plant. One property that is unique to ammonia compared to all other refrigerants is that it is less dense than air, so a leakage of ammonia results in it rising above the plant room and into the atmosphere. If the plant room is outside or on the roof of a building, the escaping ammonia will drift away from the refrigeration plant.

The safety aspects of ammonia plants are well documented and there is reason to expect an increase in the use of ammonia as a refrigerant.

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

3.6 Refrigerant blends

Many of the new, alternative refrigerants are 'blends', which have two or three components, developed for existing and new plants as comparable alternatives to the refrigerants being replaced. They are 'zeotropes' with varying evaporating or condensing temperatures in the latent heat of vaporization phase, referred to as the 'temperature glide'. Figure 3.3 shows the variation in evaporating and condensing temperatures.

To compare the performance between single component refrigerants and blends it will be necessary to specify the evaporating temperature of the blend to point A on the diagram and the condensing temperature to point B.

The temperature glide can be used to advantage in improving plant performance, by correct design of the heat exchangers. A problem associated with blends is that refrigerant leakage results in a change in the component concentration of the refrigerant. However, tests indicate that small changes in concentration (say less than 10%) have a negligible effect on plant performance.

The following recommendations apply to the use of blends:

- The plant must always be charged with liquid refrigerant, or the component concentrations will shift.
- Since most blends contain at least one flammable component, the entry of air into the system must be avoided.
- Blends which have a large temperature glide, greater than 5K, should not be used for flooded-type evaporators.

3.7 Lubricants

Choosing the right lubricating oil for the compressor has become more complex with the introduction of new refrigerants. Table 3.4 gives some indication as to the suitability of the traditional and new lubricating oils. Compressor manufacturers should be consulted with regards to changing the specified oil for a particular compressor.

Table 3.4 Choice of compressor lubricant

Refrigerant / *Lubricant*	*(H)CFC*	*Service blends*	*HFC + blends*	*Hydro-carbons*	*Ammonia*
Traditional oils					
Mineral	*	**	X	* V	*
Alkyl benzene	*	*	**	* V	**
Poly-apha-olefin	**	X	X	* V	**
New lubricants					
Polyol-ester	** M V	* M V	*	* V	X
Poly-glycol	X	X	** M	** M	** M
Hydro-treated mineral oil	X	X	X	X	*

Key
* Good suitability
** Application with limitations
X Not suitable
M Especially critical with moisture
V Possible correction of basic viscosity

Those lubricants marked 'M' easily absorb moisture and great care must be taken to prevent exposure to air when adding new oil. The moisture in the air will be absorbed into the oil and will lead to contamination of both refrigerant and oil. With hermetic compressors this can lead to motor winding failure.

3.8 Health and safety

When dealing with any refrigerant, personal safety and the safety of others are vitally important. Service and maintenance staff need to be familiar with safety procedures and what to do in the event of an emergency. Health and safety requirements are available from manufacturers of all refrigerants and should be obtained and studied.

Safety codes are available from the Institute of Refrigeration in London, for HCFC/HFC refrigerants (A1 and A2), ammonia (B2) and hydrocarbons (A3).

In the UK and most of Europe, it is illegal to dispose of refrigerant in any other way than through an authorized waste disposal company. The UK legislation expects that anyone handling refrigerants is competent to do so and has the correct equipment and containers. Disposal must be through an approved contractor and must be fully documented. Severe penalties may be imposed for failure to implement these laws.

4 Compressors

4.1 General

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

Compressors may be of the positive displacement or dynamic type. The general form of positive displacement compressor is the piston type, being adaptable in size, number of cylinders, speed and method of drive. It works on the two-stroke cycle (see Figure 4.1). As the piston descends on the suction stroke, the internal pressure falls until it is lower than that in the suction inlet pipe, and the suction valve opens to admit gas from the evaporator. At the bottom of the stroke, this valve closes again and the compression stroke begins. When the cylinder pressure is higher than that in the discharge pipe, the discharge valve opens and the compressed gas passes to the condenser. Clearance gas left at the top of the stroke must re-expand before a fresh charge can enter the cylinder (see

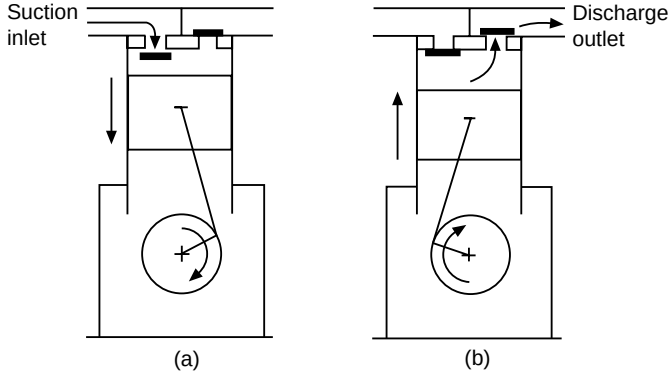


Figure 4.1 *Reciprocating compressor. (a) Suction stroke. (b) Discharge stroke*

Figure 4.2 and also Chapter 2, for theoretical and practical cycles on the Mollier chart and for volumetric efficiency).

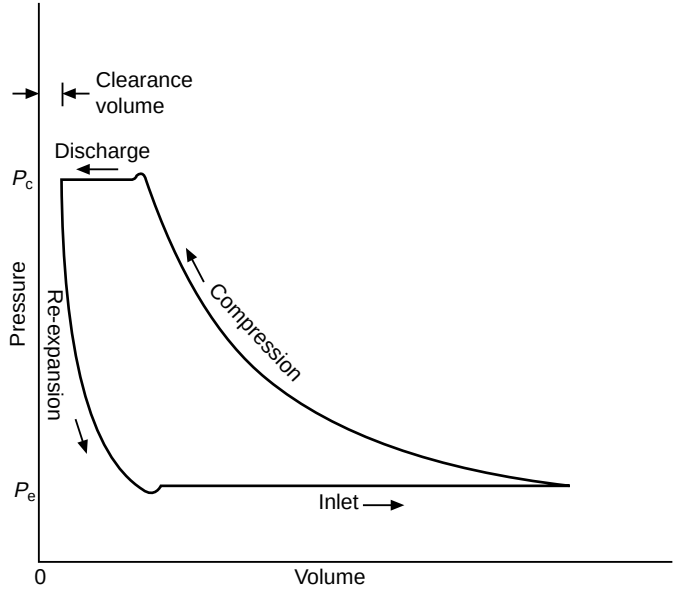


Figure 4.2 *Reciprocating compressor, indicator diagram*

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

4.2 Multicylinder compressors

In the first century of development, compressors for higher capacity were made larger, having cylinder bores up to 375 mm, and running at speeds up to 400 rev/min. The resulting component parts were heavy and cumbersome. To take advantage of larger-scale production methods and provide interchangeability of parts, modern compressors tend to be multicylinder, with bores not larger than 175 mm and running at higher shaft speeds. Machines of four, six and eight cylinders are common. These are arranged in a multibank configuration with two, three or four connecting rods on the same

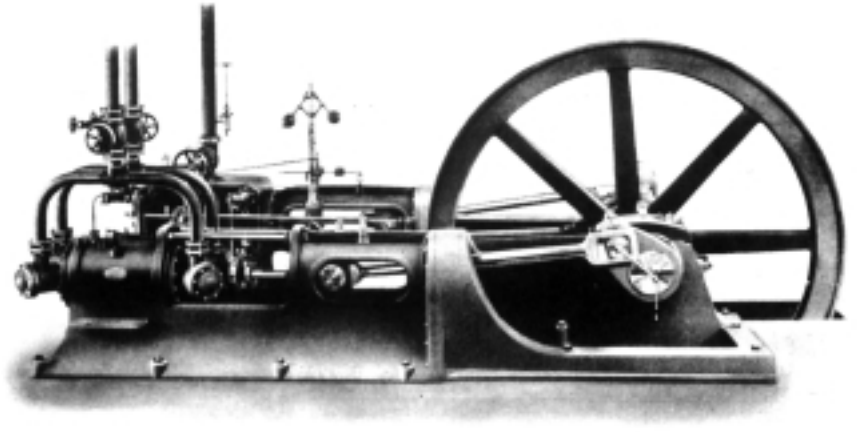


Figure 4.3 *Double-acting ammonia compressor and steam engine (Courtesy of Vilter Manufacturing Corporation)*

throw of the crankshaft to give a short, rigid machine (see Figure 4.4).

This construction gives a large number of common parts – pistons, connecting rods, loose liners and valves – through a range of compressors, and such parts can be replaced if worn or damaged without removing the compressor body from its installation.

Compressors for small systems will be simpler, of two, three or four cylinders (see Figure 4.5).

4.3 Valves

Piston compressors may be generally classified by the type of valve, and this depends on size, since a small swept volume requires a proportionally small inlet and outlet gas port. The smallest compressors have spring steel reed valves, both inlet and outlet in the cylinder head and arranged on a valve plate (Figure 4.6). Above a bore of about 40 mm, the port area available within the head size is insufficient for both inlet and outlet valves, and the inlet is moved to the piston crown or to an annulus surrounding the head. The outlet or discharge valve remains in the central part of the cylinder head. In most makes, both types of valve cover a ring of circular gas ports, and so are made in annular form and generally termed ring plate valves (Figure 4.7). Ring plate valves are made of thin spring steel or titanium, limited in lift and damped by light springs to assist even closure and lessen bouncing.

Although intended to handle only dry gas, liquid refrigerant or traces of oil may sometimes enter the cylinder and must pass out

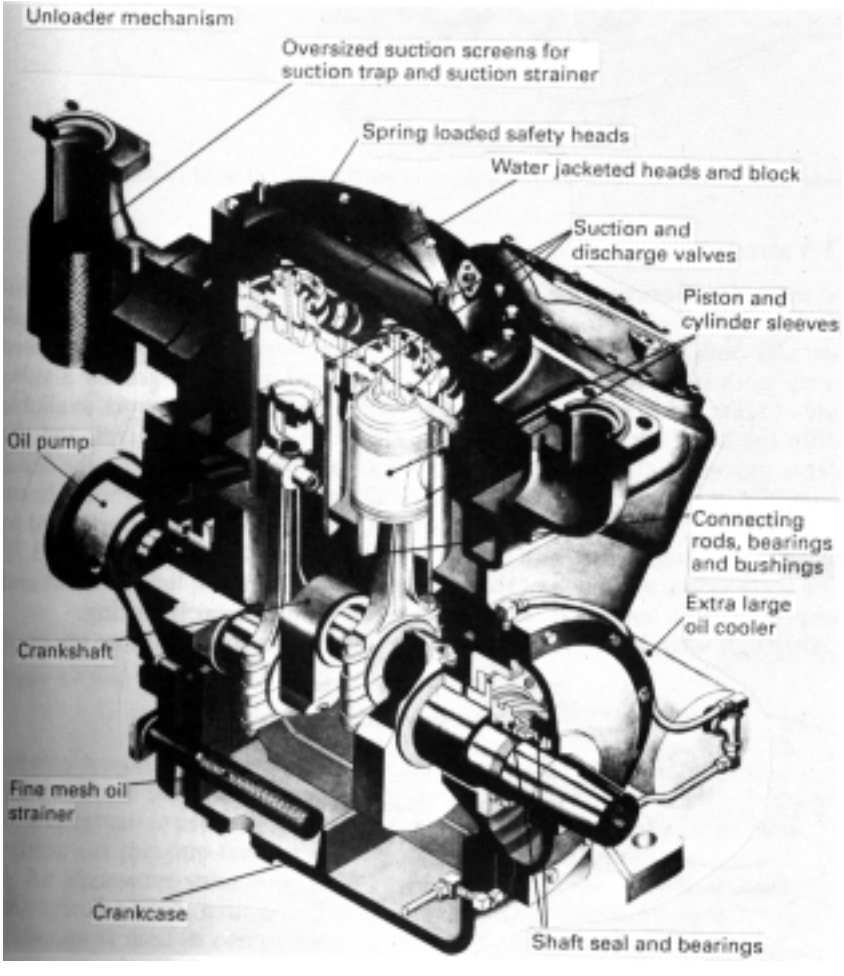


Figure 4.4 *A $1\frac{3}{8}$ in bore × 1 in stroke, two-cylinder compressor (Courtesy of APV Baker Ltd (Hall Division))*

through the discharge valves. These may be arranged on a spring-loaded head, which will lift and relieve excessive pressures. Some makes also have an internal safety valve to release gas pressure from the discharge back to the suction inlet.

An alternative valve design uses a conical discharge valve in the centre of the cylinder head, with a ring plate suction valve surrounding it. This construction is used in compressor bores up to 75 mm.

Valve and cylinder head design is very much influenced by the need to keep the volumetric clearance (q.v.) to a minimum.

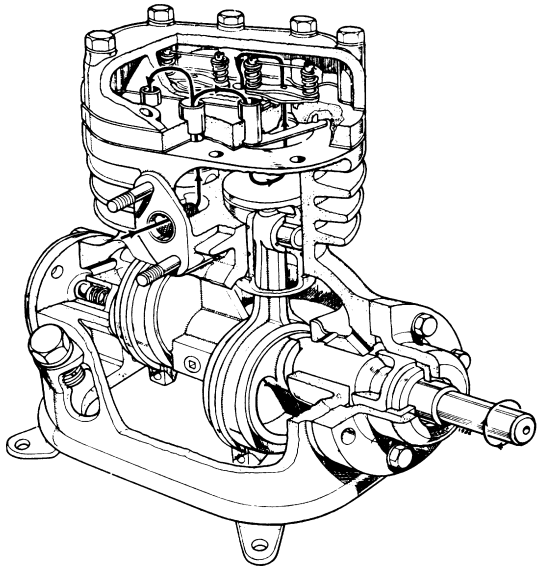


Figure 4.5 *Multicylinder compressor (Courtesy of APV Baker Ltd (Hall Division))*

4.4 Capacity reduction

A refrigeration system will be designed to have a maximum duty to balance a calculated maximum load, and for much of its life may work at some lower load. Such variations require capacity reduction devices, originally by speed control (when steam driven) or in the form of bypass ports in the cylinder walls.

The construction of multicylinder machines gives the opportunity to change the working swept volume by taking cylinders out of service with valve-lifting mechanisms. The ring plate suction valve which is located at the crown of a loose liner can be lifted by various

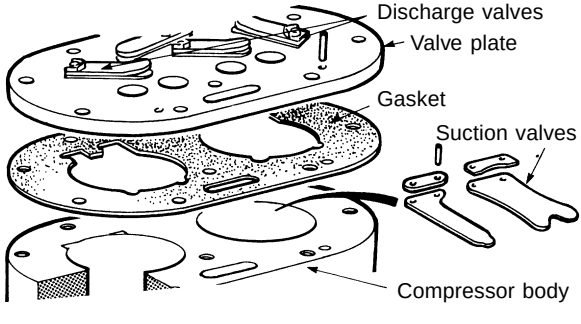


Figure 4.6 *Reed valves on valve plates (Courtesy of Prestcold Ltd)*

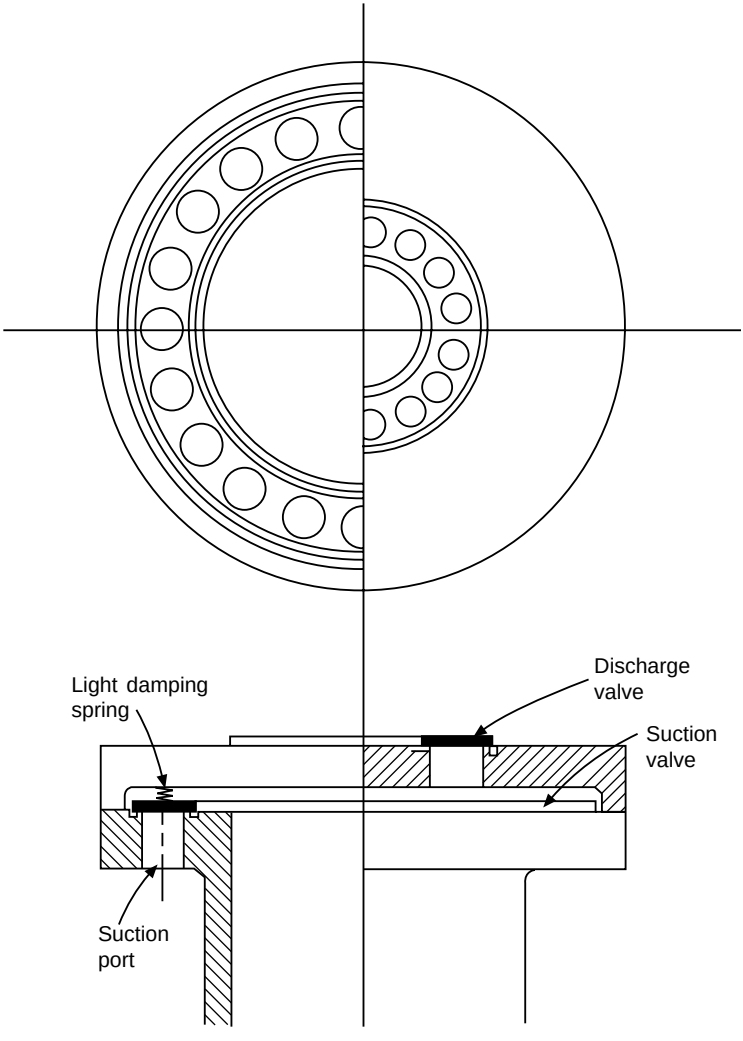


Figure 4.7 *Ring plate valves*

alternative mechanical systems, actuated by pressure of the lubricating oil and controlled by solenoid valves (see Figure 4.9). Typically, an annular piston operates push rods under the valves. In this way a multicylinder machine (see Figure 4.10) can have any number of its cylinders unloaded for capacity reduction and, in addition, will start unloaded until the build-up of oil pump pressure depresses the valve lifters.

Smaller machines may have a valved bypass across the inlet and outlet ports in the cylinder head, or a variable clearance pocket in

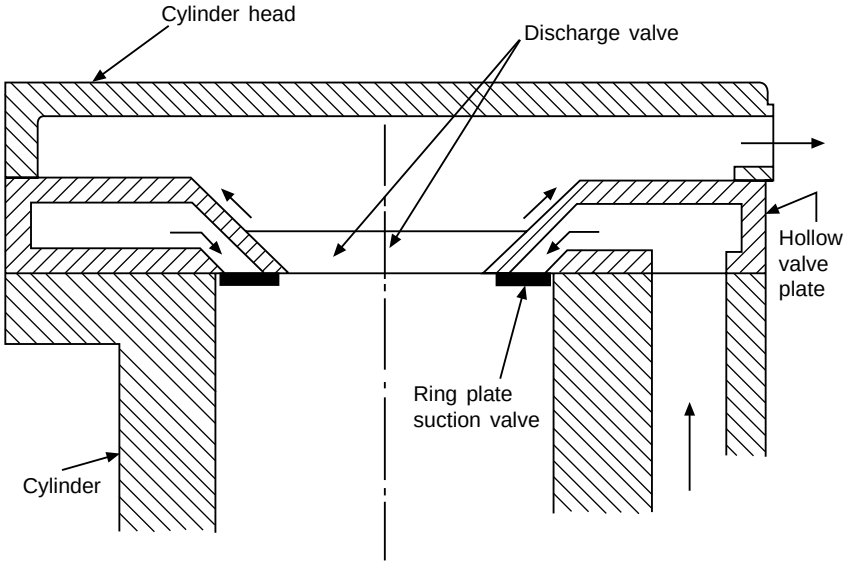


Figure 4.8 *Concentric cylinder head valves with cone-seated discharge valve*

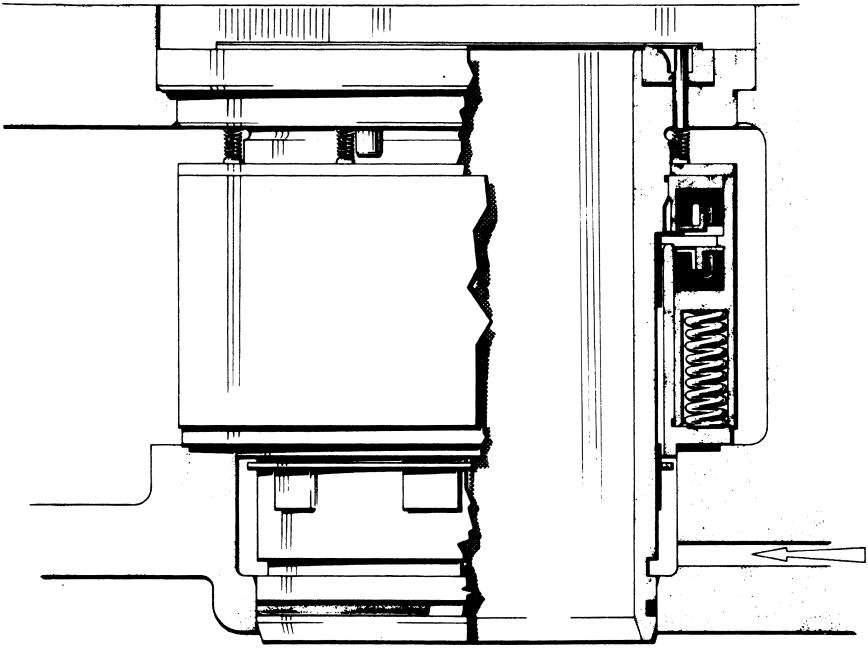


Figure 4.9 *Lifting mechanism for ring plate suction valve (Courtesy of APV Baker Ltd (Hall Division)*

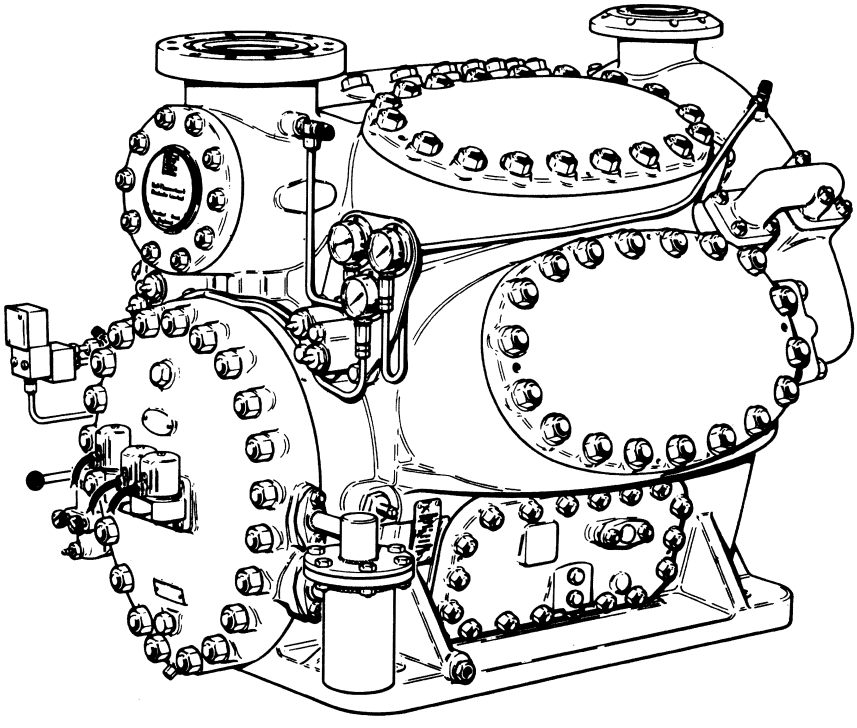


Figure 4.10 *Multicylinder compressor, outer views (Courtesy of APV Baker Ltd)*

the head itself. Capacity may be reduced by external bypass piping (see Chapter 9).

The compressor speed may be reduced by two-speed electric motors or by electronic variation of the motor speed, down to a lower limit dictated by the inbuilt lubrication system. Many high-speed industrial machines are still driven by steam turbines and this gives the opportunity for speed control within the limits of the prime mover.

4.5 Cooling

Cold suction gas provides cooling for the compressor and is sufficient to keep small machines at an acceptable working temperature. Refrigerants having high discharge temperatures (mainly ammonia) require the use of water-cooled cylinder heads. Oil coolers are needed under some working conditions which will be specified by the manufacturer. These may be water cooled or take refrigerant from the system.

Compressors will tend to overheat under low mass flow conditions resulting from abnormally low suction pressures or lengthy running with capacity reduction. Detectors may need to be fitted to warn against this condition.

4.6 Strainers. Lubrication

Incoming gas may contain particles of dirt from within the circuit, especially on a new system. Suction strainers or traps are provided to catch such dirt and will be readily accessible for cleaning on the larger machines.

All but the smallest compressors will have a strainer or filter in the lubricating oil circuit. Strainers within the sump are commonly of the self-cleaning slot disc type. Larger machines may also have a filter of the fabric throwaway type, as in automobile practice. Reciprocating compressors operate with a wet sump, having splash lubrication in the small sizes but forced oil feed with gear or crescent pumps on all others. A sight glass will be fitted at the correct working oil level and a hand pump may be fitted to permit the addition of oil without stopping or opening the plant, the sump being under refrigerant gas pressure.

4.7 Crankcase heaters

When the compressor is idle, the lubricating oil may contain a certain amount of dissolved refrigerant, depending on the pressure, temperature, and the refrigerant itself. At the moment of starting, the oil will be diluted by this refrigerant and, as the suction pressure falls, gas will boil out of the oil, causing it to foam.

To reduce this solution of refrigerant in the oil to an acceptable factor, heating devices are commonly fitted to crankcases, and will remain in operation whenever the compressor is idle.

4.8 Shaft glands. Motors

Compressors having external drive require a gland or seal where the shaft passes out of the crankcase, and are termed open compressors. They may be belt driven or directly coupled to the shaft of the electric motor or other prime mover.

The usual form of shaft seal for open drive compressors comprises a rotating carbon ring in contact with a highly polished metal facing ring, the assembly being well lubricated. The carbon ring is spring-loaded to maintain contact under all working crankcase pressures, and to allow for slight movement of the shaft.

When first started, a refrigeration system will operate at a higher suction temperature and pressure than normal operating conditions and consequently a higher discharge pressure, taking considerably more power. Drive motors must be sized accordingly to provide this pulldown power, and an allowance of 25% is usual. As a result, the drive motor will run for the greater part of its life at something under 80% rated output, and so at a lower efficiency, low running current and poor power factor. Electrical protection and safety devices must take this into account and power factor correction should be fitted on large motors. See also Chapter 8 on maximum operating pressure expansion valves.

Recent developments in electronic motor power and speed controls have provided the means to reduce the power input at normal speed to balance this reduced load requirement, and also to modulate both power and speed as a method of capacity reduction. It is improbable that electronic speed control will be economical for motors above 100 kW.

There is a need for small compressors to be driven from low-voltage d.c. supplies. Typical cases are batteries on small boats and mobile homes, where these do not have a mains voltage alternator. It is also possible to obtain such a supply from a bank of solar cells. This requirement has been met in the past by diaphragm compressors driven by a crank and piston rod from a d.c. motor, or by vibrating solenoids. The advent of suitable electronic devices has made it possible to obtain the mains voltage a.c. supply for hermetic compressors from low-voltage d.c.

4.9 Hermetic drives

The possible slight leakage of refrigerant through a shaft gland may be acceptable with a large system but would lead to early malfunction of a small circuit. The wide use of small refrigeration systems has led to the evolution of methods of avoiding shaft seals, provided that the working fluid is compatible with the materials of electric motors and has a high dielectric strength.

The *semi-hermetic* or *accessible-hermetic* compressor (Figure 4.11) has the rotor of its drive motor integral with an extended crankshaft, and the stator is fitted within an extension of the crankcase. Suction gas passes through the motor itself to remove motor waste heat. Induction motors only can be used, with any starting switches outside the crankcase, since any sparking would lead to decomposition of the refrigerant. Electrical leads pass through ceramic or glass seals. Small compressors will be *fully hermetic*, i.e. having the motor and all working parts sealed within a steel shell, and so not accessible for

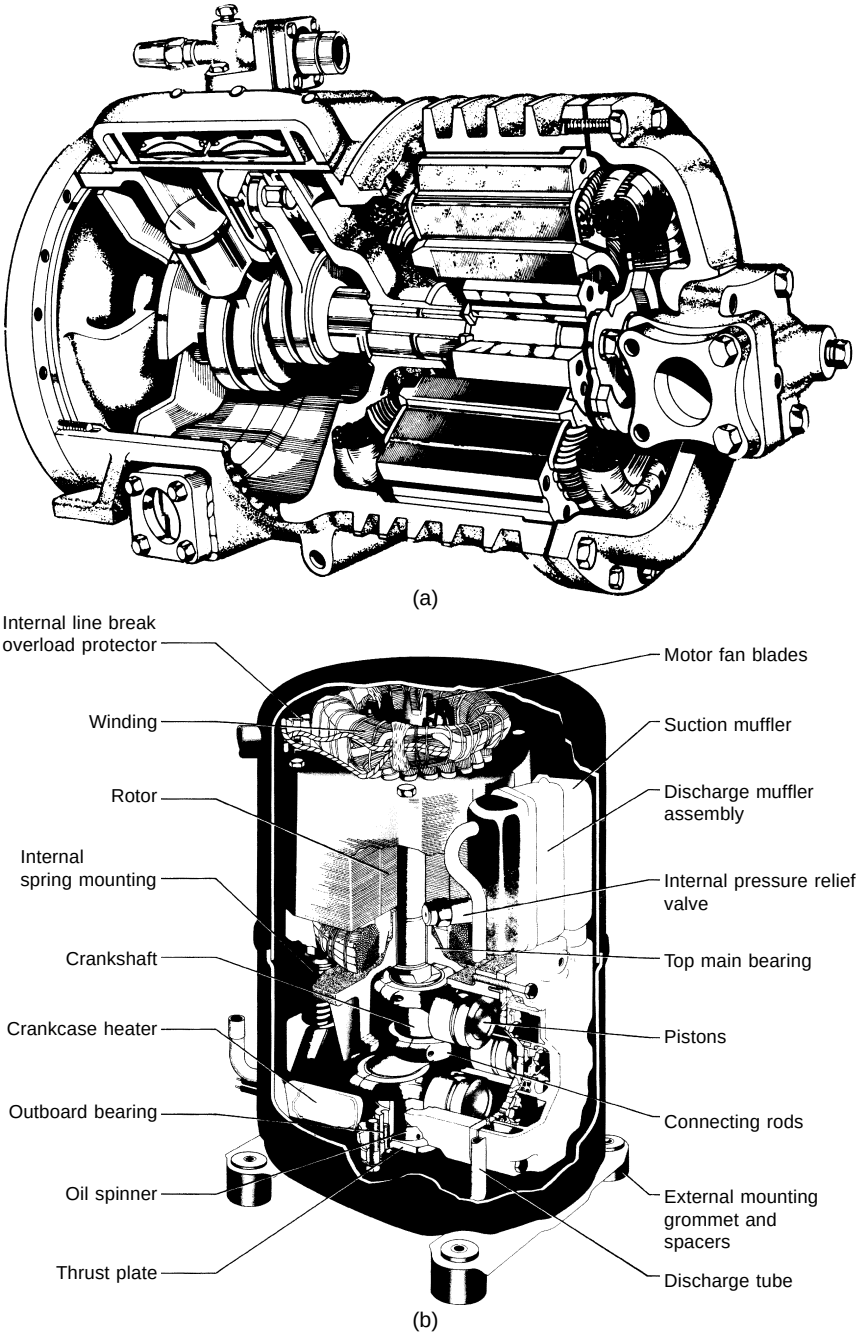


Figure 4.11 *(a) Semi-hermetic compressor (Courtesy of Dunham-Bush Ltd). (b) Welded-hermetic compressor (Courtesy of L'Unité Hermétique S.A.)*

repair or maintenance. The application of the full hermetic compressor is limited by the amount of cooling by the incoming cold gas, heat loss from the shell, and the possible provision of an oil cooler.

The failure of an inbuilt motor will lead to products of decomposition and serious contamination of the system, which must then be thoroughly cleaned. Internal and external motor protection devices are fitted with the object of switching off the supply before such damage occurs.

4.10 Sliding and rotary vane compressors

The volumes between an eccentric rotor and sliding vanes will vary with angular position, to provide a form of positive displacement compressor (Figure 4.12). Larger models have eight or more blades and do not require inlet or outlet valves. The blades are held in close contact with the outer shell by centrifugal force, and sealing is improved by the injection of lubricating oil along the length of the blades. Rotating vane machines have no clearance volume and can work at high pressure ratios.

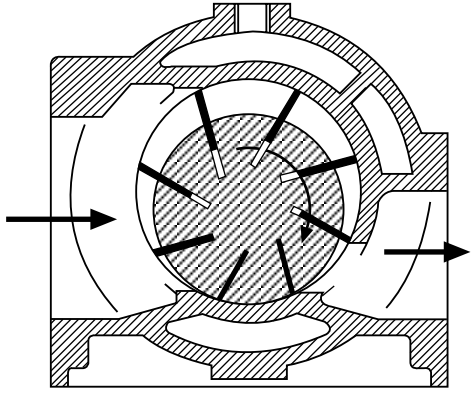


Figure 4.12 *Rotary vane compressor (Courtesy of Hick, Hargreaves & Co. Ltd)*

Larger rotating vane compressors are limited in application by the stresses set up by the thrust on the tips of the blades, and are used at low discharge pressures such as the first stage of a compound cycle. Smaller compressors, up to 110 kW cooling capacity, are now available for the full range of working pressures. These also incorporate a spring-loaded safety plate to relieve excess pressure if liquid refrigerant enters (see Figure 4.13).

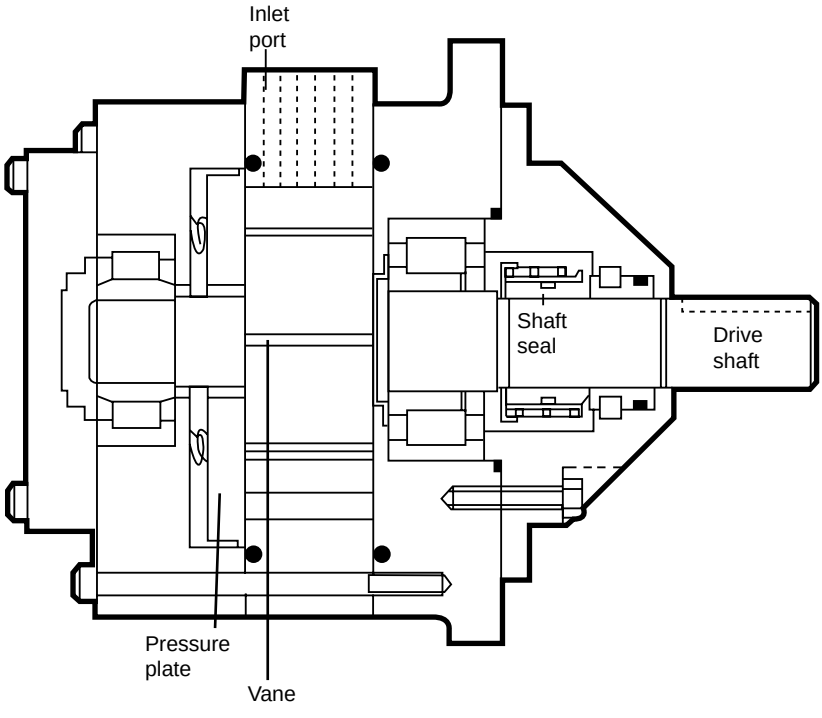


Figure 4.13 *Rotary vane compressor (Courtesy of Rotocold Ltd)*

Sliding vane or *rolling piston* compressors have one or two blades, which do not rotate, but are held by springs against an eccentric rotating roller. These compressors require discharge valves. This type has been developed extensively for domestic appliances, packaged air-conditioners and similar applications, up to a cooling duty of 15 kW (see Figure 4.14).

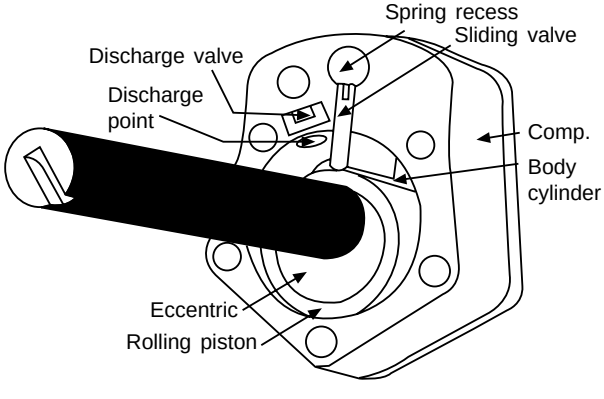


Figure 4.14 *Rolling piston compressor (Courtesy of Rotorx Company)*

4.11 Screw compressors

The screw compressor can be visualized as a development of the gear pump. For gas pumping, the rotor shapes are modified to give maximum swept volume and no clearance volume where the rotors mesh together, and the pitch of the helix is such that the inlet and outlet ports can be arranged at the ends instead of at the side. The solid portions of the screws slide over the gas ports to separate one stroke from the next, so that no extra inlet or outlet valves are needed.

The more usual form has twin meshing rotors on parallel shafts (see Figure 4.15). As these turn, the space between two grooves comes opposite the inlet port, and gas enters. On further rotation, this pocket of gas is cut off from the inlet port and moved down the barrels. A meshing lobe of the male rotor then compresses the pocket, and the gas is finally released at the opposite end, when the exhaust port is uncovered by the movement of the rotors. Sealing between the working parts is usually assisted by the injection of oil along the length of the barrels. This extra oil must be separated from the discharge gas, and is then cooled and filtered before returning to the lubrication circuit (see Chapter 5).

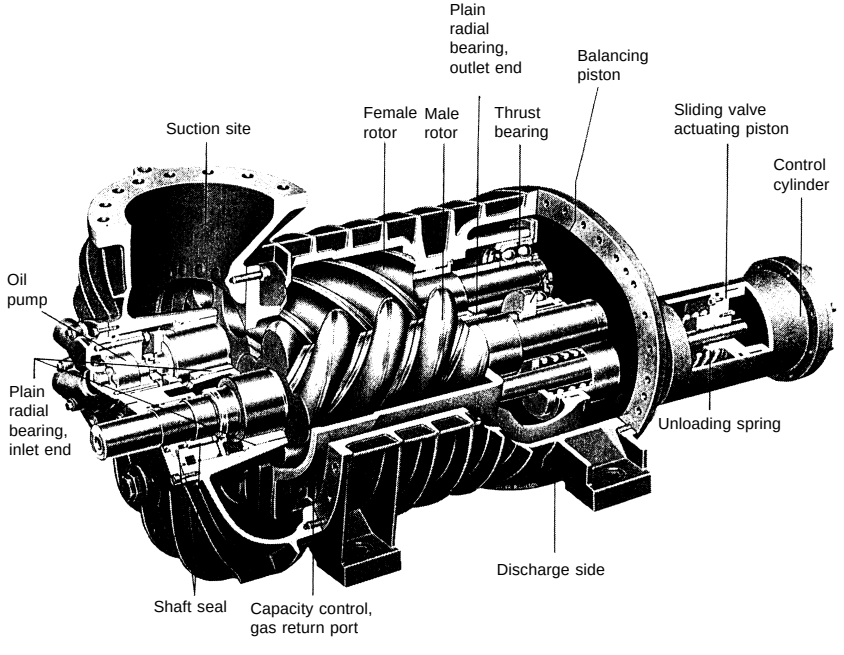


Figure 4.15 *Twin-screw compressor. (Courtesy of Stal Refrigeration AB)*

The other form has a single grooved rotor, with rotating star tooth seal vanes to confine the pockets of gas as they move along the rotor slots (see Figure 4.16). Gas sealing at these surfaces is effected by injecting a small amount of the liquid refrigerant. This obviates the need for the oil lubrication and cooling circuit, with its pumps, and leaves the compressor and the circuit oil-free [17].

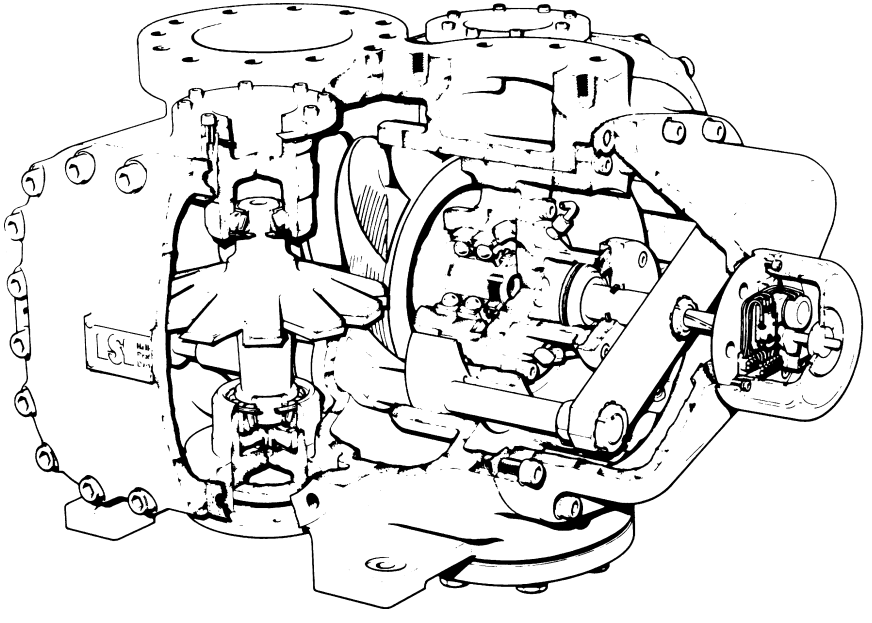


Figure 4.16 *Single-screw compressor (Courtesy of APV Hall Products Ltd)*

Screw compressors have no clearance volume, and may work at high compression ratios without loss of 'volumetric efficiency'. In all screw compressors, the gas volume will have been reduced to a pre-set proportion of the inlet volume by the time the outlet port is uncovered, and this is termed the built-in pressure ratio. At this point, the gas within the screws is opened to condenser pressure and gas will flow inwards or outwards through the discharge port if the pressures are not equal.

The absorbed power of the screw compressor will be at its optimum only when the working pressure ratio is the same as that of the built-in one. This loss of efficiency is acceptable since the machine has no valves and no working parts other than the screws and sealing vanes.

Capacity reduction of the twin-screw compressor is effected by a sliding block covering part of the barrel wall, which permits gas to

pass back to the suction, so varying the working stroke. Variation down to 10% of maximum is usual.

The oil separation, cooling and filtering for a twin-screw compressor adds to the complexity of an otherwise simple machine. Some commercial screw compressors are available which have the oil-handling circuit built into the assembly, with a small loss of overall efficiency.

4.12 Scroll compressor

A positive displacement gas compressor can be constructed with a pair of nesting volutes, one stationary and one orbiting (see Figure 4.17). Gas enters from the surrounding enclosure (Figure 4.17*a*), is trapped between the volutes and moved inwards (*a, b, c, d* etc.)

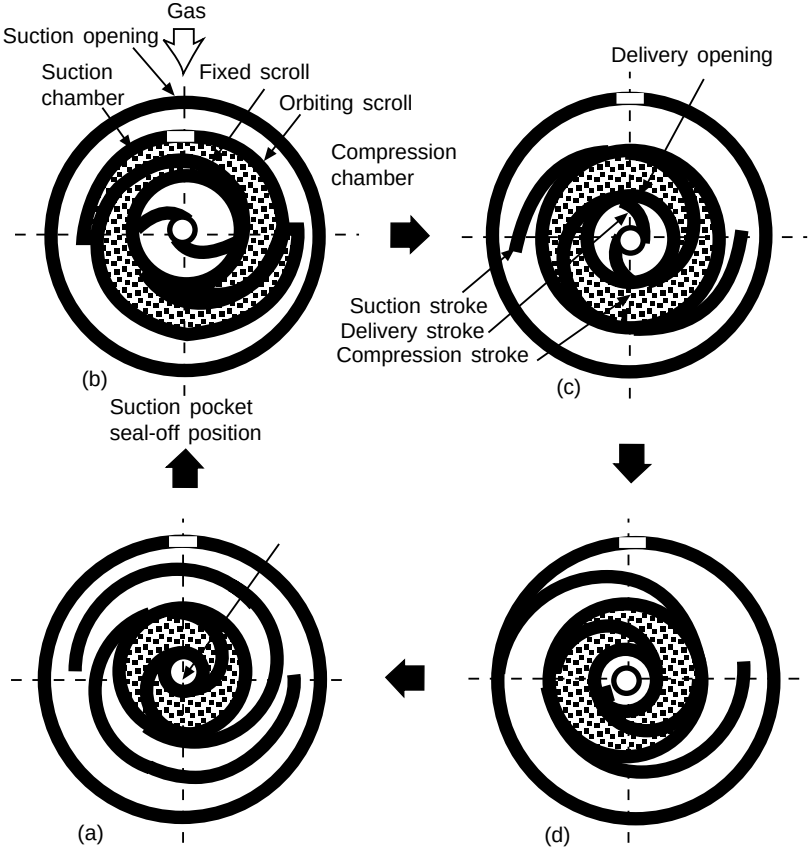


Figure 4.17 *Scroll compressor (Courtesy of Climate Equipment Ltd)*

until it is finally forced out through the central discharge port. Owing to the close manufacturing tolerances the scroll compressor is built only in hermetic enclosed models. The dynamic and gas pressure loads are balanced so that it is free of vibration. It is currently available in cooling capacities up to 60 kW, and is being made in larger sizes as development proceeds.

Capacity control of these compressors is achieved by varying the compressor speed by means of an inverter motor.

4.13 Dynamic compressors

Dynamic compressors impart energy to the gas by velocity or centrifugal force and then convert this to pressure energy. The most common type is the centrifugal compressor. Suction gas enters axially into the eye of a rotor which has curved blades, and is thrown out tangentially from the blade circumference.

The energy given to gas passing through such a machine depends on the velocity and density of the gas. Since the density is already fixed by the working conditions, the design performance of a centrifugal compressor will be decided by the rotor tip speed. Owing to the low density of gases used, tip speeds up to 300 m/s are common. At an electric motor speed of 2900 rev/min, a single-stage machine would require an impeller 2 m in diameter. To reduce this to a more manageable size, drives are geared up from standard-speed motors or the supply frequency is changed to get higher motor speeds. The drive motor is integral with the compressor assembly, and may be of the open or hermetic type. On single-stage centrifugal compressors for air-conditioning duty, rotor speeds are usually about 10 000 rev/min.

Gas may be compressed in two or more stages. The impellers are on the same shaft, giving a compact tandem arrangement with the gas from one stage passing directly to the next. The steps of compression are not very great and, if two-stage is used, the gas may pass from the first to the second without any intercooling of the gas.

Centrifugal machines can be built for industrial use with ammonia and other refrigerants, and these may have up to seven compression stages. With the high tip speeds in use, it is not practical to build a small machine, and the smallest available centrifugal compressor for refrigeration duty has a capacity of some 260 kW. Semi-hermetic compressors are made up to 7000 kW and open drive machines up to 21 000 kW capacity.

Systems of this size require large-diameter refrigerant suction and discharge pipes to connect the components of the complete

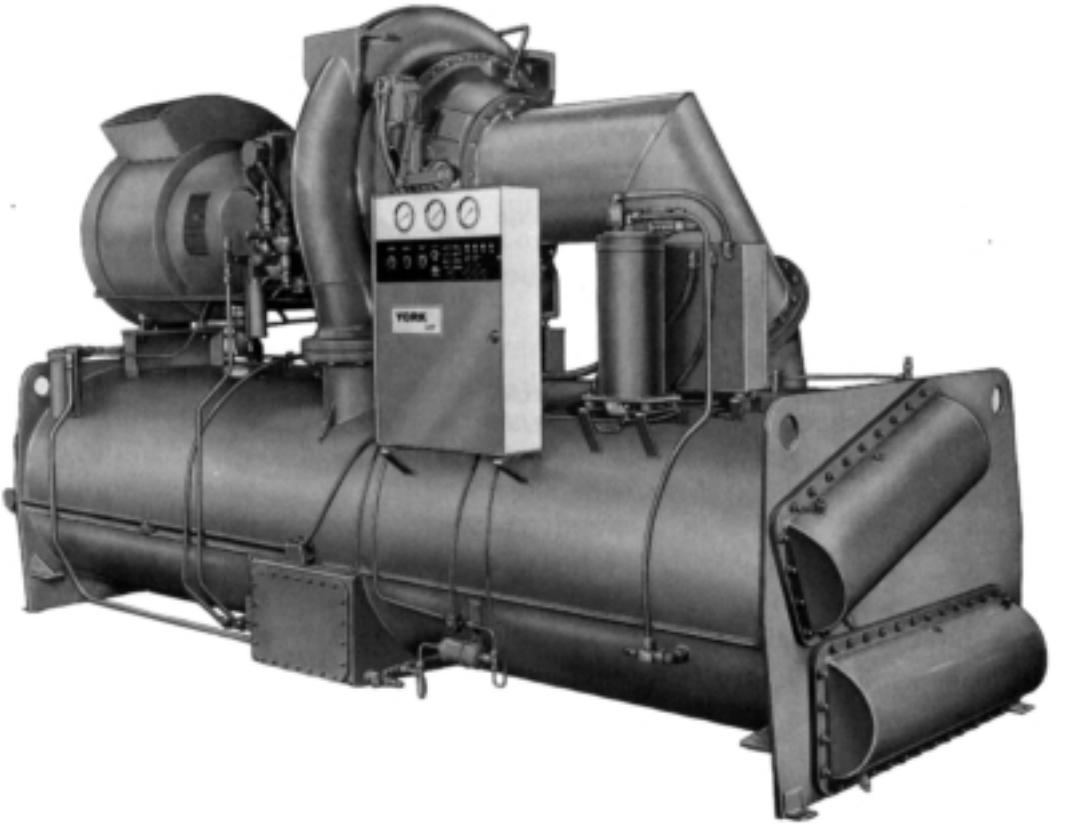


Figure 4.18 *Centrifugal compressor unit (Courtesy of York Division of Borg-Warner Ltd)*

Table 4.1 Model No. PLE08, R.502

Suction		*Condensing at*							
		30°C		35°C		40°C		45°C	
Temperature	*Pressure*	*Capacity*	*Power*	*Capacity*	*Power*	*Capacity*	*Power*	*Capacity*	*Power*
– 50	0.82	5.64	5.70	4.78	5.59	3.87	5.41	3.01	5.18
– 45	1.04	8.49	7.00	7.49	7.00	6.48	6.94	5.48	6.81
– 40	1.31	11.9	8.29	10.7	8.41	9.56	8.48	8.40	8.40
– 35	1.63	15.9	9.65	14.5	9.92	13.1	10.1	11.9	10.3
– 30	2.00	20.6	11.0	19.1	11.5	17.6	11.6	15.9	11.9

system. As a result, and apart from large-scale industrial plants, they are almost invariably built up as liquid-cooling, water-cooled packages with the condenser and evaporator complete as part of a factory-built package (Figure 4.18).

The main refrigerant for packaged water chillers of the centrifugal type are R123 and R134a.

Since centrifugal machines are too big to control by frequent stopping and restarting, some form of capacity reduction must be inbuilt. The general method is to throttle or deflect the flow of suction gas into the impeller. With most models it is possible to reduce the pumping capacity down to 10–15% of full flow. There are no components which require lubrication, with the exception of the main bearings. As a result, the machine can run almost oil-free.

The pumping characteristic of the centrifugal machine differs from the positive displacement compressor since, at excessively high discharge pressure, gas can slip backwards past the rotor. This characteristic makes the centrifugal compressor sensitive to the condensing condition, giving higher duty and a better coefficient of performance if the head pressure drops, while heavily penalizing performance if the head pressure rises. This will vary also with the angle of the capacity reduction blades. Excessive pressure will result in a reverse flow condition, which is followed a fraction of a second later by a boosted flow as the head pressure falls. The vapour surges, with alternate forward and reverse gas flow, throwing extra stress on the impeller and drive motor. Such running conditions are to be avoided as far as possible, by designing with an adequately low head

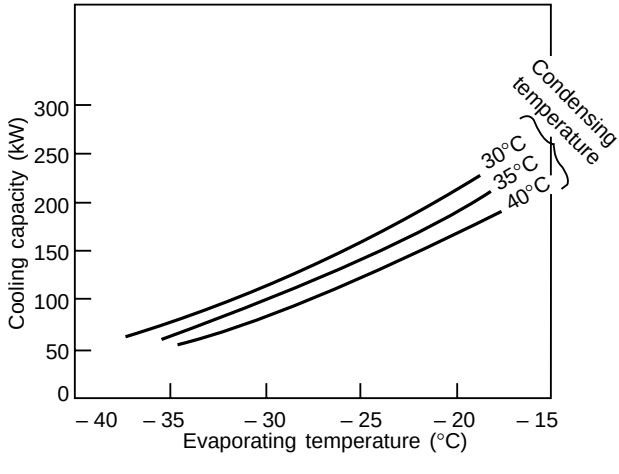


Figure 4.19 *Compressor capacity ratings in graph form*

pressure and by good maintenance of the condenser system. Rating curves indicate the stall or surge limit.

4.14 Capacity ratings

For the convenience of users, the refrigerating effect of compressors is usually tabulated (Table 4.1) or given in graphical form (Figure 4.19), and is shown as the net cooling capacity based on the evaporating and condensing temperatures or pressures. Such published data will include absorbed power and indicate any limitations of the application.

Ratings of this sort may be standardized to certain conditions at the suction, which may not apply to a particular use and need to be interpreted. (See also Chapter 28.)

5 Oil in refrigerant circuits

5.1 Oil specifications

The behaviour of lubricating oil in a refrigerant circuit and its physical interaction with the refrigerant itself is a dominant factor in the design of circuits in general and evaporators in particular.

Refrigeration compressors are mechanical devices with component parts which slide together, so requiring lubrication to reduce friction, remove frictional heat and assist with gas sealing.

Lubricants for general commercial systems are based on mineral oils, and the following properties are required of the lubricant selected:

1. It must be compatible with the refrigerant, i.e. not form any compounds or promote chemical activity.
2. The mixture with the refrigerant in the lubrication circuit must provide adequate lubrication of the working parts.
3. It must not solidify or throw out any solids such as waxes, within the working range, or clog strainers or driers.
4. It must be free of water or other contaminants which will affect the system.
5. It must not be prone to foaming.
6. It must be resistant to oxidation (high flash-point).
7. It must have a low vapour pressure.
8. For hermetic and semi-hermetic compressors, it must have a high dielectric strength.

A large variety of oils is available, and recommendations for any set of conditions, compressor type and refrigerant can be obtained from the refiners. They are naphthene or paraffin-based oils. Synthetic lubricants have been developed for ultra-low- and high-temperature systems, especially for process heat pumps.

5.2 Oil separators

During the compression stroke of a reciprocating machine, the gas becomes hotter and some of the oil on the cylinder wall will pass out with the discharge gas. To reduce the amount of this oil which will be carried around the circuit, an oil separator is frequently fitted in the discharge line (see Figure 5.1). The hot entering gas is made to impinge on a plate, or may enter a drum tangentially to lose much of the oil on the surface by centrifugal force. Some 95–98% of the entrained oil may be separated from the hot gas and fall to the bottom of the drum, and can be returned to the crankcase. The oil return line will be controlled by a float valve, or may have

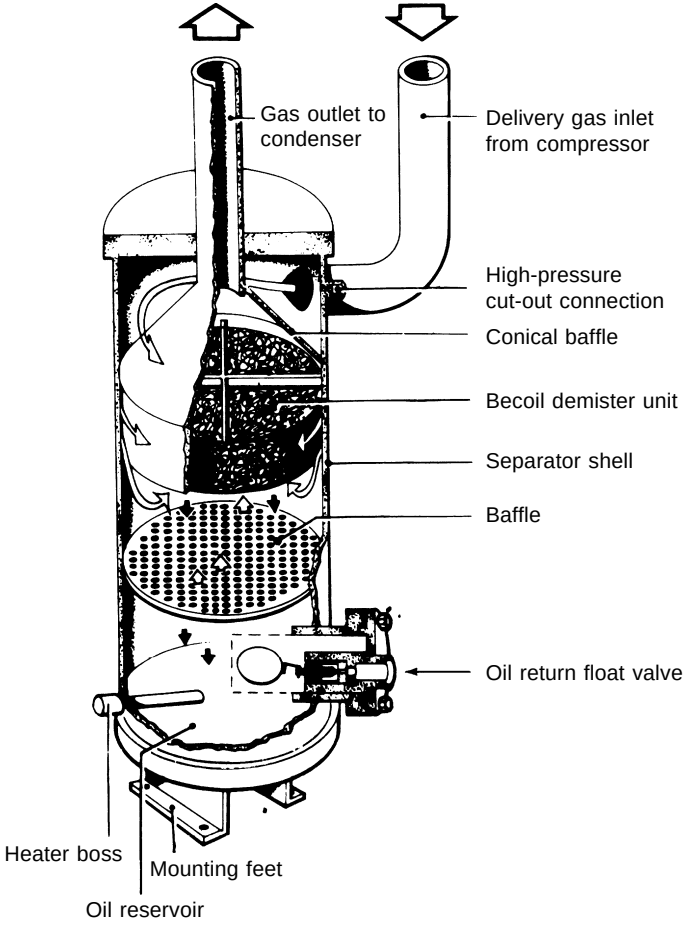


Figure 5.1 *Oil separator (Courtesy of APV Baker Ltd (Hall Division))*

a bleed orifice. In either case, this metering device must be backed up by a solenoid valve to give tight shut-off when the compressor stops, since the separator is at discharge pressure and the oil sump at suction.

On shut-down, high-pressure gas in the separator will cool and some will condense into liquid, to dilute the oil left in the bottom. When the compressor restarts, this diluted oil will pass to the sump. In order to limit this dilution, a heater is commonly fitted into the base of the separator.

For installations which might be very sensitive to accumulations of oil, a two-stage oil separator can be fitted. The second stage cools the gas to just above condensing temperature, and up to 99.7% of the entrained oil can be removed. Even so, a small quantity will be carried over. Sliding vane and screw compressors may have extra oil injected into the casing to assist with sealing, and this must be separated out and re-cooled.

5.3 Oil circulation

Traces of oil which enter the condenser will settle on the cooling surfaces and fall to the bottom as a liquid with the condensed refrigerant. The two liquids will then pass to the expansion valve and into the evaporator. Here, the refrigerant will change to a vapour but most of the oil will remain as a liquid, slight traces of the latter passing out as a low-pressure vapour with the suction gas. It is necessary to limit the build-up of liquid oil in the evaporator, since it would quickly concentrate, reducing heat transfer and causing malfunction.

Methods of limiting oil accumulation in the evaporator depend on the ease with which the liquids mix, and their densities. These properties (see Table 5.1) indicate that different problems exist

Table 5.1 Miscibility of oil with liquid refrigerants

Refrigerant	*At* 0°C	*At* 35°C	*Specific mass* (kg/m^3)
R134a	Fully miscible	Fully miscible	1295
R.22	Separates into oil-rich mixture at top and refrigerant-rich mixture at bottom	Fully miscible	1177
R.717	Non-miscible	Non-miscible	596
Oil			910

with refrigerants in general use. The extent of miscibility and the consideration of liquid density divides the problem of oil separation and circulation into two distinct classes.

With *ammonia*, oil sinks to the bottom and does not go into solution with the refrigerant. Ammonia condensers, receivers and evaporators can be distinguished by the provision of oil drainage pots and connections at the lowest point. Automatic drainage and return of the oil from these would have to depend on the different densities, and is very rarely fitted. The removal of oil from collection pots and low-point drains is a periodic manual function and is carried out as part of the routine maintenance. The *halocarbons* are all sufficiently miscible with oil to preclude the possibility of separate drainage in this way.

Evaporators containing a large body of R.22 will have a greater concentration of oil in the upper layers. By bleeding off a proportion of the mixture (about 10% of the mass flow) and separating the oil from this by distillation, the concentration can be held to an acceptable working limit (see Figure 5.2). Since the addition of outside heat for this distillation would be a direct waste of energy, the heat is obtained from the warm liquid passing from the condenser to the expansion valve.

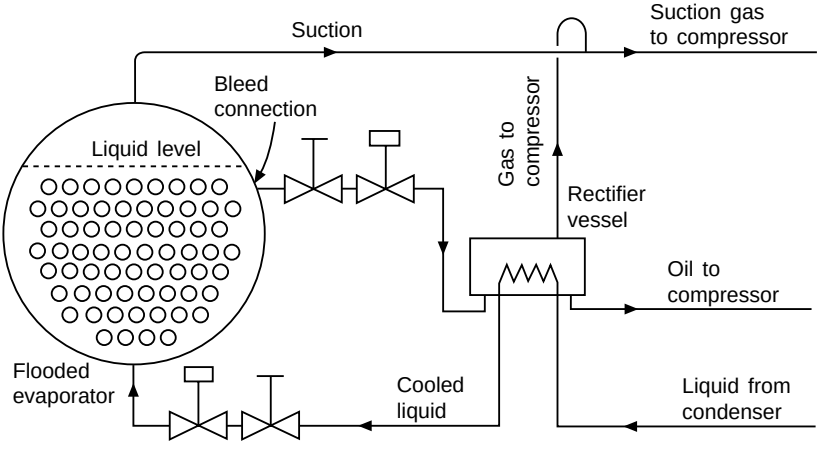


Figure 5.2 *Oil bleed and rectifier for R.22 flooded evaporator*

5.4 Dry expansion circuit

The alternative method of returning oil from the evaporator to the compressor is to keep it moving, by ensuring a minimum continuous fluid velocity in all parts of the circuit. This is termed the dry expansion circuit. This dynamic circulation method is the decisive

factor in the design of nearly all halocarbon evaporators, the exceptions being 'flooded' evaporators (see Chapter 7).

The critical section of the circuit (Figure 5.3) is where there is no liquid refrigerant left to help move the oil, i.e. the evaporator outlet and the suction pipe back to the compressor. Entrainment velocities of 5–7 m/s are required to ensure that oil droplets will be carried back by the dry refrigerant gas to the compressor. The principle of continuous fluid velocity means that the evaporator will be in a continuous circuit. This does not imply that it has to be one pipe, since many pipes may be arranged in parallel to get the required heat transfer surface, providing the minimum velocity criteria are met.

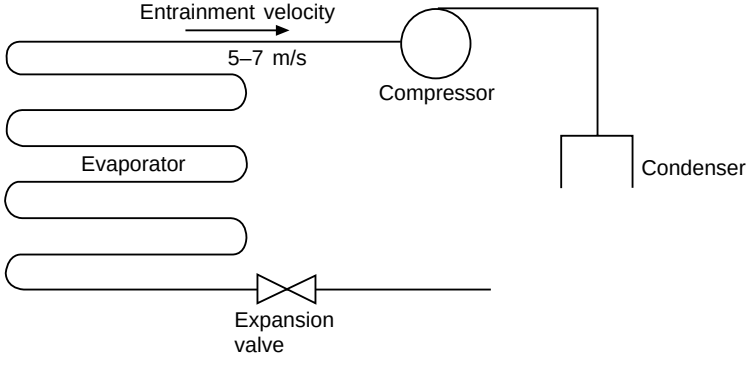


Figure 5.3 *Dry expansion circuit*

Some small cooling circuits have reversing refrigerant flow (i.e. cooling/heat pump) and may work at reduced gas flow for capacity control. Under such conditions it may not be possible to maintain the minimum velocity to carry oil back to the compressor, and it will settle in the circuit. Arrangements must be made to increase or reverse the gas flow periodically to move this oil.

5.5 Contaminants in oil

The oil in a refrigeration system should remain as clean as it is when it enters the compressor (unlike that of the automobile engine which is quickly contaminated by fuel, water, carbon and atmospheric dust). The condition of the compressor oil is therefore a direct indication of the physical and chemical cleanliness of the system.

Lubricating oil should be kept in tightly sealed containers to exclude atmospheric moisture. Oil drained from oil pots and drains is not used again unless it can be properly filtered and kept dry.

The oil as seen through the crankcase sight glass should remain

transparent. If it takes on a white, emulsified appearance it is wet and should be drained and discarded.

Overheating or an electrical fault in the winding of a hermetic or semihermetic compressor motor will produce contaminants, including the halogen acids, which can be detected by their acrid smell, litmus paper or other tests [18]. Eye goggles and rubber gloves should be worn when handling such suspect oil. If shown to be acid, the oil must be removed and carefully disposed of, and the system thoroughly cleaned out [19, 20].

6 Condensers and water towers

6.1 General

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

6.2 Heat to be removed

The total heat to be removed in the condenser is shown in the *p–h* diagram (Figure 6.1) and, apart from comparatively small heat losses and gains through the circuit, will be

Heat taken in by evaporator + heat of compression

This latter, again ignoring small heat gains and losses, will be the net shaft power into the compressor, giving

Evaporator load + compressor power = condenser load

Condenser rating is correctly stated as the rate of heat rejection. Some manufacturers give ratings in terms of the evaporator load, together with a 'de-rating' factor, which depends on the evaporating and condensing temperatures.

Evaporator load × factor = condenser load

Example 6.1 The following figures from a compressor catalogue give the cooling capacity in British thermal units per hour $\times 10^{-3}$ and the shaft horsepower, for a range of condensing temperatures

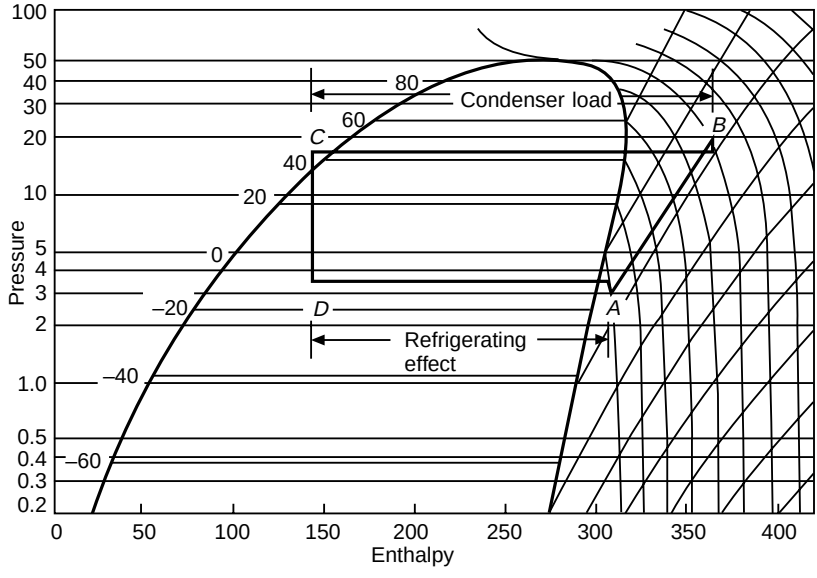


Figure 6.1 *Condenser load p–h diagram*

and one evaporator temperature. Calculate the condenser capacities in each case.

	Condensing temperature (°F)					
	75	80	85	90	95	100
Btu/h $\times 10^{-3}$	874.6	849.7	824.3	798.3	771.7	744.6
Shaft h.p.	54.3	58.1	61.7	65.1	68.3	71.4

Dividing the British thermal units per hour by 3412 to get kilowatts, and multiplying the shaft horsepower by 0.746, also to get kilowatts, and then adding, we get the condenser capacity:

Ambient	75°F	80°F	85°F	90°F	95°F	100°F
Cooling load (kW)	256	249	242	234	226	218
Compressor power (kW)	41	43	46	49	51	53
Condenser load (kW)	297	292	288	283	277	271

Example 6.2 A condenser manufacturer gives a 'heat rejection capacity factor' at 26°C wet bulb temperature of 1.22. What is the condenser duty if the cooling capacity is 350 kW?

Condenser duty = cooling capacity × factor

$$= 350 \times 1.22$$

$$= 427 \text{ kW}$$

The provision of a separate oil cooler will reduce condenser load by the amount of heat lost to the oil and removed in the oil cooler. This is of special note with twin-screw compressors, where a high proportion of the compressor energy is taken away in the oil. This proportion varies with the exact method of oil cooling, and figures should be obtained from the compressor manufacturer for a particular application.

6.3 Air-cooled condensers

The simplest air-cooled condenser consists of a plain tube containing the refrigerant, placed in still air and relying on natural air circulation. An example is the condenser of the domestic refrigerator, which may also have some secondary surface in the form of supporting and spacer wires.

Above this size, the flow of air over the condenser surface will be by forced convection, i.e. fans. The high thermal resistance of the boundary layer on the air side of the heat exchanger leads to the use, in all but the very smallest condensers, of an extended surface. This takes the form of plate fins mechanically bonded onto the refrigerant tubes in most commercial patterns. The ratio of outside to inside surface will be between 5 : 1 and 10 : 1.

Flow of the liquefied refrigerant will be assisted by gravity, so the inlet will be at the top of the condenser and the outlet at the bottom. Rising pipes should be avoided in the design, and care is needed in installation to get the pipes level.

The flow of air may be vertically upwards or horizontal, and the configuration of the condenser will follow from this (see Figure 6.2). Small cylindrical matrices are also used, the air flowing radially inwards and out through a fan at the top.

Forced convection of the large volumes of air at low resistance leads to the general use of propeller or single-stage axial flow fans. Where a single fan would be too big, multiple smaller fans give the advantages of lower tip speed and noise, and flexibility of operation in winter (see Section 6.12). In residential areas slower-speed fans may be specified to reduce noise levels. A smaller air flow will de-rate the condenser, and manufacturers will give ratings for ‘standard’ and ‘quiet’ products.

It will be recognized that the low specific heat capacity and high

Figure 6.2 *Air-cooled condenser (Courtesy of Techni-Coils Ltd)*

specific volume of air implies a large volume to remove the condenser heat. If the mass flow is reduced, the temperature rise must increase, raising the condensing temperature and pressure to give lower plant efficiency. In practice, the temperature rise of the air is kept between 9 and 12 K. The mass flow, assuming a rise of 10.5 K, is then

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

where 1.02 is the specific heat capacity of ambient air.

As an example of these large air flows required, the condenser for an air-conditioning plant for a small office block, having a cooling capacity of 350 kW and rejecting 430 kW, would need 40.85 kg/s or about 36 m^3/s of air. This cooling air should be as cold as possible, so the condenser needs to be mounted where such a flow of fresh ambient air is available without recirculation.

The large air flows needed, the power to move them, and the resulting noise levels are the factors limiting the use of air-cooled condensers.

Materials of construction are aluminium fins on stainless steel tube for ammonia, or aluminium or copper fins on aluminium or copper tube for the halocarbons. Aluminium tube is not yet common, but its use is expected to increase.

In view of the high material cost for air-cooled condensers

compared with other types, a higher ln MTD is usually accepted, and condensing temperatures may be 5–8 K higher for a given cooling medium temperature. Air-cooled condensers must, of course, be used on land transport systems. They will also be used in desert areas where the supply of cooling water is unreliable.

6.4 Water-cooled condensers

The higher heat capacity and density of water make it an ideal medium for condenser cooling and, by comparison with the 350 kW plant cited above, the flow is only 9.8 litre/s. Small water-cooled condensers may comprise two concentric pipes ('double pipe'), the refrigerant being in either the inner tube or the annulus. Configurations may be straight, with return bends or headers, or coiled (Figure 6.3). The double-pipe condenser is circuited in counterflow (media flowing in opposite directions) to get the most subcooling, since the coldest water will meet the outgoing liquid refrigerant.

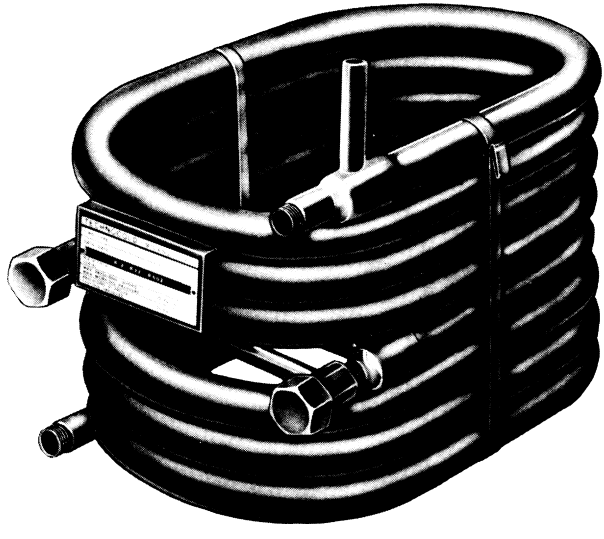


Figure 6.3 *Double-pipe water-cooled condenser (Courtesy of Hubbard Commercial Products Ltd)*

Larger sizes of water-cooled condenser require closer packing of the tubes to minimize the overall size, and the general form is shell-and-tube, having the water in the tubes (Figure 6.4). This construction is a very adaptable mechanical design and is found in all sizes from 100 mm to 1.5 m diameter and in lengths from 600 mm to 6 m, the

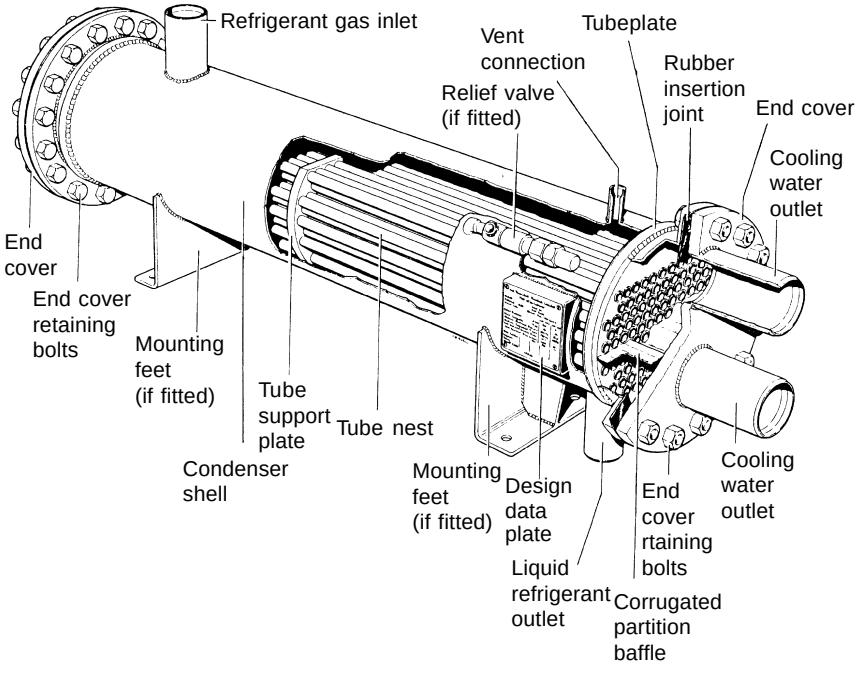


Figure 6.4 *Shell-and-tube condenser (Courtesy of APV Baker Ltd (Hall Division))*

latter being the length of commercially available tubing. Materials can be selected for the application and refrigerant, but all mild steel is common for fresh water, with cupronickel or aluminium brass tubes for salt water.

Some economy in size can be effected by extended surfaces on the refrigerant side, usually in the form of low integral fins formed on the tubes. On the water side, swirl strips can be fitted to promote turbulence, but these interfere with maintenance cleaning and are not much in favour. Water velocity within the tubes is of the order of 1 m/s, depending on the bore size. To maintain this velocity, baffles are arranged within the end covers to direct the water flow to a number of tubes in each 'pass'. Some condensers have two separate water circuits (double bundle, Figure 6.5), using the warmed water from one circuit as reclaimed heat in another part of the system. The main bundle rejects the unwanted heat. Where the mass flow of water is unlimited (sea, lake, river or cooling tower), the temperature rise through the condenser may be kept as low as 5 K, since this will reduce the ln MTD with a lowering of head pressure at the cost only of larger water pumps and pipes.

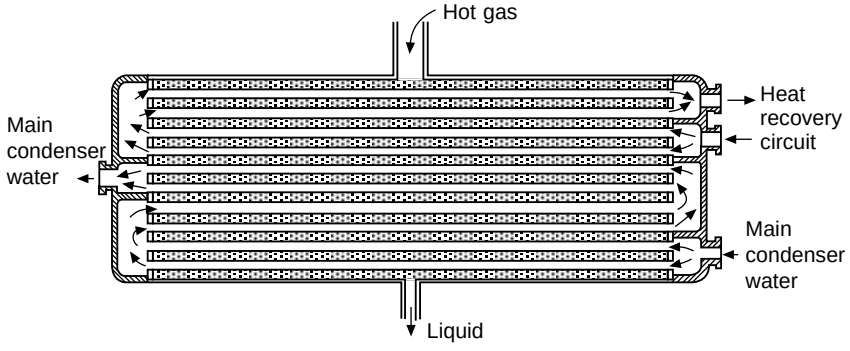


Figure 6.5 *Double-bundle shell-and-tube condenser*

Example 6.3 A condenser uses water from a river with a temperature rise of 5.2 K. Total duty at the condenser is 930 kW. How much water flow is required?

$$\frac{930}{5.2 \times 4.187} = 43 \text{ kg/s}$$

If, however, water is used once through only, and is then rejected to a drain, the range will be much higher, possibly 10–12 K.

Example 6.4 A small water-cooled condenser uses mains water at 13°C and heats this to 24°C before it goes to waste. The evaporator duty is 4.2 kW and the motor output is 1.7 kW. What is the water mass flow?

Condenser load = 4.2 + 1.7

= 5.9 kW

$$\text{Mass flow} = \frac{5.9}{(24 - 13) \times 4.187}$$

= 0.13 kg/s

Shell-and-tube condensers can be installed with the axis vertical and will be one-pass, the water falling to an outlet tank below. This arrangement permits tube cleaning while the plant is operating.

The supply of water is usually limited and requires the use of a cooling tower. Other possibilities are worth investigation; for example, in the food industries, large quantities of water are used for processing the product, and this could be passed first through the condensers if precautions are taken to avoid contamination. Also, where ground water is present, it could be taken from a borehole and afterwards returned to the ground at some distance from the suction. In both

these cases, water would be available at a steady temperature and some 8–10 K colder than summer water from a cooling tower.

6.5 Cooling towers

In a cooling tower, cooling of the main mass of water is obtained by the evaporation of a small proportion into the airstream. Cooled water leaving the tower will be 3–8 K warmer than the incoming air *wet bulb* temperature. (See also Chapters 24 and 25.) The quantity of water evaporated will take up its latent heat equal to the condenser duty, at the rate of about 2430 kJ/kg evaporated, and will be approximately

$$\frac{1}{2430} = 0.41 \times 10^{-3}\ \text{kg/(s kW)}$$

and, for the plant capacity in Example 6.2, would evaporate at 0.18 kg/s.

Cooled water from the drain tank is taken by the pump and passed through the condenser, which may be built up with the compressor as part of a compressor–condenser package (condensing unit). The warmed water then passes back to sprays or distribution troughs at the top of the tower and falls in the upgoing airstream, passing over packings which present a large surface to the air. Evaporation takes place, the vapour obtaining its latent heat from the body of the water, which is therefore cooled (see Figure 6.6).

6.6 Evaporative condensers

This cooling effect of the evaporation of water can be applied directly to the condenser refrigerant pipes in the evaporative condenser (Figure 6.7). The mass flow of water over the condenser tubes must be enough to ensure wetting of the tube surface, and will be of the order of 80–160 times the quantity evaporated. The mass flow of air must be sufficient to carry away the water vapour formed, and a compromise must be reached with expected variations in ambient conditions. An average figure is 0.06 kg/(s kW).

Example 6.5 A water tower serves a condenser rated at 880 kW and the water-circulating pump takes another 15 kW. What will be the evaporation rate, the approximate circulation rate, and the air mass flow?

Total water tower duty = 880 + 15

= 895 kW

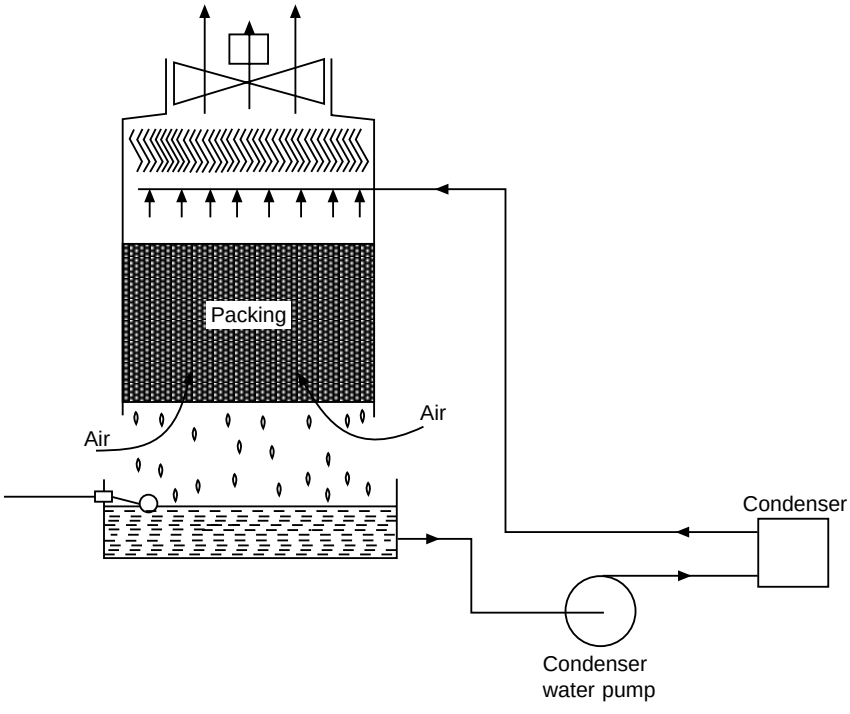


Figure 6.6 *Water tower circuit*

$$\text{Evaporation rate} = 895 \times 0.41 \times 10^{-3}$$
$$= 0.37 \text{ kg/s}$$
$$\text{Circulation rate, 80 times} = 30 \text{ kg/s } (\Delta T = 7.1 \text{ K})$$
$$160 \text{ times} = 60 \text{ kg/s } (\Delta T = 3.6 \text{ K})$$
$$\text{Air flow} = 895 \times 0.06$$
$$= 54 \text{ kg/s}$$

It will be seen that the water and air mass flow rates over a cooling tower are roughly equal.

Evaporative condensers have a higher resistance to air flow than cooling towers and centrifugal fans are often used, ganged together to obtain the required mass flow without undue size. This arrangement is also quieter in operation than axial flow fans. Most types use forced draught fans (Figure 6.7).

Cooling towers and evaporative condensers may freeze in winter if left operating on a light load. A common arrangement is to switch off the fan(s) with a thermostat, to prevent the formation of ice. The water-collection tank will have an immersion heater to reduce

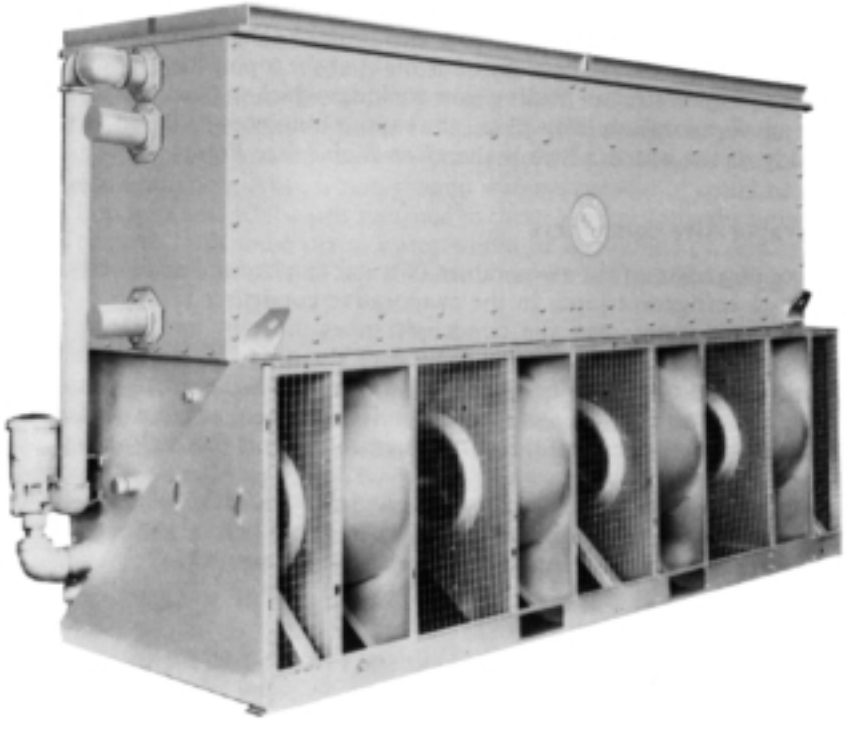


Figure 6.7 *Evaporative condensers (Courtesy of Baltimore Aircoil Ltd)*

the risk of freezing when the equipment is not in use or the tank may be located inside the building under the tower structure, if such space is conveniently available.

Materials of construction must be corrosion resistant. Steel should be hot galvanized, although some resin coatings may suffice. GRP casings are used by some manufacturers. The water-dispersal packing of a cooling tower is made of treated timber or corrugated plastic sheet.

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

6.7 Water treatment

All water supplies contain a proportion of dissolved salts. These will

tend to be deposited at the hottest part of the system, e.g. the furring of a kettle or hot water pipes. Also, these impurities do not evaporate into an airstream, so where water is being evaporated as part of the cooling process, the salts will remain in the circuit and increase in concentration, thus hastening the furring process.

It is possible to remove all solids from the make-up water, but it is much cheaper to check the concentration by other means. Two general methods are employed. The first relies on physical or chemical effects to delay deposition of scale on the hot surfaces; the second restricts the concentration to a level at which precipitation will not occur. In both cases, the accumulation of solids is removed by bleeding off water from the circuit to drain, in addition to that which is evaporated (see Figure 6.8).

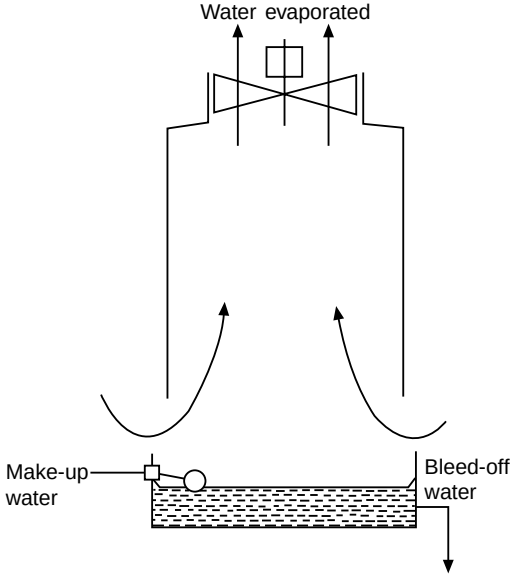


Figure 6.8 *Limitation of solids concentration by bleed-off*

The concentration of solids in the circulating water will increase until the amount carried away by the bleed water compensates for that not carried away in the water vapour. So, if

c_m = concentration of solids in make-up water (kg/kg)

c_b = concentration of solids in bleed-off (kg/kg)

w_e = mass flow of water evaporated (kg/s)

w_m = mass flow of make-up water (kg/s)

Mass of solids entering = mass of solids leaving

$$c_m \times w_m = c_b \times (w_m - w_e)$$

$$w_m = w_e \left(\frac{c_b}{c_b - c_m} \right)$$

The concentration of mains make-up water, c_m, is obtained from the water supply authority. The permissible concentration, c_b, will be decided by the method of water treatment or the assumed concentration of untreated water which will prevent precipitation.

Example 6.6 The hardness of water in Coventry is given as a maximum of 560 ppm (parts per million) and the water treatment can permit a concentration of solids to 1200 ppm. The cooling capacity is 700 kW and the compressor power 170 kW. How much water should be bled to waste and what is the total make-up required?

Cooling tower capacity = 700 + 170

= 870 kW

Latent heat of water vapour = 2420 kJ/kg

$$\text{Rate of evaporation} = \frac{870}{2420}$$

= 0.36 kg/s

$$\text{Rate of make-up} = 0.36 \left(\frac{0.0012}{0.0012 - 0.00056} \right)$$

= 0.68 kg/s

Rate of bleed-off = 0.68 – 0.36

= 0.32 kg/s

In all cases where water is used for cooling, but more especially where it is being evaporated, the hardness figure should be obtained from the local water supply authority. Enquiries should also be made as to possible variations in the supply, since many cities draw their water from two or more catchment areas, and the type and quantity of hardness may change.

Many suppliers now offer water treatment for use in refrigeration condenser circuits, and the merits of different methods need to be assessed before making a choice. The reader is referred to specialist works on the subject [10, 19, 21].

There are several methods of providing a percentage 'bleed-off' from the water circuit:

1. The make-up ball valve can be set a little high so that some water always goes down the overflow pipe. This is rather difficult to set initially, but is reliable and cannot easily be tampered with. It will work at all times, and so will waste water if the plant is not running.
2. A small bleed-off pipe is taken from the pump discharge, with an adjusting valve, and led to waste. This can be more easily adjusted and works only when the condenser is running, but is subject to interference by unauthorized persons.
3. A tundish, having an area possibly 1% of the cross-sectional area of the tower, is located just above the water level and is led to the drain, forming part of the overflow fitting. This will bleed off 1% of the water falling through the tower.

All these methods provide the maximum required rate of bleed-off at all times of the year, and so will waste water at light load conditions. The user should be aware of the essential nature of bleed-off, since cases often occur in dry weather of misguided persons closing off the bleed to 'save water'.

In some locations, it is necessary to drain the tank frequently to clear other contaminants. With careful control, this can be used as the necessary bleed-off.

6.8 Rating and sizing of condensers

Catalogue ratings show heat rejected at a stated condensing temperature and related to the following:

Ambient dry bulb temperature for air-cooled condensers

Available water temperature for water-cooled condensers

Ambient wet bulb temperature for evaporative types

Choice of equipment based on first cost only will almost certainly result in an undersized condenser and a high head pressure.

Example 6.7 In Example 6.1, the required plant capacity is 218 kW and the running time is 2000 h/year at an electricity cost of 5 p/(kW h) and a motor efficiency of 75%. In order to achieve the condensing temperature of 85°F (29.4°C) the condenser would cost £7250, while a smaller condenser for a temperature of 100°F (37.8°C) would cost £4600. (Prices of evaporative condensers at April 1987.) Estimate the break-even time if the larger condenser is fitted.

Condensing temperature	29.4°C	37.8°C
Rated capacity of plant (kW)	242	218
Running time for 218 kW × 2000 h	1802	2000
Compressor power (kW)	46	53
Electricity cost per year (£)	5526	7067
Electricity saving per year (£)	1551	

$$\text{Break-even time} = \frac{7250 - 4600}{1551} = 1.7 \text{ years}$$

This is a rough calculation, based on direct capital cost and not on interest rates, and needs to be analysed in terms of the general plant economics. It should also be borne in mind that this is based on present-day electricity costs, and a greater saving will be made as fuel costs rise.

Tendering contractors and prospective users should make themselves aware of alternatives of this sort.

6.9 Condenser maintenance

As with any mechanical equipment, condensers should never be located where they are difficult of access, since there will then be less chance of routine maintenance being carried out. Periodic maintenance of a condenser is limited to attention to the moving parts – fans, motors, belts, pumps – and cleaning of water filters, if fitted.

The overall performance will be monitored from the plant running log (see Chapter 33) and the heat exchange surfaces must be kept clean for maximum efficiency – meaning the lowest head pressure and lowest power.

Air-cooled surfaces may be cleaned by brushing off the accumulation of dust and fluff where the air enters the coil, by the combination of a high-pressure air hose and a vacuum cleaner, or, with the obvious precautions, by a water hose. Foaming detergents are also used.

Advance warning should be had from the plant running log of any build-up of scale on water-cooled surfaces. Scale within the tubes of a straight double-pipe or shell-and-tube condenser can be mechanically removed with suitable wire brushes or high-pressure water lances, once the end covers have been removed. Tubes which cannot be dealt with in this way must be chemically cleaned (see also Chapter 33).

It will be appreciated that, where air and water are present, as in a water cooling tower or evaporative condenser, the apparatus will act as an air washer, removing much of the dust from the air passing

through it. Such dirt may be caught in a fine water filter, but is more commonly allowed to settle into the bottom of the tank and must be flushed out once or twice a year, depending on the severity of local contamination. Where heavy contamination is expected, it is good practice to provide a deeper tank than usual, the pump suction coming out well clear of the bottom, and tanks 3 m deep are in use. Where plant security is vital, the tank is divided into two parts, which may be cleaned alternately.

Algae and other organisms will tend to grow on wet surfaces, in particular those in daylight. Control of these can be effected by various proprietary chemicals [21].

Cooling towers and evaporative condensers release into the atmosphere fine droplets of water, which may carry sources of contamination such as algae and bacteria. Many of these thrive at the temperatures to be expected in water cooling systems and one of them, *Legionella pneumophila*, has been identified as a particular hazard to health. Cooling apparatus should be cleaned and disinfected frequently to reduce these risks of contamination and should not be located where water droplets can be drawn into ventilation air intakes.

It is now some 20 years since the recognition of *Legionella pneumophila* in condenser water, and the measures taken by industry to combat the hazards to human health. Worldwide indications are that the initial vigilance and care have lessened in recent years and attention is now drawn again to such precautions [27, 28, 28a, 28b].

6.10 Condenser fittings

The inlet pipe bringing high-pressure gas from the compressor must enter at the top of the condenser, and adjacent piping should slope in the direction of flow so that oil droplets and any liquid refrigerant which may form will continue in the right direction and not back to the compressor.

The outlet pipe must always be from the lowest point, but may have a short internal upstand so that any dirt such as pipe scale or metal swarf will be trapped and not taken around the circuit.

Condensers for ammonia systems may have an oil trap, usually in the form of a drain pot, and the liquid outlet will be above this.

Water connections to a shell-and-tube condenser must always be arranged so that the end covers can easily be removed for inspection, cleaning, and repair of the tubes. Heavy end covers require the use of lifting tackle, and supports above the lifting points should be provided on installation to facilitate this work.

Condensers having a gross volume of more than 285 litre are

required under BS.4434: 1980 to have two pressure relief valves or two bursting discs, one always in service. Vessels between 85 litre and 285 litre must have one such relief device. Those below 85 litre but larger than 76 mm inside diameter may have a fusible plug to relieve pressure in a fire. Manufacturers will be aware of the requirements of this BS and similar Standards, and proprietary products will be correctly equipped.

6.11 Other forms of condenser

In a cascade system, the evaporator of the high stage is the condenser for the low stage (see Figure 2.10*a*). Construction of this heat exchanger will be a combination of the design factors for evaporators and condensers, and no general rules apply apart from these. The intercooler of a two-stage or compound system (see Figure 2.9*a*) de-superheats the discharge gas from the first stage so that it will not be too hot on entering the high stage. In practice, it will leave the intercooler only slightly superheated above the interstage saturation point. The normal fluctuations in an operating system may lead to actual condensation at times, but is not so intended.

The small condensing surface required by a domestic appliance such as a deep-freeze may allow the use of the outside metal skin of the appliance itself as a surface condenser. In such a construction, the condenser tube is held in close mechanical contact with the skin, so that heat is conducted through to the outside air, where it is lost by natural convection. This system is restricted to a few hundred watts.

6.12 Winter operation

Condensers are sized so that they can reject the system heat load under maximum conditions of air or water temperature. In colder weather, the condensing temperature will fall with that of the cooling medium and this may cause difficulties in correct operation of the plant. In particular, the pressure across the expansion valve (see also Chapter 8) may be too low to circulate the required mass flow of the refrigerant. Under such circumstances, artificial means must be used to keep the head pressure up, always remembering that the condensing pressure should be kept as low as practical for power economy.

Various systems are used:

1. Air-cooled condensers having two or more fans (Figure 6.2) may have a pressure switch or thermostatic control to stop the fans one by one. This method is simple, cheap, and effective.

2. The fans on such condensers may be fitted with two-speed motors or other speed control. It should be borne in mind that, if one fan of a pair stops, the noise level will fall by 3 dB, but if both fans drop to half speed, the noise drops by 15 dB. This method is of special use in residential areas where the greater noise level will be tolerated in the daytime when condensing air is warmest, but a lower fan speed can be used at night.
3. Evaporative condensers and water cooling towers with two or more fans on separate drive may be controlled in the same way. If a single motor drives several fans on one shaft, speed control or dampers will be required. Evaporative condensers and cooling towers should be fitted with antifreeze thermostats which will stop all fans before the water reaches freezing point.
4. Cooling air flow can be restricted by blanking flaps, baffles or winter enclosures, providing that, if not automatic, the operating staff are aware of their presence and will restore the air flow when the weather turns warm again.
5. Water flow may be restricted by throttling valves. One such device is operated directly by head pressure, but electric or pneumatic throttling or flow diversion valves can be applied for the purpose (see Chapter 9).
6. A set pressure bypass valve can be fitted across the condenser, so that hot gas will pass directly to the receiver in cooler weather. This will cause the condenser to partially fill with liquid refrigerant, thus decreasing the heat transfer surface available for condensation. Sufficient refrigerant must be available for this, without starving the rest of the circuit (see Chapter 9).
7. Where a complex system is served by two or more condensers, a complete condenser can be taken off line by a pressure switch.

Apart from such requirements for head pressure control, winter precautions are needed to prevent freezing of the water while the plant is not rejecting heat to it. These commonly take the form of an electric immersion heater in the water tank, together with lagging and possible trace heating of exposed pipes. In some systems, the evaporative condenser itself may be within the building, with air ducts to the outside. In severe climates, external tanks need to be lagged to conserve the heat provided by the immersion heater.

6.13 Receivers

The total refrigerant charge required in a circuit will vary with different operating loads and ambients, and must be sufficient at all times so that only liquid enters the expansion valve. This implies

that, at times, the circuit would have too much charge, which would back up in the condenser and reduce its efficiency. A drain tank is required directly after the condenser which can hold this reserve of liquid, and is termed the *receiver* (see Figure 6.9).

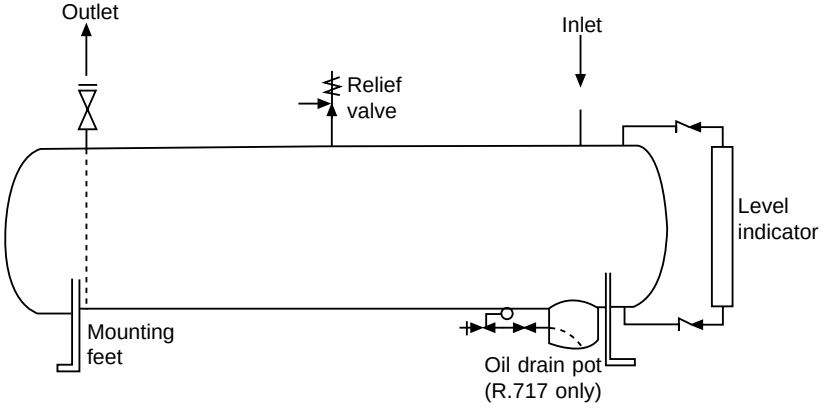


Figure 6.9 *Liquid receiver*

Receivers also act as pump-down tanks, and should be capable of holding enough of the total refrigerant charge to permit evacuation of any one vessel for maintenance, inspection or repair. They should never be more than 85% full, to allow for expansion and safety.

Receivers are commonly made of steel tube with welded dished ends, and are located horizontally. Small receivers may be vertical, for convenience of location. The liquid drain pipe from the condenser to the receiver should be amply sized, and any horizontal runs sloped to promote easy drainage. Shut-off valves in this line should not be in a horizontal outlet from the condenser, since their slight frictional resistance will cause liquid back-up in the condenser. Outlet pipes from the receiver may be from the bottom or, by means of an internal standpipe, may leave at the top. A valve is invariably fitted at this point.

Ammonia receivers may have an oil drum pot, and the receiver will slope slightly down towards this.

Receivers are pressure vessels covered by the provisions of BS.4434:1980 and require safety pressure relief devices as outlined in Section 6.10. In cases where there is no shut-off valve between the condenser and receiver, such protection may be fitted to one or the other, providing the total volume is considered.

In practice, receivers will operate about one-sixth full during normal running. Some means are usually provided to indicate the liquid level inside. These are as follows:

1. An external, vertical sight glass, of suitable pattern, having self-closing shut-off valves.
2. A number of bull's-eye glasses arranged at different heights in the shell.
3. A pair of bull's-eye glasses, arranged on the same cross-section and some 45° up from the horizontal diameter. A light is shone through one and the observer looks through the other.

Example 6.8 The evaporator and condenser of a system hold a total of 115 kg of R.717. Determine the receiver size and dimensions, pressure relief specification, and the total refrigerant charge for the plant.

Required working refrigerant mass = 115 kg

[This must be accommodated in a space 68%
(85% less one-sixth) of the proposed receiver shell.]

$$\text{Gross capacity of receiver} = \frac{115}{68\%}$$

$$= 169 \text{ kg of R.717}$$

$$\text{Specific mass of liquid R.717} = 596 \text{ kg/m}^3$$

$$\text{Volume of receiver for 169 kg gross} = 0.283 \text{ m}^3$$

$$= 283 \text{ litre}$$

The nearest catalogue standard receiver is 240 mm diameter by 3.25 m long and has a gross capacity of 314 litre or 187 kg. Being over the limit of 285 litre, it must have dual relief valves.

$$\text{l system charge} = 115 + 0.15 \times 187$$

$$= 143 \text{ kg}$$

6.14 Dry coolers

The *water-cooled condensers, cooling towers* and *evaporative condensers* described in 6.4, 6.5 and 6.6 all use water at a temperature which will promote the growth of the bacterium *Legionella pneumophila*. While correct water treatment and other precautions are 100% safe against this trouble, users may prefer to avoid any risk by using a sealed water system. Such heat exchangers are generally termed *dry coolers*. They are of similar construction to air cooled condensers, illustrated in Figure 6.2.

The cooling of water or industrial fluid, other than refrigerant, will follow a similar design. The fluid is circulated through the

tubes of the heat exchanger, while blowing air over the outside. The tubes will usually have an extended surface.

The use of dry coolers cannot take advantage of the lower cooling temperatures available by the evaporation of the cooling water, and is limited by the ambient *dry bulb* temperature, rather than the *wet bulb*. Higher power is therefore required, and a given size compressor will perform less cooling duty.

7 Evaporators

7.1 General

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

7.2 Flow pattern and function

The refrigerant flow pattern is dependent on the method of ensuring oil removal from the evaporator and, possibly, its return to the crankcase.

Flooded evaporators (Figure 7.1) have a body of fluid boiling in a random manner, the vapour leaving at the top. In the case of ammonia, any oil present will fall to the bottom and be drawn off from the drain pot or oil drain connection. With the halocarbons, a proportion of the fluid is bled off and rectified (see Figure 5.2).

Evaporators which keep the oil moving by means of continuous fluid velocity, until it gets back to the compressor suction, are termed dry expansion. In these, the refrigerant is totally evaporated.

The function of the evaporator will be to cool gas, liquid or other product load. In most cases air or a liquid is first cooled, and this is then used to cool the load. For example, in a coldroom air is cooled and this air cools the stored produce and carries away heat leaking through the structure; in a water chiller system, the water is circulated to cool the load, etc.

7.3 Air cooling evaporators

Air cooling evaporators for coldrooms, blast freezers, air-conditioning,

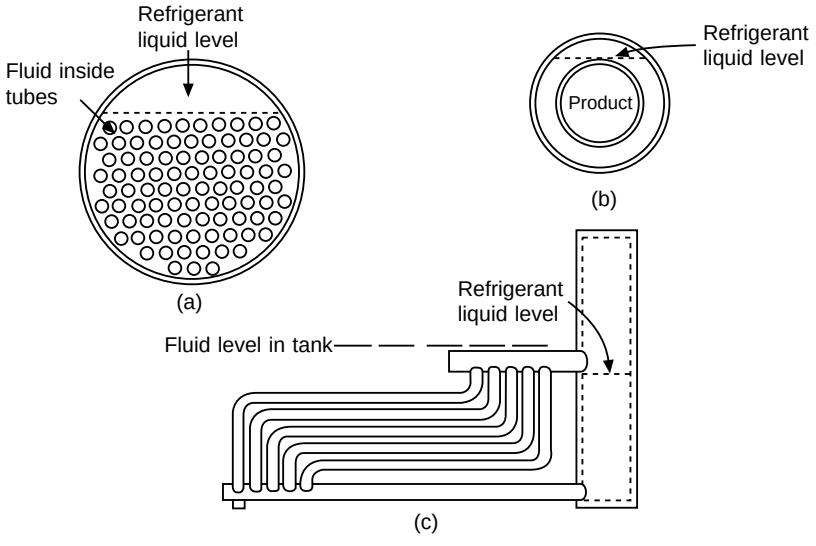


Figure 7.1 *Flooded evaporators. (a) Shell-and-tube. (b) Jacketted. (c) Raceway*

etc., will have finned pipe coils (see Figure 7.2). In all but very small coolers, there will be fans to blow the air over the coil.

Construction materials will be the same as for air-cooled condensers. Aluminium fins on copper tube are the most common for the halocarbons, with stainless steel or aluminium tube for ammonia. Frost or condensed water will form on the fin surface and must be drained away. To permit this, fins will be vertical and the air flow horizontal, with a drain tray provided under.

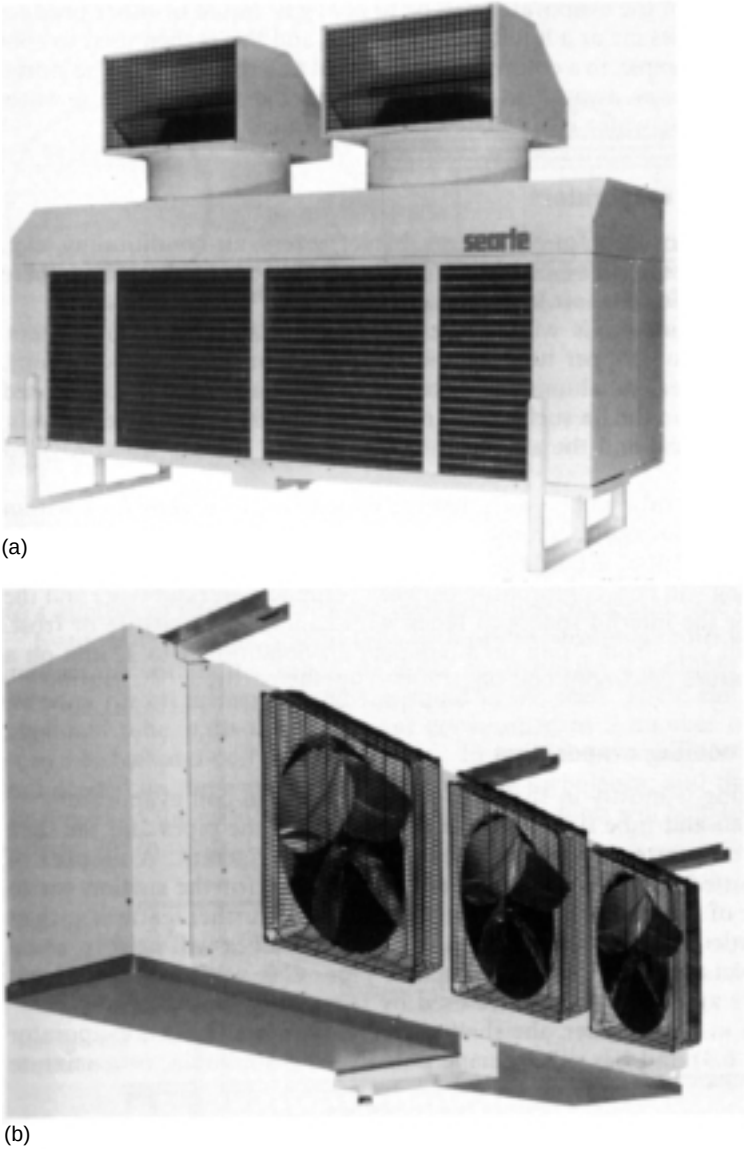
The size of the tube will be such that the velocity of the boiling fluid within it will cause turbulence to promote heat transfer. Tube diameters will vary from 9 mm to 32 mm, according to the size of coil.

Fin spacing will be a compromise between compactness (and cost) and the tendency for the interfin spaces to block with condensed moisture or frost. Spacings will vary from 2 mm on a compact air-conditioner to 12 mm on a low-temperature coldroom coil [8].

7.4 Liquid cooling evaporators

Liquid cooling is mostly in shell-and-tube or shell-and-coil evaporators.

In the shell-and-tube type, the liquid is usually in the pipes and the shell is some three-quarters full of the liquid, boiling refrigerant. A number of tubes is omitted at the top of the shell to give space for the suction gas to escape clear of the surface without entraining



(a)

(b)

Figure 7.2 *Air cooling evaporators. (a) Floor mounted. (b) Ceiling mounted (Courtesy of Searle Manufacturing Co.)*

liquid. Further features such as multiple outlet headers, suction trap domes and baffles will help to avoid liquid droplets entering the main suction pipe. Gas velocities should not exceed 3 m/s and lower figures are used by some designers.

Operated in this manner, the shell-and-tube type is a flooded evaporator (see Figure 7.3) and has oil drainage pots if using ammonia, or a mixture bleed system if the refrigerant is one of the halocarbons. The speed of the liquid within the tubes should be about 1 m/s or more, to promote internal turbulence for good heat transfer. End cover baffles will constrain the flow to a number of passes, as with the shell-and-tube condenser. (See Section 6.4.)

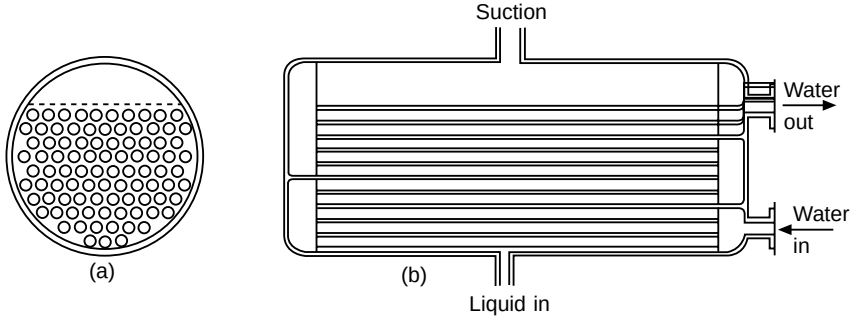


Figure 7.3 *Shell-and-tube evaporator, flooded*

Evaporators of this general type with dry expansion circuits will have the refrigerant within the tubes, in order to maintain a suitable continuous velocity for oil transport, and the liquid in the shell. These can be made as shell-and-tube, with the refrigerant constrained to a number of passes, or may be shell-and-coil (see Figure 7.4). In both these configurations, baffles are needed on the water side to improve the turbulence, and the tubes may be finned on the outside. Internal swirl strips or wires will help to keep liquid refrigerant in contact with the tube wall.

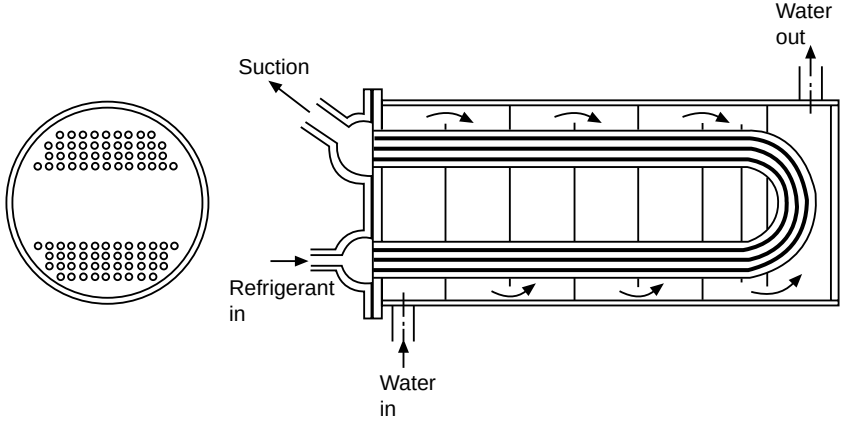


Figure 7.4 *Shell-and-coil evaporator*

Liquid cooling evaporators may comprise a pipe coil in an open tank, and can have flooded or dry expansion circuitry. Flooded coils will be connected to a combined liquid accumulator and suction separator (usually termed the surge drum), in the form of a horizontal or vertical drum (see Figures 7.1c and 7.5). The expansion valve maintains a liquid level in this drum and a natural circulation is set up by the bubbles escaping from the liquid refrigerant at the heat exchanger surface. Dry expansion coils for immersion in an open tank will be in a continuous circuit or a number of parallel circuits (see Figure 7.6). Liquid velocity over such coils can be increased by tank baffles and there may be special purpose agitators, as in an ice-making tank (see Figure 12.1). Coils within an open tank can be allowed to collect a layer of ice during off-load periods, thus providing thermal storage and giving a reserve of cooling capacity at peak load times (see also Chapter 12).

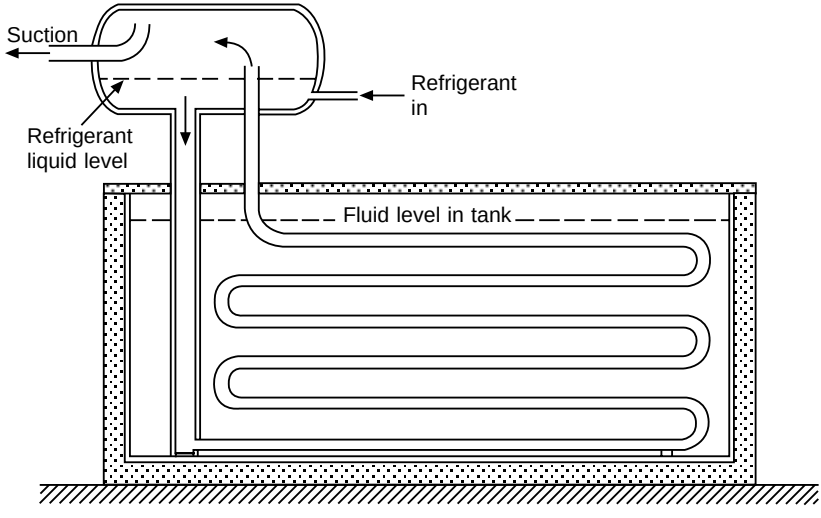


Figure 7.5 *Flooded tank evaporator*

Another type comprises a bank of corrugated plates, forming alternative paths for refrigerant and liquid, similar to that shown in Figure 17.1, of brazed or welded construction.

Where water is to be cooled close to its freezing point without risk of damage to the evaporator, the latter is commonly arranged above the water-collection tank and a thin film of water runs over the tubes. Heat transfer is very high with a thin moving film of liquid and, if any ice forms, it will be on the outside, free to expand, and it will not damage the tube. Such an evaporator is termed a *Baudelot cooler* (Figure 7.7). It may be open, enclosed in dust-tight

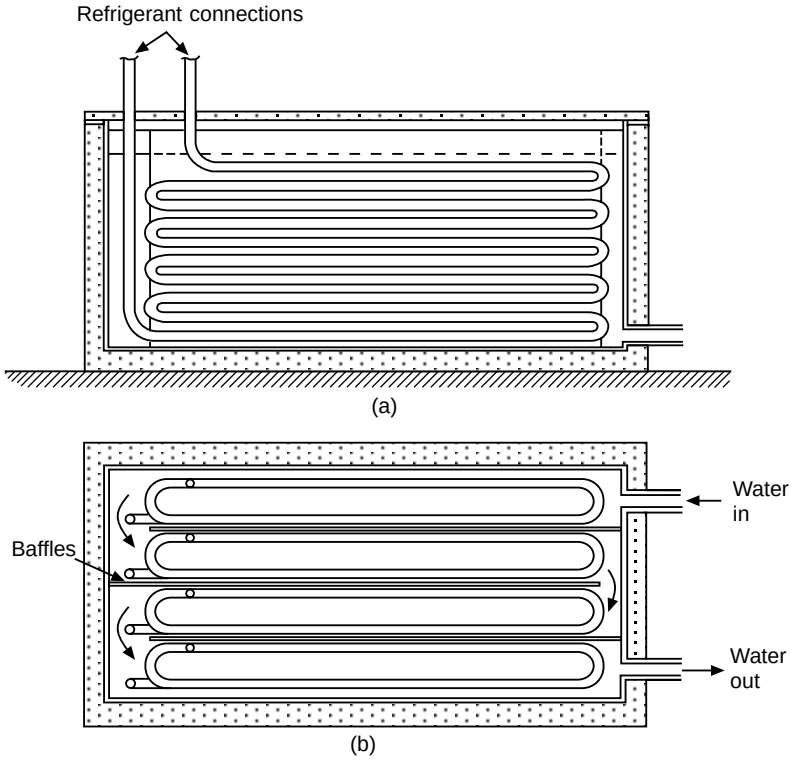


Figure 7.6 *Dry expansion tank evaporator. (a) Section. (b) Elevation*

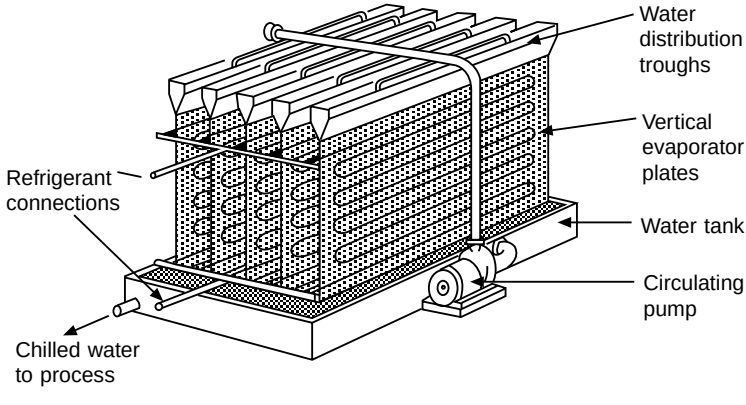


Figure 7.7 *Baudelot cooler*

shields to avoid contamination of the product (as in surface milk and cream coolers), or may be enclosed in a pressure vessel as in the Mojonniér cooler for soft drinks, which pressurizes with carbon dioxide at the same time (Figure 17.5).

Some liquids, such as vegetable fats and ice-cream mixes, increase considerably in viscosity as they are cooled, sticking to the heat exchanger surface. Evaporators for this duty are arranged in the form of a hollow drum (see Figure 7.1*b*) surrounded by the refrigerant and having internal rotating blades which scrape the product off as it thickens, presenting a clean surface to the flow of product and impelling the cold paste towards the outlet.

7.5 Plate evaporators

Plate evaporators (Figure 7.8) are formed by cladding a tubular coil with sheet metal, welding together two embossed plates, or from aluminium extrusions.

The extended flat face may be used for air cooling, for liquid cooling if immersed in a tank, or as a Baudelot cooler.

The major use for flat plate evaporators is to cool a solid product by conduction, the product being formed in rectangular packages and held close between a pair of adjacent plates.

In the horizontal plate freezer (Figure 7.9*a*), the plates are arranged in a stack on slides, so that the intermediate spaces can be opened and closed. Trays, boxes or cartons of the product are loaded between the plates and the stack is closed to give good contact on both sides. When the necessary cooling is complete, the plates are opened and the product removed.

The vertical plate freezer (Figure 7.9*b*) is used to form solid blocks of a wet product, typically fish. When frozen solid, the surfaces are thawed and the blocks pushed up and out of the bank.

To ensure good heat transfer on the inner surface of the plates and achieve a high rate of usage, liquid refrigerant is circulated by a pump at a rate 5–12 times the rate of evaporation.

If a plate evaporator is partially filled with brine (see Figure 7.8*d*) this can be frozen down while the plate is on light load, and the reserve of cooling capacity used at other times. The freezing point of the brine can be formulated according to the particular application and the plate can be made as thick as may be required for the thermal storage needed. The major application of this device is the cooling of vehicles. The plates are frozen down at night, or other times when the vehicle is not in use, and the frozen brine keeps the surface of the plate cold while the vehicle is on the road. The refrigeration machinery may be on the vehicle or static.

7.6 Defrosting

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

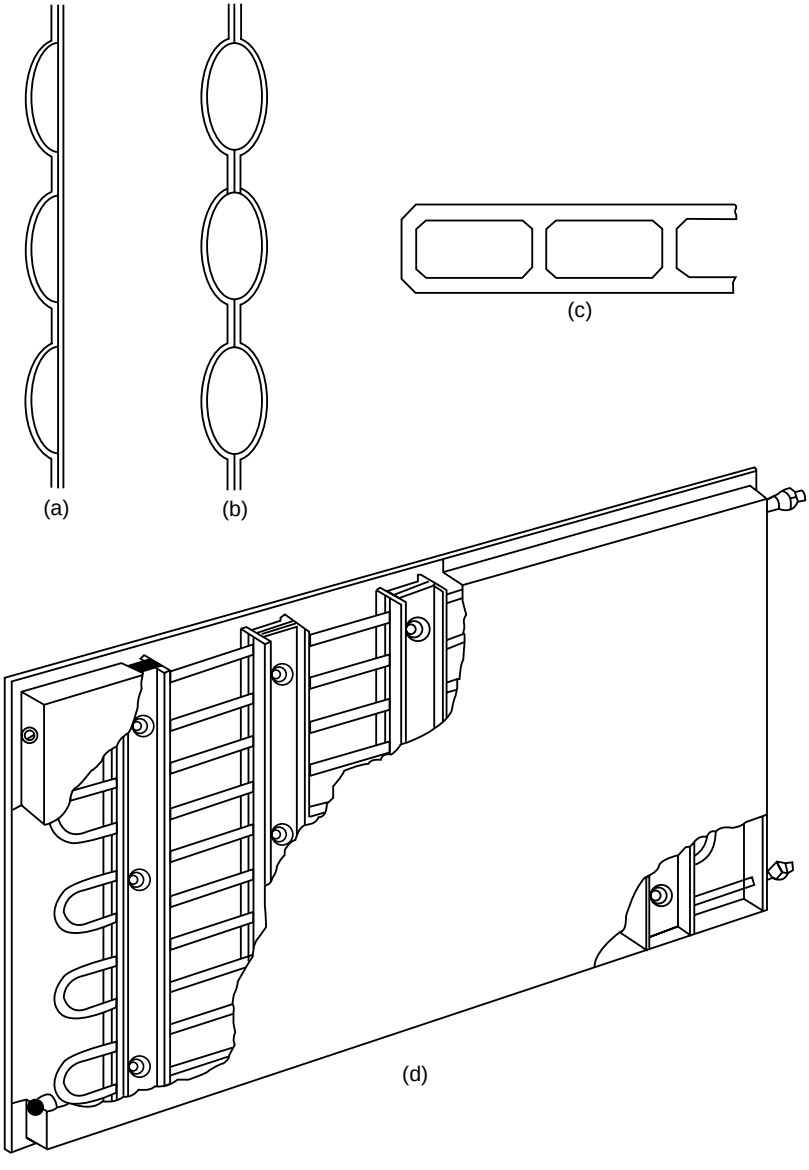
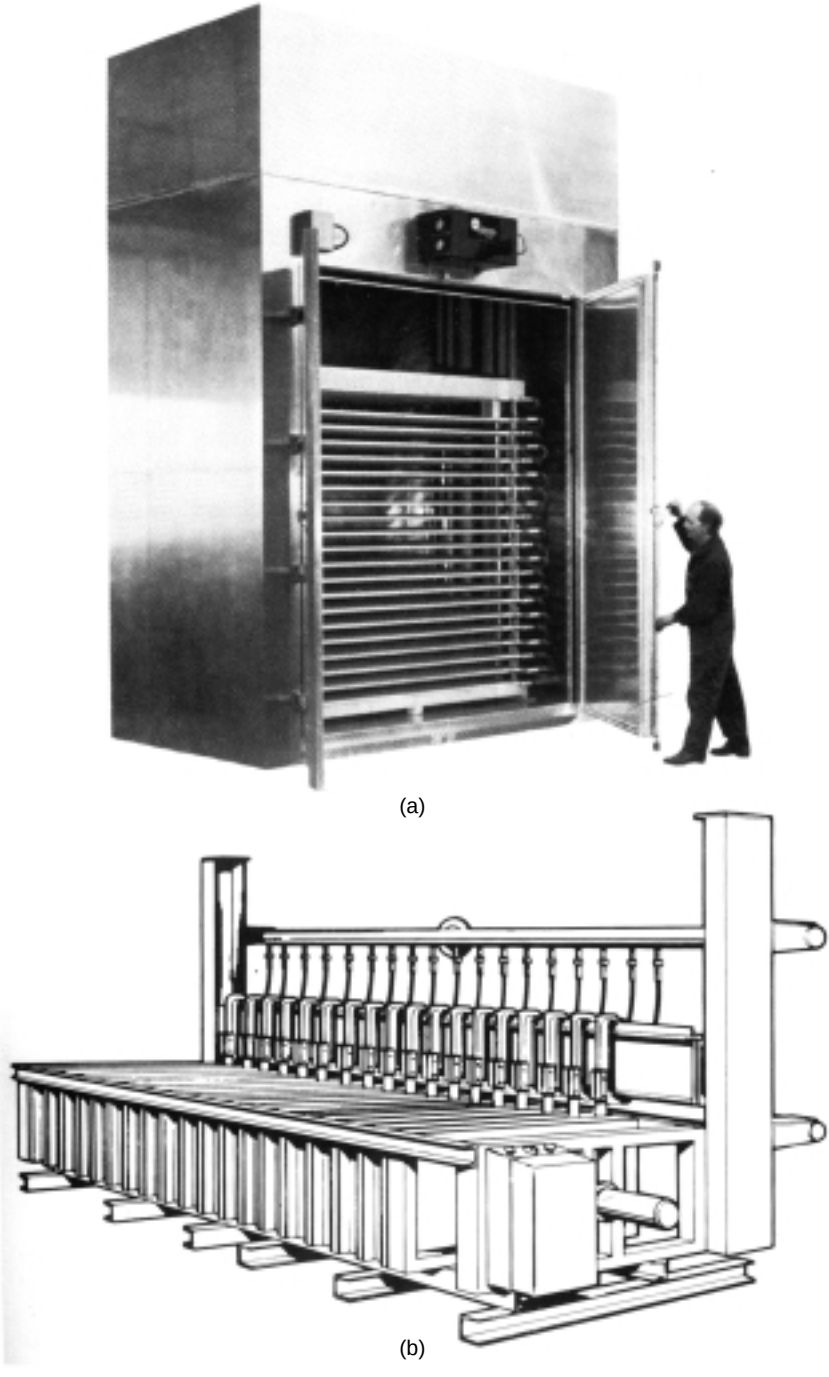


Figure 7.8 *Plate evaporators. (a) Single embossed. (b) Double embossed. (c) Extruded. (d) Holdover (brine filled) (Courtesy of Elliott Turbomachinery Ltd)*

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



(a)

(b)

Figure 7.9 *Plate freezers. (a) Horizontal. (b) Vertical (Courtesy of APV Parafreeze Ltd)*

Where the surrounding air is always at +4°C or higher, it will be sufficient to stop the refrigerant for a period and allow the frost to melt off (as in the auto-defrost domestic refrigerator). This method can be used for coldrooms, packaged air-conditioners etc., where the service period can be interrupted.

For lower temperatures, heat must be applied to melt the frost within a reasonable time and ensure that it drains away. Methods used are as follows:

1. Electric resistance heaters. Elements are within the coil or directly under it.
2. Hot gas. A branch pipe from the compressor discharge feeds superheated gas to the coil. The compressor must still be working on another evaporator to make hot gas available. Heat storage capsules can be built into the circuit to provide a limited reserve of heat for a small installation.
3. Reverse cycle. The direction of flow of the refrigerant is reversed to make the evaporator act as a condenser. Heat storage or another evaporator are needed as a heat source.

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

7.7 Condensate pumps

Condensed water will run down the evaporator fins to a collection tray below the coil. From there, drain pipes will take this water to a drain. If plastic pipe is used, it should be black to exclude daylight, or slime will grow inside the tube. Drain pipes passing through rooms below freezing point need to be fitted with trace heaters.

Where the outlet drain is higher than the coil, the water needs to be pumped away for disposal. Condensate pumps are fitted to lift this water to drain by gravity. Such pumps are usually of the peristaltic type.

8 Expansion valves

8.1 General

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

8.2 Low-pressure float valves

Flooded evaporators require a constant liquid level, so that the tubes remain wetted. A simple float valve suffices, but must be located with the float outside the evaporator shell, since the surface of the boiling liquid is agitated and the constant movement would cause excessive wear in the mechanism. The float is therefore contained within a separate chamber, coupled with balance lines to the shell (see Figure 8.1).

Such a valve is a metering device and may not provide positive shut-off when the compressor is stopped. Under these circumstances, refrigerant will continue to leak into the evaporator until pressures have equalized, and the liquid level might rise too close to the suction outlet. To provide this shut-off, a solenoid valve is needed in the liquid line.

8.3 Low-pressure float switches

Since the low-pressure float needs a solenoid valve for tight closure, this valve can be used as an on–off control in conjunction with a pre-set orifice and controlled by a float switch (Figure 8.2).

The commonest form of level detector is a metallic float carrying an iron core which rises and falls within a sealing sleeve. An induction coil surrounds the sleeve and is used to detect the position of the

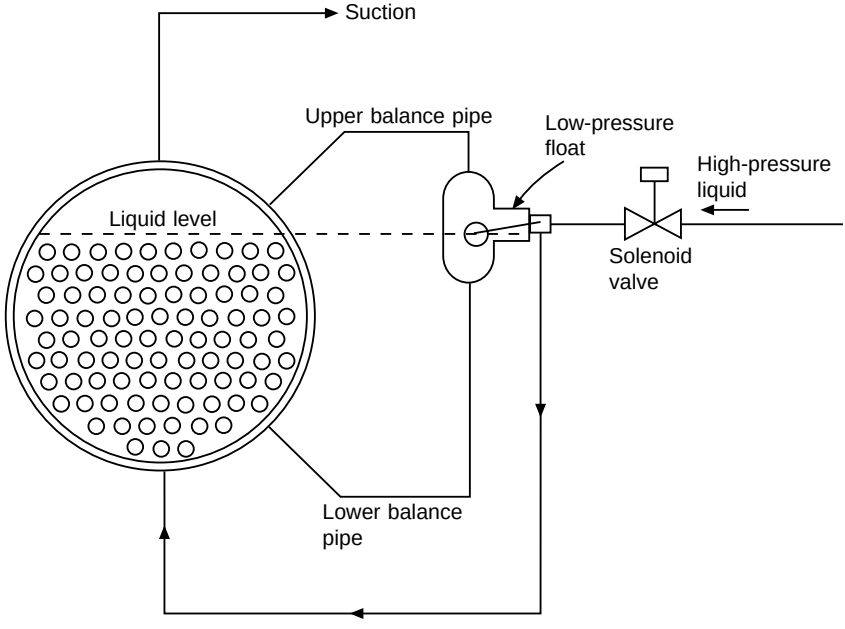


Figure 8.1 *Low-pressure float valve on flooded cooler*

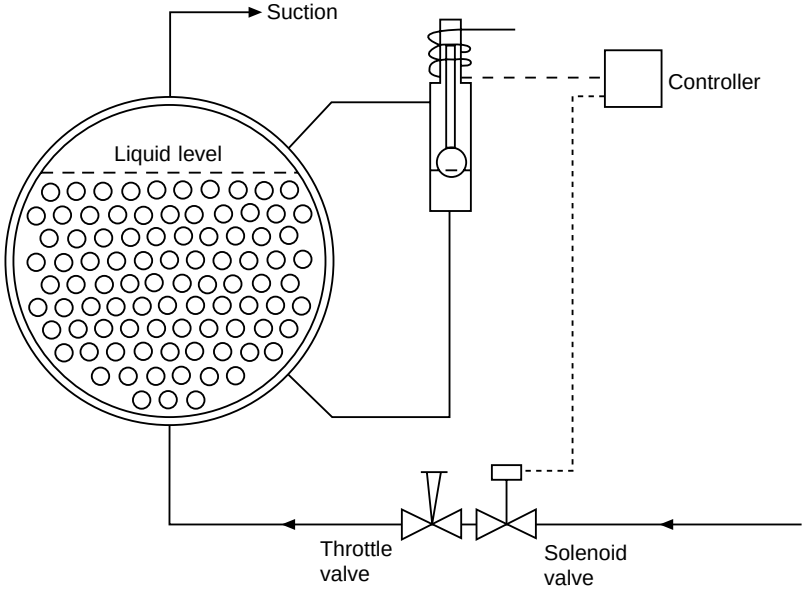


Figure 8.2 *Low-pressure float switch*

core. The resulting signal is amplified to switch the solenoid valve, and can be adjusted for level and sensitivity. A throttle valve is fitted to provide the pressure-reducing device.

Should a float control fail, the level in the shell may rise and liquid pass into the compressor suction. To warn of this, a second float switch is usually fitted at a higher level, to operate an alarm and cut-out.

Where a flooded coil is located in a liquid tank, the refrigerant level will be within the tank, making it difficult to position the level control. In such cases, a gas trap or siphon can be formed in the lower balance pipe to give an indirect level in the float chamber. Siphons or traps can also be arranged to contain a non-volatile fluid such as oil, so that the balance pipes remain free from frost.

8.4 High-pressure float valve

On a single-evaporator flooded system, a float valve can be fitted which will pass any drained liquid from the condenser direct to the evaporator. The action is the same as that of a steam trap. The float chamber is at condenser pressure and the control is termed a high-pressure float (Figure 8.3).

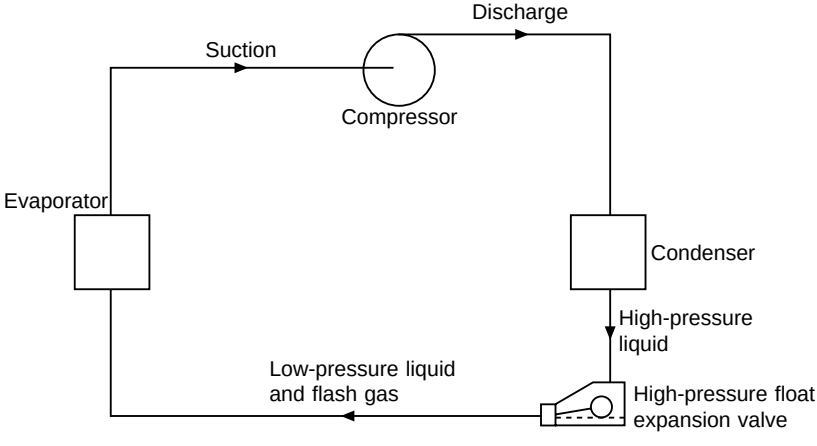


Figure 8.3 *High-pressure float valve*

The refrigerant charge of such a system is critical, since it must not exceed the working capacity of the evaporator. It is not possible to have a receiver in circuit and this control cannot feed more than one evaporator, since it cannot detect the needs of either.

The difficulty of the critical charge can be overcome by allowing any surplus liquid refrigerant leaving the evaporator to spill over

into a receiver or accumulator in the suction line, and boiling this off with the warm liquid leaving the condenser. In this system, the *low-pressure receiver circuit,* liquid is drained from the condenser through the high-pressure float, but the final step of pressure drop takes place in a secondary expansion valve after the warm liquid has passed through coils within the receiver. In this way, heat is available to boil off surplus liquid leaving the evaporator (see Figure 8.4). Two heat exchangers carry the warm liquid from the condenser within this vessel. The first coil is in the upper part of the receiver, and provides enough superheat to ensure that gas enters the compressor in a dry condition. The lower coil boils off surplus liquid leaving the evaporator itself. With this method of refrigerant feed, the evaporator has a better internal wetted surface, with an improvement in heat transfer.

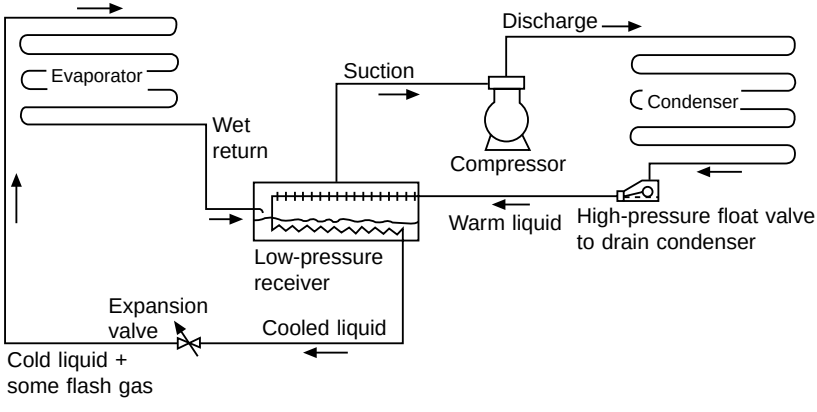


Figure 8.4 *Low-pressure receiver circuit*

The low-pressure receiver system can be adapted to compound compression and can be fitted with hot gas defrost by reverse gas flow. In both circuits the low-pressure receiver provides the safety vessel to prevent liquid entering the compressor. Providing the high-pressure float is correctly sized, this system can operate at low condenser pressures, saving compressor energy in cool weather. Where the halocarbon refrigerants are used in this system, an oil distillation device is fitted, working on the same principle as shown in Figure 5.2.

8.5 Thermostatic level control

If a small heater element is placed at the required liquid level of a flooded evaporator, together with a heat-sensing element, then the

latter will detect a greater temperature if liquid refrigerant is not present. This signal can be used to operate a solenoid valve.

8.6 Expansion valves for dry expansion circuits

The dry expansion circuit does not have a liquid level which can be detected, and another type of signal must be used to control the valve. Dry expansion circuits must be designed and installed so that there is no risk of liquid refrigerant returning to the compressor. To ensure this state, extra heat exchange surface is added to that needed, in order to heat the dry saturated gas into the superheat region. The amount of superheat is usually of the order of 5 K.

Expansion valves for such circuits embody a mechanism which will detect the superheat of this gas leaving the evaporator (Figure 8.5). Refrigerant boils in the evaporator at T_e and p_e, until it is all vapour, and then superheats to a condition T_s, p_e, at which it passes to the suction line. A separate container of the same refrigerant at temperature T_s would have a pressure p_s, and the difference $p_s - p_e$ is a signal directly related to the amount of superheat.

The basic thermostatic expansion valve (Figure 8.6) has a detector and power element, charged with the same refrigerant as in the circuit. The pressure p_s generated in the phial by the superheated gas passes through the capillary tube to the top of the diaphragm. An adjustable spring provides the balance of $p_s - p_e$ at the diaphragm, and the valve stem is attached at the centre. Should the superheat fall for any reason, there will be a risk of liquid reaching the compressor. The T_s will decrease with a corresponding drop in p_s. The forces on the diaphragm are now out of balance and the spring will start to close the valve.

Conversely if the load on the evaporator increases, refrigerant will evaporate earlier and there will be more superheat at the phial position. Then p_s will increase and open the valve wider to meet the new demand.

The phial must be larger in capacity than the rest of the power element or the charge within it may all pass into the valve capsule and tube, if these are colder. If this happened, the phial at T_s would contain only vapour and would not respond to a position T_s, p_s on the T–p curve.

Use can be made of this latter effect. The power element can be limit charged so that all the refrigerant within it has vaporized by a predetermined temperature (commonly 0°C). Above this point, the pressure within it will follow the gas laws:

$$\frac{p_1}{p_2} = \frac{T_1}{T_2}$$

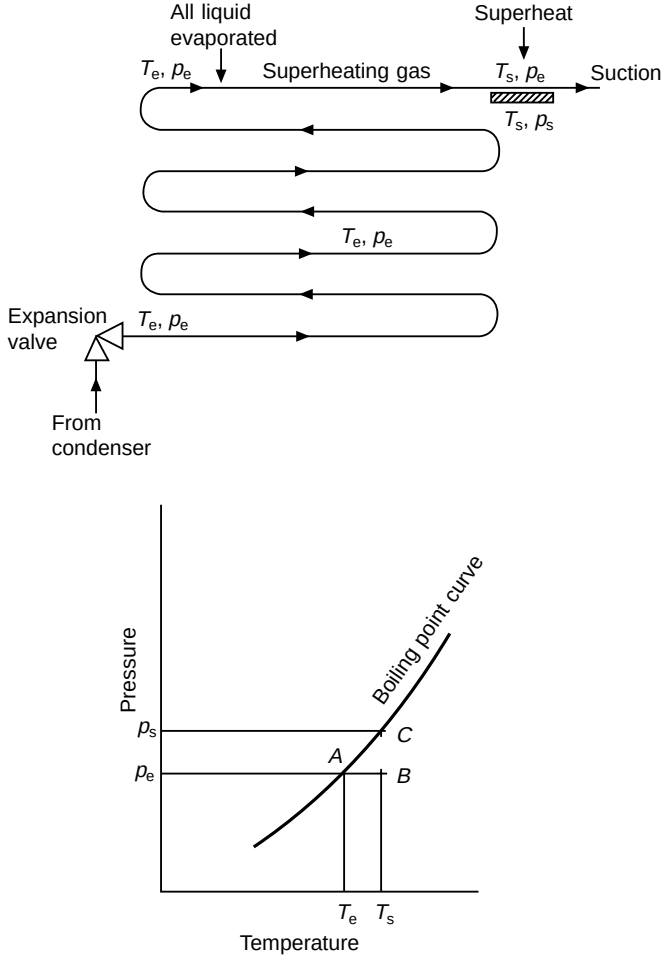
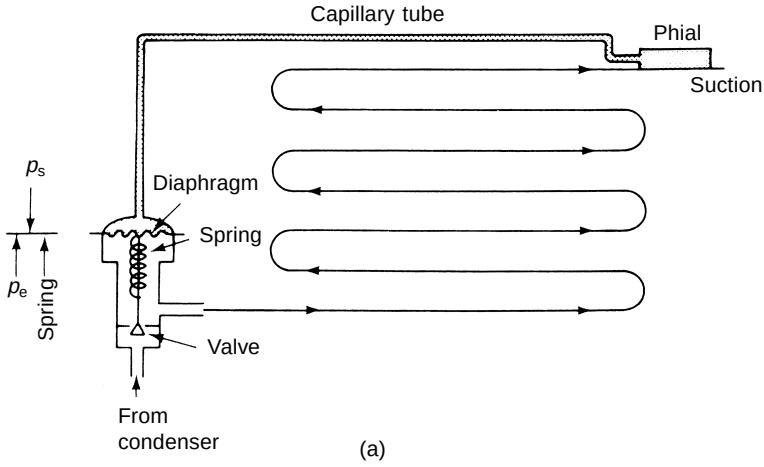


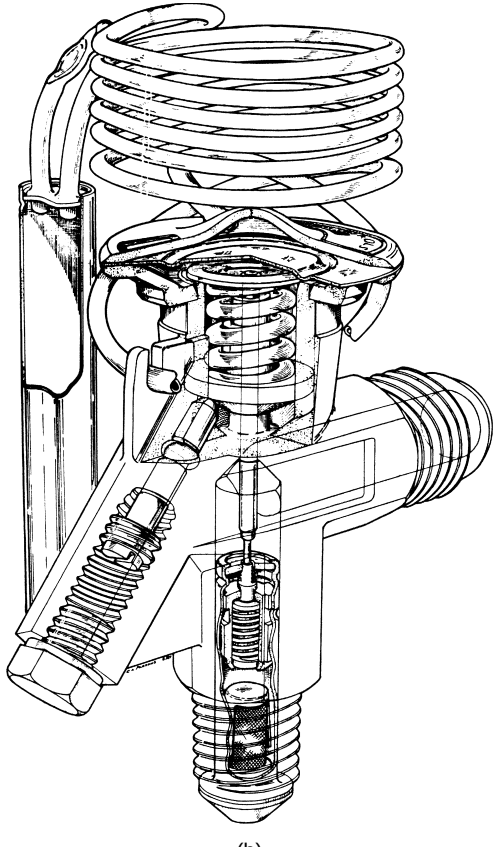
Figure 8.5 *Superheat sensor on dry expansion circuit*

and the valve will remain closed. This is done to limit the evaporator pressure when first starting a warm system, which might overload the drive motor. This is termed limit charging or maximum operating pressure. Such valves must be installed so that the phial is the coldest part (see Figure 8.7).

The slope of the *T–p* curve is not constant, so that a fixed spring pressure will result in greater superheat at a higher operating temperature range. To allow for this and provide a valve which can be used through a wide range of applications, the phial may be charged with a mixture of two or more volatile fluids to modify the characteristic curve.



(a)



(b)

Figure 8.6 *Thermostatic expansion valve. (a) Circuit. (b) Cross-section (Courtesy of Teddington Controls Ltd)*

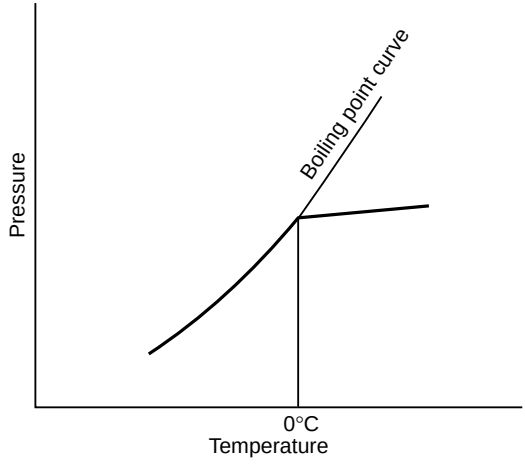


Figure 8.7 *Detector pressure for limit charged valve*

Some manufacturers use the principle of the adsorption of a gas by a porous material such as silica gel or charcoal. Since the adsorbent is a solid and cannot migrate from the phial, these valves cannot suffer reversal of charge.

8.7 External equalizer

The simple *thermostatic expansion valve* relies on the pressure under the diaphragm being approximately the same as that at the coil outlet, and small coil pressure drops can be accommodated by adjustments to the spring setting.

Where an evaporator coil is divided into a number of parallel passes, a distribution device with a small pressure loss is used to ensure equal flow through each pass. Pressure drops of 1–2 bar are common. There will now be a much larger finite difference between the pressure under the diaphragm and that at the coil inlet. To correct for this, the body of the valve is modified to accommodate a middle chamber and an *equalizing connection* which is taken to the coil outlet, close to the phial position. Most thermostatic expansion valves will have provision for an external equalizer connection (see Figure 8.8).

The thermostatic expansion valve is substantially an undamped proportional control and hunts continuously, although the amplitude of this swing can be limited by correct selection and installation, and if the valve always works within its design range of mass flow. Difficulties arise when compressors are run at reduced load and the refrigerant mass flow falls below the valve design range. It is helpful

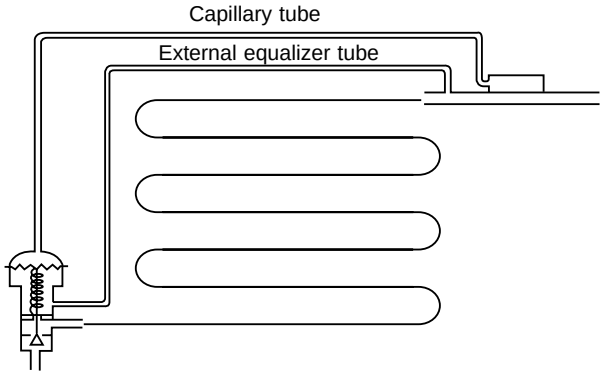


Figure 8.8 *Thermostatic expansion valve with external equalizer*

to keep the condensing pressure steady, although it does not have to be constant and can usually be allowed to fall in colder weather to save compressor power. Valves on small systems may be seen to fully close and fully open at times. The continual hunting of the thermostatic expansion valve means that the evaporator surface has an irregular refrigerant feed with a resulting slight loss of heat transfer effectiveness. It is probable that this valve will be superseded by the electronic expansion valve for many systems.

8.8 Thermostatic liquid level control

The thermostatic expansion valve can also be used to maintain a liquid level. The phial and a heater element are both clamped to a bulb at the required liquid level. If liquid is not present, the heater warms the phial to a superheat condition and the valve opens to admit more liquid.

8.9 Electronic expansion valve

Evaporator superheat can be sensed by two thermistors, one on the main pipes of the evaporator and the other on the suction outlet, and the signal used to control refrigerant flow. The final control element is a pulsing or modulating solenoid valve. The controller can also accept other signals, such as load temperature, discharge temperature, condensing pressure and motor current, and use these to provide optimum coil effectiveness for minimum input power (see Figure 8.9).

The electronic expansion valve has been fitted for some years onto factory-built packages but is now available for field installations, and its use will become more general. The extent of its future

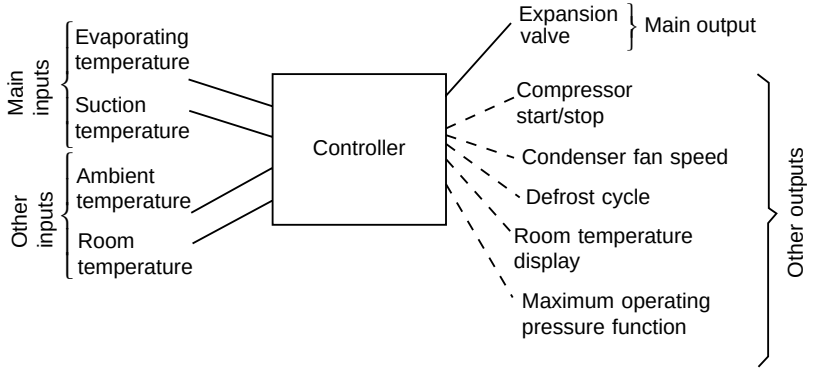


Figure 8.9 *Electronic expansion valve*

success will depend on the ability of installation mechanics to set the controller correctly.

Electronic expansion valves are now widely used on small automatic systems, mainly as the refrigerant flow control device (evaporating or condensing) in an integrated control circuit.

8.10 Thermal electric expansion valve

The signal from a suitable thermistor placed at the evaporator outlet will vary, depending on whether it senses dry refrigerant gas or traces of liquid. This can be used directly to control the current through a thermal element to modulate the expansion valve. This device usually has no separate adjustable controller and so cannot be incorrectly set (see Figure 8.10).

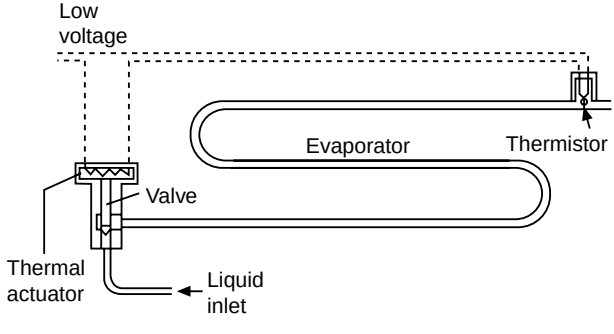


Figure 8.10 *Thermal electric expansion valve*

8.11 Capillary tube restrictor

The variable orifice of the expansion valve can be replaced, in small systems, by a long thin tube. This is a non-modulating device and has certain limitations, but will give reasonably effective control over a wide range of conditions if correctly selected and applied. Mass flow is a function of pressure difference and the degree of liquid subcooling on entry.

Tube bores of 0.8–2 mm with lengths of 1–4 m are common. The capillary tube is only fitted on factory-built and tested equipment, with exact refrigerant charges. It is not applicable to field-installed systems [16, 22].

9 Controls and other circuit components

9.1 General

A refrigeration system can be built with only the four essential components:

1. Evaporator
2. Compressor
3. Condenser
4. Expansion valve

For ease, economy, and safety of operation, and to assist the maintenance function, other system controls and components will be fitted.

9.2 Thermostats

Since the purpose of a refrigeration or air-conditioning system will be to reduce or maintain temperature, a thermostat will usually be fitted to stop the equipment or reduce its capacity when the required condition is reached. The following types are in use:

1. Movement of a bimetallic element
2. Expansion of a fluid
3. Vapour pressure of a volatile fluid

   The above produce a mechanical effect which can be used directly to operate an electric switch or modulate the pressure of an air jet (pneumatic system).

4. Electric resistance
5. Electronic – various types

These last two produce an electric signal which must be measured and amplified to operate the controlled device.

9.3 Humidistats

Where the equipment is required to maintain a predetermined level of humidity, a humidistat may be used instead of, or in addition to, a thermostat. The function will normally be to operate an electrical switch.

Mechanical humidistats employ materials which change dimension with humidity, such as animal hair, plastics, cellulosics, etc. These can work a switch directly.

Electronic humidistats generally depend on the properties of a hygroscopic salt. The signal has to be measured and amplified.

9.4 Pressure switches

The compressor, as a pump, will be limited mechanically to maximum safe operating pressures and must be stopped before such pressures are reached (see Figure 9.1).

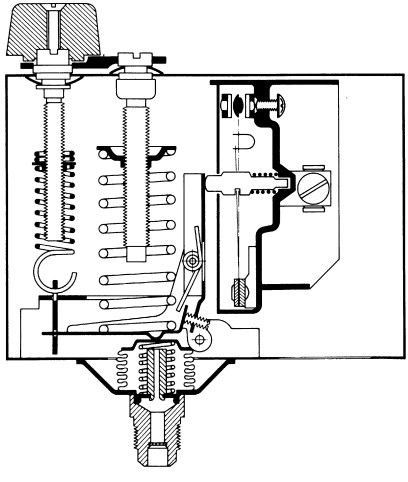


Figure 9.1 *Pressure control (Courtesy of Teddington Controls Ltd)*

High-pressure cut-outs are fitted to all but the smallest of systems. The compressor outlet pressure is brought to one side of a bellows or diaphragm, and balanced by an adjustable spring. A scale on the control indicates the pressure setting to commercial accuracy and is checked on commissioning the system.

If the spring pressure is overcome, the switch will open and stop the compressor. Normally open contacts on the cut-out can then operate a warning. The cut-out point only needs to be some 2 bar higher than the expected summer operating pressure but there is a

tendency to set such controls much higher – sometimes as much as 8 bar above summer pressures. At this setting, the user will not get a warning of abnormal running until the fault has reached serious proportions.

Since excess pressure indicates malfunction of part of the system – usually a condenser fault or incorrect closure of a valve – the high-pressure switch should be reset manually, not automatically.

Where the refrigeration system is providing an essential service which should not be interrupted, one high-pressure switch may be set at a warning level and operate an alarm, without stopping the compressor. A second switch, set somewhat higher, will stop the equipment if this warning is ignored and if excessive pressures are reached. All high-pressure cut-outs should be checked at least once a year, for correct setting and operation.

Abnormally low suction pressures will lead to high discharge temperatures, owing to the high compression ratio, and possible malfunction of other components. Air cooling coils may frost excessively, or water chillers freeze.

A *low-pressure cut-out* switch is usually fitted to stop the compressor under these circumstances. Settings may be 0.6–1.0 bar below the design evaporator pressures, but depend very much on the type of system. The cut-out setting should be above atmospheric pressure if possible to avoid the ingress of air through any leaks.

Abnormally low pressure may not be an unsafe condition and the low-pressure switch may be automatic reset, closing again at a pressure corresponding to a temperature just below that of the load.

If a plant has been shut down long enough for all pressures to equalize and is then restarted, the suction pressure will pull down below normal until the liquid refrigerant has begun to circulate. Under such circumstances the low-pressure switch may operate. This is a normal occurrence, but may require the addition of a delay timer to prevent frequent starting of the compressor motor.

A low-pressure switch can also be used in conjunction with a thermostat and a solenoid valve in the *pump-down circuit*. In this method of control, the thermostat does not stop the compressor but de-energizes the liquid line solenoid valve to stop the supply of refrigerant to the evaporator. The compressor continues to run and pumps down the evaporator until stopped by the low-pressure switch. When the thermostat again calls for cooling, it opens the solenoid valve, liquid enters the evaporator and the low-pressure switch will close again to restart the compressor. This method is used to ensure that the evaporator is kept clear of liquid when the plant is off. If there is any leak at the solenoid valve, it will cause the

compressor to restart periodically to remove the surplus liquid from the coil (see Figure 9.2). Pressure switches are also made in miniature encapsulated versions, mainly pre-set for use in integrated control circuits.

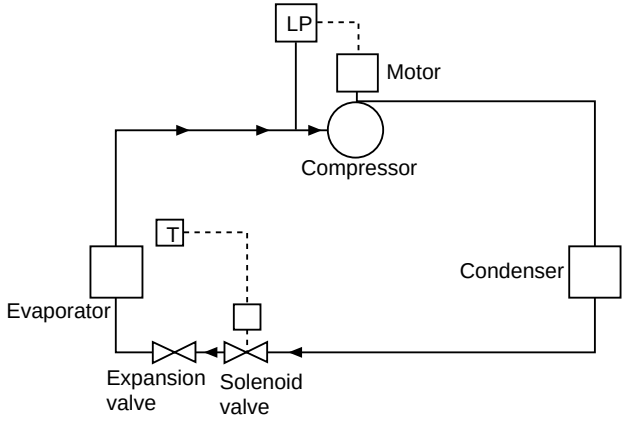


Figure 9.2 *The pump-down circuit*

9.5 Oil safety

All compressors except the smallest have mechanical lubrication and will fail if the oil pressure falls because of a pump fault or oil shortage. A safety cut-out is required which will stop the compressor. This takes the form of a differential pressure switch with a starting time delay.

Since the oil pump inlet is at sump (suction) pressure, a pressure gauge on the pump discharge will indicate the total pressure at that point above atmospheric, i.e. suction (gauge) plus pump head. Any detection element for true oil pump pressure must sense both suction and pump outlet pressures and transduce the difference. Oil safety cut-outs have pipe connections to both sides of the oil pump and two internal bellows are opposed to measure the difference.

Since there will be no oil pressure at the moment of starting, a time delay must be fitted to allow the oil pressure to build up. This timer may be thermal, mechanical or electric.

Operation of the oil safety cut-out indicates an unsafe condition and such controls are made with hand reset switches. Normally open contacts on the switch can be used to operate an alarm to warn of the malfunction.

9.6 Pressure gauges

Direct indication of the operating conditions of a compressor is by

pressure gauges at suction, discharge and oil delivery. Such gauges are mounted on or near the compressor.

Since the pressure losses along the discharge and suction lines are comparatively small on most systems, these pressures will also approximate to the conditions in the condenser and evaporator, and the equivalent saturation temperatures will be the condensing and evaporating temperatures. To indicate these temperatures for the common refrigerants, pressure gauges will have further calibrations showing these equivalent temperatures (see Figure 9.3).

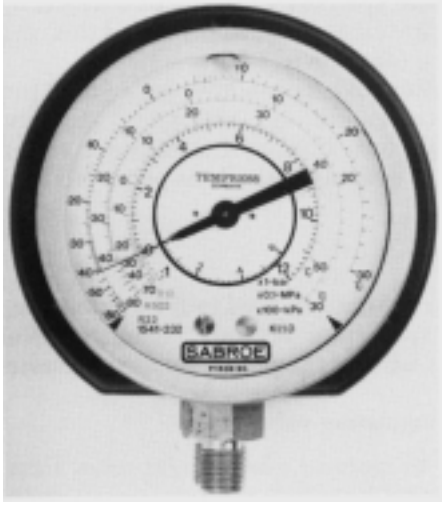
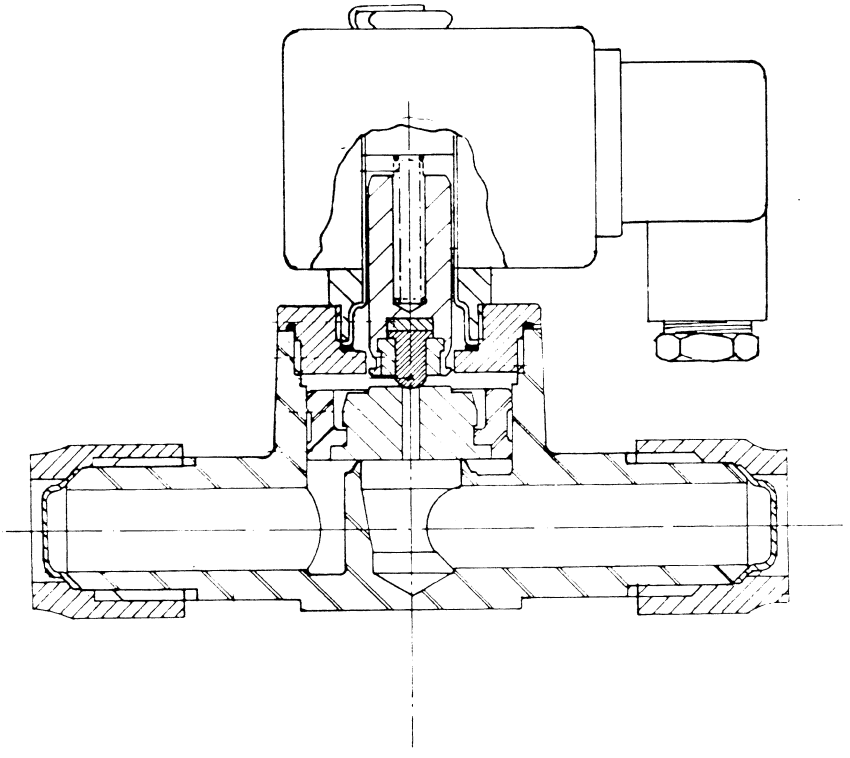


Figure 9.3 *Pressure gauge – suction and oil (Courtesy of AB Sabroe)*

A pressure gauge fitted at the oil pump outlet connection will show the sum of two pressures, that generated by the pump itself plus the crankcase pressure. True oil pump pressure can only be indicated by a dual gauge, in which the oil pump discharge rotates a circular inner scale (see Figure 9.3). On this gauge, the suction pressure is read off the perimeter scale and the oil pump pressure by observing the position of the needle relative to the inner disc scale.

Gauge mechanisms are mostly of the bourdon tube type, having a flattened tube element, which distorts under pressure change. Gas pulsations from the compressor will be transmitted along the short connecting pipes and may lead to early failure of the needle mechanism. These can be damped by restricting the tube with a valve or orifice, or oil filling the gauge, or both. Gauge needles should not be allowed to flicker noticeably from gas pulsations. Miniature pre-set pressure transducers are now made as components of an integrated control circuit.



(a)

(b)

Figure 9.4 *Solenoid valves. (a) Shut-off (Courtesy of Bailey Gill Products Ltd). (b) Change-over (Courtesy of Ranco Controls Ltd)*

9.7 Solenoid valves

Electrically operated shut-off valves (Figure 9.4*a*) are required for refrigerant and other circuits. These take the form of a plunger operated by a solenoid and working directly on the valve orifice or through a servo. The usual arrangement is to energize the solenoid to open the valve and de-energize to close. Sizes up to 50 mm bore tube connections are made. Beyond this, the solenoid acts as a pilot to a main servo (see Figure 9.5).

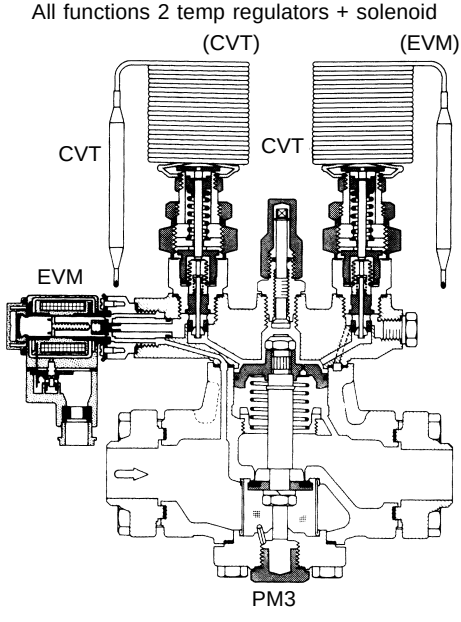


Figure 9.5 *Back pressure regulation valve assembly, all functions – two temperature regulators (CVT) and solenoid (EVM) (Courtesy of Danfoss)*

Solenoid valves are used in refrigeration and air-conditioning systems for refrigerant lines, oil pressure pipes (to control oil return and capacity reducers), and water and compressed air lines. Four-port changeover valves (Figure 9.4*b*) are used to reverse flow in defrosting and heat pump circuits. A de-energized expansion valve will act as a solenoid valve.

9.8 Back pressure regulation valves

Back pressure regulation valves (Figure 9.5) can be used in the suction line, and their function is to prevent the evaporator pressure falling below a predetermined or controlled value, although the compressor suction pressure may be lower.

The application of a back pressure regulating valve is to:

1. Prevent damage to a liquid chilling evaporator which might result from freezing of the liquid.
2. Prevent frost forming on an air cooling evaporator, where this is close to freezing point, or where a temporary malfunction cannot be permitted to interrupt operation.
3. Permit two or more evaporators, working at different load temperatures, to work with the same compressor.
4. Modulate the evaporator pressure according to a varying load, controlled by the load temperature.
5. Act as a solenoid valve, controlled by a pilot solenoid valve.

The simplest back pressure regulating valve is spring-loaded, balancing the thrust of the spring, plus atmospheric pressure, on one side of a diaphragm or piston, against the inlet or evaporator pressure. For working pressures below atmospheric, a helper spring is fitted below the diaphragm. Slight variations will result from changes in atmospheric pressure, but these are too small to materially affect a refrigeration control system.

A service gauge is usually fitted adjacent to the valve or as part of the valve assembly, to facilitate setting or readjustment. Above about 40 mm pipe size, the basic back pressure regulation valve is used as a pilot to operate a main servo valve. Other pilot signals can be used on the same servo.

Figure 9.5 shows a main servo controlled by two thermostatic pilots sensing load temperature (type CVT) and a solenoid valve (type EMV). Any of these pilots may be used separately with the servo valve.

9.9 Suction-to-liquid heat exchangers

Cold gas returning from the evaporator to the compressor can be used to pre-cool the warm liquid passing from the condenser to the expansion valve, using a suction-to-liquid heat exchanger (Figure 9.6). In cooling the liquid and reducing its enthalpy, a greater refrigerating effect will be obtained. This gain is offset to a greater or lesser extent by the superheating of the suction gas and the resultant reduction of mass flow into the compressor. The overall effect of fitting a suction-to-liquid heat exchanger in terms of thermodynamic efficiency will vary with the refrigerant and the operating conditions.

The suction-to-liquid heat exchanger will supply the suction superheat necessary for safe operation of a dry expansion evaporator, and the coil superheat may be less, giving more efficient use of the evaporator surface. The phial should be located before the heat

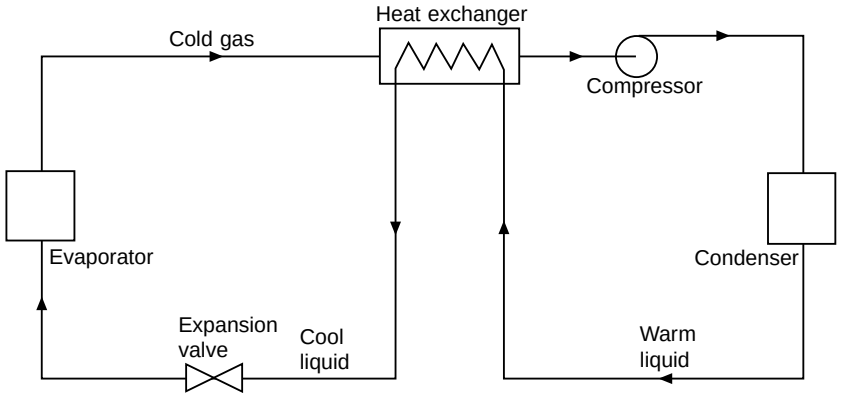


Figure 9.6 *Suction-to-liquid heat exchanger*

exchanger, in which case the superheat setting is reduced. It can be located after the heat exchanger, but an external equalizer is then necessary to allow for the gas pressure drop through the exchanger.

9.10 Condenser pressure regulators

Systems are normally designed to work satisfactorily during maximum ambient conditions, and the condenser will be sized for this. In colder weather, the condensing temperature and pressure will fall and the resulting lower pressure difference across a *thermostatic* expansion valve may lead to malfunction. A drop of pressure difference to half the normal figure may reduce mass flow below that required, and it will be necessary to prevent the condenser pressure from falling too low.

With air-cooled condensers and water cooling towers it is possible to reduce the air flow by automatic dampers, fan speed control, or switching off fans, where two or more drives are fitted. The control should work from pressure but can be made to work from temperature (see Section 6.12).

Water-cooled condensers can be fitted with a directly controlled water-regulating valve operated by condenser pressure, or may have a three-way blending valve in the water circuit.

A condenser pressure regulator can be in the form of a pressure-operated bleed valve in a bypass across the condenser, to divert hot gas to the receiver. The valve diaphragm is balanced by a pre-set spring and will open the bypass if the condensing pressure falls. A similar effect can be obtained by a pressure-operated valve between the condenser and the receiver, to restrict the flow and allow liquid to accumulate in the condenser, reducing its efficiency. For operating

economy, it is important that such valves are not set at too high a pressure [23].

Where evaporative condensers and water cooling towers have only one fan (or fan drive motor), coarse control can be effected by on–off switching. The time lag will then depend on the mass of water in the circuit, and the sensing element needs to have a wide differential to prevent frequent motor starts. Towers should have thermostatic control of the fan to prevent water freezing on the packing in winter.

An integrated control circuit with an electronic expansion valve can be arranged to permit the condensing pressure to fall, providing the valve can pass the refrigerant flow required to meet the load. This gives lower compressor energy costs.

In all forms of condenser pressure control, the minimum maintained pressure should be the lowest which will give satisfactory operation, in the interests of running economy. An indication of the relative electricity costs for a 350 kW air-conditioning plant is given in Table 9.1.

Table 9.1

Condensing temperature (°C)	*Coefficient of performance*	*Weekly electricity costs* (£, @ 5 p/unit)
35 (summer maximum)	3.41	256
30	4.00	219
25 (probable minimum)	4.73	184

9.11 Capacity reduction injection valves

Where a compressor does not have any capacity reduction device and on–off switching will not give the degree of control required by the process, the cooling capacity can be regulated by injecting discharge gas back into the suction (see Figure 9.7). It has the effect of keeping the evaporator pressure constant, regardless of the load, and can have a wide range of capacity reduction, down to 10% of full load. It is a constant pressure valve, balancing the suction pressure against a pre-set spring.

However, since the suction gas to the compressor would then be hotter than its normal slightly superheated condition, the compressor may overheat and the discharge gas become too hot for correct and safe working. This form of capacity reduction is usually combined with a liquid injection valve, thermostatically operated, which introduces liquid also into the suction to keep it cool. The fitting of dual interdependent controls of this sort, both of which have inherent fail–unsafe possibilities, should be approached with caution.

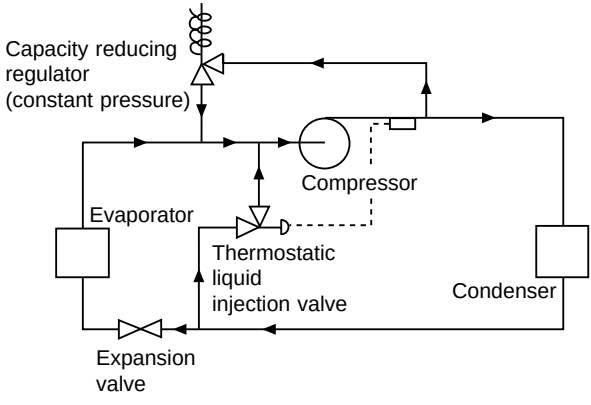


Figure 9.7 *Capacity reduction by hot gas injection, with compensating liquid injection*

A safer circuit injects the discharge gas directly after the expansion valve or into the evaporator outlet and before the sensor of the expansion valve. With this arrangement, the expansion valve will admit extra refrigerant, and gas entering the compressor will be normally cool. These control methods are wasteful of energy.

9.12 Relief valves

Under several possible conditions of malfunction, high pressures can occur in parts of the system and mechanical relief devices are advised or mandatory. The standard form of relief valve is a spring-loaded plunger valve. No shut-off valve is permitted between the relief valve and the vessel it protects, unless two such valves are fitted, when the shut-off may isolate one at a time [13]. Two valves are required on a vessel greater than 285 litres in volume.

In all cases, the outlet of the valve must be led to the open air, in a location where the sudden discharge of refrigerant will not cause annoyance or danger. Under certain circumstances, a relief valve from the high-pressure side may enter the low side of the same system. Small vessels may have a plug of a low melting point metal, which will melt and release the pressure in the event of fire. Plunger-type relief valves, if located outdoors, should be protected from the ingress of rain, which may corrode the seat. Steel valves, when installed, should have a little oil poured in to cover the seat as rust protection.

To prevent overpressure within a compressor, a relief valve or bursting disc is often fitted between the inlet and discharge connections.

9.13 Shut-off valves

Manual stop valves are required throughout a circuit to permit isolation during partial operation, service or maintenance (see Figure 9.8).

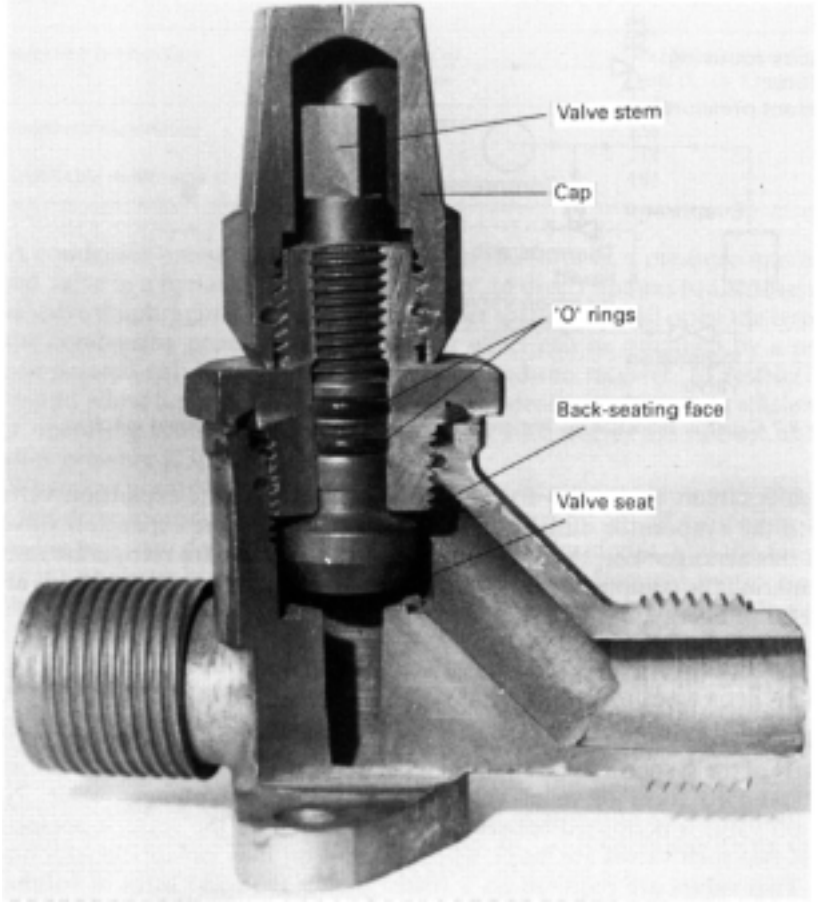


Figure 9.8 *Seal cap shut-off valve*

Small valves which are to be operated frequently have a packless gland, either a diaphragm or bellows, and a handwheel.

Valves of all sizes which are only used occasionally will be sealed with 'O' rings. As a safeguard against leakage, they have no handwheel fitted and the stem is provided with a covering cap which is only removed when the valve is to be operated. The stem will have flats for operation by a spanner. Most such valves can be back-seated to permit changing the 'O' rings.

Valves should not be installed with the stem downwards, as any internal dirt will fall into the spindle thread.

Under low-temperature conditions, ice will form on the spindle and will be forced into the gland if the valve is operated quickly. Under such circumstances, the spindle should be well greased, or the ice melted off first.

Service stop valves on small compressors may also carry a connection for a pressure cut-out or gauge, or for the temporary fitting of guages or charging lines when servicing. The valve back-seats to close off this port while gauges are being fitted. Valve seats are commonly of soft metal or of a resistant plastic such as PTFE.

9.14 Strainers

Piping circuits will usually contain a small quantity of dirt, scale and swarf, no matter what care is taken to keep these out. A strainer is fitted in the compressor suction to trap such particles before they can enter the machine. Such strainers are of metal mesh and will be located where they can be removed for cleaning. In some configurations two strainers may be fitted.

As an extra safeguard, on new compressors a fabric liner may be fitted inside the mesh strainer to catch fine dirt which will be present. Such liners must be removed at the end of the running-in period, as they create a high resistance to gas flow.

Oil strainers may be of metal mesh and within the sump, in which case the sump must be opened for cleaning. Self-cleaning disc strainers are also used, the dirt falling into a drain pot or into the sump itself. There is an increasing tendency to provide replaceable fabric oil filters external to the compressor body, following automobile practice.

9.15 Strainer-driers

With the halocarbons, it is essential to reduce the water content of the refrigerant circuit to a minimum by careful drying of components and the fitting of drying agents in the system. The common form of drier is a capsule charged with a solid desiccant such as silica gel, activated alumina or zeolite (molecular sieve), and located in the liquid line ahead of the expansion valve. These capsules must have strainers to prevent loss of the drying agent into the circuit, and so form an effective strainer-drier to also protect the valve orifice from damage by fine debris.

Large driers are made so they can be opened, and the spent drying agent removed and replaced with new. Small sizes are throwaway. Driers may also be used in the suction line.

9.16 Sight glasses

Pipeline sight glasses can be used to indicate whether gas is present in a pipe which sould be carrying only liquid. The main application in refrigeration is in the liquid line from the receiver to the expansion valve. If the equipment is running correctly, only liquid will be present and any gas bubbles seen will indicate a refrigerant shortage (see also Chapters 11 and 33).

Sight glasses for the halocarbons are commonly made of brass, and may have solder or flare connections. For ammonia, they are made of steel or cast iron.

Since the interior of the system can be seen at this point, advantage is taken in most types to insert a moisture-sensitive chemical which will indicate an excess of water by a change of colour. When such an indication is seen, the drier needs changing or recharging, and the colour should then revert to the 'dry' shade.

9.17 Charging connection

In order to admit the initial refrigerant charge into the circuit, or add further if required, a charging connection is required. The safest place to introduce refrigerant will be ahead of the expansion valve, which can then control the flow and prevent liquid reaching the compressor. The usual position is in a branch of the liquid line, and it is fitted with a shut-off valve and a suitable connector with a sealing cap or flange. A valve is needed in the main liquid line, just upstream from the branch and within reach. For the method of use, see Chapter 11.

The relative positions of all these components are shown in the complete circuit in Figure 9.9.

9.18 Auxiliary components

More complex refrigeration systems may have components for specific purposes which are not encountered in simple circuits. Non-return or *check* [24] *valves* will be found in the following positions:

1. On heat pump circuits, to prevent flow through expansion valves which are not in service on one cycle
2. On hot gas circuits, to prevent the gas entering another evaporator
3. Where several compressors discharge into one condenser, to prevent liquid condensing back to an idle compressor
4. Where two or more evaporators work at different pressures, to prevent suction gas flowing back to the colder ones

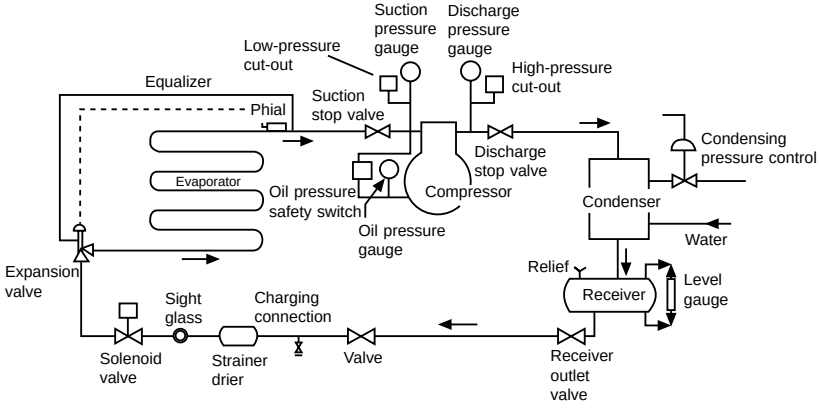


Figure 9.9 *Dry expansion circuit showing components*

9.19 Liquid refrigerant pumps

In a flooded evaporator, the movement of the liquid may be sluggish, with resulting low heat transfer. Liquid pumps can be used to circulate refrigerant from the suction separator (or 'surge drum'), through the evaporator(s) and back. In the separator, remaining liquid falls back and is recirculated, while vapour goes to the compressor (see Figure 9.10). These pumps are found mainly on low-temperature coldrooms, blast freezers and process applications [25].

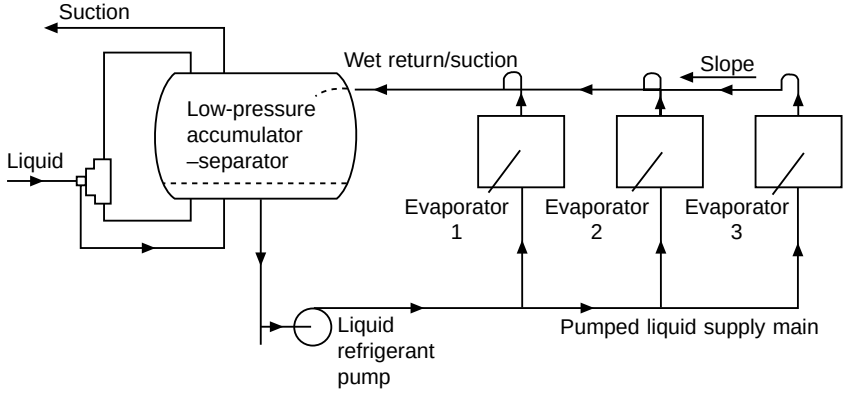


Figure 9.10 *Pumped liquid circuit*

9.20 Suction separators

Suction line accumulators are sometimes inserted in halocarbon circuits, to serve the same purpose of separating return liquid and prevent it passing over to the compressor. Since this liquid will be carrying

oil, and this oil must be returned to the compressor, the outlet pipe within the separator dips to the bottom of this vessel and has a small bleed hole, to suck the oil out (see Figure 9.11).

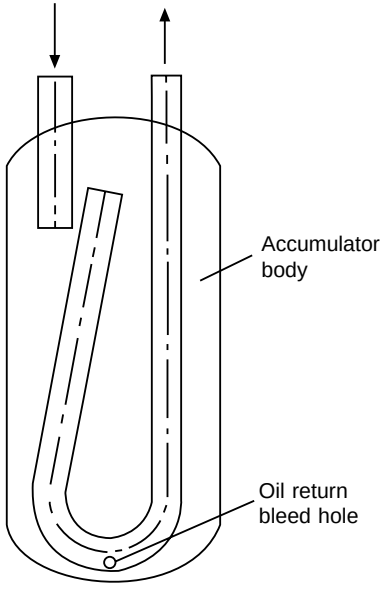


Figure 9.11 *Suction line accumulator or liquid trap*

Suction traps are now widely used, particularly on rolling piston and scroll compressors, to prevent liquid passing into the compressor.

9.21 Liquid separators

Separation vessels can be inserted in a liquid line. Liquid will fall to the bottom and pass through an expansion device to an evaporator. High pressure gas will rise to the top of the vessel and can then be used for heating or for hot gas defrost of another heat exchanger.

9.22 Overheat protection

Small compressors will have motor overheat protection adjacent to the hermetic shell or built into the winding (see Section 4.8) and larger motors will have contactor–starters with overcurrent devices. Overheat protection is also fitted on many machines, to guard against high motor winding, cylinder head or oil temperatures. These usually take the form of thermistor detectors, connected to stop the motor.

9.23 Integrated control systems

The purpose of the various electromechanical elements of a circuit is to effect monitoring, safety and automatic control, and these may be connected separately into a custom-built system. The availability of electronic logic circuits gives the possibility of integrated systems and superior control, using a large number of input signals. Observed parameters are:

Electrical supply
Load temperature
Air and water flows
Number of compressors running, and loading stages
Condenser pressure
Number of condensers running
Condenser fan speed
Evaporator temperature
Discharge temperature
Cylinder head temperature
Motor current
Expansion valve opening
Refrigerant shortage

Control may then be effected of:

Number and stages of compressors running
Limitation of motor start frequency
Limitation of maximum electrical demand
Number of fans or condensers running
Fan speeds
Number of fans or evaporators running
Warning of faulty plant
Shutting down faulty plant
Starting standby plant
Monitoring energy used
Printed running logs
Scheduling maintenance
Remote alarm systems

Integrated control systems are mainly found on factory-assembled equipment, but the increased use of programmable logic controllers for process control is giving designers and installation mechanics the experience to apply these methods to custom-built refrigeration systems [26].

10 Selection and balancing of components

10.1 Balanced system design

The four main components of a vapour compression refrigeration circuit – the evaporator, the compressor, the condenser and the expansion valve – must be selected to give a balanced system.

Each of these items must:

1. Be suitable for the application
2. Be correctly sized for the duty
3. Function as required in conjunction with the other components

The system designer must consider these components and examine the options which may be available in order to determine a best selection with reference to first cost, installation, operation, running cost, maintenance and expected life. The following factors are some of those affecting the final decision:

1. If the initial capital cost is the deciding factor, then the plant will almost certainly be more expensive to operate.
2. Installation of a new plant may cause serious disruption of the user's ongoing business, and the extent of this disruption should be determined before it is too late. Apart from the installation of the plant itself, there is the associated builders' work and the temporary disconnection of other services. The use of factory-built packaged equipment helps to reduce this nuisance.
3. Most systems are now automatic in operation, but the user's staff must be aware of the control system and have facilities to run on manual control, as far as this may be possible, in the event of a control failure.
4. Operators must understand the function of the system. If not, they will not have the confidence to work on or with it, and the

plant will not be operated at its best efficiency. Also, if it breaks down for any reason, they will be unable to put it right.

5. The cost of electricity, other fuels, water, spare parts and operating and maintenance labour represents the greater part of the owning costs of a refrigeration system. It is probable that a small extra expenditure on some items, especially heat exchangers, will reduce running costs.
6. A lot of modern equipment is almost maintenance-free but the user must be aware of what maintenance functions are required, whether these are within the scope of his own staff and where to get assistance. Where maintenance is contracted out, it is important that this should be carried out, at least for the warranty period, by the supplier.
7. Life expectancies are 15–20 years for refrigeration systems, and somewhat less for small packaged equipment. Where the need is for a shorter period, such as a limited production run or for a temporary building, equipment of lower quality or second-hand plant could be considered.

10.2 Evaporating temperature

The next step is to decide a suitable evaporating temperature. This will be set by the required load condition and the appropriate temperature differential (ΔT) across the evaporator. In the context of evaporator selection, the ΔT used is the difference between the evaporating refrigerant and the temperature of the fluid entering the cooler, not the log mean temperature difference (see [1–5]).

In systems where the evaporator cools air, the air itself becomes the heat transfer medium and its temperature and humidity must be considered in relation to the end product. Where the product cannot suffer dehydration, the ΔT may be high, so as to reduce the size and cost of the coil, but the lower the evaporating temperature falls, the lower will be the capacity of the compressor and its COP. In these circumstances, a first estimate might be taken with a ΔT of 10–12 K and cross-checked with alternative plant either side of this range. In each case, the 'owning' cost, i.e. taking into account the running costs, should be considered by the user. For a cold store example, running 8760 hours per year, see Table 10.1.

Unsealed products will be affected by low humidity of the air in the cooled space and may suffer dehydration. Conversely, some food products such as fresh meat will deteriorate in high humidities. Since the dew point of the air approaches the fin surface temperature of the evaporator (see also Chapter 24), the inside humidity is a function of the coil ΔT. That is to say, the colder the fin surface, the

Table 10.1

Cooler size	*Cost* (£)	ΔT	*Annual electricity costs*		
			Fans	*Compressor*	*Total*
65	627	11.7	58	2140	2198
85	845	10.0	69	1970	2039
120	982	8.2	110	1820	1930

more moisture it will condense out of the air, and the lower will be the humidity within the space. Optimum conditions for all products likely to be stored in cooled atmospheres will be found in the standard reference books, or may be known from trade practice. The following may be taken as a guide:

Products that dehydrate quickly, such as most fruits and vegetables	ΔT = 4 K
Products requiring about 85% saturated air	ΔT = 6 K
Products requiring 80% saturation or drier	ΔT = 8 K
Materials not sensitive to dehydration	ΔT = 10 K upwards

A further consideration may be the possibility of reducing ice build-up on the evaporator, whether this is in the form of frost on fins or ice on the coils of a liquid chilling coil. Where temperatures close to freezing point are required, it may be an advantage to design with an evaporator temperature high enough to avoid frost or ice – either for safety or to simplify the defrost method.

10.3 Evaporator

Once the evaporating temperature has been provisionally decided, an evaporator can be selected from catalogue data or designed for the purpose. Catalogue ratings are usually in the form of cooling capacity for a given temperature difference between the entering fluid and the evaporating refrigerant, since the user cannot easily determine the ln MTD. Units will be in kW/K (Btu/(h °F) or kcal/(h °C) in old catalogues).

This factor, the *basic rating,* is assumed constant throughout the design working range of the cooler and this approximation is good enough for equipment selection. The basic rating will change with fluid mass flow and, to a lesser extent, with working temperature. It may change drastically with fluids such as the glycol brines, since the viscosity and hence the convection heat transfer factor alter at

lower temperatures. In unusual applications, the supplier should be consulted. (See also Section 35.4.)

10.4 Compressor

The choice of compressor type is now a wide one, and at least two alternatives should be considered before making a final selection.

Compressor capacities may be shown in tables or curves, and will be for a given refrigerant and a range of condensing pressures (see Section 4.13). They may also show the power taken. At this stage, a first guess must be taken for the condensing temperature, and this might be 15 K above the summer dry bulb for an air-cooled condenser or 12 K above the wet bulb temperature in the case of water or evaporative cooling.

The balance condition between the evaporator and the compressor can be visualized in a graphic solution, superimposing the basic rating of the cooler on the compressor curves (see Figure 10.1).

While it is usual to consider only the balance at the maximum summer ambient, the application engineer should be aware of the running conditions in cooler weather. If this is not favourable to

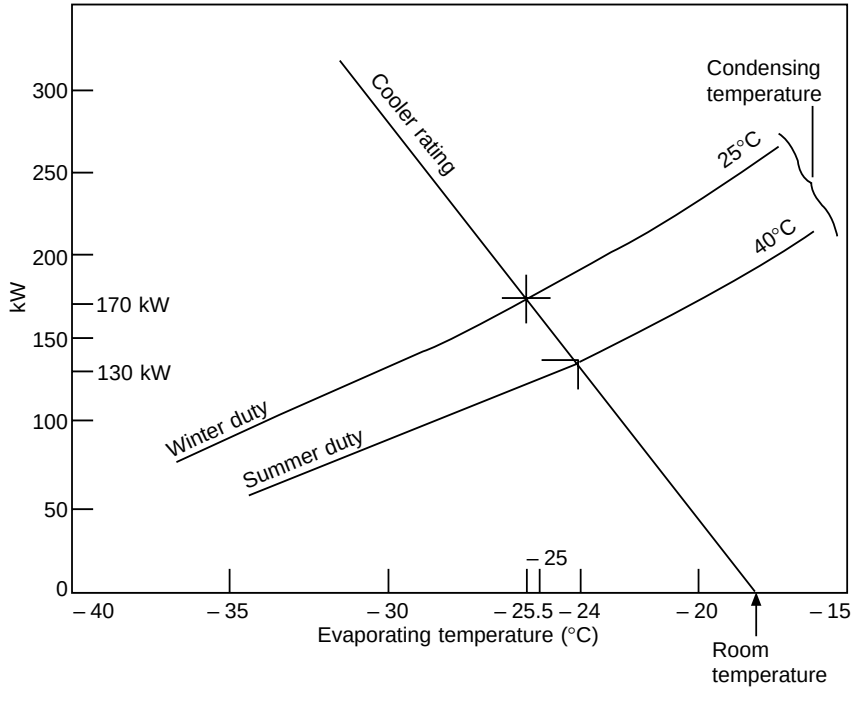


Figure 10.1 *Balance condition between compressor and evaporator*

the product, some average choice may be made, or a back pressure valve inserted to prevent the evaporating temperature dropping too low (see Figure 10.2). A different set of conditions will also occur if the compressor has capacity control. If this is likely to cause problems, then a compressor with 50% capacity control may be connected to two equal evaporators, and one of these shut off at half load.

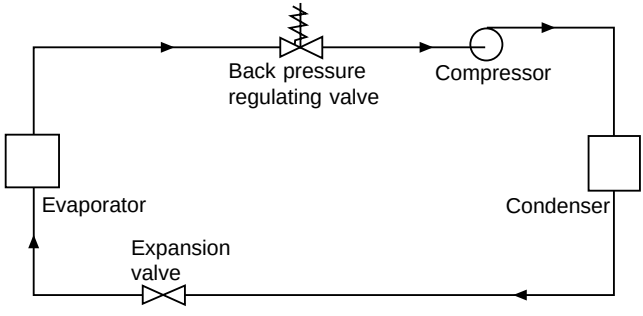


Figure 10.2 *Use of back pressure regulating valve to maintain evaporator pressure (and temperature)*

10.5 Condenser

A first guess of a condensing temperature has already been taken as a rough guide. Users should be aware of the wide difference in owning costs arising from the choice of condenser, so the options should be compared. The buyer who is influenced only by first cost will almost certainly face higher fuel bills. Certain machines, such as the centrifugal compressor, are very sensitive to high condensing conditions, and the correct choice (in this case, of a cooling tower) can give a considerable gain in COP.

Users seeking tender quotations should demand relative running costs and make their decision on the basis of their anticipated running times and so of the expected fuel costs, taking into account the slow inflation of electricity tariffs. Buyers must be aware of the tendency of a contractor to offer only one make or type of equipment, and where this situation arises, alternative tenders should be sought.

In most climates the wet bulb temperature is well below the dry bulb temperature and there is an advantage in using water or evaporative cooling for larger plant. These options need to be investigated and compared. The present concern over spray-borne diseases may indicate a preference for air cooling in the vicinity of institutions but correct maintenance of water cooling towers and evaporative condensers will permit their use elsewhere. Table 10.2, based on the tentative temperature differences of 15 K and 12 K

Table 10.2

Climate	*Air-cooled*		*Evaporative*	
	Dry bulb (°C)	*Condenser* (°C)	*Wet bulb* (°C)	*Condenser* (°C)
South UK	27	42	21	33
Scotland	24	39	18	30
Mediterranean	32	47	24	36
Desert	47	62	24	36
Tropical humid	33	48	28	40

given above, shows that such figures need to be reconsidered in extreme cases. For example, if it is necessary to use an air-cooled condenser in the desert, because there is no water available, then there will be considerable economy in oversizing the condenser to reduce the condensing temperature from a first guess of 62°C down to, possibly, 56°C.

Some manufacturers of air-cooled packaged condensing units offer a range of condenser sizes for each compressor, and these should be closely compared in terms of higher duty and lower running costs.

The maximum design condensing temperature will only apply when the ambient is at its hottest, and full advantage should always be taken to allow this temperature to drop at cooler times, down to its minimum working limit (see also [8–10]). Systems should be allowed to drop to a condensing temperature of 25°C when the cooling medium permits this, and some systems can go a lot lower. A true estimate of owning cost should take this into account.

The performance of alternative condensers with a compressor–evaporator system can be shown graphically but the curves will have to be plotted, since manufacturers cannot be expected to supply these figures for all conditions of working. In this construction (Figure 10.3) the rating curves are the rejected heat from the compressor, i.e. cooling duty plus compressor power. These are plotted against the basic rating of the condenser. Some condenser manufacturers provide rating curves based on the cooling capacity of the compressor and using typical factors for the power (see Example 6.2).

Air-cooled condensers require a large air flow for a given heat rejection duty and the limitation on their use is reached on account of their size and the need to get enough air. Water or evaporative cooling should always be considered as a possibility, except for smaller sizes or where using packaged condensing units.

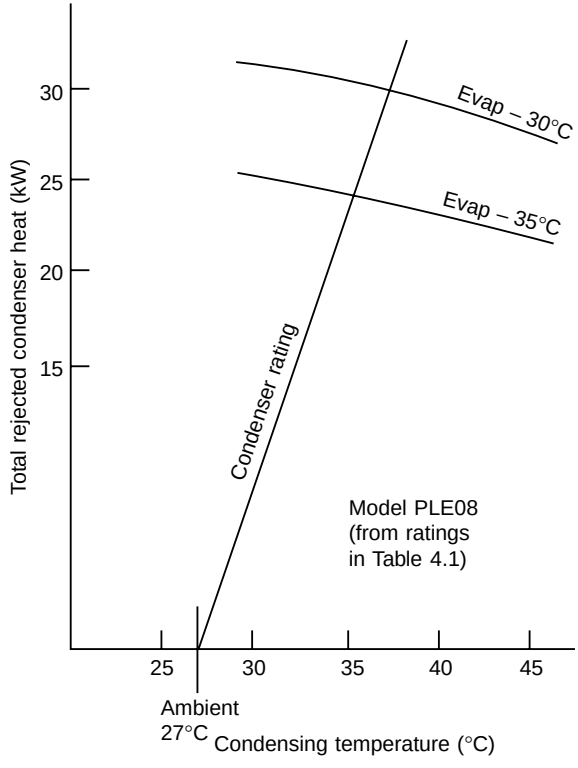


Figure 10.3 *Balance with condenser*

10.6 Expansion valve

The expansion valve is a passive orifice, through which the liquid refrigerant is forced by the pressure difference between the condensing and evaporating conditions. Capacity ratings are given in the catalogues of manufacturers and suppliers. Types in general use are:

1. Capillary tubes, for small hermetic systems. These are factory selected and cannot be adjusted.
2. Solenoid valves with liquid level sensors or liquid level valves for most flooded evaporators.
3. High-pressure float valves plus handset throttle valves for some flooded and low-pressure receiver circuits.
4. Thermostatic expansion valves or electronic expansion valves for most dry expansion circuits.

Troubles arise with the selection of thermostatic expansion valves, since this is the type generally used in custom-built systems and, for these, selected outside a factory.

It is usual to select a thermostatic expansion valve for the maximum duty and at the summer condensing condition, taking into account the pressure drop through a liquid distributor in the case of a multiple-feed coil. Valve ratings are given for a range of pressure differences, i.e. for a range of condensing conditions, in Table 10.3. It might be thought that the duty varies with pressure difference according to fluid flow laws, but this valve capacity is plotted against the expected mass flow curve in Figure 10.4. It is seen that the valve capacity is greater. This is because the refrigerant can absorb more heat if it is colder on entry. This means that the valve may be able to pass the required amount of liquid at a much lower condensing pressure. Conversely, if the valve is selected at a lower pressure difference (possibly corresponding to a condensing condition in the UK of 20–25°C), the valve will not be grossly oversized at the maximum summer condition.

Table 10.3

Pressure difference (bar)	2	4	6	8	10	12	14
Valve duty (kW)	0.77	0.95	1.08	1.16	1.22	1.24	1.26

Unless a thermostatic expansion valve is very tightly rated, the system will operate satisfactorily at a lower condensing condition in cool weather, with a gain in compressor duty and lower power input. A growing awareness of energy economy is leading to more careful application of this component. Suppliers are ready to help with advice and optimum selections.

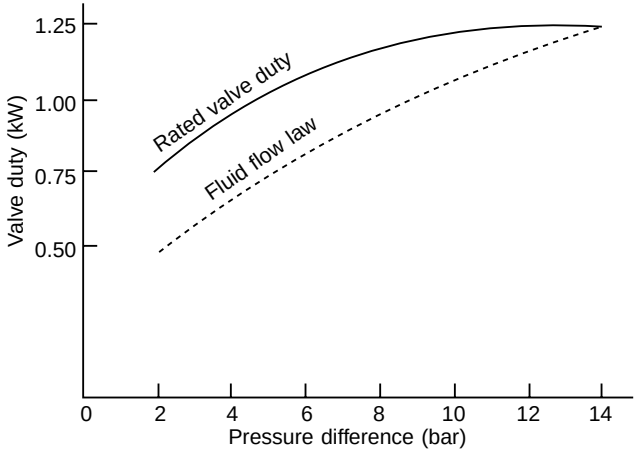


Figure 10.4 *Rating curve for expansion valve*

A greater difficulty arises where the compressor may go down to 33% or 25% capacity and the thermostatic expansion valve is called upon to control a much reduced flow. Under such conditions, the thermostatic expansion valve may be unstable and 'hunt', with slight loss of evaporator efficiency. Since the required duty is less, this is of no great importance. It is possible to fit two expansion valves in parallel, one for the minimum load and both for the full load, but this arrangement is not usually necessary.

Low condensing pressure operation should present no problem with float or electronic expansion valves, since these can open to pass the flow of liquid if correctly sized.

10.7 Sizing pipe and other components

Refrigeration system pipes are sized to offer a low resistance to flow, since this reflects directly on compression ratio, commensurate with economy of pipe cost and minimum flow velocities to ensure oil return with the halocarbons.

Pressure losses due to pipe friction can be calculated from the basic formulas established by Reynolds and others. However, as with the calculation of heat transfer factors, this would be a time-consuming process and some of the parameters are not known accurately. Recourse is usually made to simplified estimates or tables published in works of reference [32, 33].

Example 10.1 A suction pipe for an R.502 system, evaporating at – 40°C and having a cooling duty of 50 kW, is to be run in copper tube. What size should it be? Reference 32, Chapter 3, Table 3, shows that a copper tube of 79 mm nominal bore ($3\frac{1}{8}''$ o.d.) will carry R.502 suction gas for a cooling capacity of 51.86 kW, with a pressure drop of 23 kPa per 100 m run. This is given as a commercially acceptable pressure loss.

Pressure drops on the high-pressure side will be small enough to have little effect on the performance of the complete system. Pressure losses in the suction pipe and its fittings, especially if this is long, should be checked, and a correction made for the actual compressor suction pressure. For low-temperature applications, pipe sizes may have to be increased to avoid excessive frictional losses at these low pressures.

Flow control valves, such as back pressure valves, will not necessarily be the same nominal size as the pipe in which they are fitted. Manufacturers' data for selection of their products is usually very comprehensive, and their guidance should be sought in case of any doubt.

10.8 Recheck components

In the course of carrying through an equipment selection of this sort, several options may be tried. It is essential to make a final check on those selected to ensure that the correct balance has been achieved. Predicted balance figures should be noted, to guide the final commissioning process and subsequent operation.

11 Materials. Construction. Site erection

11.1 Materials

Materials used in the construction of refrigeration and air-conditioning systems are standard engineering materials, but there are a few special points of interest:

1. Compressors are generally of gray cast iron, but some makes are fabricated from mild steel.
2. Compressor pistons are of cast iron or aluminium, the latter following automobile practice.
3. Piping for the smaller halocarbon installations is usually of copper, because of the cleanliness and the ease of fabrication and jointing.
4. Some stainless steel pipe is used, mainly because of its cleanliness, although it is difficult to join.
5. Most other piping will be of mild steel. For working temperatures below – 45°C, only low-carbon steels of high notch strength are used (mainly to BS.3603).
6. Aluminium tube is used to a limited extent, with the common halocarbons and also with ammonia.
7. Copper and its alloys are not used with ammonia.
8. Sheet steel for ductwork, general air-conditioning components, and outdoor equipment is galvanized.

Specific guidance on materials and their application may be had from various works of reference [4, 16, 29, 30].

11.2 Pressure tests for safety

Factory-built equipment will be constructed to the relevant Standards and will be pressure tested for safety and leaks at the works. In cases of doubt, a test certificate should be requested for all such items.

Design and test pressures will depend on the refrigerant or other fluids used.

Site-assembled plant will be pressure tested for safety and leakage after erection (see Section 11.11).

11.3 Erection programme

Successful site erection of plant demands coordination of the following:

1. Site access or availability
2. Supply on time, and safe storage, of materials
3. Availability of layout drawings, flow diagrams, pipework details, control and wiring circuits, material lists and similar details
4. Availability at the correct time of specialist trades and services – builders, lifting equipment, labourers, fitters, welders, electricians, commissioning engineers, etc.

Site work is now mostly carried out by a number of subcontractors representing specialist trades. It is essential that authority and executive action are in the hands of a main contractor and that this authority is acknowledged by the subcontractors. If this is not so, delays and omissions will occur, with divided responsibility and lack of remedial action.

The controlling authority must, well before the start of site erection, draw up a material delivery and progress chart and see that all subcontractors (and the customer) are in agreement and that they are kept informed of any changes [31].

11.4 Pipe-joining methods

Steel pipe is now entirely welded, except for joints which need to be taken apart for service, which will be flanged. It is essential that welding is carried out by competent craftsmen and is subject to stringent inspection [29, 30].

Flanges for ammonia (and preferably, also, for other refrigerants) must be of the tongued-and-grooved type which trap the gasket. Mechanical joints for copper tube up to $\frac{3}{4}$ inch outside diameter can be of the flare type, in which the tube end is coned out to form its own gasket. This must be carried out with the proper flare tools, and it may be necessary to anneal the tube to ensure that the resulting cone gasket is soft.

Copper tube can be bent to shape in the smaller sizes and the use of bending springs or formers is advised, to retain the full bore.

Where fittings are required, these should be of copper or brass to give a correct capillary joint gap of not more than 0.2 mm, and joined with brazing alloy. This, again, is a craft not to be entrusted to the semi-skilled.

The brazing of copper tube will leave a layer of copper oxide inside, which may become detached and travel around the circuit. The best practice is to pass nitrogen into the pipe before heating, to avoid this oxidation. The use of special grades of oxygen-free or moisture-free nitrogen is not necessary.

11.5 Piping for oil return

The sizing and arrangement of suction and discharge piping for the halocarbons is dominated by the need to ensure proper entrainment of oil, to return this to the compressor. Pipes for these gases usually have a higher velocity at the expense of a greater pressure drop than those for ammonia. Pipe sizes may only be increased in runs where the oil will be assisted by gravity to flow in the same direction as the gas.

Horizontal pipes should slope slightly downwards in the direction of flow, where this can be arranged. If a suction or discharge line has to rise, the size may be decreased to make the gas move faster. In the case of a lift of more than 5 m, a trap should be formed at the bottom to collect any oil which falls back when the plant stops [33].

Suction and discharge risers (Figure 11.1) will normally be sized for full compressor capacity, and velocities will be too low if capacity reduction is operated. In such installations, double risers are required, the smaller to take the minimum capacity and the two together to carry the full flow. Traps at the bottom and goosenecks at the top complete the arrangement. At part capacity, any oil which is not carried up the main riser will fall back and eventually block the trap at the bottom, leaving the smaller pipe to carry the reduced flow, with its quota of oil. When the system switches back to full capacity, the slug of oil in this trap will be blown clear again.

11.6 Pipe supports. Valve access

Piping must be properly supported at frequent intervals to limit stress and deflection [10]. Supports must allow for expansion and contraction which will occur in use. In particular, pipework which might form a convenient foothold for persons clambering about the plant should be protected from damage by providing other footholds and guarding insulation.

Stop valves, especially those which might need to be operated in

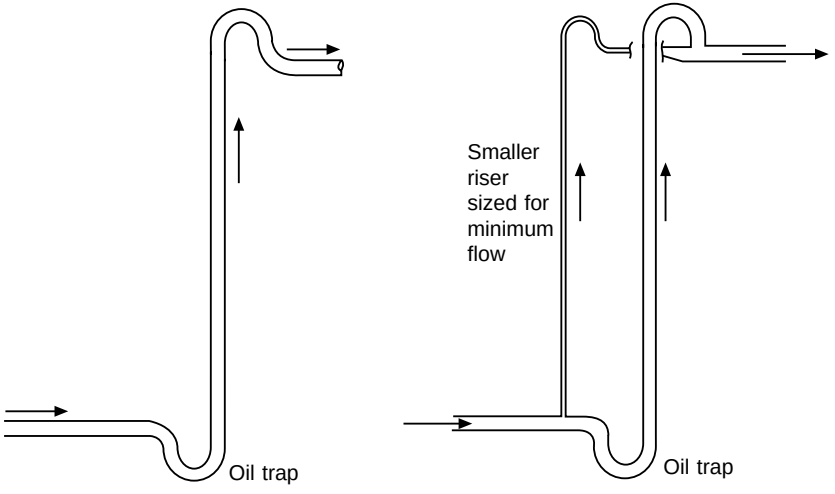


Figure 11.1 *Gas risers for oil return*

a hurry (and this means most, if not all, of them), should have easy access. Where they are out of reach, reliance should not be placed on moveable ladders, which may not be there when needed, but permanent access provided. Chain-operated wheels can be fitted to the larger valves, to permit remote operation.

Emergency stop valves must not be placed in tunnels or ducts, since personnel may be subject to additional danger trying to operate them.

11.7 Instruments

Until recently it has been the custom to fit thermometer wells at various points in the pipework, to enable check temperatures to be taken during initial commissioning and also during the life of the plant. The advent of the electronic probe thermometer has simplified commissioning work, and the fitting of thermometer wells is less important. Even so, such facilities are worth considering when the pipe is being erected, and will be necessary with insulated pipes if true temperatures are to be taken without damaging the insulation.

Wells should slope downwards into the pipe, so that they can be part filled with liquid to provide better thermal contact. Where a pipe temperature is a critical factor in the operation of a system, it is usually worth fitting a permanent thermometer.

The monitoring of temperatures for electronic control systems is now mainly by thermocouples, secured onto the outside of the pipe with self-adhesive tape and the pipe then insulated over.

Pressure gauges should always be fitted on the discharge side of liquid pumps, to check performance and give warning of a possible drop in flow resulting from dirty strainers. Manometer pressure gauges are required across air filters (see Chapter 27).

11.8 Rising liquid lines

If liquid refrigerant has to rise from the condenser or receiver to an expansion valve at a higher level, there will be a loss of static head, and the refrigerant may reach its boiling point and start to flash off. Under such circumstances, bubbles will show in the sight glass and will not be dispersed by adding more refrigerant to the system.

Example 11.1 R.22 condenses in a circuit at 34°C and is subcooled to 30°C before it leaves the condenser. How much liquid lift can be tolerated before bubbles appear in the liquid line?

Saturation pressure at 34°C = 13.21 bar

Saturation pressure at 30°C = 11.92 bar

Permissible pressure drop = 1.29 bar (129 000 Pa)

Specific mass of liquid = 1162 kg/m^3

$$\text{Possible loss in static head} = \frac{129\,000}{9.81 \times 1162}$$

(where $g = 9.81$ m/s^2)

= 11.3 m approximately

Where a high lift cannot be avoided, the liquid must be subcooled enough to keep it liquid at the lower pressure. Subcooling can be accomplished by fitting a subcooling coil to the condenser, a water-cooled subcooling coil, a suction-to-liquid heat exchanger before the lift, or a refrigerated subcooler.

To reduce the risk of these troubles, the condenser should always be higher than the evaporator, if this can be arranged.

The same effect will occur where the liquid line picks up heat on a horizontal run, where it may be in the same duct as hot pipes, or pass through a boilerhouse. If the sight glass flashes even with the addition of refrigerant, the possibility of such extra heating should be investigated. To cure this, insulate the pipe.

11.9 Vibration

Compressors and pumps will transmit vibration to their connecting pipework.

Water and brine pumps may be isolated with flexible connectors. For small-bore pipes, these can be ordinary reinforced rubber hose, suitably fastened at each end. For larger pipes, corrugated or bellows connectors of various types can be obtained. In all cases, the main pipe must be securely fixed close to the connector, so that the latter absorbs all the vibration. Flexible connectors for the refrigerant usually take the form of corrugated metal hose, wrapped and braided. They should be placed as close to the compressor as possible.

A great deal of vibration can be absorbed by ordinary piping up to 50 mm or 65 mm nominal bore, providing it is long enough and free to move with the compressor. Three pieces, mutually at right angles and each 20 diameters long, will suffice. At the end of these vibration-absorbing lengths, the pipe must be securely fixed.

In all instances of antivibration mounting of machinery, care must be taken to ensure that other connections – water, electrical, etc. – also have enough flexibility not to transmit vibration.

11.10 Cleanliness of piping

All possible dirt should be kept out of pipes and components during erection. Copper pipe will be clean and sealed as received, and should be kept plugged at all times, except when making a joint. Use the plastic caps provided with the tube – they are easily seen and will not be left on the pipe. Plugs of paper and rag tend to be forgotten and left in place. Steel pipe will have an oily coating when received, and it is important that this should be wiped out, since the oil will otherwise finish up in the sump and contaminate the proper lubricating oil. If pipe is not so cleaned, the compressor oil should be changed before the plant is handed over.

Rusty pipe should not be used. The rust and loosened mill scale will travel around the circuit to block the suction strainer and the drier. Other avoidable debris are loose pieces of weld, flux, and the short stubs of welding rod often used as temporary spacers for butt welds. Pipe should only be cut with a gas torch if all the oxidized metal can be cleaned out again before closing the pipe.

It should be borne in mind that all refrigerants have a strong solvent effect and swarf, rust, scale, water, oils and other contaminants will cause harm to the system, and possible malfunction, and shorten the working life.

11.11 Site pressure safety tests

Site-erected pipework, once complete, must be pressure tested for safety and freedom from leaks. Pressures will be 1.3 times the

maximum working pressure, and usual figures for the UK are 27.5 bar for the high side of air-cooled plants, 22.75 bar for water-cooled plants, and 13.75 bar for the evaporator side. These figures are for R.22 and R.717.

It is necessary to hold a Safe Handling of Refrigerants Certificate to work with refrigerants. This can be obtained through short training courses. Maintenance engineers must keep themselves updated on safety procedures with existing new refrigerants.

Factory-built components and pressure vessels which have already undergone test should not be retested, unless they form part of the circuit which cannot be isolated, when the test pressure must not exceed the original figure. Site hydraulic testing is considered unnecessary, owing to the extreme difficulty of removing the test fluid afterwards. However, it must always be appreciated that site testing with gases is a potentially dangerous process, and must be governed by considerations of safety. In particular, personnel should be evacuated from the area and test personnel themselves be protected from the blast which would occur if a pressure vessel exploded [30].

Testing should be carried out with nitrogen, and the use of grades of gas having very low levels of water or oxygen is not necessary. Air may be used where no oil is present but cannot be recommended, as it will bring with it a quantity of moisture, which is difficult to remove.

Nitrogen is used from standard cylinders, supplied at about 200 bar, and a proper reducing valve must always be employed to get the test pressure required. A separate gauge is used to check the test pressure, since that on the reducing valve will be affected by the gas flow.

If the high side is being tested, the low side should be vented to the atmosphere, in case there is any leakage between them which could bring excessive pressure onto the low side. It may be necessary to remove relief valves. Other valves within the circuit will have to be open or closed as necessary to obtain the test pressure. Servo-operated valves will not open on a 'dead' circuit, and must be opened mechanically.

After the test gas has gone in, there may be a slight change in pressure with a change of temperature. In particular, if left overnight, the pressure may drop as much as 1 bar. This is not significant.

The test pressure should be maintained for at least an hour. In this period a thorough test is made of each joint with soapy water. This method is no more tedious than a refrigerant leak test and saves the time and loss of refrigerant. Large leaks will be heard.

11.12 Evacuation

It is now necessary to remove as much as possible of the original air, with its moisture content, and the test gas, before introducing the refrigerant. The principle of evacuation is to reduce the pressure of any water vapour left in the piping to a saturation temperature well below the operating condition of the system. If this is not done, water will condense when the piping gets down to working temperature. These low pressures are expressed in a number of units, all as absolute pressures, as shown in Table 11.1.

Table 11.1

Temperature (°C)	*Vapour pressure of water, abs*			
	(Pa)	(mb)	(mm Hg) torr	(μmHg)
–60	1	0.01	0.01	7.5
–50	4	0.04	0.03	30
–40	13	0.13	0.10	96
–30	38	0.38	0.29	285
–20	103	1.03	0.77	775
–10	260	2.60	1.95	1950
0	611	6.11	4.5	4585

The test pressure is released and a vacuum pump connected to draw from all parts of the circuit. This may require two connections, to bypass restrictions such as expansion valves, and all valves must be opened within the circuit, requiring electrical supplies to solenoid valves and the operation of jacking screws, where these are fitted.

On small systems, such as factory packages, a final pressure of 50 μmHg (7 Pa) should be reached, but larger and site installations for air-conditioning temperatures are acceptable at a final vacuum of 170 Pa [31]. Vacuum pumps of this quality can be hired if not immediately available. Evacuation of a large system may take a couple of days. During this time, checks should be made around the pipework for cold patches, indicating water boiling off within, and heat applied to get this away.

Care should be taken that the pump used will tolerate the refrigerant gas.

11.13 Charging with refrigerant

Refrigerant may be charged as a liquid through the connection shown in Figure 11.2. The cylinder is connected as shown and the

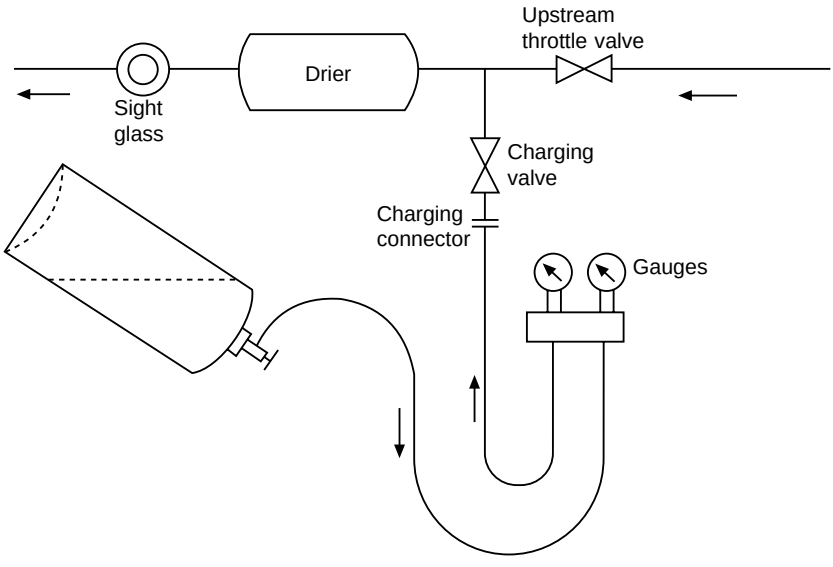


Figure 11.2 *Charging connection*

connecting pipe purged through with a little of the gas to expel air from it. For small charges, the bottle may be supported on a weighing machine, or a calibrated charging cylinder may be used.

The compressor must not be started while the system is under vacuum, so refrigerant is admitted first up to cylinder pressure. At this point, the compressor may be started, assuming that all auxiliary systems (condenser fan, pump, tower, cooler fan, etc.) are running. The liquid-line valve upstream of the charging connection is partially closed to reduce the line pressure at this point below that of the supply cylinder, and the refrigerant will flow in. While the refrigerant can be safely admitted in this way, the system is not running normally, since the throttle valve is reducing the pressure across the expansion valve. At intervals during charging, the cylinder valve must be closed and the throttle valve opened fully. Only under these conditions can correct running be observed. When fully charged, the sight glass will be clear [34].

As an alternative method, the cylinder pressure can be increased by gently heating it. Any heating of this sort should only be done while keeping a careful watch on the cylinder pressure. Serviceman's charging cylinder sets can be obtained with built-in heater elements. Raising the cylinder pressure in this way avoids the use of the throttle valve and the charging process is much quicker.

If no receiver is fitted, extra charge may be added, possibly another 5% above that already in the plant, to allow for seasonal and load

variations. If a receiver is in circuit, this should be about one-sixth full under normal running conditions.

Systems having high-pressure float expansion valves, and those without sight glasses, must be charged gradually, observing the frost line or using a contact thermometer to measure superheat.

Small packages will have the charge marked on the nameplate and must be carefully charged to this weight, which will be critical.

Systems may need to have further lubricating oil added, to make up for that which will be carried around with the refrigerant. In the absence of any firm guidance from the supplier, the crankcase must be topped up gradually during normal running, until it is level with the middle of the sight glass under operating conditions. This is not so with the small hermetic systems, where there is usually sufficient oil in the compressor to supply the needs of the circuit, and which, in any case, have no oil level sight glass.

11.14 Insulation

Pipework and other components should be insulated after the safety pressure test, but usually before prolonged running of the plant, since it is very difficult to remove water and frost once it has formed. Only the low-pressure piping is insulated, where it does not form part of the evaporator, i.e. after the expansion valve, where this may be outside the cooled space, and from the evaporator back to the compressor. Basic materials are cork and the expanded plastics. These are sufficiently rigid to be moulded to the correct shape, remain firmly in place, and support the external vapour barrier which is essential to prevent the ingress of water vapour (see also Chapter 15).

Insulants for pipework and curved pressure vessels can be obtained ready shaped, so that they fit tightly to the surface. All surfaces should be quite dry before the material is applied, even if the adhesive is a water-based emulsion, and the water or other solvent must be given ample time to dry or set before any outer wrapping is applied.

Any air spaces within the insulation should be avoided, since this air will contain moisture, which will condense and freeze, leading to early deterioration of the insulant.

The essential part of the insulation system is the vapour barrier, which must be complete and continuous over the outer (warm) surface. Even materials such as coconut fibre, rice husks, sawdust and wood shavings are successfully used as insulants if the vapour barrier is good.

The application of insulating materials is a specialist trade and justifies careful supervision and inspection.

Much use is made of flexible foamed plastic material, which can be obtained in tubular form for piping up to 114 mm diameter and in flat sheets of various thicknesses for tailoring onto other shapes. This material has a vapour-tight outer skin, but must be sealed at all joins and the ends. The manufacturers are helpful in advising users.

11.15 Water circuits

Water and other fluid circuits will be pressure tested for safety and leakage, using water at a pressure of 1.5 times the working pressure, or as required.

The opportunity is taken, while filling for testing, to ensure that the circuits can be filled without airlocks. Air vents at high points of the circuit may be automatic or manual. While the pipes are full, pumps should be run if possible to dislodge any dirt before draining down and cleaning the strainers. If a lot of dirt is found, the pipework should be filled again and reflushed. In any case, the pumps should be run at the earliest opportunity and the strainers cleaned out.

Fluids, if other than water, are not put in until this pressure testing and flushing has been carried out.

11.16 Non-condensible gases

Other gases, mainly ambient air, may enter a refrigeration system as a result of incomplete evacuation before charging, opening of parts for maintenance or repair, or inward leaks on circuits operating below atmospheric pressure. These gases will be circulated with the refrigerant vapour until they are all in the condenser and receiver. They cannot move further around the circuit because of the liquid seal at the outlet to the expansion valve.

Within the confines of the condenser and receiver, the gases will diffuse together and will exist in the same proportions throughout. The non-condensibles may therefore be removed through purge valves on either vessel, but such valves are commonly fitted on or near to the hot gas inlet to the condenser. The presence of non-condensible gas will be shown as an increase of condenser pressure (Law of Partial Pressures) and may be detected during normal operation if the running log is accurate. The effect of this higher condenser pressure is to increase the compression ratio and so reduce the volumetric efficiency and increase the power. There will also be the effect of the gas blanketing the condenser surface, reducing heat flow.

Where the presence of such gas is suspected, a cross-check can be made, providing the high-pressure gauge is of known accuracy.

The method is to switch off the compressor after a short running period, and so stop the flow of thermal energy into the condenser, but continue to run the condenser until it has reached ambient conditions. The refrigerant vapour pressure can then be determined from the coolant temperature, and any increase indicates non-refrigerant gas in the system.

The bleeding of gas from the purge valve will release a mixture which can be estimated from the total pressure.

Example 11.2 A system containing R.22 is cooled to an ambient temperature of 20°C and the condenser gauge then indicates 10.5 bar. What is the partial pressure of the non-condensible gas, and how much R.22 must be lost to purge 1 kg of this gas? Assume it is air.

Vapour pressure of R.22 at 20°C = 9.09 bar

Observed pressure = 10.50 bar

Partial pressure of non-condensible gas = 1.41 bar

Gas	*Proportion by pressure*	*Molecular mass*	*Proportion by weight*	*Weight ratio*
Air	1.41	28.97	40.85	1
R.22	9.09	86.5	786.3	19.2

So 19.2 kg of R.22 must be wasted to purge 1 kg of non-condensible gas.

Ammonia has a much lower molecular mass and the proportion by weight in this example would only have been 3.8 kg of ammonia lost. Also, ammonia is much cheaper than R.22.

11.17 Automatic gas purgers

Wastage of refrigerant can be reduced by cooling the mixture of gases and thus reducing the ratio. By means of a refrigerated purge device, which cools the mixture down to the evaporator temperature (for example, – 35°C in a blast freezer plant), the ratio would become

Vapour pressure of R.22 at – 35°C = 1.318 bar

Partial pressure of non-condensible gas

(10.50 – 1.318) = 9.182 bar

The ratio now becomes 0.43 kg of R.22 lost per kilogram of air.

Automatic gas purgers comprise a collection vessel for the gas mixture with an inbuilt cooling coil connected to the main suction, or with its own refrigeration system. Condensed refrigerant returns to the condenser, and any gas remaining in the vessel will be non-condensible and can be vented by an inverted bucket trap.

Purging of gases must always be to the open air. The release of ammonia–air mixture is usually made through a flexible tube into a container of water. The water will absorb the ammonia, and any bubbles seen to rise to the surface will be other gases.

12 Liquid chillers. Ice. Brines. Thermal storage

12.1 Distributed cooling

A building or process having a large number of separated cooling loads could have a refrigeration system for each of these loads. It will usually be more convenient to concentrate the cooling into one plant.

The cooling effect of a central refrigerating system can be distributed by a heat-transferring liquid or secondary refrigerant.

Where the working temperatures are always above 0°C, such as in air-conditioning, water is commonly used. At temperatures below this, non-freezing liquids are used.

12.2 Liquid chillers

The preferred secondary refrigerant will be water, if this can possibly satisfy the temperature requirement, i.e. if the load temperature is sufficiently above 0°C that water can be circulated without risk of freezing.

The greatest demand for chilled water is in air-conditioning systems (see also Chapters 23–28). For this duty, water is required at a temperature usually not lower than 5°C and, for this purpose, the evaporator will be of the shell-and-tube type, operating with refrigerant temperatures close to freezing point. A very wide range of factory-built package chillers is available and models are mainly one-piece units with integral water-cooled condensers as shown in Figures 4.18 and 13.2. Other types may have air-cooled or evaporative condensers, and so require refrigerant pipe connections on site to these condensers. Sizes range from 14 kW to 35 000 kW, most installations being within the range 100–1500 kW.

At water temperatures close to freezing, and with evaporators

which are vulnerable to ice damage, it is important to have adequate safety controls, to check the calibration of these frequently and to avoid interference by unauthorized persons. Nearly all troubles from packaged water chillers arise from a failure of safety controls. Several types of controls are in use, frequently three or more on the same equipment, but there should never be less than two of them:

1. Water flow switch, to stop the machine if flow stops in the chilled water circuit
2. Refrigerant low-pressure cut-out
3. Water outlet low-temperature cut-out
4. Back pressure regulation valve (see Section 9.8)
5. Hot gas bypass valve, to keep the evaporating temperature up above freezing point.

Most packaged water chillers are large enough to have capacity control devices in the compressor. The main control thermostat will unload the compressor as the water temperature approaches a lower safe limit, so as to keep the water as cold as possible without risk of freeze damage.

In all but the smallest installations, two or more chillers will be used, or one chiller with two separate circuits. This arrangement gives some continuity of the service if one machine is off-line for maintenance or another reason, gives better control and provides economy of running when loads are light.

If water is required below 5°C, the approach to freezing point brings considerable danger of ice formation and possible damage to the evaporator. Some closed systems are in use and have either oversize heat exchange surfaces or high-efficiency-type surfaces. In both of these, the object is to improve heat transfer so that the surface in contact with the water will never be cold enough to cause ice layers to accumulate.

12.3 Baudelot coolers and ice bank coils

Water can be cooled safely to near freezing point, using evaporators which have the refrigerant inside, with space for ice to form on the outside of the surface without causing damage. Two types are used:

1. Baudelot coolers (see Figures 7.7 and 7.8*a*). The evaporator stands above a collection tank, and the water runs down the outside surface in a thin layer. Evaporator construction can be pipe coils or embossed plates. The latter are now usually of stainless steel, to avoid corrosion troubles. Evaporators may be flooded or dry expansion. During operation, a Baudelot cooler

may sometimes build up a thin layer of ice, but this does no damage to the evaporator, and should melt off again when the load changes.

2. Pipe coils within a water tank (see Figures 7.1*c*, 7.5 and 7.6). Both flooded and dry expansion evaporators are in use. Water is circulated by pumps and/or special agitators. This type of water chiller may be operated without formation of ice, or ice may be allowed to accumulate intentionally (see below).

Water chillers of these two types are not usually made as single packages with their condensing unit, owing to the bulk of the system and subsequent difficulty of installation.

12.4 Ice manufacture

Ice may be made and transported to where the cooling effect is required. The refrigeration energy available in this way will be mainly its latent heat of melting, 334 kJ/kg, as it changes back to water.

Ice can be made as thin slivers on the surface of evaporator drums, and removed mechanically when the correct thickness has been formed. Either the drum or the scraper may rotate. This is a continuous process and the ice flakes fall directly onto the product or into a storage bin below the machine. Smaller units are made as packages with the bin integral and cooled by a few turns of the suction line or by a separate evaporator. Small pieces of ice can be formed in or on tubes or other prismatic shapes made as evaporator tubes, arranged vertically. Water is pumped over the surface to freeze to the thickness or shape required. The tube is then switched to 'defrost' and the moulded section thaws sufficiently to slide off, possibly being chopped into short pieces by a rotating cutter. The machine itself will be made as a package, and the smaller sizes will include the condensing unit.

The manufacture of ice in large blocks is by the can method (see Figure 12.1), where a number of mould cans, filled with water, are immersed to just below the rim in a tank of refrigerated brine. The smallest block made in this way is 25 kg and will freeze in 8–15 h, using brine at –11°C. Blocks up to 150 kg are made by this method. When frozen, the moulds are lifted from the tank and slightly warmed to release the ice block from the sides of the moulds, when they can be tipped out. Blocks may go into storage or for direct use.

Where the available water has a high proportion of solids, there are methods either of pretreating the water or, by agitating the water in the centre of the block (which freezes last), of removing the concentrated dirty water before it becomes solid. The core is then refilled with fresh water [30].

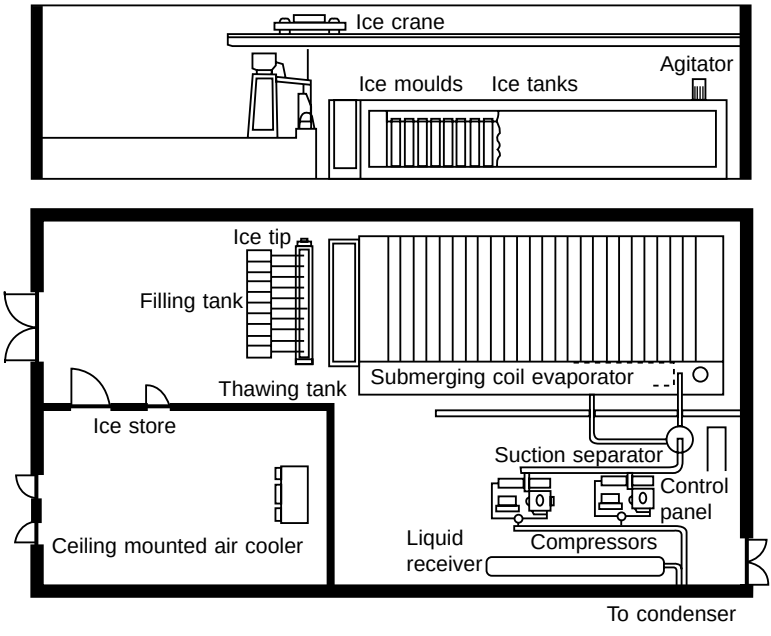


Figure 12.1 *Can ice plant (Courtesy of Hall-Thermotank Products Ltd)*

Block ice can be made from sea water but the central core cannot be frozen.

12.5 Brines

Where a secondary refrigerant fluid is to be circulated, and the working temperatures are at or below 0°C, then some form of non-freeze mixture must be used. These are collectively termed brines.

Brines may be, as the name suggests, solutions of inorganic salts in water, and the two in general use are sodium chloride and calcium chloride. Of these, the former is compatible with most foodstuffs and can be used in direct contact or in circumstances where the brine may come into contact with the product. Calcium chloride has an unpleasant taste and cannot be permitted to contaminate foods.

12.6 Physical properties

With any solution, there will be one concentration which remains liquid until it reaches a freezing point, and then will freeze solid. This is the eutectic mixture, and its freezing point is the eutectic point of the solute (see Figure 12.2). At all other concentrations, as

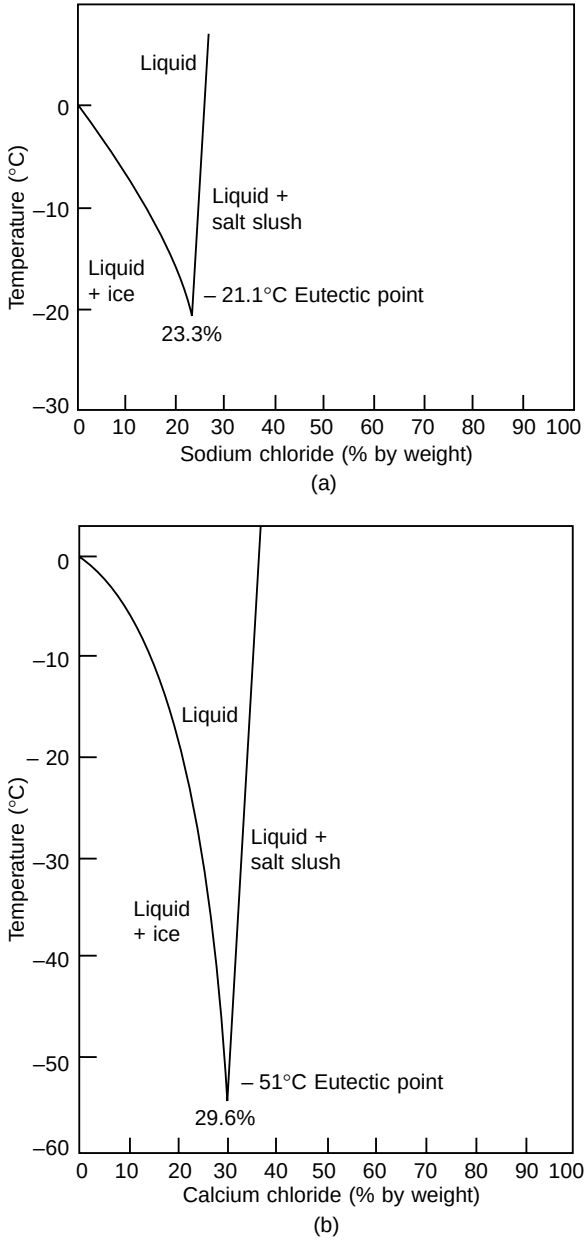


Figure 12.2 *Eutectic curves. (a) Sodium chloride in water. (b) Calcium chloride in water*

the solution is cooled it will reach a temperature where the excess water or solute will crystallize out, to form a slushy suspension of

the solid in the liquid, until the eutectic point is reached, when it will all freeze solid.

For economy of cost, and to reduce the viscosity (and so improve heat transfer), solutions weaker than eutectic are normally used, provided there is no risk of freezing at the evaporator.

In salt brines, the water may be considered as the heat transfer medium, since the specific heat capacity of the salt content is low (see Figure 12.3). The specific heat capacity of the brine will therefore

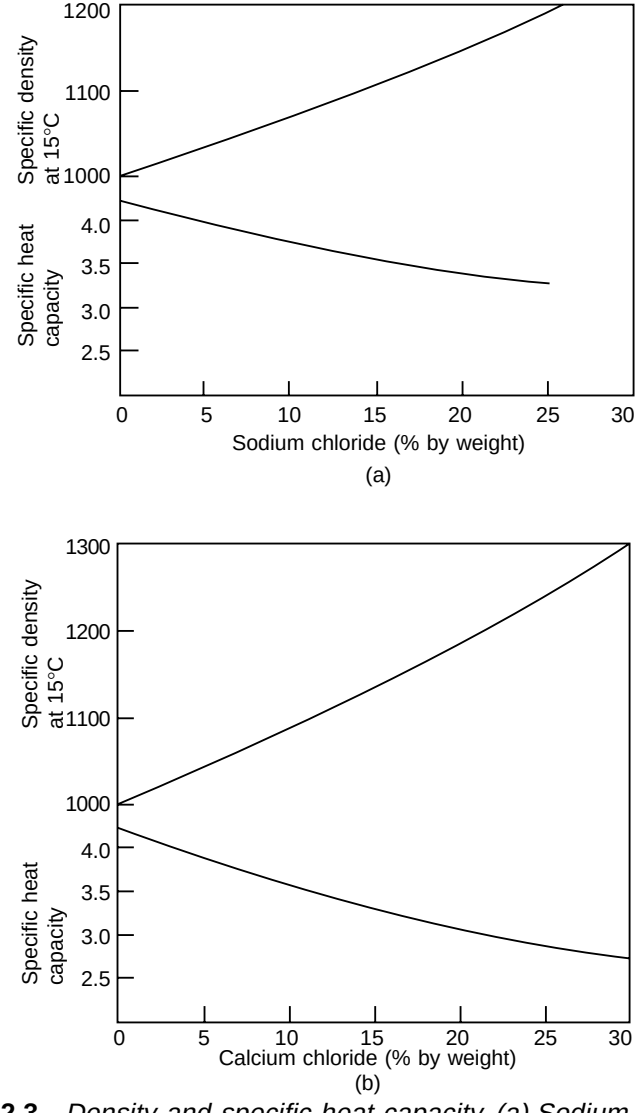


Figure 12.3 *Density and specific heat capacity. (a) Sodium chloride. (b) Calcium chloride*

decrease as the salt concentration increases. At the same time, the specific mass will increase.

Non-freezing solutions can also be based on organic fluids, principally the glycols, of which ethylene and propylene glycol are in general use. Where contact with food is possible, propylene glycol (see Figure 12.4) should be used.

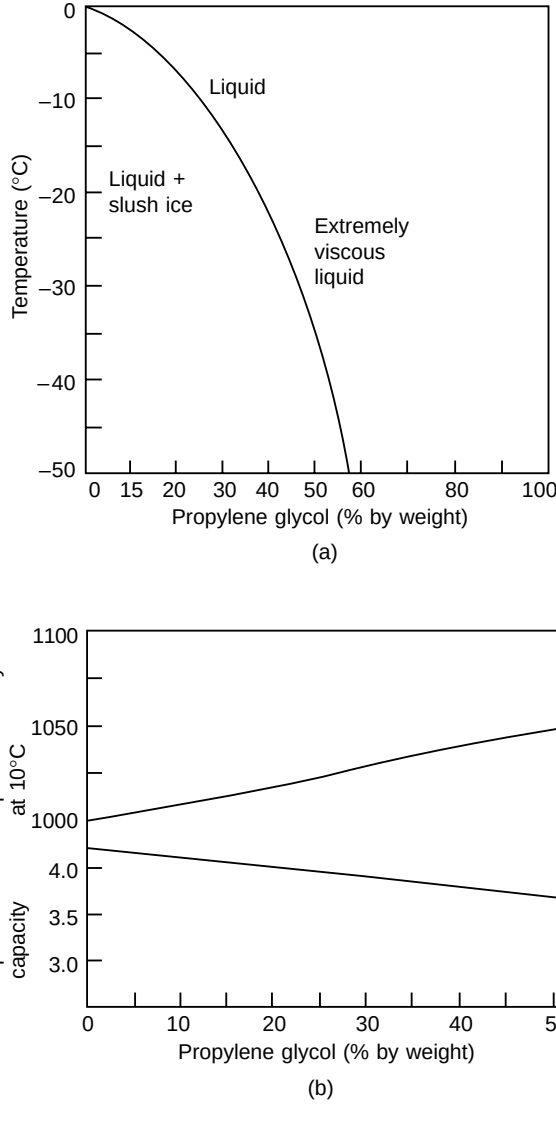


Figure 12.4 *Propylene glycol in water. (a) Eutectic curve. (b) Density and specific heat capacity*

The concentration of a solute has a considerable effect on the viscosity of the fluid and so on the surface convective resistance to heat flow. There is little published data on these effects, so applications need to be checked from basic principles. Industrial alcohol (comprising ethyl alcohol with a statutory addition of methyl alcohol to render it poisonous) may be used as a secondary refrigerant, either at 100% concentration or mixed with water. The fluid has a low viscosity and good heat transfer, but is now little used on account of its toxicity and the fire risk in high concentrations. Other non-freeze heat transfer fluids are used in specialist trades.

12.7 Brine circuits

Brine may be pumped to each cooling device, and the flow controlled by means of shut-off or bypass valves to maintain the correct temperature (see Figure 12.5)

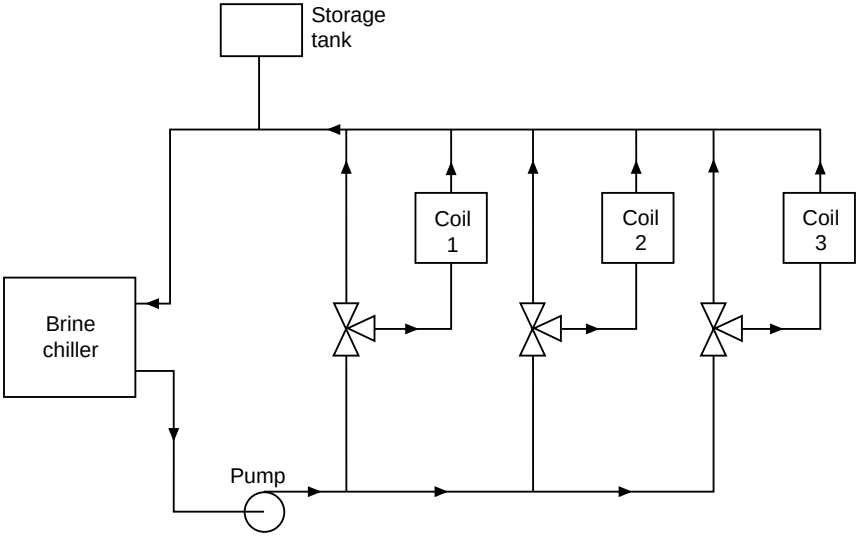


Figure 12.5 *Brine circuit for separate rooms*

Where a brine system services a multiple-temperature installation such as a range of food stores, the coolant may be too cold for some conditions, causing excessive dehydration of the product. In such cases, to cool these rooms the brine must be blended. A separate three-way blending valve and pump will be required for each room (see Figure 12.6).

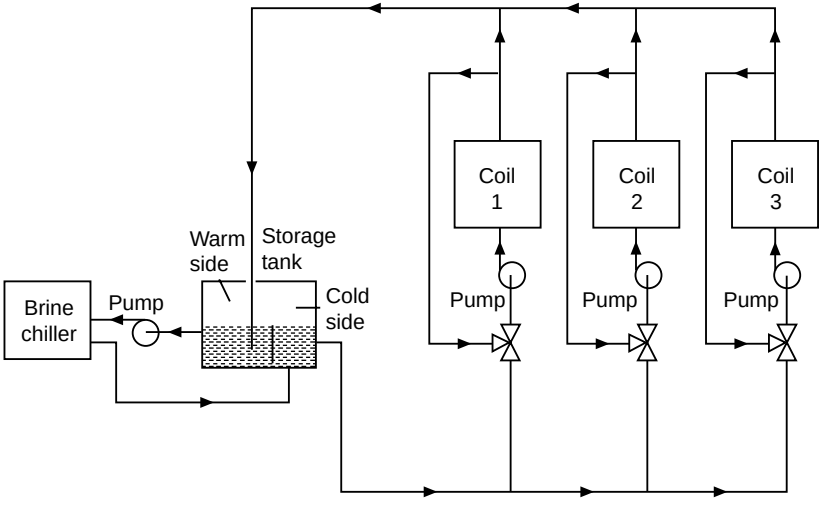


Figure 12.6 *Brine circuit for rooms at different temperatures*

12.8 Corrosion

If brine circuits are open to the atmosphere, air may be entrained, with consequent oxidation, and the solution will become acid. This will promote corrosion and should be prevented as far as possible by ensuring that return pipes discharge below the tank surface.

To reduce the effects of corrosion, inhibitors are added, typically sodium chromate in the salt brines and sodium phosphate in the glycols. These are alkaline salts and help to counteract the effects of oxidation, but periodic checks should be taken, and borax or similar alkali added if the pH value falls below 7.0 or 7.5 [1].

Brines are hygroscopic and will weaken by absorbing atmospheric moisture. Checks should be made on the strength of the solution and more salt or glycol added as necessary to keep the freezing point down to the required value.

The preferred brine circuit is that shown in Figure 12.5, and having the feed and expansion tank out of the circuit, which is otherwise closed. This avoids entrainment of air and too much surface exposure. The same arrangement can be used with the divided storage tank shown in Figure 12.6, except that the tank will be enclosed, with a separate feed and expansion tank.

12.9 Thermal storage by frozen brines and ice

Variations in cooling load can be provided from the latent heat of melting of ice or a frozen eutectic. Ice can be formed by allowing it

to build up on the outside of evaporator coils in a tank. Brines are more normally held in closed tanks or plates, again with evaporator coils inside, the outside of the tank forming the secondary heat exchange surface. Eutectics can be formulated according to the temperature required (see Figure 7.8).

A variation is to have a pumpable fluid such as one of the glycols, and to contain a eutectic solution within capsules in a storage tank. The capsules are in the form of plastic balls and the eutectic within may be formulated to suit any required thermal storage temperature. The capsules are frozen solid by pumping the glycol through a normal shell-and-tube cooler and then through the tank. When the stored cooling effect is to be used, the glycol flow is diverted to the load, and the capsules then melt again. This system has the advantage of avoiding the corrosion effects of salt brine, and can be used at almost any required storage temperature, depending on the eutectic temperature of the mixture within the capsules.

A similar product is available for domestic use. Plastic containers hold a eutectic solution, and these are frozen down by placing these elements in the domestic deep-freeze cabinet. Once frozen, they can be used in picnic baskets, etc., for the short-term storage of cold foods, wines, ice cream, etc.

The use of ice cubes to cool beverages by contact or immersion is well known.

In commercial use, thermal storage has three main applications:

1. To handle a peak cooling load with a reduced size of refrigeration plant, typically to make ice over a period of several hours and then use ice water for the cooling of a batch of warm milk on a dairy farm. This is also used at main creameries, to reduce peak electricity loads. The available water is very close to freezing point, which is the ideal temperature for milk cooling.
2. To run the refrigeration system at night, or other times when electricity is cheaper, to avoid premium electricity rates, or to avoid maximum demand charges. It is also in use in areas where the electricity supply is unreliable. Where the cold water is to be used at a higher temperature, such as in air-conditioning, the circuit will require three-way blending valves.
3. As hold-over cooling plates in transport (see Figure 7.8*d* and Chapter 20).

13 Packaged units

13.1 General

A high proportion of the total cost of a refrigeration or air-conditioning system is made up of work which can be carried out quicker, cheaper and under better managerial control within a factory rather than on the installation site. This work includes the following:

1. Procurement, inspection and storage of bought-out items
2. Storage of manufacturing materials
3. Manufacture of in-house components
4. Assembly of parts into systems
5. Piping, wiring, charging, testing

A wide range of factory-built packaged units is now made, and covers most requirements except the larger or more specialized installations.

The advantages of packaged units are as follows:

1. Correct selection and balance of components
2. Assembly, leak testing, processing and charging under factory conditions
3. Inspection and testing of the complete unit before it leaves the factory
4. Delivery to the site complete and in working order, so avoiding site delays for materials
5. Simplified site installation, with a minimum of disruption, inconvenience and cost

Disadvantages are that the unit may not be exactly the right size for the duty, since a stock unit may be used, and the risk of misapplication.

13.2 Condensing units

The basic condensing unit is a single package comprising the

compressor with its drive, the condenser (either air- or water-cooled) and all connecting piping, and the necessary controls to make the set functional (Figure 13.1).

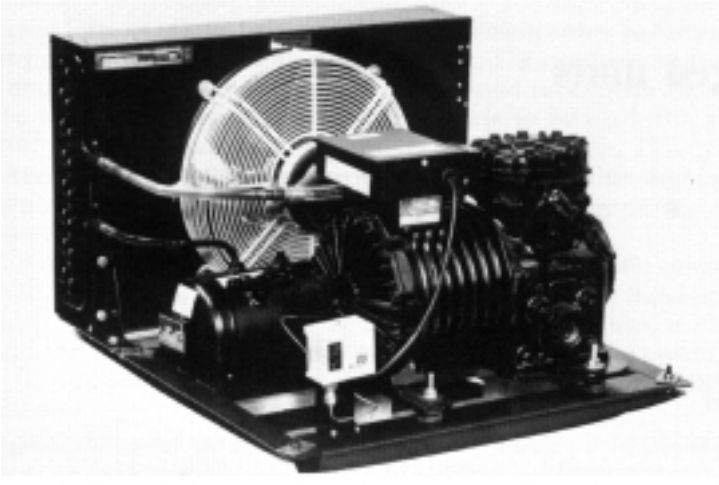


Figure 13.1 *Air-cooled condensing unit (Courtesy of Prestcold Ltd)*

Such assemblies might have the compressor and drive only, for site connection to a remote air-cooled condenser. As such, they are correctly termed compressor units. Compressor and condensing units will be site connected to evaporators, and these components must be matched in capacity (see Chapter 10).

Cooling capacity data will be based on various condensing conditions, in terms of air or water temperature onto the condenser, and for a range of evaporating conditions for which the set may be suitable [35].

Example 13.1 In the rating curves for an air-cooled condensing unit shown in Figure 13.3, what is the cooling capacity at an evaporating temperature of – 25°C and with air onto the condenser at 25°C? By how much does this drop with condenser air at 35°C?

From curves rating at – 25 to + 25°C is 1310 W

rating at – 25 to + 35°C is 1085 W

Example 13.2 In the rating table shown in Table 13.1 for a water-cooled condensing unit, what is the cooling duty at –20°C evaporation, with water onto the condenser at 25°C?

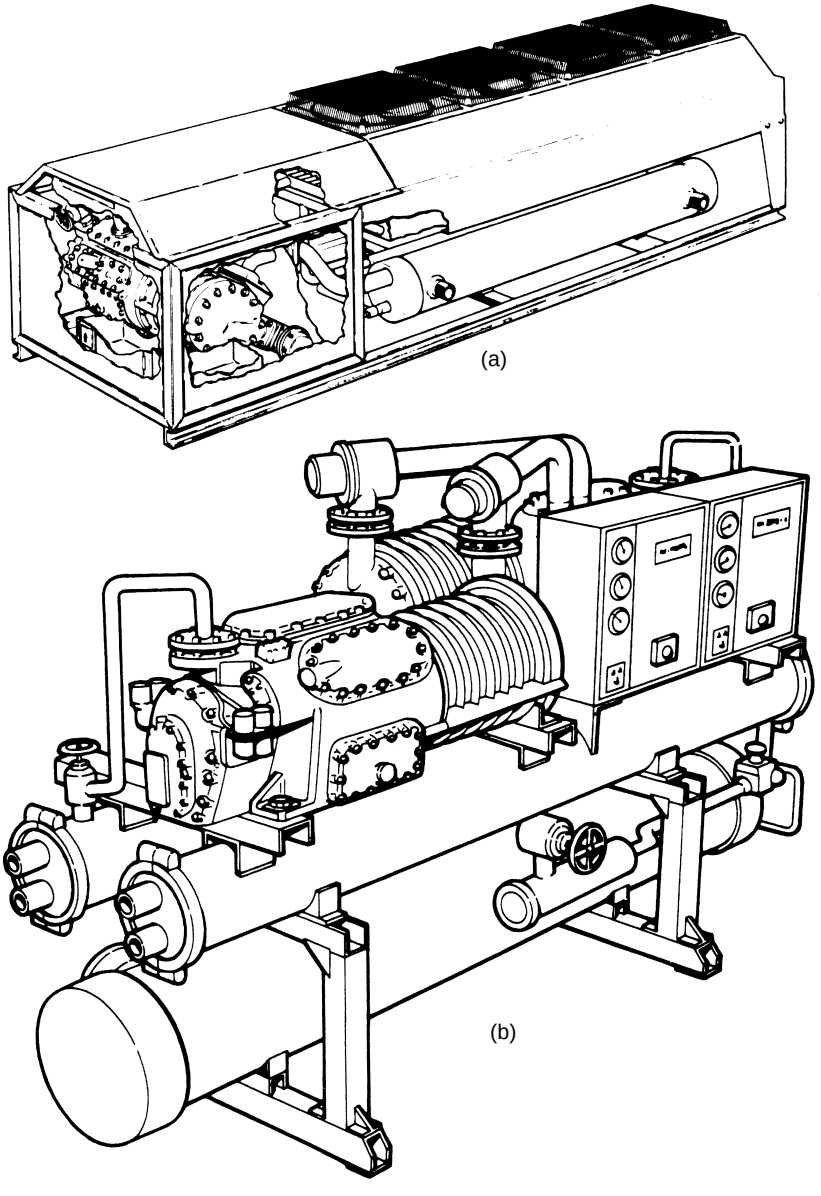


Figure 13.2 *Packaged water chillers. (a) Air cooled. (b) Water cooled (Courtesy of APV Baker Ltd)*

From table, rating at – 20 to + 25°C is 18.6 kW

Since compressor and condensing units do not include an evaporator, they are not complete systems and will not be charged with refrigerant, but may have a holding charge of dry nitrogen, or a little of the

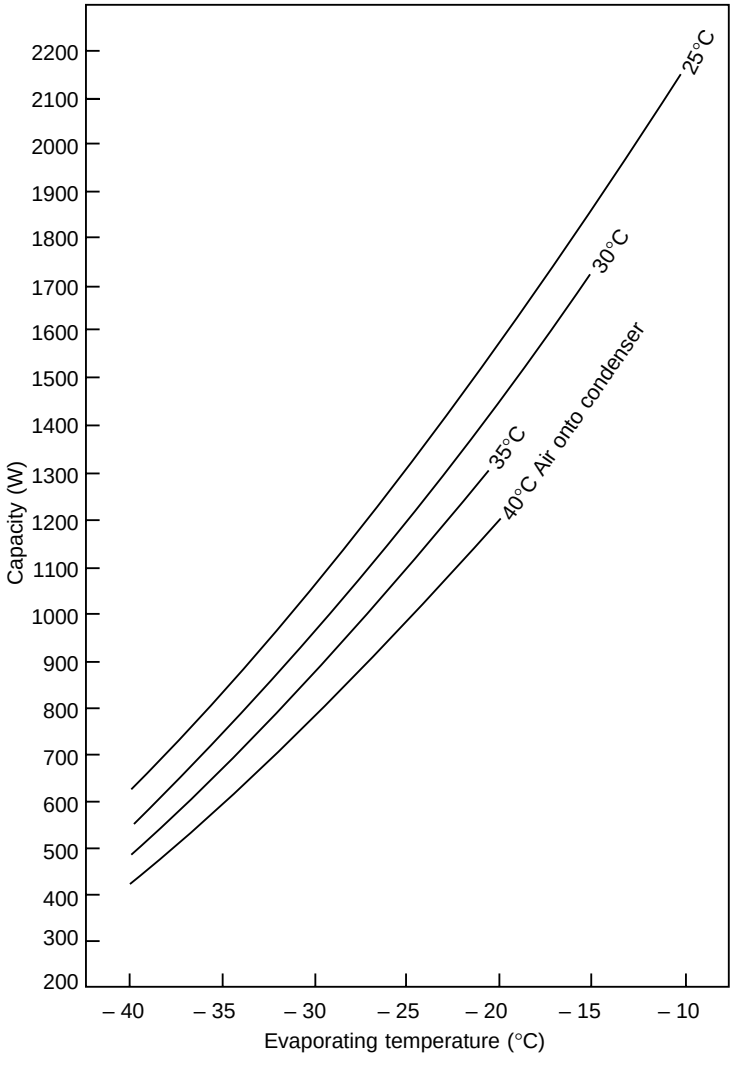


Figure 13.3 *Capacity curves for AS75 condensing unit*

refrigerant gas to maintain a slight positive pressure for transit. Suction and liquid interconnecting lines and wiring will have to be installed on site.

13.3 One-piece packages

The true packaged unit will have all the parts of the system and will be factory tested in the complete state. There are four basic types:

Table 13.1 Capacity, in kW, of water-cooled condensing

Water onto condenser (°C)	*Evaporating temperature*			
	– 30°C	– 20°C	– 10°C	0°C
25	10.5	18.6	30.6	45.1
30	9.7	17.2	28.1	41.8
35	9.0	15.3	25.1	37.4

Air cooling, air cooled
Air cooling, water cooled
Liquid cooling, air cooled
Liquid cooling, water cooled

Ratings for such units will be published in terms of the entering fluid on both the evaporator and condenser side (see also Chapter 35).

The siting of a packaged unit is more critical than that of separate plant, since all components are together, and a compromise may have to be reached between the convenience of having the unit close to the load and the difficulty of obtaining condenser air or water, transmitting extra noise, or creating new safety aspects.

13.4 Split packages

To avoid the constraint of having all parts in one package, the evaporator set may be split from the condenser, the compressor going with either (see Figure 13.4). The unit will be designed as a complete system but the two parts are located separately and connected on site. On some small units, flexible refrigerant piping may be provided.

If the system is of a range up to about 5 kW, coils of precharged soft copper tube, with self-sealing couplings, may be supplied for connection within a limited distance of 5–15 m. This facility enables full factory processing to be carried out to the standards of a one-piece unit. It is limited to the availability of suitable tubing, usually $\frac{5}{8}$ inch outside diameter. In such systems, the total charge is suitable for the final assembly, and pipes should not be extended beyond the factory-supplied length without prior consultation with the supplier.

Larger split packages must be piped on site by normal methods, and then processed and charged as an open plant. Split unit evaporators should not be located more than 5 m higher than their condensers (see Section 11.8) See also *multi-splits*, Section 28.8.

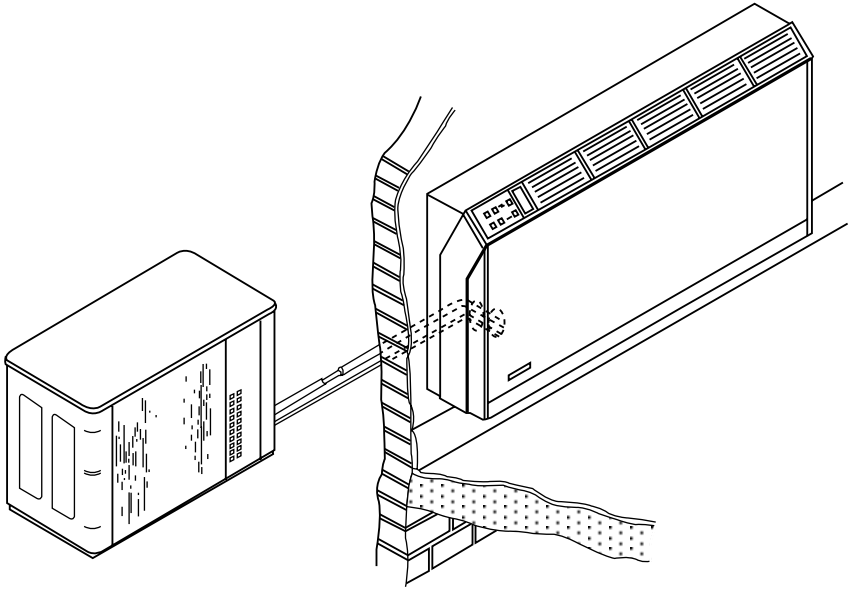


Figure 13.4 *Split package air-conditioner (Courtesy of Qualitair)*

13.5 Evaporator units

Evaporator sets, as supplied as part of a split package or for application to a condensing unit, will be of three main types:

1. Air-conditioning, having the air cooling coil with drip tray under, expansion valve, fan and motor, air filters, inlet and outlet grilles. They may also include dampers and duct connections for return and fresh air, heaters, humidifiers and various controls.
2. Cold store evaporators having the coil with drip tray under, fans, and possibly the expansion valve.
3. Cold store evaporators for use below + 2°C must also have some means of defrosting the coil. If this is to be by electric heat, the elements will be inbuilt.

13.6 Application data

Comprehensive application data should be made available for all marketed packaged units, to allow designers or sales engineers to make the correct selections for their purposes. However, it should be borne in mind that manufacturers or sales outlets are frequently not aware of all the parameters of an installation, and the interpretation of catalogue data has many pitfalls.

Errors in application stem mainly from a lack of understanding

of the requirement and a tendency to buy at the lowest price without the protection of a clear specification. Once the application is fully understood and assessed, a specification needs to be drawn up, and the possibility of error and dispute is reduced (see also Chapter 32).

13.7 Testing of packaged units

Manufacturers' test procedures for packaged units take several forms:

1. Rating test data on the prototype, which forms the basis for the published capacity and application leaflets.
2. Rating check tests on a proportion of production units, to verify that standards are being maintained.
3. Function tests on all production units, to verify correct operation of components.
4. A short running test at normal conditions to check for reliability of operation plus, possibly, an approximate capacity check. This will not be possible on some types.
5. Safety tests at maximum operating conditions – usual on all or a high proportion of production units.

These test schedules are costly, requiring expensive equipment, and are reflected in the anticipated high quality.

Factory records will be kept of all such tests and, in the case of larger units, manufacturers will, if asked, provide a certified copy of the test on the equipment supplied.

13.8 Mobile application units

The requirement for transport air-conditioning or refrigeration is for an air cooling, air-cooled unit with reliable availability of service.

For long-distance travel, the prime mover is usually a built-in petrol or diesel motor, driving the compressor through belts and a clutch. An electric motor may also be provided which can be connected to mains supply when the vehicle is not moving. Other prime movers used are direct drives from the vehicle engine or indirect drives through hydraulic piping. The evaporator fan is usually electric for convenience, running off a 24 V d.c. feed with an auxiliary dynamo on the cooling unit (see Figure 13.5).

Most of these methods allow essential maintenance and repairs to be carried out under workshop conditions without taking the vehicle off the road, providing a spare unit is available. A dominant feature of this market has become the wide availability of spares and service wherever such vehicles may go.

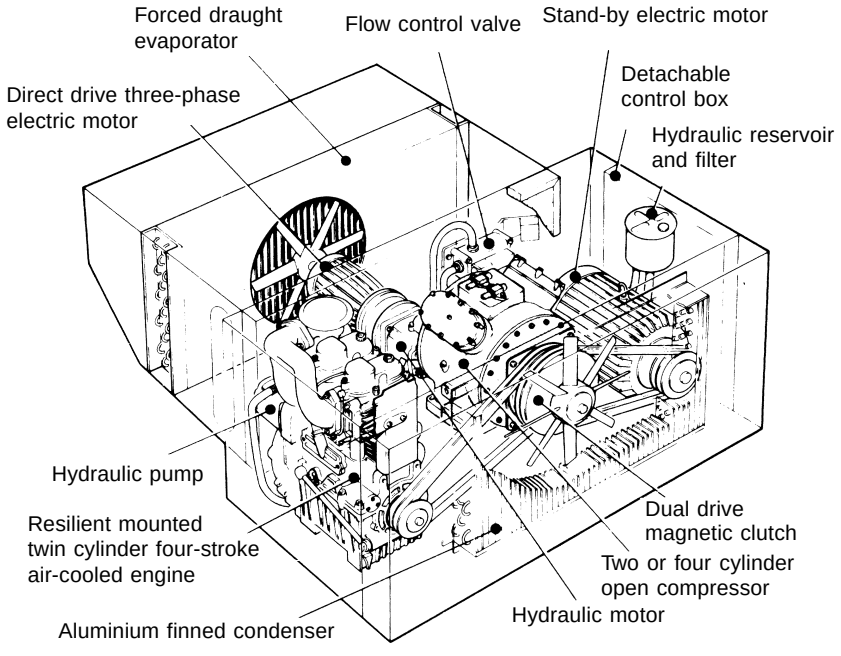


Figure 13.5 *Packaged vehicle cooler (Courtesy of Transfrig Ltd)*

13.9 Other packages

A very large variety of self-contained refrigeration and air-conditioning packages are made, mainly for the consumer durable market and small domestic applications. They include:

1. The domestic refrigerator and freezer.
2. Ice-cream conservators.
3. Retail display cold and freezer cabinets and counters.
4. Cooling trays for bottles (beer, soft drinks, wines).
5. Instantaneous draught beer coolers. These usually comprise a tank of constantly chilled water, through which the beverage flows in stainless steel piping.
6. Ice makers – cubes and flakes.
7. Cooled vending machines.
8. Soft ice-cream freezers.
9. Dehumidifiers, in which air is passed first over the evaporator to remove moisture, and then over the condenser to re-heat and lower the humidity (see Figure 29.1).
10. Drinking water chillers.

With the advantages of factory-built packaged cooling devices, this list cannot be exhaustive.

14 Refrigeration of foods. Cold storage practice

14.1 Principles of cooling for preservation

A major use of refrigeration is in the preservation, storage and distribution of perishable foods. Although the use of low temperatures for this purpose has been known and practised for many thousands of years, it was not until the last century that Pasteur and others determined the bacteriological nature of food spoilage and the beneficial effect of cooling. An immediate advantage was to make it possible to provide the extra food required by the growing urban populations. A large international trade has built up, starting with the transport of frozen meats to Europe in 1873 and 1876.

The edibility of foodstuffs is prolonged by lowering the temperature, since this slows chemical reactions and breakdown by bacteria. Some products can be frozen, and when they are in the solid state all movement in the individual cells will cease, inhibiting further reactions.

The decision whether to just chill or to freeze solid depends on the type of product and the length of time it must be stored. Freezing results in some structural change, since ice crystals are formed inside the cells, and the final foodstuff may be of different texture when thawed out.

As a general rule, foods which are not to be frozen are handled and stored at a temperature just above their freezing point, providing this does no damage (exceptions are fruits such as bananas and lemons). Produce which is to be frozen must be taken down below the freezing point of the constituents. Since foodstuffs contain salts and sugars, the freezing process will continue down to –18°C and lower.

A distinction must be drawn between the cooling process and subsequent storage. Careful control of temperature and humidity is

needed when cooling warm produce, or there may be serious losses in weight and quality. Considerable research has been carried out to find optimum methods for different foodstuffs, especially meats, for cooling and for short-term and long-term storage.

14.2 Pre-storage treatment

Cooling and freezing cannot improve a product, and the best that can be achieved is to keep it near to the condition in which it entered the cooling process. This means that only the best produce should be used, and this should be as fresh as possible. (This general principle must, of course, be interpreted in the light of local conditions and needs. In some countries of the world, preservation in cold stores is essential to prevent wastage, regardless of the quality of the crop.)

All foods must be clean on entry. Some, such as fish, leaf vegetables and some fruits, may be washed and left wet. Fish will tend to dry out and lose its fresh appearance, so it is packed wet or given a sprinkling of ice chips to keep the surface moist.

Other products, especially the meats, must be dry, or bacteria will live on the moisture and make the skin slimy.

Potatoes will start to sprout after a long period in storage. This can be checked by spraying the freshly lifted tubers with a chemical sprout depressant.

Certain fruits, notably grapes and dates, may have some surface contamination or infestation when first picked, and they are fumigated with sulphur dioxide or some other gas. They must, of course, then be thoroughly ventilated before going into storage.

The techniques of this processing will be known to the user or can be found in sources from the particular branch of the food industry [36, 37].

Handling conditions must be hygienic. Some types of food, such as milk, can be kept sealed within the processing system. If the food will be exposed to the air during handling, the conditions of the surrounding air – in terms of temperature, humidity and cleanliness – must be the best that can be maintained. This is especially the case with fresh meats.

14.3 Pre-cooling

If warm produce is taken into a cold store, moisture will evaporate from its surface and may condense on the cold produce already there. This will be of no consequence with wet products such as fish and leaf vegetables but cannot be permitted with meat or poultry.

For these meats, pre-cooling is carried out in a separate room under controlled conditions so that the product is reduced to near-final storage temperature, the surface remaining dry all the time [37].

A lot of pre-cooling can be achieved by allowing produce to stand in ambient air, especially at night. For example, apples and pears picked in the daytime at 25°C may cool down to 12°C by the following morning, halving the final refrigerated cooling load.

Wet products can be pre-cooled in chilled water, or by the addition of flake ice. Ice is also used with fish and leaf vegetables to help maintain freshness in transit to storage. Leaf vegetables can be cooled by placing them in a vacuum chamber and so evaporating surface water at low pressure.

14.4 Freezing

Most products will keep longer and fresher in the frozen condition, and this process is used for those whose sale value will support the extra process cost.

The cells of animal and vegetable products contain a watery solution of salts and sugars. If this solution starts to freeze, surplus water will freeze out until the eutectic mixture is reached (see Section 13.2). If freezing is not carried out quickly, these ice crystals will grow and pierce the cell walls; then when the product thaws out, the cells will leak and the texture will be spoiled. This is of no great consequence with the meats, whose texture is changed by cooking, but will not be suitable for fresh fruits or vegetables.

As a general rule, any product which will be eaten without cooking, or only very brief cooking (such as green peas), should be quick-frozen in a blast-freezing tunnel or similar device. Other foodstuffs need not be frozen so quickly, and may be left in a coldroom at a suitable temperature until frozen.

Frozen confections (ice-cream and ice-lollies) rely on speed of freezing to obtain a certain consistency and texture, and they require special treatment (see Chapter 17).

Once a product has been frozen, it must never be allowed to warm and then be re-cooled, or partial thawing may take place with slow re-freezing.

14.5 Packing and handling

Cold storage packing must contain and protect the product, while allowing the passage of cooling air to keep an even temperature.

Packages generally will be small enough to be lifted by hand if required, and of a suitable shape to be stacked on pallets for

mechanical handling by fork-lift trucks. Stacking on a pallet should allow the passage of air between the individual packs.

Fruits and vegetables which give off heat of respiration need to have perforated cases so that air may pass through the product.

Carcase meat does not lend itself to regular packages and, in any case, needs to be out of contact with other surfaces, including other carcases, or slime may form. Carcases are hung from overhead rails on roller hooks so that they can travel along the rail system (see Figure 14.1). Special cage pallets are also used for carcase meat.

Figure 14.1 *Meat store with rails at Baxters (Butchers) Ltd (Courtesy of Gordon-Johnson Ltd)*

Potatoes are cold stored in bulk or in large boxes of $\frac{1}{2}$ or 1 t capacity (also in sacks in some countries). They are always stored on their own, so special handling methods have evolved.

Pallets are now mostly standardized at 1.2×1.0 m and the total weight will be between $\frac{1}{2}$ and $1\frac{1}{2}$ t, depending on the product. Handling in small cold stores can be by hand pallet trucks or hand-steered electric drive trucks. These can transport but not lift one pallet onto another. The usual fork-lift truck is a ride-on vehicle, electric driven, and can lift to form a stack of two, three, four or even five pallets high, according to the length of the telescopic lift and the skill of the driver.

Methods of arranging the product in the store will depend on the number of varieties and the storage life (see Figure 14.2). With short-term storage it may be necessary to get to any pallet, so access gangways will be required with only one row of pallets on each side.

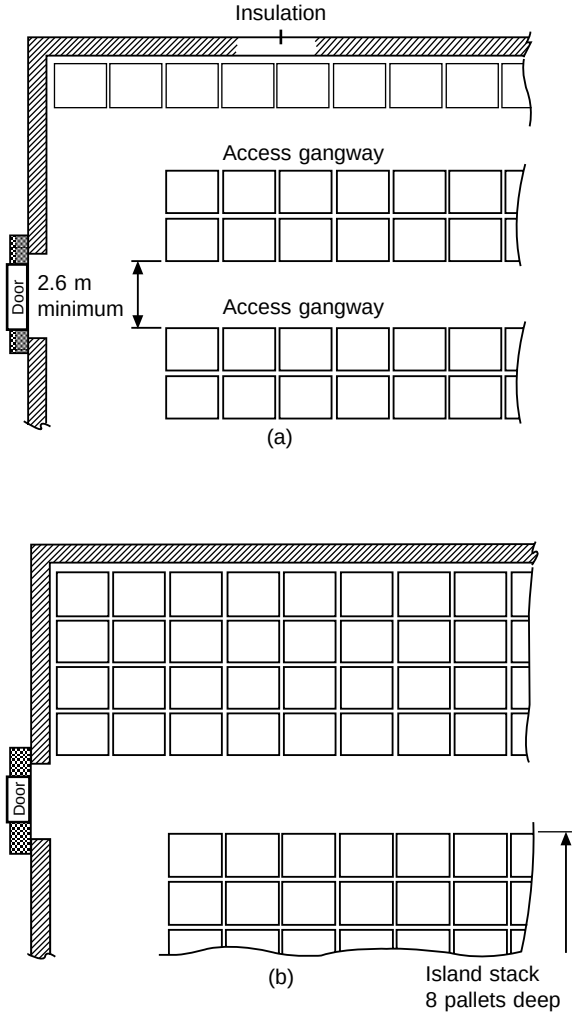


Figure 14.2 *Pallet storage. (a) Full access. (b) Restricted access*

The gangway width for a high-lift truck is at least 2.6 m and may be as much as 3.7 m, so some 55% of the available floor space will be gangways.

Some sacrifice of perfect accessibility is usually made in the interest of economy. Where storage life is long, pallets may be stacked as many as four rows deep, requiring one gangway for eight rows of pallets. In this case, a gangway of 3.7 m is required to shuffle pallets to get to those at the rear, and the usable floor space comes up to 68%.

Racking can be installed to support the pallets above floor level,

and permit a pallet to be removed without disturbing those above it. Alternatively, *post-pallets* having corner pillars to support upper pallets provide a firm stacking method. The height per pallet is about 1.4 m. (See Figure 14.3.)

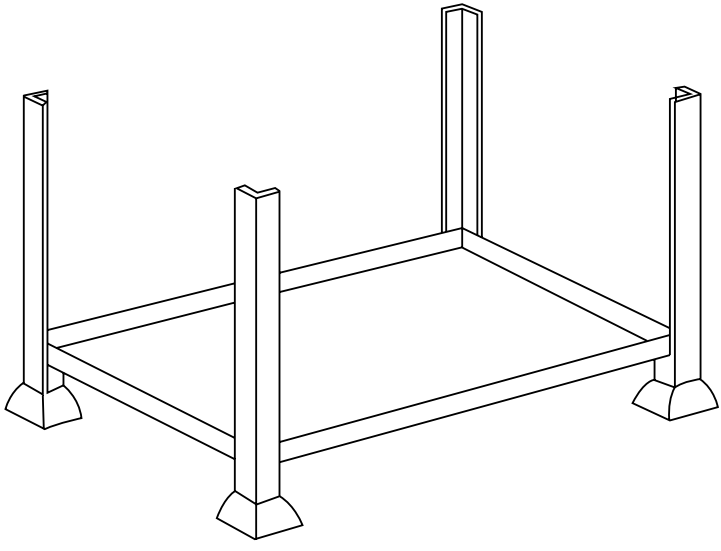


Figure 14.3 *Post-pallet*

Meat pallets, for hanging carcases and sides, will have about the same floor area but will be up to 2 m high, with cage sides.

Where the product is in rigid boxes or cartons, it is possible to stack pallets up to three high without auxiliary support, i.e. one on the other.

Fork-lift drivers need to be skilled, experienced, and safety-conscious, since the misplacement of a pallet on a high stack can present a serious hazard. They work in well-insulated protective clothing and in short spells within a shift if the room is much below 0°C. Fork-lift trucks are available with enclosed, heated driver's cabs. Extra staff are required if the traffic is constant. Large stores will have a wide loading platform outside at floor level to permit fork-lift trucks to manoeuvre their loads onto vehicles.

Stock control must present a clear picture of the contents and location at any time, to ensure correct rotation of stocks.

14.6 Grouping of products

Most cold storage installations will have a wide variety of products to hold, with several different types in each chamber. Apart from

obvious separation by storage temperature, some foodstuffs are not compatible with others, e.g.:

1. Wet fish will impart its smell to butter, cheese, eggs and fresh meat.
2. Citrus fruits such as lemons do the same.
3. Onions must not be stored too long at the high humidity of other vegetables, or they will rot.
4. Frozen meat for medium-term storage at – 10°C is the only product kept at this temperature.
5. Frozen foods below – 20°C may be mixed with impunity.

14.7 Storage conditions

Table 14.1 shows recommended storage conditions for a few basic products. Comprehensive tables will be found in standard works of reference [36, 37]. In the event of a product being encountered which cannot be found in general references, information can often be obtained from agricultural and other research establishments.

It will be noted that fruits and vegetables, with the exception of dried fruit and onions, are stored in high humidity to prevent drying out through the skin. Meats generally must be in drier air, or slime may form with the growth of bacteria.

14.8 Post-storage operations

As a general principle, products leaving cold storage for ultimate consumption may be allowed to rise slightly in temperature but, if so, must not be re-cooled. It follows that they should be kept at the storage temperature as long as possible down the chain of delivery. This requires prompt handling and the use of cooled vehicles up to the final retail outlet.

Some products require special treatment, for which provision should be made, e.g.:

1. Frozen meat coming out of long-term storage to be sold chilled must be thawed out under controlled conditions. This is usually carried out by the retail butcher, who will hang the carcase in a chill room (–1°C) for two or three days. On a large scale, thawing rooms use warmed air at a temperature below 10°C.
2. Potatoes and onions coming out of storage will collect condensation from the ambient air and must be left to dry or they will rot.
3. Fruits of various sorts are imported in a semi-green state and must be ripened off under the right conditions for sale.
4. Some cheeses are frozen before they have matured. On thawing out for final distribution and sale, they need to mature.

Table 14.1 Storage conditions for foodstuffs

Products	*Temperature*	*Humidity*	*Life*
Apples	1–4*	85–90	2–8 months
Bananas, green	12–14	90	10–20 days
ripe	14–16	90	5–10 days
Beer, barrel	2–12	65	3–6 months
Cabbage	0–1	95	3–5 weeks
Carrots, young	0–1	95	1–2 months
old	0–1	95	5–8 months
Celery	0–1	95	1–2 months
Cucumber	10–12	90–95	10–14 days
Dairy products, milk	0–1	–	2–4 months
cream	– 23–(– 28)	–	6–12 months
cheese	1–4	65–70	6–18 months
yoghurt			
Dried fruits	0–1	Low	6 months up
Eggs, shell	– 1–0	80–85	5–6 months
Fish, wet	1–2	90–95	5–15 days
Fruit soft (berries)	0–1	90–95	5–7 days
Grapefruit	10–14	85–90	4–6 weeks
Grapes	0–1	90–95	2–5 months
Lemons, green	14–15	85–90	1–6 months
Lettuce	0–1	90–95	1–2 weeks
Meats, bacon	1–4	85	1–3 months
beef	– 1–(+ 1)	85–90	1–6 weeks
ham, fresh	0–1	85–90	7–14 days
lamb, mutton	0–1	85–90	5–14 days
pork, fresh	0–1	85–90	3–7 days
poultry	– 1–0	85–90	1 week
frozen	– 12	90–95	2–8 months
frozen	– 18	–	4–12 months
Melons	4–10†	85–90	1–4 weeks
Mushrooms	0	90	1–4 days
Onions	0–1	65–70	1–8 months
Oranges	0–9†	85–90	3–12 weeks
Pears	– 1–(+ 1)*	90–95	2–6 months
Pineapples	7–10	90	2–4 weeks
Plums	0–1	85–90	2–8 weeks
Potatoes, new crop	10–12	85–95	3–6 weeks
main crop	1–3	90–95	6–10 months
Tomatoes, green	12–15	85–90	3–5 weeks
ripe	10	85–90	8–12 days
Wine unfortified	8–10	–	Indefinite

*See also Section 18.2.
†Depending on variety, harvest time and other factors.

15 Cold store construction

15.1 Size and shape

The purpose of a cold store is to provide an insulated and refrigerated enclosure suitable for the handling and storage of perishable goods, at some predetermined temperature.

The shape and size of the cold store will depend on:

1. The quantity and variety of goods to be stored
2. The size and shape of the packaging
3. Proposed methods of handling
4. Storage times

Since the handling methods require power-driven fork-lift trucks, nearly all cold stores are now erected as single-storey buildings with the insulation in the form of factory-made panels. The main loading bay will probably be at tailboard height, so that products can be taken directly in and out of insulated vehicles.

Example 15.1 What will be the internal dimensions of a cold room to store 900 t of boxed frozen meat if the box size is $700 \times 450 \times 150$ mm and the net weight 30 kg?

$$\text{Number of boxes in store} = \frac{900\,000}{30} = 30\,000 \text{ boxes}$$

These can be stacked on pallets 1200×1000 mm, the height per pallet being 1400 mm, say a box height of 1250 mm, allowing for the base of the pallet. Boxes would be loaded flat, three per layer and eight layers high, making 24 boxes per pallet:

$$\text{Number of pallets} = \frac{30\,000}{24} = 1250 \text{ pallets}$$

These pallets can be stacked three high without auxiliary support, and this will be the cheapest and most flexible arrangement, leaving

the floor clear for any other type of produce if there is less than a full load in the store:

$$\text{Floor space required, pallets} = \frac{1250}{3} = 420$$

The net area will be 420×1.2 m^2, plus a space of some 75 mm between pallets to allow clearance in handling, giving an occupied floor area of about 575 m^2.

Since it is a one-product store, it seems unlikely that any one pallet must be accessible, so stacking the pallets in rows three deep should be acceptable.

One or two sketch layouts can be tried based on these figures, unless the shape of the store is already decided by the available site. One such layout is shown in Figure 15.1. It will hold a maximum of 1278 pallets and requires a floor area 34.5 m long by 27 m wide, a gross floor area of 932 m^2. The stacks will be 4.05 m high. To this must be added a clearance for lifting and the depth of the evaporators. If the latter are 1.1 m, the store internal height will be 5.25 m. The volume is 4890 m^3 and the storage density is 5.4 m^3/t or 184 kg/m^3.

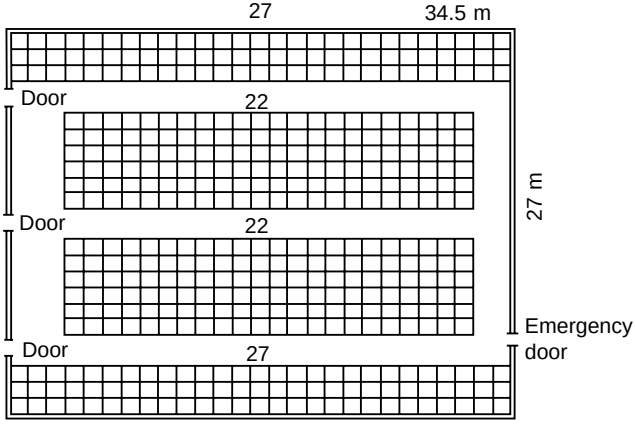


Figure 15.1 *Schematic layout of 900-t pallet store*

Example 15.2 What would be the volume of this store if the pallets were arranged on racks?

The solution to this would require constructional details of the proposed racking system and each pallet will now require an area of about 1.400×1.075 m. Height clearance must be allowed for each tier – a total of 1.600 m each pallet high. Also, it is not possible for a fork-lift truck to reach three pallets deep into a stack, so the arrangement would now be two deep along the sides and four-deep

islands. However, with racking, the pallets could be stacked four high and possibly five.

A provisional arrangement would possibly be with 30 pallets along the walls and two islands 24 pallets long. The internal dimensions will now be approximately as follows:

$$\text{Length, } 30 \times 1.400 = 42 \text{ m}$$

$$\text{Width, } 12 \times 1.075 + (3 \times 3) = 22 \text{ m}$$

$$\text{Floor area} = 924 \text{ m}^2$$

$$\text{Height, } (4 \times 1.600) + 1.2 = 7.6 \text{ m}$$

$$\text{Volume} = 7022 \text{ m}^3$$

$$\text{Density, } 900\,000/7022 = 130 \text{ kg/m}^3$$

A store of the same capacity, designed to give immediate access to any one pallet, would have a floor area of 1260 m^2 and a volume of 9600 m^3, giving a storage density of 94 kg/m^3.

Cold stores intended only for carcase meat will have the product hung from an overhead rail system (see Figure 15.2). The meat is hung on hooks on roller carriers – possibly one side of beef on a single hook or smaller carcases and cuts on multiple-hook carriers.

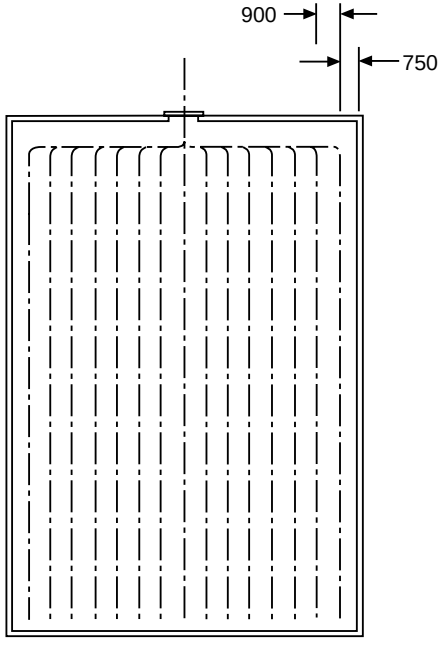


Figure 15.2 *Schematic plan of rail layout*

Rails for beef sides will be spaced 900–1200 mm apart, and the length for a side is 450 mm of rail. The height to the top of the rail may be up to 3.35 m, but may be less. The average side weight in the UK is 140 kg, with a variation of 80–225 kg throughout the world; the local trend should be taken as a guide. Space is required for the service rails and junctions at the door end, possibly 1 m being taken up by this. Rails should be well clear of walls so that air can circulate and the meat cannot touch the wall.

Example 15.3 What will be the floor area of a railed coldroom to take 500 head of beef in sides?

Number of sides = $500 \times 2 = 1000$

Rail length required = 1000×0.450 m = 450 m

A possible arrangement would be 15 rails, each 30 m long.

Width, say 1 m spacing, $15 \times 1 = 15$ m

Extra wall clearance on each long side = 0.5 m

Overall = 15.5 m

Length, 30 m + clearance for service rails = 32 m

Giving a floor area of $32 \times 15.5 = 496$ m^2

A room as large as this may require an access door at each end, with rail points and an exit rail, adding another 1 m to the length. This would be decided by the amount of traffic in and out and the direction, if part of a factory production line.

Small stores for miscellaneous products must allow for random stacking of a variety of different packages. The storage density may be between 100 and 450 kg/m^3 when the store is fully loaded, allowing for access passages. A closer estimate could be made if probable loadings and package shapes were known [29, 33].

More than one chamber may be required, based on the separation of products by type and by storage temperature. These will be sized on the probable individual contents. Where a low-temperature room is required as well as some at higher temperatures, it should be placed between them, to reduce heat gains.

Old stores, especially those occupying valuable land in city centres, were built several stories high. Access was by lifts and most of the handling was by hand or with hand pallet trucks. Such stores are occasionally built today, but the general use of mechanical handling has led to the single-storey building close by a main road but away from a city centre.

The majority of transport of frozen foods is by 12-m-long articulated

trailers. Access, turning, docking and parking space is needed for such vehicles and the loading dock should be at the tailboard height, with adjustable ramps to allow for small differences in this. The loading platform usually runs across the full side or end of the store with doors opening onto it. The absolute minimum width is 3 m and many docks are as wide as 12 m. The check-in office will be on the dock and may have a weighbridge or rail scale for carcases. The refrigeration machine room should have separate access.

15.2 Insulation

The purpose of insulation is to reduce heat transfer from the warmer ambient to the store interior. Many different materials have been used for this purpose but most construction is now with the following:

1. Cork, a natural material – the bark of the Mediterranean cork oak tree. It is largely air cells and the fibrous cell walls have a high resin content. When baked, the resin softens and welds the pieces of bark into a comparatively homogeneous mass, which is sliced into blocks, commonly 50, 75 and 100 mm thick.
2. Expanded polystyrene. The plastic is formed into beads containing an expanding agent. When placed in a mould and heated they swell and stick together. The blocks are then cut into thicknesses as required.
3. Foamed polyurethane. The basic chemicals are mixed in the liquid state with foaming agents, and swell into a low-density foam which sets by polymerization into a rigid mass. As the swelling material will expand into any shape required, it is ideal for the core of sandwich panels, and the sheet material skins may be flat or profiled. When the panels are manufactured the mixture is injected between the inner and outer skins and expands to the thickness required, adhering to the lining materials.

The value of an insulant to reduce heat flow is expressed as resistivity or its reciprocal conductivity. The units of the latter are watts per metre kelvin (W/(m K)). Values for these materials used are approximately as follows:

Corkboard 0.04 W/(m K)

Expanded polystyrene 0.034 W/(m K)

Foamed polyurethane 0.026 W/(m K)

Example 15.4 What is the heat conduction through a panel of foamed polyurethane 125 mm thick, 46.75 m long and 6 m high if the inside temperature is – 25°C and the ambient is 27°C?

$$\text{Area} = 46.75 \times 6 = 280.5 \text{ m}^2$$

$$\Delta T = 27 - (-25) = 52 \text{ K}$$

$$Q = 280.5 \times 52 \times \left(\frac{1}{0.125}\right) \times 0.026 = 3034 \text{ W}$$

This assumes that a wall of that size could be made of an unbroken sheet of the insulant. Since there will be some structural breaks, an allowance of some 5% should be added, making the leakage 3.2 kW.

Insulation thicknesses used are 50, 75, 100, 125 and 150 mm, but insulants can be obtained in non-standard thicknesses for special applications. A general guide to determine the possible thickness for a required temperature difference is to design for a conductance of 9 W/m^2. This gives the values in Table 15.1.

There will be exceptions to this rule, such as thicker insulation where electric power is expensive, or thinner insulation for a chamber only used infrequently. Ceiling panels may be thicker to give added structural strength. In cases of doubt, the insulation costs must be resolved as the optimum owning cost.

Table 15.1

	Corkboard	*Expanded polystyrene*	*Foamed polyurethane*
50 mm	11 K	13 K	17 K
75 mm	16 K	19 K	25 K
100 mm	22 K	25 K	33 K
125 mm	27 K	32 K	42 K
150 mm	32 K	38 K upwards	50 K upwards
200 mm	43 K upwards		

In most cases, the insulation will be the greatest resistance to heat flow and other materials in the construction and surface resistances are ignored in estimating heat gains through cold store walls, ceilings and floors.

Conductivity figures for other materials will be found in standard references [2].

15.3 Vapour barriers

When the evaporator begins to cool a cold store, surplus moisture in the air in the room will condense on the coil and, if cold enough, will freeze. This will continue until the water vapour pressure inside

the room approaches the saturation pressure at the coil fin temperature, e.g. with a coil temperature of – 20°C the vapour pressure would be 0.001 bar. Since this is lower than the vapour pressure of the ambient air, water vapour will try to diffuse from the hot side to the cold, through the wall (see Figure 15.3). At the same time, heat is passing through the wall, and the temperature at any point within the insulation will be proportional to the distance through it.

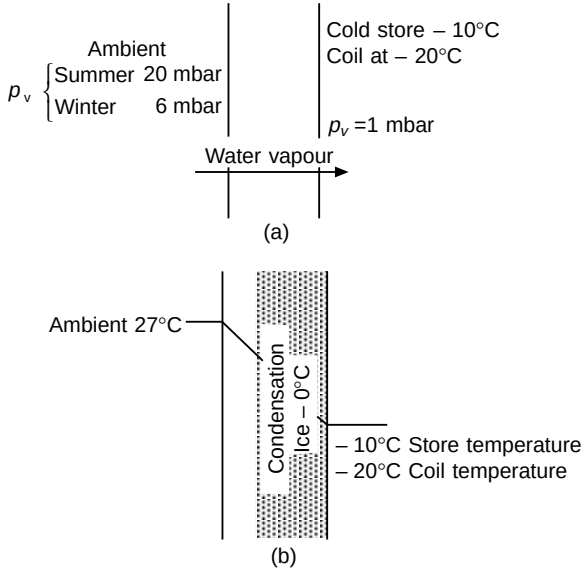


Figure 15.3 *Section through coldroom insulation. (a) Vapour diffusion. (b) Thermal gradient*

At some point through the wall, the temperature will be equal to the saturation temperature of any water vapour passing through it, and this vapour will condense into liquid water within the insulation. This process will continue and the water will travel inwards until it reaches that part of the insulation where the temperature is 0°C, where it will freeze. The effect of water is to fill the air spaces in the material and increase its conductivity. Ice, if formed, will expand and split the insulant.

To prevent this deterioration of the insulation, a vapour barrier is required across the warm face. This must be continuous and offer the best possible barrier to the transmission of water vapour. The traditional vapour barrier was bituminous emulsion or hot bitumen, applied in two or more layers. More recent materials are heavy-gauge polythene sheet, metal foil and metal sheet. It is sometimes

thought that the plastic insulants, since they do not easily absorb moisture, are vapour barriers. This is not so, and no reliance should be placed on the small resistance to vapour transmission which they may have.

Any small amount of vapour which might enter through faults in the vapour barrier should be encouraged to pass through the inner (cold side) skin of the structure to the coil, rather than be trapped within the insulation. It follows that, if the vapour barrier is at all suspect, the inner wall coating should be more porous. In traditional construction, this was provided by an inner lining of cement plaster or asbestos cement sheet, both of which transmit vapour. The modern use of impervious materials on both skins requires meticulous attention to the sealing of any joints.

Great care must be exercised at wall-to-floor junctions and all changes of direction of walls and ceilings. In the case of a wall-to-floor junction, this will often occur at two dissimilar types of construction, i.e. preformed wall panels to *in situ* floor insulation. A satisfactory continuous vapour barrier needs careful design.

Any conductive material, such as masonry and metal structural members or refrigerant pipes, which must pass through the insulation, will conduct heat, and the outer part may become cold enough to collect condensation and ice. Such heat bridges must be insulated for some distance, either inside or outside the main skin, to prevent this happening. If outside, the vapour barrier must, of course, be continuous with the main skin vapour barrier.

15.4 Sectional coldrooms

Small coldrooms can be made as a series of interlocking and fitting sections, for assembly on site on a flat floor (see Figure 15.4). Standard ranges are made up to about 70 m^3, but larger stores can be made on this principle. The floor section(s) is placed on a flat floor and the sides erected on this, located, sealed and pulled up together. The roof sections then bridge across the walls. Such packages are supplied complete with all fittings. They can be dismantled and moved to another location if required. Specialist site work is restricted to cutting necessary holes for pipework and fitting the cooling equipment.

Stores of this size can be built, using standard size factory-made sandwich panels, cutting these to size, jointing and sealing on site. This form of construction is prone to fitting errors, with subsequent failure of the insulation, if not carried out by skilled and experienced craftsmen. The best system can be ruined if the base is uneven or by inexpert finishing of pipe entries, sealing, etc.

Figure 15.4 *Assembly of section coldroom (Courtesy of Hemsec (Construction) Ltd)*

15.5 Inbuilt construction

Traditional cold store construction was to build an insulated lining within a masonry shell. The outer skin would be erected in brick and concrete, and rendered as smooth as possible inside with cement plaster, to take the insulation. When the surface was dry, it would have several coats of bitumen applied as a vapour barrier and slabs of insulation material stuck to this with hot bitumen. This was normally carried out in two or more layers so that joints did not pass right through the insulant, but were staggered. The inner skin would be finished with cement plaster, reinforced with wire mesh. The usual insulant was slab cork.

Any columns passing through coldrooms would be insulated, at least partially, to reduce conduction along the heat bridge and the build-up of condensation and ice. Floors would have a layer of hard concrete on the floor insulation. Ceilings were stuck to a concrete ceiling or fixed to a false timber ceiling.

This form of construction is seen to be quite sound, and there are still many such stores in service which were built 50 and more years ago. The method is still used in countries where cork is cheap and craft labour available at an economic price.

15.6 Factory panel systems

The plastic insulants are rigid, homogeneous materials, suitable as the core of sandwich panels. Such a method of fabrication is facilitated when using foamed rigid polyurethane, since the liquids can be made to foam between the inner and outer panel skins and have a good natural adhesion, so making a stiff structural component [40].

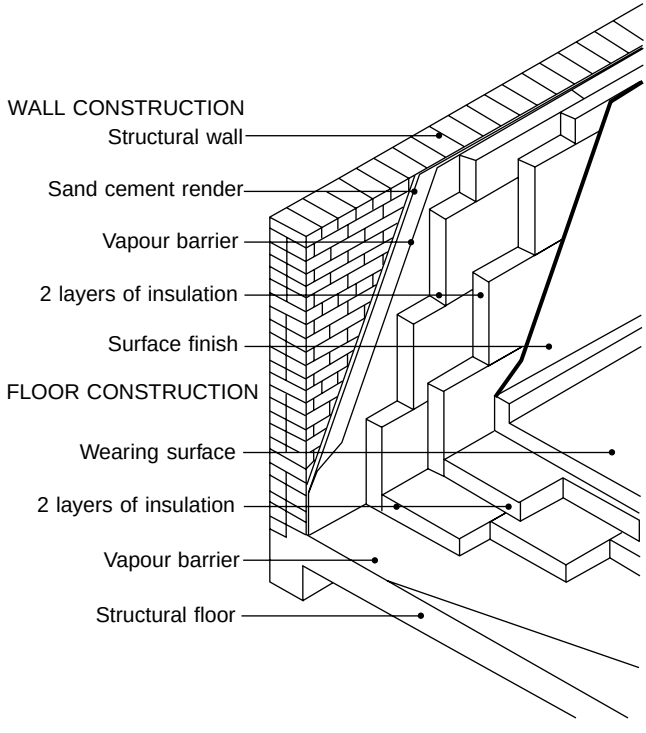


Figure 15.5 *Inbuilt coldroom (Courtesy of F. A. Wallis)*

Panels made in this way for cold store and other structures are usually 1.2 m wide and can be made in lengths of up to a maximum of about 14 m. The manufacture incorporates interlocking edging pieces and other fittings (see Figure 15.6). Such panels are used for walls and ceilings, although not for floors above a certain store size. The inner and outer skins are of aluminium or rustproofed steel sheet, usually finished white, and may be flat or profiled. The edge seals are plastic extrusions or similar material. The panel edge locking devices may be built in or applied on site. To build such a store, the floor is first prepared (see Section 15.7), bringing the vapour barrier up at the outer face. Wall sections are erected on end on the edge

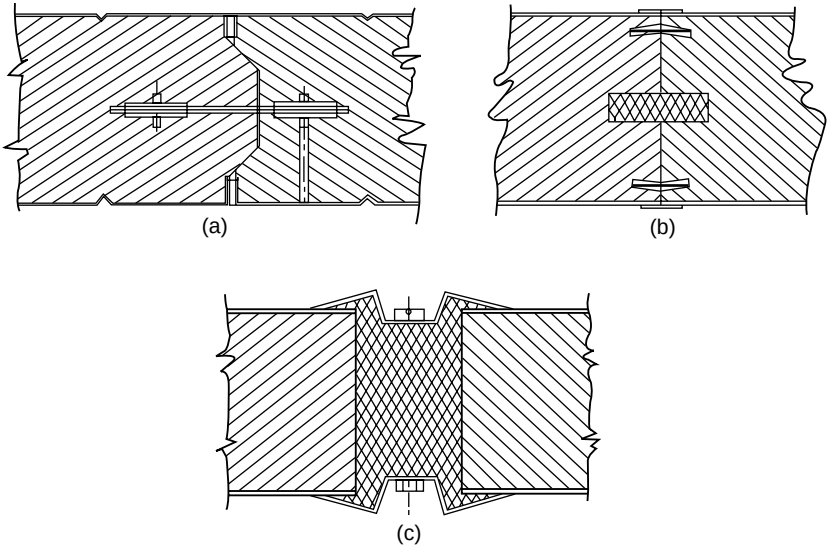


Figure 15.6 *Typical wall panel mounting systems. (a) Hemsec. (b) Isowall (O'Gorman-BTC). (c) Cape*

of the floor and locked together, making the interpanel seal at edges and corners. Ceiling panels are fitted over the tops of the walls and sealed at the warm face of the junction.

Since the panels must be rigid enough to support their own weight, thickness cannot be reduced below a minimum, and this is usually 100 mm, although less insulation might suffice for the purpose. For a large store, panels will be 125 or possibly 150 mm thick.

The insulation panels are normally erected within a frame building so that panel joints are protected from the weather. Long vertical panels can be additionally braced to the structure. It is possible,

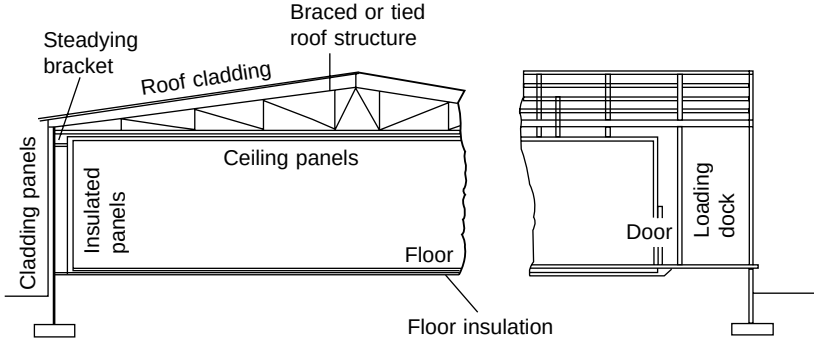


Figure 15.7 *Panel construction*

with suitable construction and finishes, to erect the insulation panels around an internal supporting framework.

Care must be taken regarding the method of supporting ceiling panels. Large portal framed steel buldings may provide a cheap outer shell but do have a considerable amount of roof movement. Panels hung from this type of structure can be subjected to movement which cannot be tolerated in cold store construction. A tied portal, however, can be acceptable [38]. The outer shell may also be required to bear the weight of the evaporators and, in the case of stores for carcase meats, the rails and the product itself.

15.7 Floors

Heavy floor loadings and the use of ride-on electric trucks demand a strong, hard-working floor surface, which must be within the insulation envelope.

Floor construction starts with a firm concrete foundation slab about 200–250 mm below the final floor level. This is covered with the vapour barrier, probably of overlapping layers of heavy-gauge polythene sheet. On this is placed the insulation board in two layers with staggered joints; this is fitted as tightly as possible. The upper joints may be covered with strips of plastic to prevent concrete running in, but a continuous layer of vapour-tight sheet must not be used on this cold side of the insulation. The concrete floor is made with granite aggregate, laid to the final level, as dry as possible, reinforced with steel mesh and in panels not more than 10 m square, to allow for contraction on cooling. Where fork-lift trucks are in use, it is best to lay these panels with no gap, to minimize cracking of the edges under load. If the floor will be wet in use, a finite gap is left, and filled with mastic to prevent water getting into the insulation.

The need for good design and expert installation of floor finishes cannot be emphasized too strongly. The floor receives the greatest wear of all the inner linings, and once the temperature has been reduced in the store, it will usually remain low for the rest of its life. Repairs are therefore very difficult.

Where a store is to take post-pallets, or will have internal racking to store pallets, careful calculation is necessary of the load on the feet. They can have a considerable point load, having the effect of punching a hole through the floor finish.

15.8 Frost-heave

It floors are laid on wet ground, the vapour pressure gradient (Figure

15.3) will force water vapour up towards the vapour seal. Given a ground temperature of 13°C in the UK, the underside slab may become as cold as 0°C after many months of store operation, and any moisture condensed under the floor insulation will freeze and, in freezing, expand. In time this layer of ice under the floor slab, unable to expand downwards, will lift the floor (frost-heave).

Frost-heave is prevented by supplying low-intensity heat to the underside of the insulation, to keep it above freezing point. This may take several forms:

1. Low-voltage electric resistance heater cables fixed to the structural floor slab and then protected within a 50 mm thickness of cement and sand to give a suitable surface on which the floor vapour barrier can be laid. The heating is thermostatically controlled, and it is usual to include a distance reading or recording thermometer to give visual indication of the temperature of the floor at several locations below the insulation.
2. Pipes buried in the structural slab. These are connected to delivery and return headers, and glycol circulated. This is heated by waste heat from the refrigeration plant. Steel pipe should not be used under the floor unless protected against corrosion.
3. Air vent pipes to allow a current of ambient air through the ground under the base slab. This is not very suitable in cold climates.
4. On very damp ground or where the finished floor level is in line with the deck of transport vehicles, the cold store floor can be raised above the existing ground level. This is done by building dwarf walls or extending the length of the piles, if these are used, to support a suspended floor at the required height. This leaves an air void of some 1 m under the cold store, through which air can naturally circulate.

15.9 Door and safety exits

Cold store doors must combine the functions of door and insulation. Small doors will be hinged and have an arrangement of double gaskets to reduce the transmission of convected heat (air leakage) and consequent ice accumulations at the door edges. Such doors are normally wood-framed to reduce conduction, but may now have plastic moulded frames. Insulation is by one of the foam plastics, and the face panels are sheet metal or GRP. In order to keep the seals in good alignment throughout the life of the door, hinges will be made adjustable. The closing latch will have a cam or lever action to compress the large gasket area and give a tight seal.

Where a flush door sill is required, the gaskets on the lower edge will be in the form of two or three flexible blades which just brush the floor.

A simpler and more adaptable method of sealing is a face-fitting or overlap door (Figure 15.8). The door itself overlaps the opening by some 150 mm all round, and two or three soft gaskets seal the overlapping surfaces. This type of door is general in rooms operating below 0°C, and may have warming tapes embedded in the wall face to prevent freezing of any vapour which penetrates. The smaller sizes, and the rebated doors, are hand operated.

Larger doors, especially those to take fork-lift trucks, must be mechanically operated for speed and convenience, and because the doors should never be left open too long. For most purposes, horizontal sliding doors are used, closing onto face gaskets in the same way as the overlap doors. The slide system is generally arranged so that the door moves out from the wall during the first part of its travel, so as to free the gaskets and make for easier sliding. Various electric and pneumatic mechanisms are used, and the switches for opening and closing are controlled by toggle ropes hanging down where the fork-lift driver can reach them without dismounting, or by electronic sensors. Protection posts each side reduce the risk of damage to the door frame or wall if the truck collides with them.

All mechanical doors are required by law to be capable of hand operation in the event of power failure, and doors of all types must have fastenings which can be opened from either side in case an operator is shut in the store. Larger rooms must have an escape door or breakout hatch or panel at the end remote from the doors, for use in an emergency. Door openings are frequently fitted, additionally, with plastic strip curtains or doors, to reduce infiltration when the main door is open.

15.10 Interior finish and fittings

The interior surface finish, to comply with EEC and other health standards, must be rustproof, cleanable, and free from any crevices which can hold dirt. Bare timber in any form is not permitted. Most liners are now aluminium or galvanized steel sheet, finished white with a synthetic enamel or plastic coating. GRP liners are also in use. Floors are of hard concrete or tiles. Very heavy working floors may have metal grids let into the concrete surface. Floor concrete is coved up at the base of the walls to form a protective curb.

In the past, timber dunnage battens were fixed around the walls to protect the surface from collision damage and ensure an air space for circulation of the air from the evaporators. Since timber

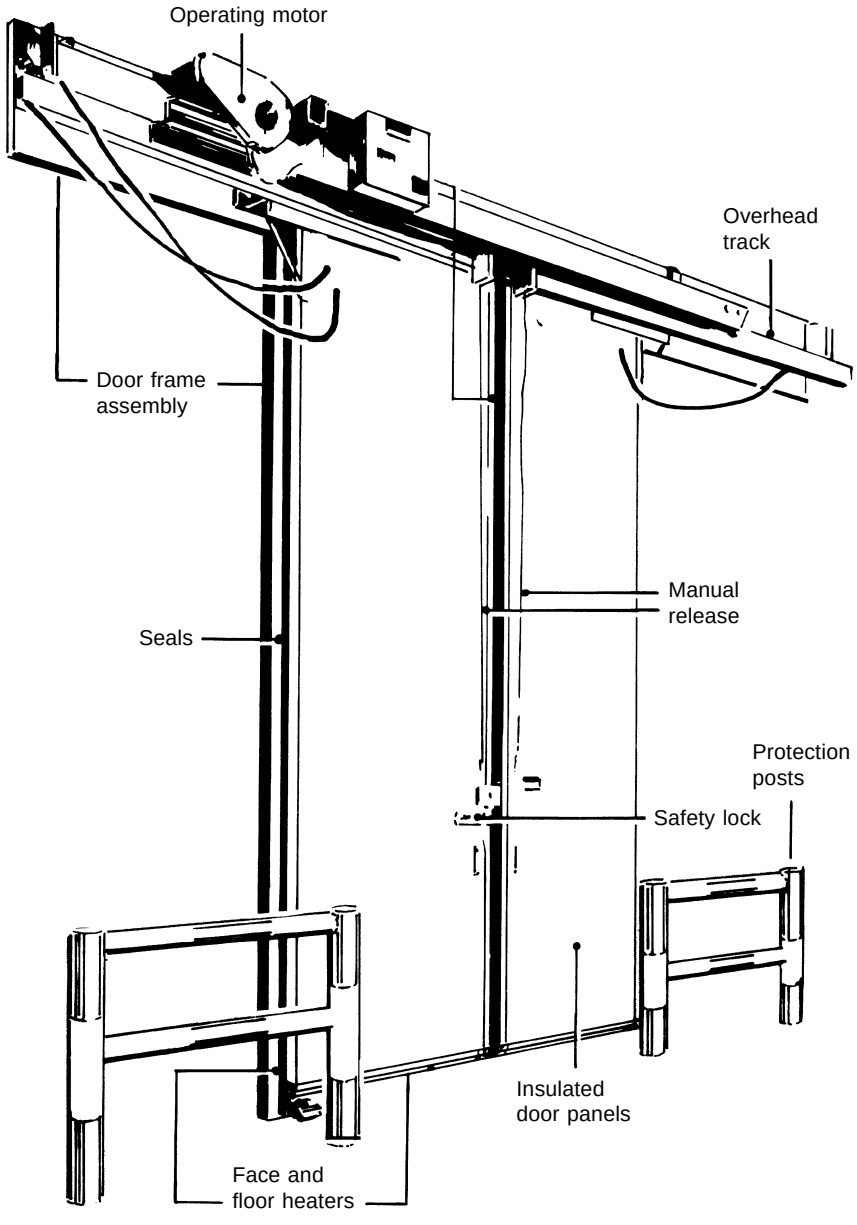


Figure 15.8 *Double sliding cold store doors, power operated (Courtesy of Clark Door Ltd)*

is no longer used, dunnage may be provided in the form of metal rails. The provision of the floor curb at the walls will ensure that pallets cannot be stacked to prevent air circulation.

Lighting in higher-temperature rooms is normally by fluorescent tubes fixed to the ceiling and having starters suitable for the temperature concerned. Low-temperature stores now mostly have sodium or mercury vapour lamps and it is possible to obtain an overall lighting intensity of 125 lux with an electrical load of 6 W/m^2 floor area. Lamps must be protected so that broken glass cannot fall onto food products. The design of efficient lighting systems merits close attention, since all energy put into the store for lighting must be removed again. Control switches are usually outside the entrance doors.

Large stores must be fitted with an emergency lighting system, battery maintained, to enable the routes to the exits to be seen clearly in the event of a mains power failure.

15.11 Evaporators

In small cold stores, the coolers will be fixed to the walls, probably blowing the air downwards, or to the ceiling, blowing sideways (see Figure 7.2).

Larger evaporators (see Figure 15.9) will also be mounted at high level if possible, to save ueful floor space. Owing to the weight, they must be supported from the outer structural roof by tie-rods passing through the insulation. Access gangways are needed in the roof void to facilitate maintenance and inspection of piping, valves and insulation. Some stores have the coolers mounted in a recess above the loading bay, providing a maintenance platform. This can only be done where the fans can cover the full length or width of the chamber.

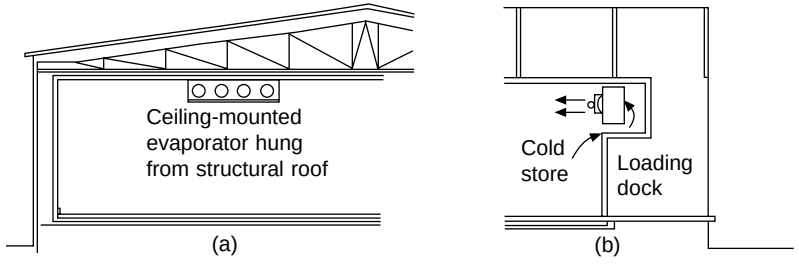


Figure 15.9 *Coldroom evaporators. (a) Ceiling hung. (b) Above loading bay*

It is sometimes necessary to assist the distribution of air from the cooler by installing air ducting. This can take the form of individual ducts, but these are prone to damage from fork-lift trucks. Alternatively, a full or partial false ceiling, below the insulated surface,

can be used. This is usually of white plastic-coated metal to match the remainder of the lining, and the light fittings can then be fitted flush with the underside.

15.12 Automated cold stores

The need for access by fork-lift trucks can require up to 60% of the floor area for gangways. There are two main methods of avoiding this wastage of store space.

Automatic stacker cranes were first used in a cold store in the USA in 1962 and there are now many installations throughout the world. The store height can be increased considerably, to 16–20 m, or even higher if the rack frame is used to support the roof of the cold store. The operation of such a store can be by using a crane with the operator inside the store, driving the crane from a heated, insulated cab, or can be fully automatically operated by a computer. One crane can service some 4000 pallet positions at the rate of 50 pallets per hour.

Mobile racking – where the lines of racking are on transverse rails, these can be closed together when access is not needed, but rolled apart to provide an aisle for a fork-lift truck. This system is best for a limited range of products moving in rotation, since the racking will not have to be moved very often. A typical small installation might have seven mobile racks, each 25 m long by four pallets high, and require an extra 3 m width for one access aisle, plus an end access of 4 m. This results in a store of 504 pallet capacity and a floor area of 270 m^2.

The tight stacking when the racks are closed impedes air flow around the pallets, so this system is not suitable where some cooling of the product may be required.

15.13 Security of operation

The value of the produce in a large cold store may be several times the cost of the store itself, and every effort should be made to maintain the refrigeration service at all times, even if plant may be inoperative for inspection, overhaul or repair. The principle of plant security is that there should be sufficient pieces of each item of plant and that they should have enough capacity for conditions to be held as required by the produce, regardless of any one item which might be stopped [29].

Usual arrangements can be summarized as follows:

1. At least two compressors, either of which can keep the store at temperature. It may run continuously to hold this.

2. Two condensers, or a condenser assembly having two separate refrigerant circuits and permitting rapair to one circuit while the other is working. If there is one assembly with forced convection, there are at least two fans.
3. All circulating pumps to be in duplicate, with changeover valves to permit immediate operation.
4. At least two evaporators, to maintain conditions if one is not working.
5. Where two compressors and two condensers are installed as independent circuits, provide changeover valves so that either compressor can work with either condenser or evaporator.

Before installation, the planned system should be analysed in terms of possible component failures to ensure that it can operate as required. Commissioning running tests should include simulated trials of plant failure, and operatives should be made aware of failure drills to keep the plant running.

16 Refrigeration in the food trades – meats and fish

16.1 Meat industry applications

In the meat industry, the main applications of mechanical refrigeration are:

Chilling of carcases directly after slaughter and dressing
Cooling of meat-handling rooms such as butcheries
Chilled water and brine for cooling poultry
Chill storage of edible meats and offal
Chilling of brine and pickling vats
Meat and poultry freezing

Animals when slaughtered, are at a body temperature of 39°C. The carcase cools slightly as it is being dressed, but must be put into refrigerated chambers as soon as possible [41, 42]. The speed of cooling depends on the thickness of the joint, so the larger carcases are usually halved into sides. While there is a need to remove body heat to check deterioration, if this process is too quick with beef or lamb, the resulting meat may be tough. A general rule for lean meat such as beef is that no part should be cooled below 10°C for at least 10 hours after slaughter, although this limit may be varied by the local producer. The total time in this chiller stage will be about 24 hours for a beef side [43]. Meat-cooling curves are shown in Figure 16.1.

During the initial cooling stage, the surface of the meat will be quite warm, and careful design of the coolers and their operation is needed to reduce weight loss by evaporation from the surface. A good air circulation is required at a humidity level of 90–94%, so as to keep the surface dry without too much dehydration. In order to maintain a good and steady air circulation around the carcases at this time, they are hung from rails (see Figures 14.1 and 16.2).

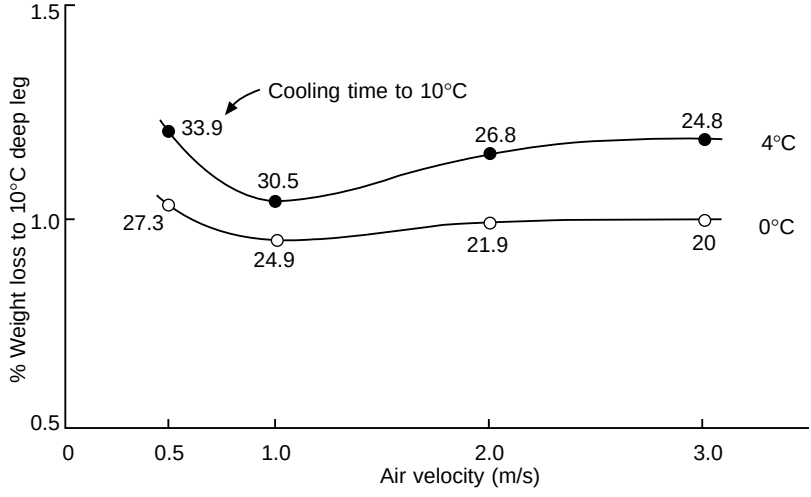


Figure 16.1 *Effect of air velocity and temperature on the weight loss of beef carcases [43]*

Storage conditions in terms of air movement and humidity will be different to those used when initially chilling the carcase. Chilled meat on the bone is stored at about 0°C, up to the point of sale. The humidity of the surrounding air is also critical in the case of fresh meats – too dry and the meat will lose weight and discolour, too humid and a slime will form on the surface.

16.2 Boned, boxed and processed meats

A lot of meat is now boned or produced as the final cuts, in the factory. For this, the meat needs to be at 0°C or just below, i.e. just above the temperature at which it starts to freeze hard.

This work must be carried out under hygienic and cool conditions. The air temperature is usually not lower than 10°C, for the comfort of the butchery staff, but some establishments work down to 2°C or 3°C. Air movement in the working area must be diffused and not too fast, to give an acceptable environment to the operators.

Cut meats are usually wrapped or vacuum packed directly after cutting. The viscera, bones and other parts not going for human consumption have a byproduct value, and will probably need to be stored at chill temperature before disposal.

Cut meats may be frozen or kept at ‘chill’ temperatures. If the latter, the shelf life is comparatively low and the product will be despatched almost immediately for sale.

In ‘protein economy’ processes, parts of the carcase which are

not to be sold as joints or cuts are made up in moulds into artificial joints, 'gigots' or meat loaf, in a pre-cooking operation. The made-up product must then be cooled to about 0°C, and may then be sliced and vacuum packed, these operations taking place in air-conditioned rooms kept at temperatures of 10°C or lower. Most such items are for 'chill' storage and immediate distribution for sale.

There are many variations in the manner of handling and processing meats, and these will be known only to specialist companies in the trade. The principles of cooling are the same for all.

Meat may be frozen on the bone, but this is not a very convenient shape for packing and handling. It is more usually boned, vacuum wrapped, boxed and then frozen. Boxed meat sizes are about 635 × 350 mm and 100, 125 or 150 mm thick, the largest of these holding some 25 kg. The freezing may be in a cold air blast and the speed of cooling will depend on the thickness of the slab (see [1–7]) and the insulation effect of the box or wrapping (Figure 16.2). Thinner pieces of meat can be frozen between refrigerated plates (see Figure 7.9*a*) [44].

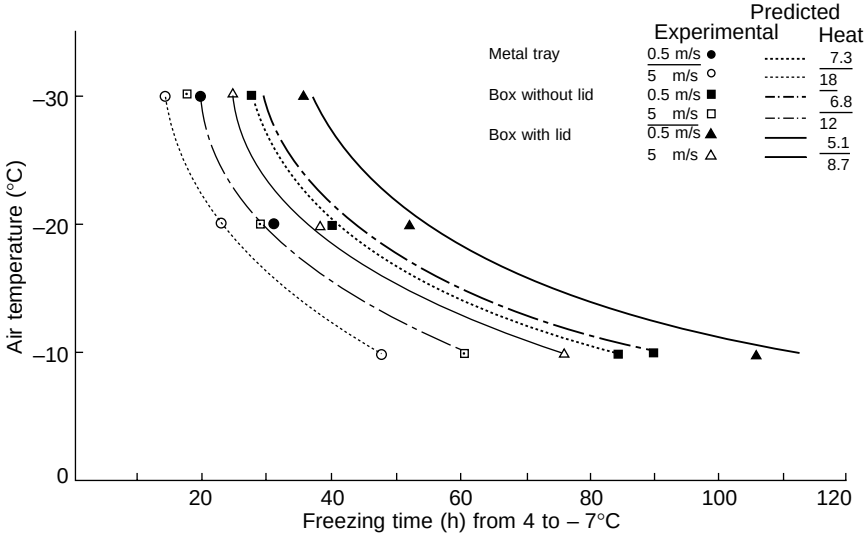


Figure 16.2 *Freezing time for 150 mm wrapped boxed beef (Courtesy of AFRC Institute of Food Research, Bristol Laboratory)*

16.3 Pork and bacon

Fresh pork has a shorter shelf life than beef, but is handled in the same way and at the same chill-room temperatures. Although no latent heat of the freezing of water content will be extracted at chill

temperatures, some heat will be removed when the fat 'sets' or crystallizes. The quantity of heat to be removed should be estimated and may be included in the sensible heat capacity in that temperature range. For example, the sensible heat capacity of pork meat averages 2.5 kJ/(kg K), but a figure as high as 3.8 may be used for carcase cooling to allow for this factor.

A high proportion of pork is pickled in brine and smoked, to make ham or bacon. The original process was to immerse the meat in a tank of cold brine for a period. A quicker method is to inject the cold pickle with hypodermic needles into the cuts. Smoking is carried out at around 52°C, so the cured bacon must be cooled again for slicing, packing and storage.

16.4 Poultry

Poultry is immersed in hot water just after slaughter, in order to loosen the feathers for the plucking process. The carcases are then eviscerated and chilled as soon as possible by cold air blast or using iced water in the form of a bath or spray.

Larger birds may be reduced to portions, so the flesh must be cooled to about 0°C to make it firm enough for cutting. Whole birds are prepared for cooking and then vacuum wrapped for hygiene.

Poultry may be chilled for the fresh chicken market, or frozen. Chilling and freezing are mainly by cold air blast. Large birds such as turkeys are wrapped and immersed in low-temperature brine until the outside is well frozen, and then put into low-temperature storage to freeze right through. Some poultry is frozen by spraying with liquid carbon dioxide.

Storage of chilled poultry is at –1°C. The shelf life is relatively short and the product will not remain in store for more than a couple of days.

16.5 Fish

Most fish is still caught at sea and must be cooled soon after it is taken on board, and kept cold until it can be sold, frozen or otherwise processed [45]. The general practice is to put the fish into refrigerated sea water tanks, kept down to 0°C by direct expansion coils or a remote shell-and-tube evaporator. The sea water must be clean and may be chlorine dosed. At this condition, fish can be kept for up to four days.

Ice is also used on board, carried as blocks and crushed when required, carried as flake, or from shipboard flake ice makers. Artisanal fishermen in hot climates may take out crushed ice in

their small boats. Fresh fish is stored and transported with layers of ice between and over the fish, cooling by conduction and keeping the product moist. Fish kept at chill temperatures in this manner can travel to the final point of sale, depending on the time of the journey. Where refrigerated storage is used, the humidity within the room must be kept high, by using large evaporators, so that the surface of the fish does not dry.

Most vessels can now freeze their catch at sea, enabling them to stay offshore without the need to run back to a port within the limited life of the chilled product. If the fish is to be cleaned and processed later, it is frozen whole, either by air blast or, more usually, in vertical plate freezers (see Figure 7.9*b*), followed by frozen storage. Some fishing vessels and the fish factory vessels will carry out cleaning, filleting and other operations on board and then freeze and store the final product.

A limited amount of fish is frozen by immersing it in a cold concentrated sodium chloride brine. This is mainly tuna for subsequent canning, or crustaceans.

Fish which is frozen in air blast will often be dipped into clean water afterwards, resulting in a layer of ice on the surface. This glazing process protects the fish from the effects of dehydration in subsequent storage.

Some freezing of fish fillets and other processed fish is carried out between or on freezer plates, in an evaporator assembly similar to that shown in Figure 7.9*a*. Flat cartons of fish and fish fillets are frozen in these horizontal plate freezers.

Health and safety requirements continue to become stricter in the maintenance of the cold chain and the latest regulations should be adhered to.

17 Refrigeration for the dairy, brewing and soft drinks industries

17.1 Milk and milk products

Milk is converted in the creamery and associated factories to whole or 'market' milk, skimmed milk, creams, butters, cheeses, dried milk, whey, yoghurts, butter oil, condensed milk, milk powder and ice cream [46].

In the dairy industry as a whole, the main needs for mechanical cooling are:

- Cooling milk directly after it leaves the cow and before transport to a central creamery
- Keeping the raw milk cool after it enters the creamery
- Chilled water for use in plate heat exchangers to cool milk and milk products directly after pasteurizing
- Chilled water to wash butter
- Chill temperature stores for milk, butter, cheese, yoghurt and other liquid milk products
- Frozen storage for butter (and sometimes cheese)
- Continuous, plate and air blast freezers for ice-cream
- Low-temperature brine for lollipop freezing

Milk comes from the cow at about 37°C, and must be cooled within two hours to 4°C or lower, and under hygienic conditions. At this temperature any micro-organisms present will not multiply at a dangerous rate and the milk can be transported to the creamery.

Dairy farms have bulk-storage tanks with their own refrigeration plants. These are usually made in the form of a double-skin, insulated tank, having the evaporator coils in the jacket, which also contains water. The refrigeration system runs throughout the 24 hours and

builds up a layer of ice on the evaporator coils when there is no milk cooling load. This stored cooling effect is available to help cool the warm milk when it comes from the cow (see also Section 12.3).

Bulk tanker vehicles will not collect milk which is warmer than 4°C. If milk can be picked up from the farm at this temperature in bulk tankers, and transported quickly enough to the creamery, then there is no need to have refrigeration equipment on the vehicle.

On arrival at the creamery the milk is tested and transferred to bulk-storage tanks, which may hold up to 150 t each. These will be heavily insulated and may have some method of cooling, so as to keep the milk down to 4°C until it passes into the processing line.

Throughout the subsequent processes, milk and milk products will require to be re-cooled down to 4°C or thereabouts. The main method of achieving this is to use chilled water at just above freezing point as the secondary refrigerant. Creameries will have a large central water-chilling system, using Baudelot coolers or evaporators in water tanks. Some older systems are in use, but are being rapidly replaced. Chilled water is piped to all the cooling loads within the plant.

Whole milk for human consumption is pasteurized at 75°C for a short time, and then re-cooled to 4°C immediately. This is done by contraflow heat exchange between milk entering and leaving the process, hot water and chilled water, in plate heat exchangers (see Figure 17.1) in the following stages:

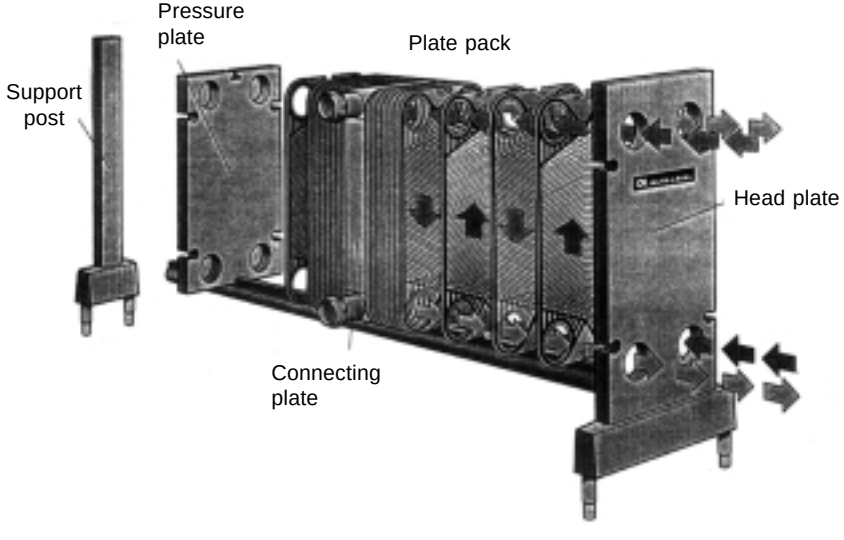


Figure 17.1 *Plate heat exchangers*

1. Raw milk at 4°C is heated by the outgoing milk up to about 71°C.
2. This milk is finally heated by hot water up to the pasteurizing temperature of 75°C (or hotter for UHT milk) and held for a few seconds.
3. The milk is cooled by the incoming milk, down to about 10°C.
4. The final stage of cooling from 10°C to 4°C is by chilled water at 2°C.

Milk for other products is treated:

1. In a centrifuge to obtain cream and skim milk
2. In churning devices to make butter and buttermilk
3. With rennet to make cheese (leaving whey)
4. With cultured bacteria to make yoghurt
5. By drying, to milk powder

Butter is made from cream in continuous churning machines. At stages during this process, the butter is washed in clean, cold water to keep it cold and remove surplus buttermilk. At the end of the churning stage, butter is still in a plastic state and, after packaging, must be stored at 5°C to crystallize the fat. Long-term storage of butter is at – 25°C.

Cheeses may be pressed into a homogeneous block, or left to settle, depending on the type and methods of manufacture. They then undergo a period of ripening, to give the characteristic flavour and texture. The cold storage of cheese during the ripening period must be under strict conditions of humidity and hygiene, or the cheese will be damaged. Some cheeses can be frozen for long-term storage, but must then be allowed to thaw out gradually and complete their ripening before release to the market.

Other processes (except milk drying) require the finished product to be cooled to a suitable storage temperature, usually 4°C or thereabouts, and kept cool until the point of sale. Conventional-type cold stores can be used for mixed dairy products, since all of them will be packaged and sealed after manufacture.

17.2 Ice-cream

Ice-cream is a product which has been developed since mechanical refrigeration became available. Ice-cream mixes comprise fats (not always dairy), milk protein, sugar and additives such as emulsifiers, stabilizers, colourings, together with extra items such as fruit, nuts, pieces of chocolate, etc., according to the particular type and flavour. The presence of this mixture of constituents means that the freezing

process covers a wide band of temperatures, starting just below 0°C and not finishing until – 18°C or lower. The manufacturing process is in three main stages – mixing, freezing to a plastic state, and hardening.

The basic mix is made up in liquid form, pasteurized, homogenized and cooled, using chilled water in plate heat exchangers. It is then 'aged' for a few hours and, for this, it will be stored at 2–3°C in jacketed tanks, with chilled water in the jacket.

The next stage is to freeze it rapidly, with the injection of a controlled proportion of air, to give it a light, edible texture. Aerated mix of about 50% air, 50% ice-cream mix by volume is passed into one end of a barrel which forms the inside of a flooded evaporator. The mix freezes onto the inside of the barrel and is then scraped off by rotating stainless steel beater blades, and passes through the barrel with a continuous process of freezing, beating and blending. The most usual refrigerant for ice-cream continuous freezers is ammonia, which will be at an evaporating temperature of – 35°C to – 30°C. About half of the total heat of freezing is removed in this stage, and the ice-cream leaves at a temperature of around – 5°C, depending on the particular type of product. A continuous ice-cream freezer is shown in Figure 17.2.

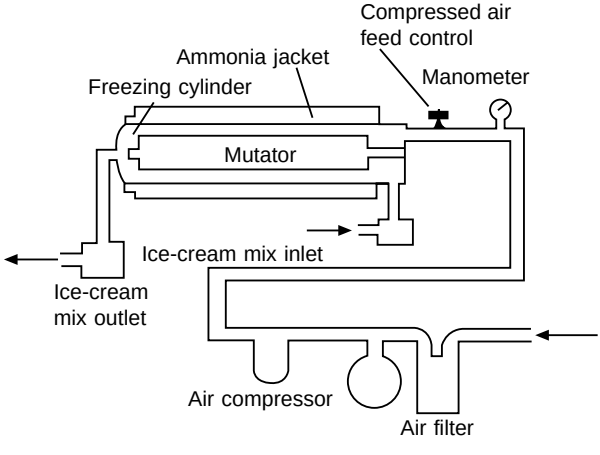


Figure 17.2 *Continuous ice-cream freezer (Courtesy of Alfa-Laval Co. Ltd)*

The ice-cream is still plastic as it comes from the freezer, and it is extruded into the final sales shape – carton, tub, box, etc. It must then be hardened by cooling down to a storage temperature of – 25°C or lower, during which the other half of its heat of freezing is removed.

Flat boxes can be hardened between refrigerated plates as shown in Figure 7.9*a*. Other shapes pass through a cold air blast, and a typical machine has a flexible conveyor belt, capable of taking a wide variety of shapes (see Figure 17.3). An important factor of this final freezing process is that it must be as rapid as possible, in order to limit the size of ice crystals within the ice-cream. Rapid freezing implies a high rate of heat transfer and, therefore, a very low refrigerant temperature. Air blast at – 40°C is common. Two-stage compression systems are used.

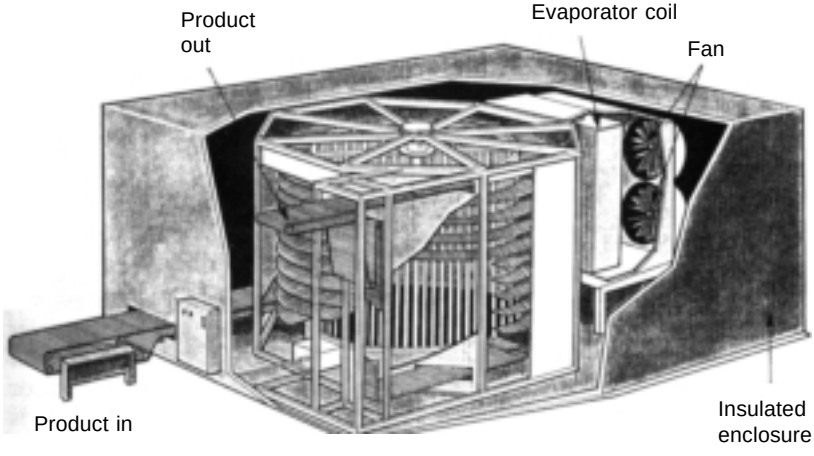


Figure 17.3 *Cross-flow spiral tunnel (Courtesy of APV Baker Ltd, Hall Division)*

Ice-cream must be kept at low temperature right up to the point of final consumption. If it is allowed to soften, the entrained air bubbles may escape and the original texture will be lost. If it softens and is then re-frozen, a hard, solid skin forms, making the product inedible. Ice-cream must always be handled quickly when passing through transit stages from the factory to consumer.

17.3 Ice lollies

Ice lollies are made from juice (water, sugar, citric acid, flavour and colour) and are frozen into shape using moulds immersed in a cold brine solution, in a similar manner to can ice making (see Section 12.4). The moulds are made from stainless steel or nickel, and pass in rows through a brine bath at – 45°C. Different layers of confection may be built up by allowing one outside layer to freeze, sucking out the unfrozen centre and refilling with another mix. The sticks are inserted before the centre freezes solid. The moulds finally pass

through a defrost section of warm brine to release the lolly from the mould, and extractor bars grab the sticks, remove the lollies and drop them into packaging bags.

17.4 Brewing

The production of beers and ciders requires the fermentation of sugary fluids by the action of yeasts, and the cooling, filtration, clarification and storage of the resulting alcohol–water mixture.

The starting mix for beers is a warm brew of grain-based sugar and flavouring. This 'wort' leaves the hot brewing process and is cooled to a suitable brewing temperature – around 10°C for lagers and 20°C for traditional bitters. This was originally carried out with Baudelot coolers, but now plate heat exchangers are mainly used, with chilled water as the coolant.

The process of fermentation gives off heat, and the tanks may need to be cooled with chilled water coils, with jackets, or by cooling the 'cellar' in which the tanks are located. When fermentation is complete, many beers are now pasteurized, in the same manner as milk (see Section 17.1). The beer is then cooled to just above freezing, filtered and left to 'age'. Before final bottling, kegging or canning it will undergo a fine filtration to improve the clarity.

Refrigeration is required for the cold storage rooms and to provide chilled water for the plate heat exchangers. The 'cellars' are very wet areas, and the cooling plant should be designed to maintain as low a humidity as possible, to help preserve the building structure.

Beers at the point of sale are traditionally stored in cellars to keep them cool. Beers are in kegs or piped into bulk tanks. Artificial cooling of these areas is now usual, using packaged beer cellar coolers, somewhat similar to the air-conditioner shown in Figure 13.4. Bulk-storage tanks may have inbuilt refrigeration plant. Drinks such as lager beer, which are normally drunk colder than other beers, are passed through a chilled water bath or double-pipe heat exchanger for final cooling.

Bottled beers and other drinks are kept on refrigerated trays, comprising a cooled base tray and an inbuilt refrigeration system.

17.5 Wines and spirits

The optimum temperature of fermentation of wine depends on the type, red wines working best at about 29°C while the white wines require a cooler condition of around 16°C. Heat is given off by the chemical process of fermentation. They are then traditionally matured and stored in caves or cellars at about 10°C. Much of the manufacture

and most of the storage is now carried out in rooms controlled by mechanical refrigeration. Spirits do not need low-temperature storage.

The clarity of the final beverage is affected by small particles of tartrates and other substances which precipitate during storage. To obtain a product which will remain clear in storage, many wines and spirits are cooled by refrigeration to a temperature just above their freezing points and then fine-filtered.

17.6 Soft drinks

The feature of most soft drinks is that they are 'carbonated', i.e. they have a proportion of dissolved carbon dioxide, which causes the bubbles and typical effervescent taste. The quantity of gas dissolved in the water will be 3.5–5 volumes, i.e. each litre of water will have dissolved 3.5–5 litres of carbon dioxide. The manufacturing technique is to dissolve the required amount of gas into the beverage, and then get it into its can or bottle.

The solubility of carbon dioxide in water depends on the pressure and temperature. The relationship between temperature and pressure for 3.5 and 5 volumes is shown in Figure 17.4. It will also be affected by the amount of air already dissolved in the water. The raw water is therefore carefully filtered and de-oxygenated under vacuum before the sugars and flavourings are added.

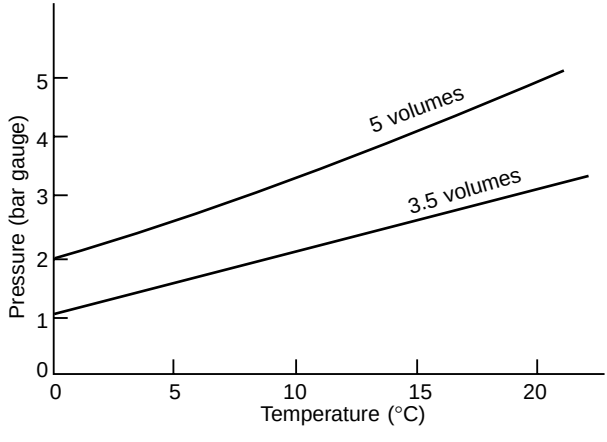


Figure 17.4 *Solubility of carbon dioxide in water*

Since the gas will dissolve at a much lower pressure at a low temperature, the beverage will be cooled to near 0°C, either before or during the introduction of the gas.

The liquid may be pre-cooled in plate heat exchangers, using

chilled water or one of the brines – formerly ethyl alcohol–water but now more usually propylene glycol–water. One of the carbonization methods is to carry out the final cooling stage over a Baudelot cooler which is fitted within a pressure vessel (see Figure 17.5). The gas is introduced at the pressure needed to dissolve the required proportion, and the gas meets the liquid as it flows in a thin film down the surface of the cooler.

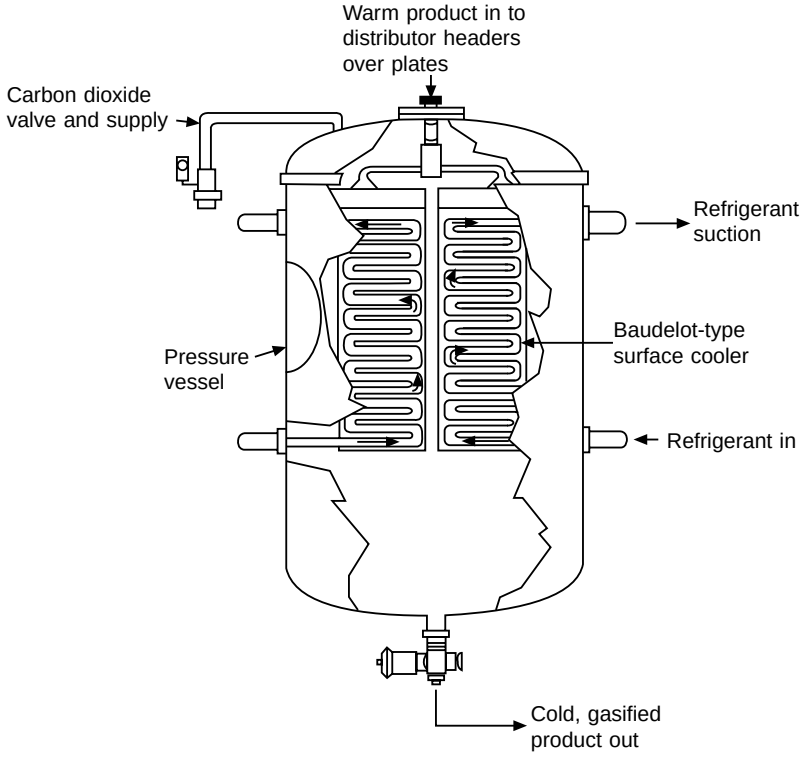


Figure 17.5 *Chiller-carbonizer for soft drinks (Courtesy of Meyer Mojonnier Ltd)*

It is then bottled as quickly as possible, before the gas has time to bubble out again. Once it is sealed in the bottle, cooling is not needed for storage.

Chilling of brines for pre-cooling will generally be in shell-and-tube evaporators. The Baudelot cooler within the pressure vessel may be cooled by flooded or dry expansion refrigerant, or by brine.

18 Refrigeration for fruit, vegetables and other foods

18.1 Fruits

Fruits are seasonal in temperate climates, and a good harvest may be followed by a shortage if there is no method of preservation. The hard fruits, apples and pears, have traditionally been stored in cool places and may then last for several months, depending on the variety. Refrigeration has extended the storage life, and made this more reliable.

Artificial cooling has made it possible for fruit grown anywhere in the world to be brought to any market willing to pay the extra price. Where transit times are long, such as in the shipment of bananas, the fruit is picked while still green and undergoes a controlled ripening on the ship. The conditions for refrigerated shipping depend on many factors, and the temperatures and humidities shown in Table 14.1 are a general indication of the ranges. More precise information must be used for the operation for a particular product.

A large amount of perishable food now travels by air. Since the cargo holds of airliners are not pressurized, the problem may be one of temporary protection against low temperatures, rather than of keeping the product cold.

18.2 Gas storage of apples and pears

All fruits respire oxygen and, in doing so, start to decay. If the oxygen concentration can be reduced, the rate of respiration will be slowed and the storage life may be extended. The maintenance of a low partial pressure of oxygen requires a gas-tight structure to prevent diffusion. Such controlled atmosphere stores are carefully constructed and sealed to achieve this, and are generally termed *gas stores*.

The fruits are loaded and the store sealed. Within a few days they consume a proportion of the available oxygen and respire carbon dioxide. Considerable research over the past 60 years, mainly in the UK [47], has determined the correct balance of gases to prolong the storage life of the different varieties of apples and pears, both home grown and imported.

Apparatus is required to monitor the atmosphere within the store and keep the right proportions by chemical removal or controlled ventilation.

18.3 Vegetables

Most vegetables contain a very high proportion of water, and wilt rapidly as they dry out. Storage conditions demand a high humidity level of 90–98% saturation and temperatures as close to their general freezing point of 0°C as possible. Some leaf vegetables are sprinkled with ice chips, to maintain this damp, cold condition. Cold stores for vegetables have very large evaporators, to provide these high humidities. Apart from the preservation of the vegetable substance itself, mould growths and insect pests are also controlled by low temperature.

A few products, such as bananas, tomatoes, melons, cucumbers and some crops of potato, are better kept at higher temperatures. These conditions vary with the variety, state of ripeness when picked and required time of storage.

Onions and garlic are susceptible to moist conditions, which encourage mould growth, and are stored at humidities of 65–70%. It is not possible to store these together with other vegetables for more than a very short time.

18.4 Frozen vegetables

The convenience of having high-quality produce, graded and ready for cooking, may justify a premium price. Peas, carrot slices, beans and some leaf vegetables are frozen in air blast (see Chapter 19). There are slight changes in the texture, but the texture is further changed by cooking, and the final result does not differ from fresh produce.

A few items, strawberries, other soft fruits and pieces of cauliflower, are quick-frozen with liquid nitrogen.

Frozen fruit and vegetables will be sealed in plastic bags and stored at –18°C or lower. The humidity at this temperature is not important.

18.5 Bakery products

Bread doughs become heated by the mixing process, and the yeast may begin to work too soon. The water content of the mix may be chilled, or the larger machines may have water-cooled jackets to take away this heat.

Doughs are prepared some time before the final baking process and will be left to 'prove', i.e. allow the yeast to commence working. The action can be retarded by cooling the dough at this stage, and this process permits the workload to be spread through the day. Typically, bread for the following morning can now be prepared on the previous day, up to the proving stage, and then kept under cold, humid storage until a few hours before baking is to commence.

Dough-retarding cabinets are now used in most bakeries. Bread doughs may be made up at any time and put into storage at a temperature between – 4° and + 3°C, depending on the required retard time, which may be up to three days. An automatic timer will terminate the cooling cycle and bring the doughs up to proving temperature when required. In this way, doughs can be ready for the oven when the bakery staff commence in the early morning. Also, stocks can be held ready for unexpected extra demands.

A high proportion of bread is sold sliced but it will be too hot for this on leaving the oven. Large-scale bakeries have cooling tunnels to reduce the bread temperature so that it can be sliced. A high degree of hygiene is necessary, or the slicer will introduce airborne spores and the bread will grow moulds.

18.6 Cook/chill

There is an increasing demand for ready-prepared foods for final re-heating or cooking in microwave ovens. Applications are for retail sale of take-away meals and factory/office and institution catering. Such foods may be frozen and will then have a longer storage life, but will require frozen storage.

It is possible to pre-cook the product to a pasteurization temperature and then cool, for short-term storage above freezing point. The required standards of temperature control and hygiene are very strict and the subsequent shelf life restricted. The process is cheaper than freezing. Product leaves the oven at 100°C and may be allowed to cool in the ambient air to 70°C, if conditions of hygiene are satisfactory. During this time it may be split into meal-size or other portions. Generally, it should then be in a thickness not more than 50 mm, or it will not cool in the specified time. Trays of the product are loaded into a chilling cabinet and all parts must

be reduced to 3°C in 90 minutes. Since it is not required to freeze any part, the air to cool the product cannot be much below 0°C, and cabinets for this purpose have a built-in refrigeration plant which will provide air at –2°C, and with a speed over the product of some 6.5 m/s.

The chilled product must be stored at 3°C or thereabouts. Shelf life may be up to a maximum of five days, but is usually only a day or so.

18.7 Chocolate enrobing

Many confections are coated in a thin layer of chocolate. The latter is a mixture of chocolate, cocoa butter and other fats, blended to form a suitable coating material. This layer melts at a temperature generally in the range 27–34°C. The manufacturer wishes to coat the confection in a thin, continuous layer, and then harden this layer so that the product can be wrapped and packed with the least delay on the production line.

Chocolate enrobing starts with the item passing through the coating process, and then through a refrigerated air blast tunnel to harden the layer. The colder the air, the quicker this will take place, but if the product leaves the tunnel too cold, atmospheric moisture may condense on the surface and spoil the glossy finish expected by the consumer. The average air temperature in the tunnel may be between 2°C and 7°C, and the air is usually cooled with refrigerated or brine coils within the tunnel. It is sometimes necessary to air-condition the entire working area so as to keep the dew point temperature (see Chapters 23–25) lower than the temperature of the surface of the confection as it leaves the tunnel.

18.8 Refrigeration of foods

The present-day food industry is almost totally dependent on refrigeration in one form or another, to manufacture, preserve, store and bring the product to the point of sale. The few examples chosen in Chapters 14–18 indicate the general principles. The history, development and current practice of refrigeration of foodstuffs is largely the history, development and current practice of the refrigeration industry itself.

19 Food freezing. Freeze-drying

19.1 Quick freezing

The liquid content of foodstuffs, containing proportions of sugars and salts, has a band of freezing temperatures from 0°C down to – 18°C and lower. If these liquids freeze slowly, long ice crystals are formed which pierce the cell walls and change the resulting texture. If this damage is to be avoided, the product must be frozen rapidly, so that the crystals do not have time to grow. The process is only applicable to products which are eaten raw or lightly cooked, such as strawberries, peas and beans. The speed of freezing is a relative matter, but produce of this sort is generally frozen in 5–10 minutes in an air blast, somewhat quicker if immersed.

Various methods have evolved, depending on the available resources, the product concerned and the premium value it might earn in an improved frozen state.

19.2 Air blast coolers and tunnels

Where the product shape is irregular, the only way to extract its heat will be by using a cold fluid surrounding it. The most common of these is air. The air temperature will be of the order of – 40°C and the air speed over the product will be high, to get good heat transfer.

Discrete pieces of product, such as peas, slices of carrot, beans and items of this size, can be conveyed on a perforated belt, with the cold air blasting up through the holes, to both cool the product and agitate it, to prevent it sticking either to the belt or to other similar pieces. This type of cooling tunnel is shown in Figure 19.1.

Flat pieces of product, such as fish fillets, would suffer a change in shape in a free air blast and are better on a flat moving belt. Here, some of the heat goes direct to the cold air and some by conduction to the belt, which is usually of stainless steel. This tunnel

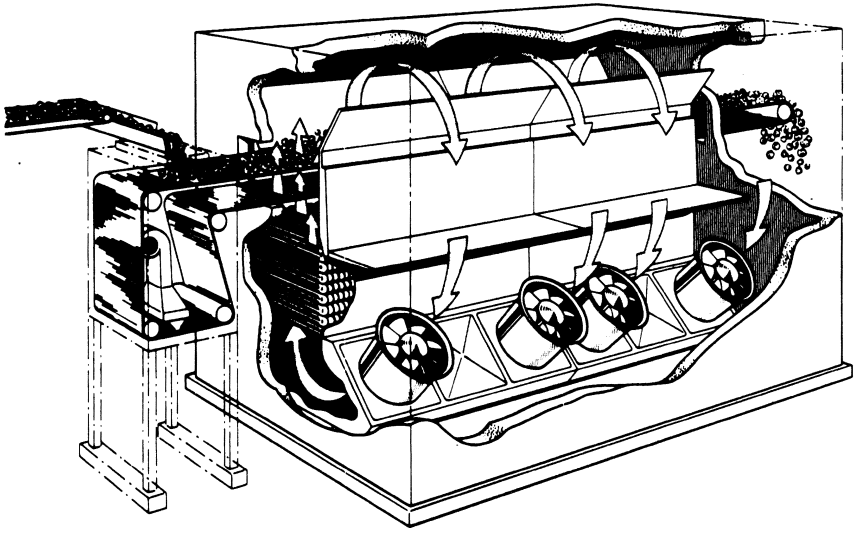


Figure 19.1 *Freezing tunnel, fluidized bed (Courtesy of APV-Parafreeze Ltd)*

can be designed to absorb much less fan power and, since fans input energy which must then be removed by the refrigeration system, the tunnels will be more energy efficient [48]. (See Figure 19.2.)

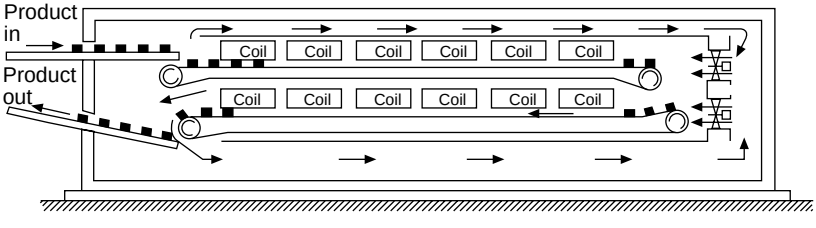


Figure 19.2 *Freezing tunnel, belt (low fan energy) (Courtesy of S. Forbes Pearson)*

Larger items, such as tubs of ice-cream, take a long time to harden and a straight conveyor would be too long for convenience. Such conveyors can be wound into a spiral shape and contained within a coldroom with air blast coil (see Figure 17.3).

19.3 Contact freezing

Products in regular-shaped packages, such as ice-cream in flat cartons, are pressed between horizontal, flat, refrigerated plates. These can

be opened apart slightly to admit the product and are then closed by hydraulic rams to give close thermal contact. When freezing is complete, the plates open again to remove the packs (see Figure 7.9*a*). The vertical plate freezer (Figure 7.9*b*) is used for a loose product such as wet fish, which is packed into the gaps between the plates. When the freezing is complete, the product is removed as a solid block and may be 75 mm or 100 mm thick.

Trays of product to be frozen can be loaded onto trollies, which are taken through an air blast tunnel. The evaporator coils will usually be in the upper part of the tunnel, with air flow across the trays.

Material to be frozen can be fully immersed in a cold liquid. This might be a brine, in which case the material may have to be wrapped in a plastic bag to avoid contact with the liquid. The sodium chloride and glycol brines cannot be used cold enough to get complete freezing, so this may be a first pre-cooling stage before a final air blast. Alternatively, liquid nitrogen (– 196°C) or carbon dioxide (–78.5°C) can be sprayed onto the surface.

19.4 Freeze-drying

Certain products cannot be kept in the liquid form for an appreciable time and must be reduced to dry powders, which can then be kept at chill or ambient temperatures. The water must be removed to make them into powders, but any heating above ambient to boil off the water would lead to rapid deterioration. The water must therefore be removed at low temperature, requiring low pressures of the order of 125 Pa.

The process is carried out in a vacuum chamber fitted with refrigerated contact freezing plates, heaters and a vacuum pump. Between the chamber and the pump may be a refrigerated separator to prevent too much of the moisture entering the pump. The product is placed in containers on the plates and frozen down to about – 25°C, depending on the product, but sometimes as low as – 50°C. The vacuum and, at the same time, a carefully controlled amount of heat, is then applied, to provide the latent heat of sublimation (ice to vapour) without allowing the temperature to rise. As the water is driven off, the product collapses to a dry powder. This is extremely hygroscopic and must be packed in air-tight containers as quickly as possible on completion of the cycle.

This process was developed for the preservation of antibiotics, but is now in widespread use for other products such as ‘instant’ coffee, tea, soup, etc.

20 Refrigerated transport, handling and distribution

20.1 The cold chain

The 'cold chain' principle of food handling and distribution is that the product will be maintained at suitable conditions all the way to the point of sale. This requires transport and various kinds of storage.

The transport of cooled produce, using mechanical refrigeration, was one of the first major uses, dating back to 1880 and only 20 years after the first static cold storage. The present annual movement of refrigerated produce exceeds 50 million tonnes.

Sea transport was originally in insulated holds built into the ships. Few of these remain, owing to the high handling costs, and most maritime trade now uses insulated containers, either with their individual cooling plants or connected to a central refrigeration system on the vessel. The type of cooling unit for a container follows the general principles of that shown in Figure 20.1, and will be accommodated within the framework of the container. Such units will carry monitoring and alarm devices, to ensure safety of the produce.

Larger road vehicles are articulated semi-trailers with a maximum length of 15.5 m, an internal volume of 73 m^3 but holding up to 40 t. The majority of the cooling units are one-piece factory-built units and have their own petrol or diesel engine for use on the road and an electric motor which can be run from mains supplies when the vehicle is static. Change of the drive is by magnetic clutches. Compressors will be open drive and the complete unit will be of rugged construction to withstand vibration from poor roads and the inbuilt drive motor. Such units will be adaptable, in being able to maintain any required temperature automatically. Heaters are also fitted, since vehicles may be working at ambient temperatures lower than that required for the produce being carried.

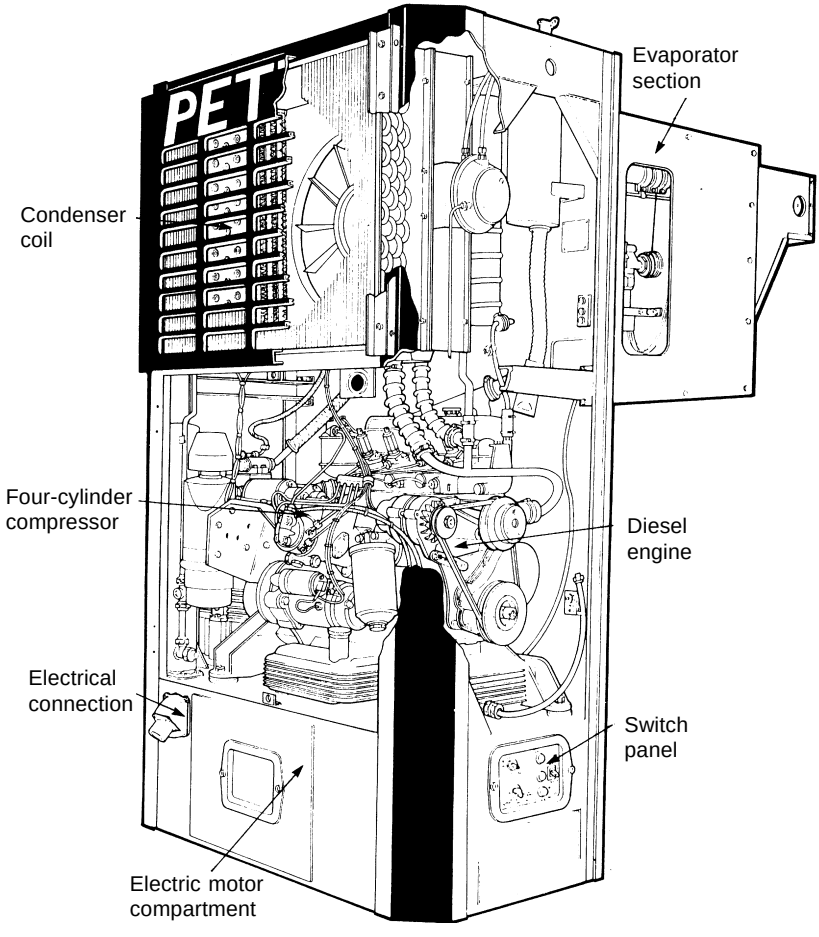


Figure 20.1 *Self-contained transport refrigeration unit (Courtesy of Petter Refrigeration Ltd)*

Direct injection of liquid nitrogen is also used on the larger vehicles. This is carried in metal vacuum flasks and the vehicle will be reliant on depots where the liquid nitrogen flask can be refilled. The only mechanical equipment will be a thermostatically controlled solenoid injection valve.

Vehicles for local delivery journeys tend to be in use only in the daytime and spend the night static. Cooling systems can run from a mains electricity supply providing they can hold a sufficiently low temperature while on the road. Use is made of eutectic plates (see Section 7.5) and of cooling the vehicle body only when in the garage, relying on the cold mass of produce and good insulation to

maintain conditions during delivery. Some local delivery vehicles use liquid nitrogen.

Rail traffic is mainly in purpose-built, insulated wagons, many of these having self-contained refrigeration systems. Some produce is pre-cooled and/or iced. Re-icing stations are available on the longer routes in Europe.

The transport of perishables by air does not require mechanical refrigeration, as low temperatures prevail at the heights flown. Fresh vegetables and flowers need to be protected from freezing, and produce will usually be in insulated containers. A feature of this traffic is the prompt and speedy handling at the airports. Coldrooms are provided at some airports to store produce immediately before and after transit. Solid carbon dioxide ('dry ice') is used for short-term cooling of airline passenger meals.

20.2 Handling

During movement of goods between static cold stores and vehicles, every effort must be made to avoid any warming. The principle is to close the vehicle right up to the cold store wall.

The ideal arrangement is to back the vehicle up to a door with a sealing collar, so that the contents may move directly into the store without exposure to ambient temperatures. If the height differs from that in the store, adjustable platforms are fitted at the door. Where fork-lift trucks have to pass in and out of a cold store, plastic strip curtains are used (see Figure 20.2).

To avoid ingress of warm air (and loss of cold air) it is useful to have an airlock. However, these need to be at least the length of a loaded fork-lift truck, and the extra space required, together with the double doors and extra movement time, should be investigated closely before such an arrangement is put into use.

20.3 Order picking

The market situation is that a few large producers of frozen and chilled foods supply a large number of retailers. This had led to the development of distribution stores, where goods are delivered in bulk, stored for a short time, 'order-picked' and then sent out to the individual supermarkets and other outlets.

Distribution stores require adjacent refrigerated storage and order-picking areas, and may operate on a 24-hour basis. For full access, the storage will be on pallet racking (see Figure 14.2*a*). This will occupy some two-thirds of the store, leaving the remainder for sorting the goods into the individual outgoing batches. The latter may be

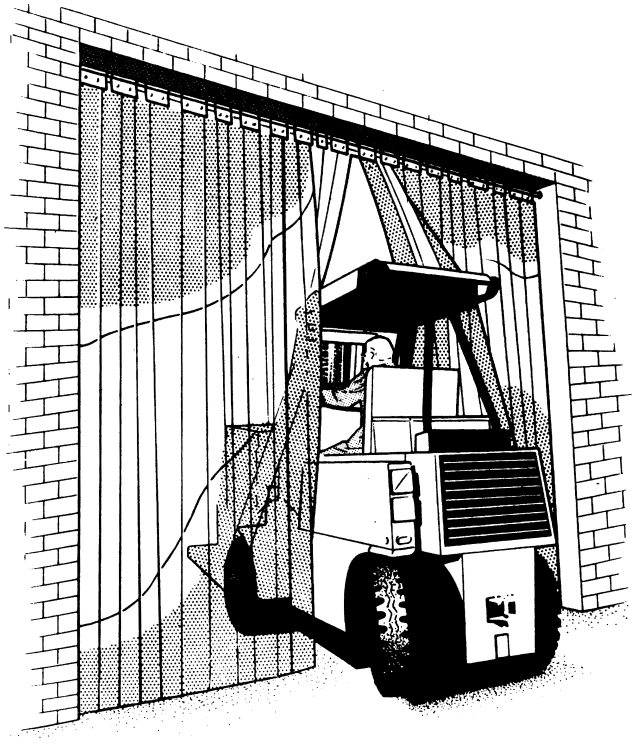


Figure 20.2 *Strip curtain at cold store door*

on pallets or wheeled racking. Operatives have to carry out the order-picking operation within the store and will have suitable protective clothing. Stores are usually 5–8 m high, so that there is less air movement from the coolers at working level. Fork-lift trucks are available with enclosed and heated cabs. Some order picking is now carried out on a more mechanized basis, using automatic handling (see Section 15.11).

20.4 Refrigerated display

It is a well-established principle that goods which can be seen are more likely to be bought than those hidden from sight. This has now reached a stage where retailers can predict the relative attractions of shelf heights and positions within a supermarket. The requirement to maintain the product at a suitable temperature at all times cannot be avoided. Refrigerated display aims to show the produce to the best advantage while still keeping it cool.

The first arrangement for frozen foods was the ice-cream conservator, a chest-freezer type of cabinet, i.e. reach-in from the

top, and with sliding or hinged glass lids. The refrigeration system is inbuilt and the evaporator is a coil of pipe in contact with the inner wall. These are still in use in confectionery shops, for ice-cream.

Providing the surrounding air is reasonably still, the lids may be omitted. It helps to have glass walls at the sides to reduce draughts, which would disturb the layer of very cold air in the cabinet. The evaporator may be pipe coils on the outside of the inner wall, but is more usually a finned coil at the back or sides. It is important that produce is kept below the design level of the cold air blanket. The construction with discrete cold trays is now taken a stage further, where several trays may be arranged one above the other.

Open-top display can gain considerable heat from air currents and radiant heat from lighting. Temporary covers are frequently used when the building is closed, to reduce these gains and help preserve the foodstuffs. This is of considerable importance where cut meats are displayed, since the radiant heat from lights and loss

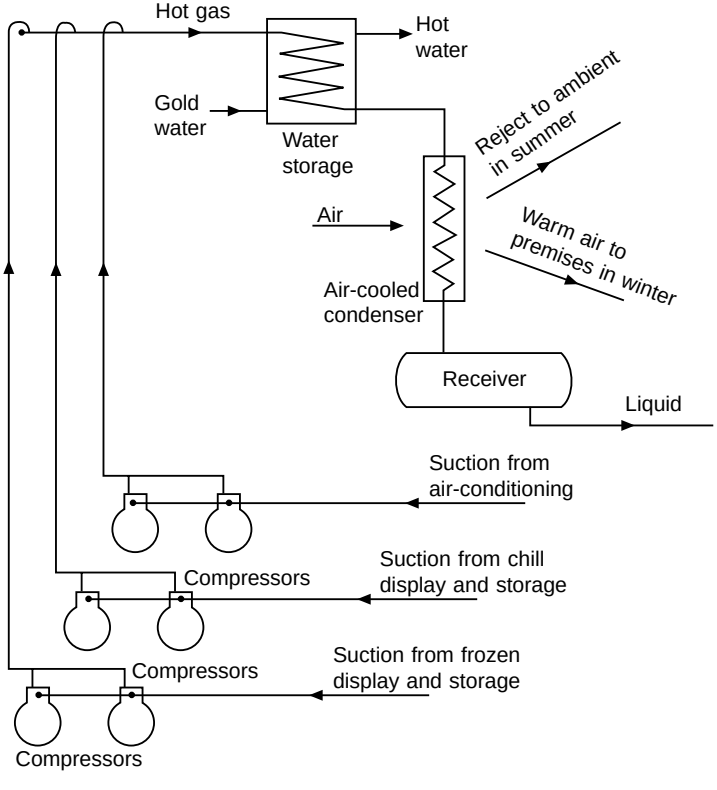


Figure 20.3 *Multiplex installation for supermarket, with heat recovery*

of the cold air blanket lead to surface moisture loss with severe darkening of the appearance.

Evaporators need to be defrosted at regular intervals and this is usually timed to take place in the early morning. Build-up of frost on the evaporators can be limited by air-conditioning the shop area and so reducing the amount of moisture in the surrounding air.

20.5 Refrigeration for display

A supermarket will have a large number of coldrooms and display cabinets, all of which require refrigeration. The original method was, as with the domestic food freezer, to have a condensing unit as part of the cabinet. This arrangement in a supermarket would mean that the condenser heat would be given off in the shopping area. To avoid this, all condensing units are remote, usually in a central plantroom. Since suction and liquid piping must now pass between the many evaporators and the plantroom, one or a group of compressors can service a large number of units (see Figure 20.3).

A bank of compressors will be provided for each suction temperature, with a common condensing pressure. This arrangement is very flexible, with the compressors switched by logic controller to maintain correct conditions, regardless of the number of units working at any one time. The grouped condensers give the opportunity to recover heat from the discharge gas for water heating, and from the condensers in winter for heating the building (see also Chapter 30).

21 Refrigeration load estimation

21.1 Load sources

Refrigeration loads are from two sources:

1. To cool something down, i.e. reduce its enthalpy
2. To keep something cool, i.e. remove incoming and internally generated heat

The components of the total cooling load will be:

1. Removal of heat, sensible or latent, from a product
2. Heat conducted in through the surfaces of the room, tank, pipe, etc., from warmer surroundings
3. Radiant heat from outside
4. Heat convected from outside (air infiltration or ventilation), both sensible and latent
5. Internal sources of heat – lights, fan motors, machinery, personnel, etc. – and heat generated by the product

Some of these can be calculated fairly accurately from known data. Others have unknown parameters, so estimates are based on a combination of available data and practical experience.

21.2 Product cooling

The total amount of sensible and latent heat to be removed in cooling a product is given by:

$$H = M((c_a \times \Delta T_a) + h_l + (c_b \times \Delta T_b))$$

where H = total quantity of heat
M = mass of product
c_a = specific heat capacity above freezing

ΔT_a = temperature decrease above freezing
h_l = latent heat of freezing
c_b = specific heat capacity below freezing
ΔT_b = temperature decrease below freezing

Some of these components will be zero if cooling does not take place through the range of temperatures above and below the freezing point. Typical specific heat capacities, freezing points and latent heats are given in Table 21.1.

Table 21.1 Specific and latent heats of foodstuffs (typical values)

Product	*Specific heat capacity above freezing*	*Highest freezing point* (°C)	*Latent heat of freezing*	*Specific heat capacity below freezing*
Apples	3.65	– 1.1	280	1.89
Bananas	3.35	– 0.8	250	1.78
Beer	3.85	– 2.2	–	–
Cabbage	3.92	– 0.9	–	–
Carrots	3.79	– 1.4	294	1.94
Celery	3.99	– 0.5	–	–
Dairy products				
milk	3.75	– 0.6	–	–
butter	1.37	down to – 20	53	1.04
ice cream	2.95	– 6	210	1.63
cheese	2.1	– 13	125	1.3
Dried fruits	1.8	– 2		
Eggs, shell	3.05	– 2.2	220	1.67
Fish, white	3.55	– 2.2	270	1.86
blue	2.9	– 2.2	210	1.63
Meats, bacon	1.5	– 2	64	1.07
beef	3.2	– 2	230	1.7
ham	2.7	– 2	188	1.55
lamb	3	– 2	215	1.65
pork	2.6	– 2.5	125	1.3
poultry	3.3	– 2.8	246	1.77
Melons	3.95	– 0.9	310	2
Mushrooms	3.89	– 0.9	304	1.98
Onions	3.8	– 0.9	295	1.95
Oranges	3.75	– 0.8	–	–
Pears	3.62	– 1.6	–	–
Potatoes	3.5	– 0.7	265	1.84
Tomatoes	3.98	– 0.5	–	–

Many of these figures will be slightly different, according to the variety, breed or location of the product.

The rate of heat extraction, i.e. the product cooling load, will be:

$Q = H/t$ where t = the time available for cooling.

Example 21.1 What is the cooling duty to freeze water from 15°C to ice at 0°C, at the rate of 20 t/day?

$$Q = \frac{20\,000[4.187 \times 15] + 334}{24 \times 3600} = 92 \text{ kW}$$

Example 21.2 What duty is required to cool 8 t of lean meat (specific heat capacity 3.1 kJ/(kg K)) in 14 h from 22°C to 1°C?

$$Q = \frac{8000[3.1 \times (22 - 1)]}{14 \times 3600} = 10.3 \text{ kW}$$

There may be several unknown quantities in an estimate. For example, a dairy farm may produce 2400 litre/day (a rate of 100 litre/h), but this will come from two milkings, possibly 1400 litre in the morning and 1000 litre in the afternoon, and the milk must be cooled in 2 h, so the peak rate is 700 litre/h. The entering temperature of a product may be uncertain, being warmer in the summer or after a long journey. The dwell time within the cooling system may vary, beer leaving an instantaneous cooler at 4°C when first tapped, but at 12°C if drawn off continuously. The exact product may not be known – a general foodstuffs cold store might contain bacon (sensible heat capacity 2.4) or poultry (sensible heat capacity 3.3).

Observations may need to be taken of the operation, to form an estimate of unknown figures, or the process analysed to decide representative rates. Assumptions should be stated and agreed by the parties concerned, since these estimates are to form the basis for the selection of the required plant.

21.3 Conducted heat

Conducted heat is that going in through cold store surfaces, tank sides, pipe insulation, etc. It is normally assumed to be constant and the outside temperature an average summer temperature, probably 25–27°C for the UK, unless some other figure is known. Coldroom surfaces are measured on the outside dimensions and it is usual to calculate on the heat flow through the insulation only, ignoring other construction materials, since their thermal resistance is small.

Example 21.3 A coldroom measures 35 m long by 16 m wide and

is 5 m high inside. Insulation is 125 mm to walls and ceiling and 75 mm under the floor, of polystyrene having a thermal conductivity of 0.035 W/(m K). Inside it is at –10°C, the ambient is 27°C, and the ground temperature is 12°C. What is the heat flow inwards?

$$\text{Area of walls} = 5.2 \times 2(35.25 + 16.25) = 535.6 \text{ m}^2$$

$$\text{Area of ceiling} = 35.25 \times 16.25 = 572.8 \text{ m}^2$$

$$\text{Area of floor} = 572.8 \text{ m}^2$$

$$\text{Heat flow, walls} = 535.6 \times \frac{0.035}{0.125} \times [27 - (-10)] = 5549 \text{ W}$$

$$\text{ceiling} = 572.8 \times \frac{0.035}{0.125} \times [27 - (-10)] = 5935 \text{ W}$$

$$\text{floor} = 572.8 \times \frac{0.035}{0.075} \times [12 - (-10)] = 5881 \text{ W}$$

$$Q = 17\,365 \text{ W, say } 17.5 \text{ kW}$$

Solar radiation may fall on outside walls or roofs, raising the skin temperature, and this must be taken into account. Most cold stores are built within an outer envelope which protects them from the elements and from direct sunshine. In cases where the insulation itself is subject to solar radiation, an allowance of 5 K higher outside temperature should be taken. Heat load must be estimated through all surfaces including piping, ducts, fan casings, tank walls, etc., where heat flows inwards towards the cooled system.

Radiant heat is not a serious factor in commercial or industrial refrigeration systems, being confined to sunshine through refrigerated display windows (which should have blinds) and radiation into open shop display cabinets from lighting. (See also Chapter 26.)

21.4 Convected heat

Warm air will enter from outside mainly during the opening of doors for the passage of goods. This must be estimated on the basis of the possible use of the doors, and such figures are based on observed practice. The parameters are the size of the store, the enthalpy difference between inside and outside air, and the usage of the doors. The latter is affected by the existence of airlocks and curtains [49].

Standard textbooks give data on which to base an estimate, and this can be summed up as

$$Q_f = (0.7V + 2)\Delta T$$

where Q_f = heat flow
V = volume in m^3
ΔT = temperature difference between room and ambient

This is for cold rooms up to 100 m^3 with normal service. For heavy service, i.e. a great deal of traffic through the doors, this figure can be increased by 20–35%.

Rooms above 100 m^3 tend to be used for long-term storage, and are probably fitted with curtains (air or plastic, see Chapter 20). For such rooms, the service heat gain by convection may be taken as

$$Q_f = (0.125V + 27)\Delta T$$

Example 21.4 Estimate the infiltration air heat gain for the coldroom in Example 21.3.

$$\text{Volume} = 35 \times 16 \times 5 = 2800 \text{ m}^3$$

$$\Delta T = 27 - (-10) = 37 \text{ K}$$

$$Q_f = (0.125 \times 2800 + 27) \times 37$$

$$= 13\,950 \text{ W, or } 14 \text{ kW} \qquad \text{say (compare 13.9 kW) [1]}$$

The amount of outside air entering a refrigerated space may be seriously affected by unbalanced air supply to adjacent areas, causing short-circuiting of ambient air through the cooled space. Such possibilities should be investigated during a site survey. Cold store staff, such as loaders and fork-lift truck drivers, may operate more carefully while they are under observation but revert to less disciplined working at other times, adding considerably to door-opening times. Some allowance may need to be made for this.

21.5 Internal heat sources

The main sources of internal heat are fan motors and circulating pumps. Where the motor itself is within the cooled space, the gross energy input to the motor is liberated as heat which must be removed. Where the motor is outside, only the shaft power is taken.

Other motors and prime movers may be present – conveyors, lifts, fork-lift trucks, stirrers, injection pumps, packaging machines, etc. The gross power input to these machines may be read from their nameplates or found from the manufacturers.

Personnel will give off about 120 W each.

All lighting within the space must be included on the basis of the gross input. The usual 80-W lighting tube takes about 100 W gross. Where the lighting load heat input is seen to be a large proportion

of the total, it is probable that the lighting system has been poorly designed, and some alterations may be necessary. (See also Section 15.9.)

Where coolers are fitted with defrosting devices, the heat input from this source must be determined.

Example 21.5 The coldroom in Example 21.3 has 12 lighting fittings labelled 280 W. The four evaporators each have three fan motors of 660 W gross per fan and 18 kW defrost heaters which operate alternately for 15 min twice a day. The fork-lift truck is rated 80 A at 24 V and will be in the store 20 min each hour during the 8-h working day. Two packers will be present for 10 min each hour. Estimate the average and peak loads (see Table 21.2).

Table 21.2

	Average over 24 h	*Peak*
Lighting, 12 × 280, 8 h/day	1.12	3.36
Fan motors, 12 × 660 W	7.78	7.92
Defrost heaters, 72 kW, $\frac{1}{2}$ h/day	1.50	18.00
Fork-lift, 1.92 kW, $\frac{1}{3}$ × 8 h	0.21	1.92
Fork-lift driver, 120 W, $\frac{1}{3}$ × 8 h	–	0.12
Packers, 240 W, $\frac{1}{6}$ × 8 h	–	0.24
	10.61	31.56

This example shows that the greatest load is the fan motors, since these run all the time, except during defrosting. There are several unknowns. For example, it is assumed that the defrosting of the evaporators will not coincide, but this may occur if badly timed, and cause a peak load which may raise the store temperature for a time. The last two items can be ignored, making the loak 11 kW average. However, the greatest heat input is still the fan motors, which indicates that any reduction in this component of the load, possibly by switching off two evaporators at night, can appreciably reduce the energy requirements, in terms of both the electricity input and the cooling load to take this heat out again.

21.6 Heat of respiration

Certain stored foodstuffs are living organisms and give off heat as their sugar or starch reserves are slowly consumed. This is known as the heat of respiration, since the products consume oxygen for the

process. The heat of respiration varies with the sugar or starch content of the product, the variety, and its temperature, and is between 9 and 120 W/t at storage temperatures. Typical figures are shown in Table 21.3. These figures increase with temperature, roughly doubling for every 10 K, so that fruits and many vegetables deteriorate very rapidly if they are warm, using up their food reserves and then decaying [29, 33, 34].

Table 21.3

Product	*Temperature* (°C)	*Heat of respiration* (W/t)
Apples	2	12
Pears	1	16
Bananas	13	48
Strawberries	0	45
Potatoes	1.5	9

21.7 Estimate analysis

It is frequently the case that very little definite information is available on which to base a heat load estimate. In these circumstances, the probable minimum and maximum should be calculated from the best available data and an average decided and agreed with the user.

Example 21.6 A dockside frozen meat store has a capacity of 1000 t stored at – 12°C, and leaving the store at a maximum rate of 50 t/day. Meat may arrive from a local abattoir at 2°C or from ships in batches of 300 t at – 10°C. Estimate a product cooling load.

Case 1
Meat goes out at the rate of 350 t/week and may arrive from local supplies. There is possibly a four-day week, allowing for odd holidays, and so there may be 90 t/day from the abattoir. Cooling load is 90 t/day from 2°C to – 12°C. Tables give the following:

Specific heat capacity above – 1°C = 3.2 kJ/(kg K)

Freezing point of meat, average = –1.0°C

Latent heat of freezing = 225 kJ/kg

Specific heat of frozen meat = 1.63 kJ/(kg K)

$$Q_f = \frac{90\,000}{24 \times 3\,600}\,[(3.2 \times 3) + 225 + (1.63 \times 11)] = 263 \text{ kW}$$

Case 2
Shipments may come in on consecutive days (unlikely, but possible if store is almost empty):

$$Q_f = \frac{300\,000}{24 \times 3\,600}\,(1.63 \times 2) = 11 \text{ kW}$$

These show a wide variation. Since meat will keep for several days at 2°C, rework case 1 on the basis of a steady input of 50 t/day all coming from the abattoir.

Case 3

$$Q_f = \frac{50\,000}{24 \times 3\,600}\,[(3.2 \times 3) + 225 + (1.63 \times 11)] = 146 \text{ kW}$$

It would seem, then, that the minimum safe cooling capacity required is 146 kW, with the possible risk of 263 kW for a day or so. Most of the time the load will be much less.

A practical approach would be to install plant having a maximum product-cooling capacity of 146 kW (to which must be added the other load components of heat leakage, internal heat, and service). After an estimate of the total cooling load has been formed, this must be converted into a refrigeration plant capacity.

General practice, after having calculated the average load over a period of 24 h, is to take the absolute maximum, or allow 50% over the average, i.e. a plant running time of 16 h in the 24. This general rule must be assessed for the particular application.

Example 21.7 The milk-cooling requirement (above) of 700 litre/h is a maximum rate. There is no need to allow for any more than this, but it cannot be any less. Alternatively, this could be cooled using an ice bank, in which case the total load of 2400 litre could be spread over 16 h of running time. With an allowance for water tank insulation heat gains and an ice water pump, the load might be reduced to a refrigeration plant one-third the size.

Example 21.8 The meat-cooling load in Example 21.2 is probably a daily batch from an abattoir and the duty will be less at night, once the meat is cooled. The maximum capacity will therefore be 10.3 kW, plus the fans and other room losses, and the plant will run continuously while the meat is being chilled only.

All assumptions regarding the load and estimated cooling duty should be recorded as the design parameters of the system, and agreed with the user.

Example 21.9 The cold store in Example 21.3 is now to be located in an ambient of 35°C, and to have the internal load of Example 20.5 and the product load of case 3 of Example 21.6. Include for infiltration and estimate plant capacity.

Answer

Product cooling load, from Example 21.6 case 3. Allow another 5% for higher ambient, in case meat warms up in transit from abattoir = 153 kW

Heat gain through walls, etc., as Example 21.3 but corrected for 35°C:

wall	6749 W
ceiling	7218 W
floor	7733 W
	21 700 W

$$21\,700 \times \frac{24}{16} = 33 \text{ kW}$$

Service or convection gains:

$$[(0.125 \times 2800) + 27]45 \times \frac{1}{1000} \times \frac{24}{16} = 25 \text{ kW}$$

Internal heat gains as Example 21.5

$$10.5 \times \frac{24}{16} = 16 \text{ kW}$$

Total plant capacity = 227 kW

These estimate figures should be included as part of the contract documents for the purchase of the plant.

22 Industrial uses of refrigeration

22.1 Air conditioning

The widest application of the refrigeration process is to provide cooling for air-conditioning. The majority of this is for personal comfort in hot climates or where heat is given off in enclosed spaces. There is an additional demand for industrial manufacturing processes where precise conditions of temperature, humidity and cleanliness are necessary.

The physical principles of air-conditioning, its methods of application and the suitable apparatus are the subject of Chapters 23–28.

22.2 Chilled liquids for cooling

The use of chilled water or a non-freeze solution for heat transfer is now replacing many applications where direct expansion of refrigerant has been used in the past. The method gives the advantage of using packaged liquid chillers.

Uses in the dairy and beverage and other food industries have already been mentioned in previous pages. Other uses are:

1. Cooling of butcheries and meat-slicing rooms, with brine coils
2. Cooling of multi-room cold stores at different temperatures, with brine coils
3. Cooling the moulds of plastic-moulding machines with chilled water

The list of such applications is extended with developing technologies.

22.3 Solvent recovery

Large quantities of solvent liquids are used in industrial and commercial processes and any loss of these creates an environmental hazard, apart from the cost of the material itself.

All these solvents are volatile liquids and will have a pressure–temperature characteristic (see Section 1.2), so can be condensed if cooled to their saturation temperature. Finned-tube evaporators are generally used, but the condensation may be at a high pressure, requiring heat exchangers of the shell-and-tube type.

The size of equipment can vary from a 200 W unit for a commercial dry-cleaning machine to systems of megawatt size for synthetic fibre processes.

22.4 Low-temperature liquid storage and transport

Many volatile liquids can only be stored or transported at reduced temperatures, or excessive pressures will build up in the vessel. The important application is in the storage and transport of liquid methane, at temperatures of around – 250°C. The types of refrigeration apparatus for this duty lie outside the scope of this book.

Liquid carbon dioxide has many industrial uses and is stored at power stations for purging boiler furnaces and in oil tankers to purge petrol tanks. The vapour pressure of carbon dioxide is high, and storage vessels might possibly reach the critical temperature of 31°C. Storage temperatures of – 20° to – 4°C are in use, corresponding to vessel pressures of 19–30 bar. Single-stage refrigeration systems are used, with the evaporator coil inside the insulated storage vessel. For safety, most cooling systems are in duplicate.

The bulk transport of volatile liquids such as ammonia can be in insulated, unrefrigerated tanks, providing the liquid is cold on entry and the journey time is limited.

22.5 Dewaxing of oils

Impurities may be removed from lubricating oils in the same way that wines and spirits are cooled and filtered (see Section 17.4). The base liquid is cooled down to a temperature at which the impurity will solidify, and then passed through a filter to take out the solids. The general principle is applied to many manufacturing and refining processes. The pre-cooling of the base liquid and its subsequent re-heating can be achieved by counterflow heat exchangers, as in the pasteurization and cooling of milk (see Figure 17.1). Most waxes have a byproduct value, and it may be necessary to chill them in a warm climate, to set the wax into blocks for packaging.

22.6 Ice rinks

Artificial ice rinks are frozen shallow ponds, formed and maintained using a brine in tubes buried under the surface. Tubes may be steel or plastic for a permanent rink or plastic for a temporary installation. The brine temperature within the pipes will be about – 11°C, and must be lower for rinks in the open air, owing to high solar radiation loads. Packaged liquid chillers are now generally used, and will be transportable, complete with brine pumps and other apparatus, for temporary installations.

22.7 Cooling concrete

The setting of concrete is an exothermic reaction, and large masses of concrete in building foundations, bridges and dams will heat up, causing expansion cracks if not checked. To counteract this heating, the materials are cooled before and as they are mixed, so that the concrete is laid some 15 K colder than ambient, and warms to ambient on setting. In practice, the final mix temperature can be held down to 10°C.

Methods are to pre-cool the aggregate with cold air, to chill the mix water, and to provide part of the mix in the form of flake ice. Chilled water pipes may be buried in the concrete mass.

22.8 Ground freezing

In mining and, more recently, the construction of underground storage tanks for liquefied natural gas, it is often necessary to sink a shaft through water-logged ground. The requirement is to form a temporary cofferdam to permit excavation and the building of a permanent liner.

The general method is to drive in a ring of vertical pipes and pass chilled brine down through an inner pipe so that it flows up the annulus, to cool and eventually freeze the surrounding wet soil. This process is continued until the ice builds up a continuous wall around the proposed excavation. Depths of over 650 m have been excavated in this way. Calcium chloride brine, cooled by surface plant, is usual, but liquid nitrogen has been used on small shafts [50].

22.9 Low-temperature testing

Mechanisms and electronics for the aerospace industry are tested at temperatures which may prevail under working conditions. A

typical specification might be to test at – 70°C. Where the component is large, it must be contained within a cold chamber which is capable of reaching this condition. The major organizations have this type of facility.

Smaller items are tested in self-contained cabinets with a chamber the size of a large domestic refrigerator. Two-stage and three-stage systems are used, with R.13 in cascade at the lower end and R.22 for the high stage(s).

Some metals change their structure, or maintain an annealed condition, at low temperature, and this may be used as part of a manufacturing process.

22.10 Chemical industry

Processes in the chemical industry require the control of temperatures of reactions where heat is liberated. Direct expansion refrigerant coils may constitute a hazard, and such heat exchangers generally use chilled water or brine. Coolers of this sort will be found in every branch of the chemical industry.

Piston, screw and centrifugal compressors are used. As many chemical processes, such as oil refining, may have cheap waste heat, large absorption systems will also be found.

Since continuity of the process and safety are prime considerations, plant security will require duplication of all items of apparatus so that a temporary shut-down for repair or maintenance will not reduce the cooling capacity.

23 Air and water vapour mixtures

23.1 General

The atmosphere consists of a mixture of dry air and water vapour. Air is itself a mixture of several elemental gases, mainly oxygen and nitrogen, but the proportions of these are consistent throughout the atmosphere and it is convenient to consider air as one gas. This has a *molecular mass* of 28.97 and the standard atmospheric pressure is 1013.25 mbar or 101 325 Pa.

Water may be present in air in the liquid form, as rain or mist, or as a solid (snow, hail). However, in general ambient and indoor conditions the water present in the air will be in the vapour form, i.e. as superheated low-pressure steam.

23.2 Calculation of properties

If air and water are present together in a confined space, a balance condition will be reached where the air has become saturated with water vapour. If the temperature of the mixture is known, then the pressure of the water vapour will be the pressure of steam at this temperature (see also Section 1.3) (Table 23.1). Dalton's Law of partial pressures (see also Section 1.5) states that the total pressure of a mixture of gases is equal to the sum of the individual pressures of the constituent gases, taken at the same temperature and occupying the same volume. Since the water saturation vapour pressure will remain constant, depending on temperature and not on volume, this pressure can be obtained from steam tables as below. The partial pressure exerted by the dry air must therefore be the remainder.

Thus, for an air–water vapour mixture at 25°C:

Table 23.1

Temperature (°C)	*Vapour pressure* (mbar)
0	6.10
10	12.27
15	17.04
20	23.37
25	31.66

Total (standard) pressure = 1013.25 mbar
Partial pressure of saturated vapour = 31.66 mbar

Partial pressure of dry air = 971.59 mbar

This calculation of the proportions by partial pressure can be converted to proportions by weight, by multiplying each pressure by the molecular mass (Avogadro's hypothesis), to give:

Proportion by mass of water = $31.66 \times 18.016 = 570.4$
Proportion by mass of dry air = $971.59 \times 28.97 = 28\,146$

$$\text{Proportion by weight of } \frac{\text{water}}{\text{dry air}} = \frac{570.4}{28146} = 0.020\,3 \text{ kg/kg}$$

Since neither dry air nor water vapour is a perfect gas, there will be a slight difference between published tables [4] (0.020 16) and this simplified calculation.

The specific enthalpy (or total heat) of the mixture can be taken from 0 K (– 273.15°C) or from any convenient arbitrary zero. Since most air-conditioning processes take place above the freezing point of water, and we are concerned mostly with differences rather than absolute values, this is commonly taken as 0°C, dry air. For conditions of 25°C, saturated, the specific enthalpy of the mixture, per kilogram of dry air, is

Sensible heat of dry air = $1.006 \times 25 = 25.15$ kJ/kg
Sensible heat of water = $0.020\,16 \times 25 \times 4.187 = 2.11$
Latent heat of water = $0.020\,16 \times 2440 = 49.19$

Total 76.45 kJ/kg

(Again, there are some slight variations in these properties within the range considered, and the published figure [4] is 76.49 kJ/kg.)

The specific volume of the mixture can be obtained, taking either of the two gases at their respective partial pressures, and using the General Gas Law. Only basic SI values must be used, so the pressures must be expressed in pascals:

$$pV = mRT$$

$$\text{or} \quad V = mRT/p$$

$$\text{For the dry air } V_a = \frac{1 \times 287 \times (25 + 273.15)}{97\,159}$$

$$= 0.880\,7 \text{ m}^3$$

$$\text{For the water vapour } V_v = \frac{0.020\,16 \times 461 \times (25 + 273.15)}{3166}$$

$$= 0.875\,2 \text{ m}^3$$

(The published figure is 0.871 5 m^3/kg) [4].

23.3 Moisture content, percentage saturation, and relative humidity

The moisture content in the example at 25°C, saturated, was given in standard tables as 0.020 16 kg/kg dry air. This is also termed its specific humidity.

Air will not always be saturated with water vapour in this way, but may contain a lower proportion of this figure, possibly 50%:

$$\frac{0.020\,16}{2} = 0.010\,08 \text{ kg/kg dry air}$$

This lower figure can be expressed as a percentage of the saturation quantity:

$$\text{Percentage saturation} = 100 \times \frac{g}{g_{ss}}$$

$$= 100 \times \frac{0.010\,08}{0.020\,16}$$

$$= 50\% \text{ sat.}$$

Properties for this new mixture can be calculated as above to obtain the specific enthalpy and specific volume.

The proportion of moisture can also be expressed as the ratio of the vapour pressures, and is then termed *relative humidity:*

$$\text{Relative humidity} = 100 \times \frac{P_s}{P_{ss}}$$

$$= 50.8\% \text{ relative humidity}$$

(for the example taken)

Since most air-conditioning calculations are based on weights of air and moisture, *percentage saturation* is usually employed, and moisture content is expressed as kilograms per kilogram of dry air. Much published data is still expressed in the original quantities of Willis H. Carrier, i.e. grains per pound, where 1 lb = 7000 grains.

23.4 Dew point

Saturated air at 25°C, having a water vapour content of 0.020 16 kg/kg, can be shown as a point *A* on a graph of moisture content against temperature (Figure 23.1). Air which is 50% saturated at this temperature will contain 0.010 08 kg/kg and will appear on this graph as point *B*. If this 50% saturation mixture is slowly cooled, the change of condition will be along the line *BC*, with constant moisture content but decreasing temperature. It will eventually reach point *C* on the saturation line, where the maximum moisture it can hold is 0.010 08 kg/kg (about 14.2°C). It cannot be cooled below this temperature and still hold this proportion of water vapour, so moisture will be precipitated as dew. The point *C* for the mixture originally at *B* is termed the *dew point temperature*.

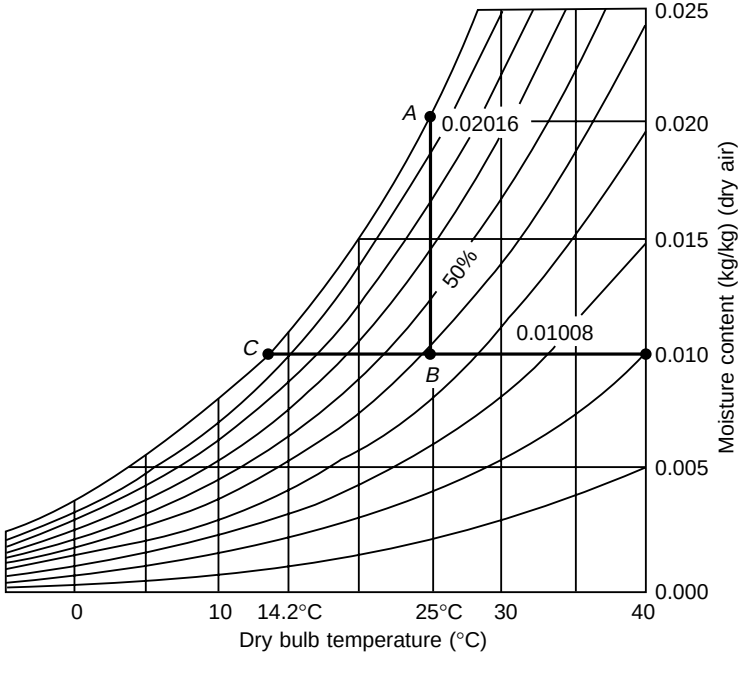


Figure 23.1 *Temperature–moisture content graph*

23.5 Wet bulb temperature

If the percentage saturation of an air sample is less than 100, i.e. it is less than saturated, and it comes into contact with water at the same temperature, there will be a difference in vapour pressures. As a result, some of the water will evaporate. The latent heat required for this change of state will be drawn from the sensible heat of the water, which will be slightly cooled. This drop in the water temperature provides a temperature difference, and a thermal balance will be reached where the flow of sensible heat from the air to the water (Figure 23.2) provides the latent heat to evaporate a part of it.

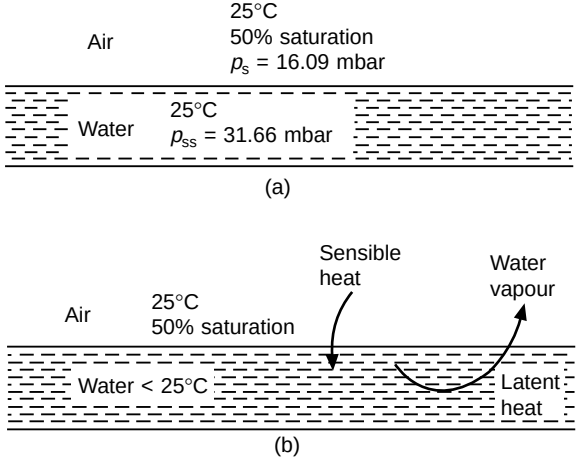


Figure 23.2 *Exchange of sensible and latent heat at water–air surface*

The effect can be observed and measured by using two similar thermometers (Figure 23.3), one of which has its bulb enclosed in a wet wick. The drier the air passing over them, the greater will be the rate of evaporation from the wick and the greater the difference between the two readings. In the case of air at 25°C, 50% saturation, the difference will be about 6.5 K. The measurements are termed the *dry bulb* and *wet bulb* temperatures, and the difference the *wet bulb depression*.

In order that consistent conditions can be obtained, the air speed over the thermometers should be not less than 1 m/s. This can be done with a mechanical aspiration fan (the Assmann psychrometer) or by rotating the thermometers manually on a radius arm (the sling psychrometer). If the thermometers cannot be in a moving airstream, they are shielded from draughts by a perforated screen and rely only on natural convection. In this case the wet bulb

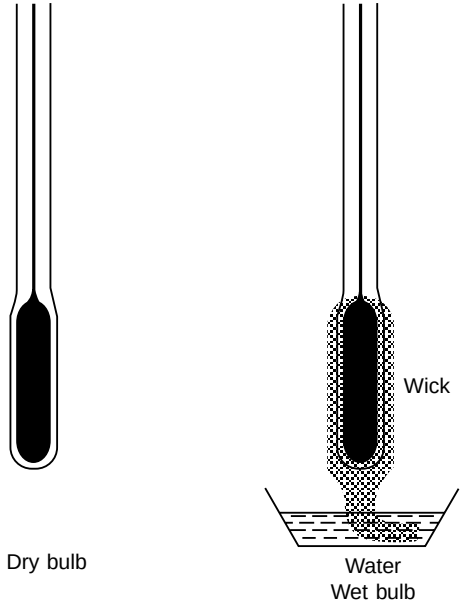


Figure 23.3 *Thermometers, dry bulb and wet bulb*

depression will be less and the reading is termed the *screen wet bulb* (see Figure 23.4).

It follows that the drier the air, the greater will be the difference between the dry bulb, wet bulb and dew point temperatures and, conversely, at 100% saturation these three will coincide.

23.6 The psychrometric chart

All the above properties may be tabulated, but can be displayed more effectively in graphical form. The basic properties to be shown are dry bulb temperature, moisture content and specific enthalpy. Within the limits of the graph required for ordinary air-conditioning processes, the grid lines can be assumed as parallel and form the basis of the psychrometric chart (Figure 23.5). (It will be seen from the full chart, Figure 23.6, that the dry bulb lines are slightly divergent. The moisture content and enthalpy grids are parallel.)

On this chart, the wet bulb temperatures appear as diagonal lines, coinciding with the dry bulb at the saturation line. If measurements are taken with the two thermometers of the sling psychrometer, the condition can be plotted on the psychrometric chart by taking the intersection of the dry bulb temperature, as read on the vertical line, with the wet bulb temperature, read down the diagonal wet bulb line.

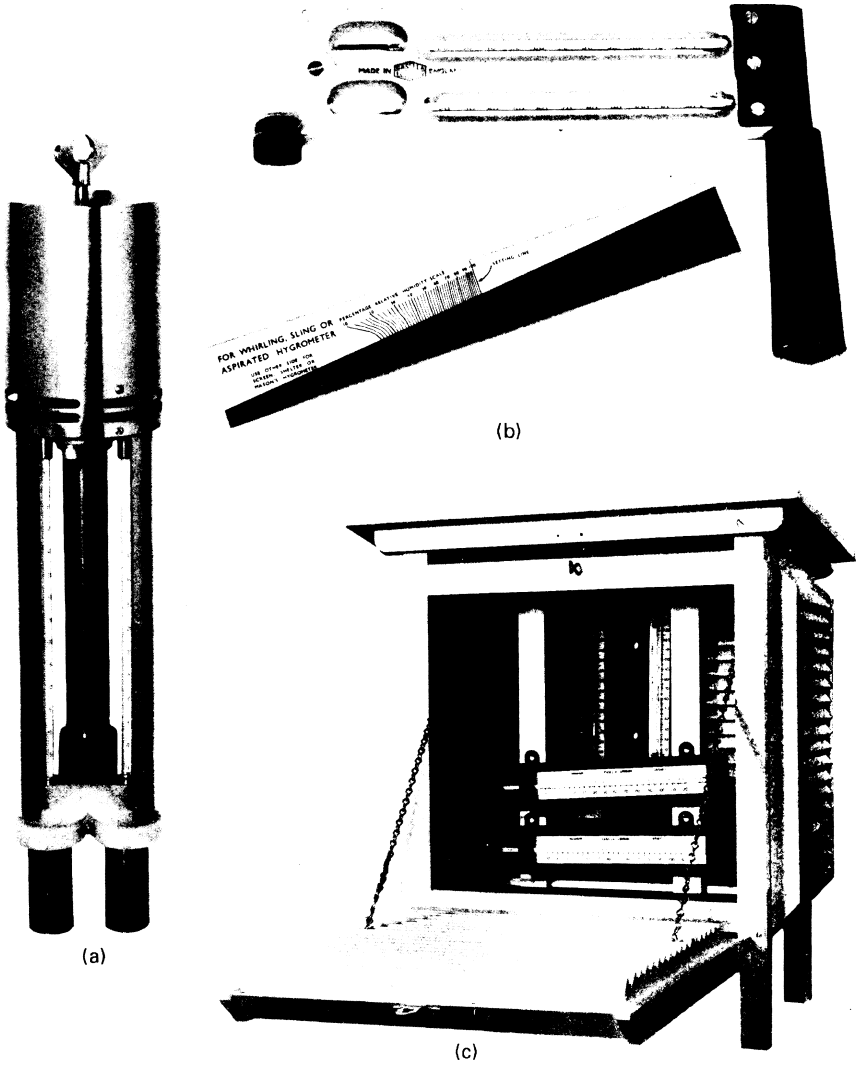


Figure 23.4 *Psychrometers. (a) Assmann. (b) Sling. (c) Screen housing (Courtesy of Casella London Ltd)*

The specific enthalpy will increase with dry bulb (sensible heat of the air) and moisture content (sensible and latent heat of the water). The adiabatic (isoenthalpic) lines for an air–water vapour mixture are almost parallel with the wet bulb lines so, to avoid any confusion, the enthalpy scale is placed outside the body of the chart, and readings must be taken using a straight-edge. (See Figure 23.7.)

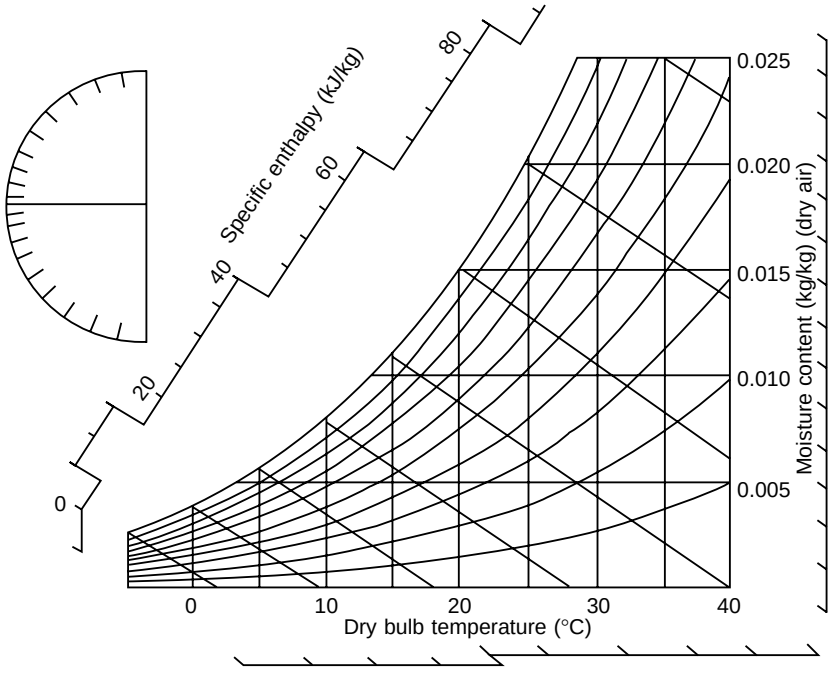


Figure 23.5 *Basic CIBSE psychrometric chart (Courtesy of the Chartered Institution of Building Services Engineers)*

A further property which is shown on the psychrometric chart is the specific volume of the mixture, measured in cubic metres per kilogram. This appears as a series of diagonal lines, at intervals of 0.01 m^3.

23.7 Effects on human comfort

The human body takes in chemical energy as food and drink, and oxygen, and consumes these to provide the energy of the metabolism. Some mechanical work may be done, but the greater proportion is liberated as heat, at a rate between 90 W when resting and 440 W when doing heavy work.

A little of this is lost by radiation if the surrounding surfaces are cold and some as sensible heat, by convection from the skin. The remainder is taken up as latent heat of moisture from the respiratory tissues and perspiration from the skin (see Table 23.2). Radiant loss will be very small if the subject is clothed, and is ignored in this table.

Convective heat loss will depend on the area of skin exposed, the air speed, and the temperature difference between the skin and the

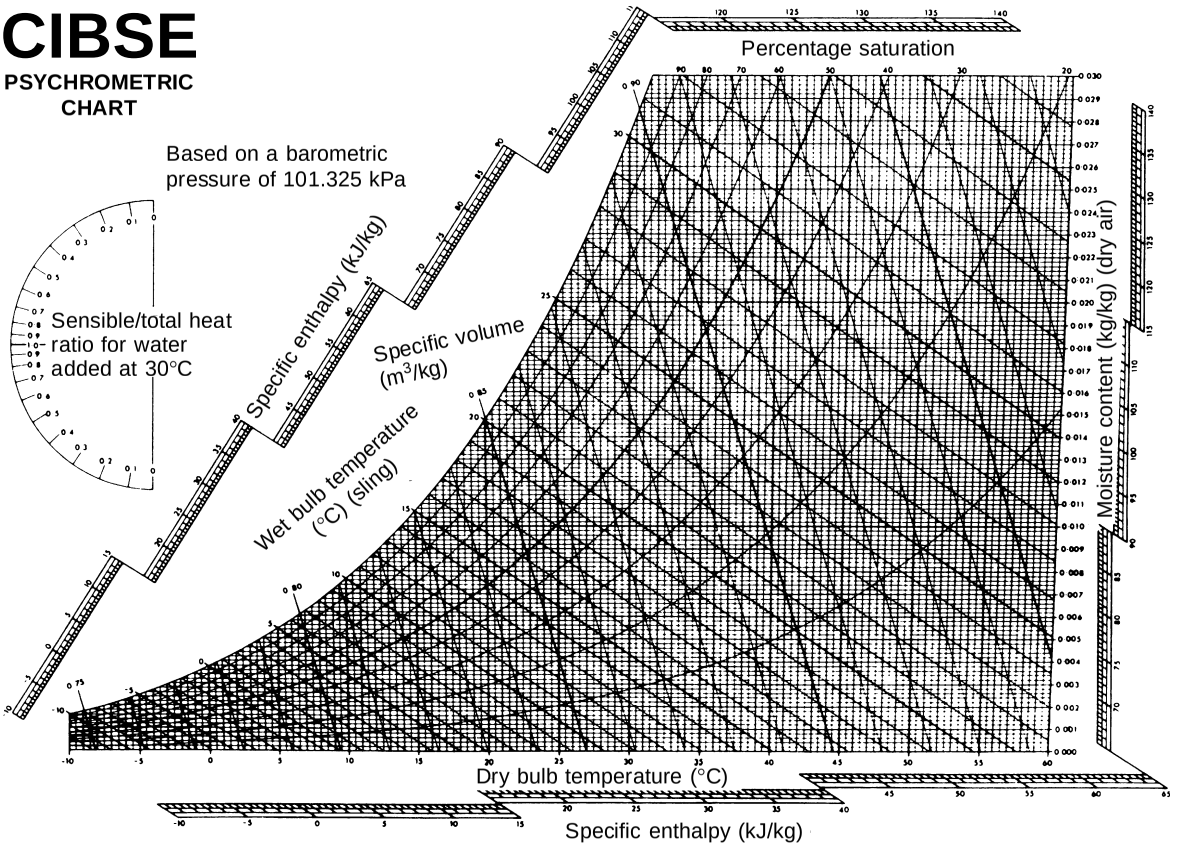


Figure 23.6 *Psychrometric chart*

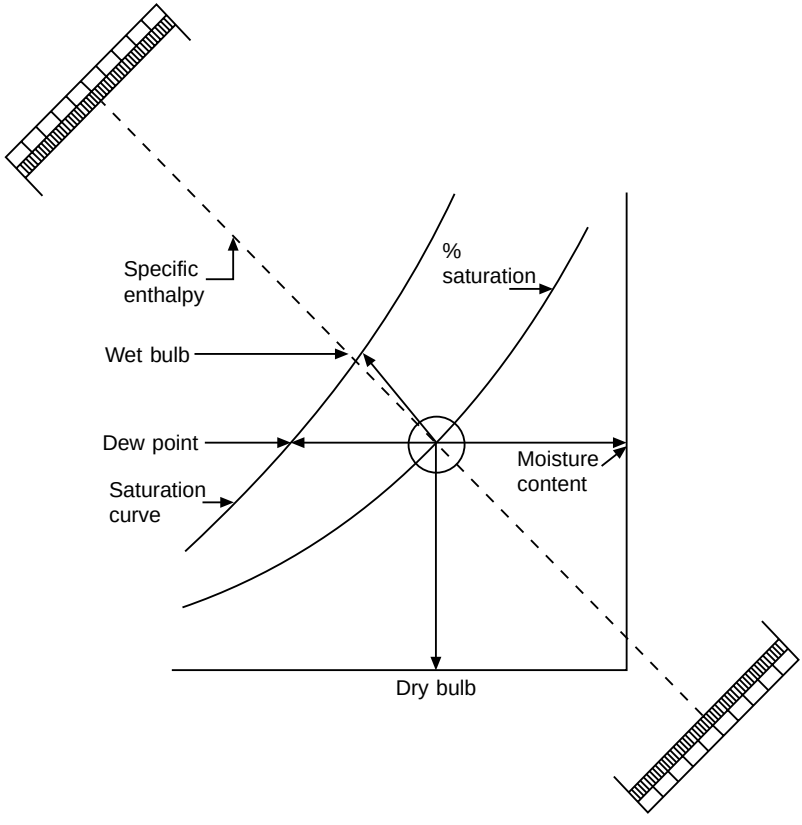


Figure 23.7 *Reading the CIBSE psychrometric chart*

ambient. As the dry bulb approaches body temperature (36.9°C) the possible convective loss will diminish to zero. At the same time, loss by latent heat must increase to keep the body cooled. This, too, must diminish to zero when the wet bulb reaches 36.9°C.

In practice, the human body can exist in dry bulb temperatures well above blood temperature, providing the wet bulb is low enough to permit evaporation. The limiting factor is therefore one of wet bulb rather than dry bulb temperature, and the closer the upper limits are approached, the less heat can be rejected and so the less work can be done.

23.8 Climatic conditions

Figure 23.8 shows the maximum climatic conditions in different areas of the world. The humid tropical zones have high humidities but the dry bulb rarely exceeds 35°C. The deserts have an arid

Table 23.2 Heat emission from the human body (adult male, body surface area 2 m^2) (From CIBSE Guidebook A)

Application	*Sensible* (s) *and latent* (l) *heat emissions, W, at the stated dry bulb temperature* (°C)							
			20		22		24	
Degree of activity	*Typical*	*Total*	(s)	(l)	(s)	(l)	(s)	(l)
Seated at rest	Theatre, hotel lounge	115	90	25	80	35	75	40
Light work	Office, restaurant	140	100	40	90	50	80	60
Walking slowly	Store, bank	160	110	50	100	60	85	75
Light bench work	Factory	235	130	105	115	120	100	135
Medium work	Factory, dance hall	265	140	125	125	140	105	160
Heavy work	Factory	440	190	250	165	275	135	305

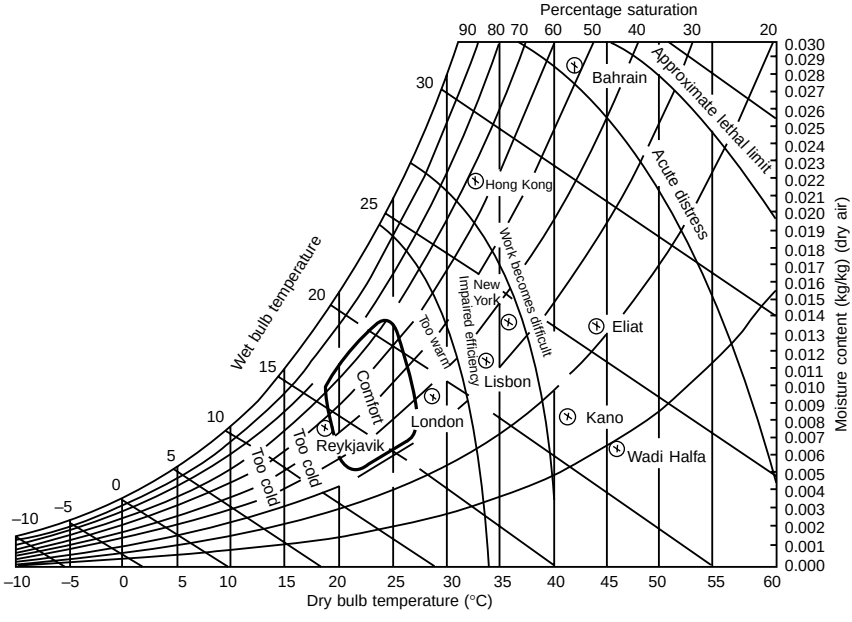


Figure 23.8 *Typical climate conditions*

climate, with higher dry bulb temperatures. Approximate limits for human activities are related to the enthalpy lines and indicate the ability of the ambient air to carry away the 90–440 W of body heat.

The opposite effect will take place at the colder end of the scale. Evaporative and convective loss will take place much more easily and the loss by radiation may become significant, removing heat faster than the body can generate it. The rate of heat production can be increased by greater bodily activity, but this cannot be sustained, so losses must be prevented by thicker insulation against convective loss and reduced skin exposure in the form of more clothing. The body itself can compensate by closing sweat pores and reducing the skin temperature.

23.9 Other comfort factors

A total assessment of bodily comfort must take into account changes in convective heat transfer arising from air velocity, and the effects of radiant heat gain or loss. These effects have been quantified in several objective formulas, to give equivalent, corrected effective, globe, dry resultant and environmental temperatures, all of which give fairly close agreement. This more complex approach is required where air speeds may be high, there is exposure to hot or cold surfaces, or other special conditions call for particular care.

For comfort in normal office or residential occupation, with percentage saturations between 35 and 70%, control of the dry bulb will result in comfortable conditions for most persons. Feelings of personal comfort are as variable as human nature and at any one time 10% of the occupants of a space may feel too hot and 10% too cold, while the 80% majority are comfortable. Such variations frequently arise from lack or excess of local air movement, or proximity to cold windows, rather than an extreme of temperature or moisture content.

23.10 Fresh air

Occupied spaces need a supply of outside air to provide oxygen, remove respired carbon dioxide, and dilute body odours and tobacco smoke. The quantities are laid down by local regulations and commonly call for 6–8 litre/s per occupant. Such buildings are usually required also to have mechanical extract ventilation from toilets and some service areas, so the fresh air supply must make up for this loss, together with providing a small excess to pressurize the building against ingress of dirt [2].

24 Air treatment cycles

24.1 Winter heating

Buildings lose heat in winter by conduction out through the fabric, convection of cold air, and some radiation. The air from the conditioning system must be blown into the spaces warmer than the required internal condition, to provide the heat to counteract this loss.

Heating methods are as follows:

1. Hot water or steam coils
2. Direct-fired – gas and sometimes oil
3. Electric resistance elements
4. Refrigerant condenser coils of heat pump or heat reclaim systems

Figure 24.1 shows the sensible heating of air.

Example 24.1 Air circulates at the rate of 68 kg/s and is to be heated from 16°C to 34°C. Calculate the heat input and the water mass flow for an air heater coil having hot water entering at 85°C and leaving at 74°C.

$$Q = 68 \times 1.02 \times (34 - 16) = 1248 \text{ kW}$$

$$m_w = \frac{1248}{4.187 \times (85 - 74)} = 27 \text{ kg/s}$$

Note: the 1.02 here is a general figure for the specific heat capacity of indoor air which contains some moisture, and is used in preference to 1.006, which is for dry air.

Example 24.2 A building requires 500 kW of heating. Air enters the heater coil at 19°C at the rate of 68 kg/s. What is the air-supply temperature?

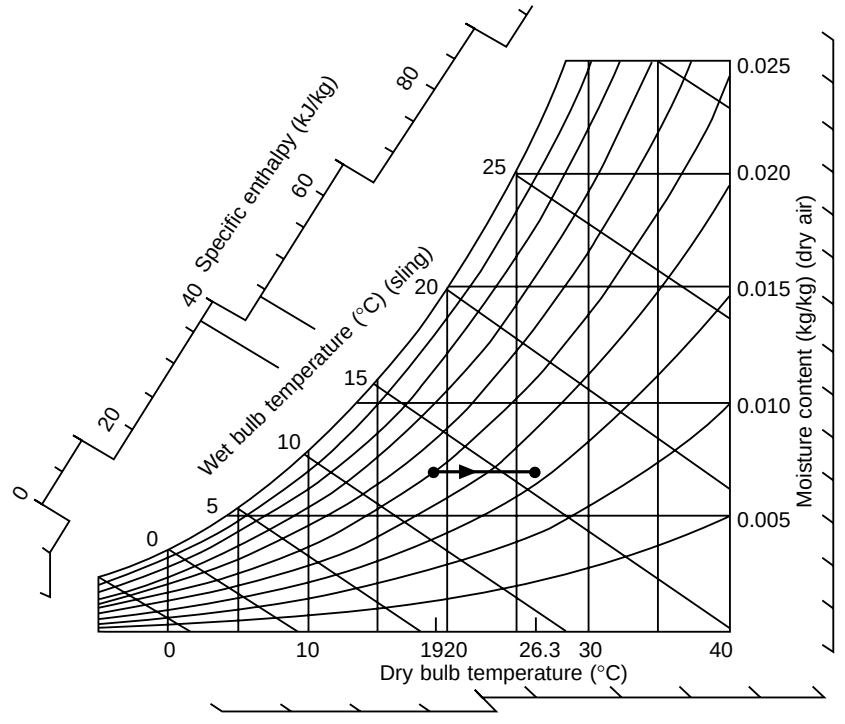


Figure 24.1 *Sensible heating of air*

$$t = 19 + \frac{500}{68 \times 1.02} = 19 + 7.2$$

$$= 26.2°\text{C}$$

If the cycle is being traced out on a psychrometric chart, the enthalpy can be read off for the coil inlet and outlet conditions. In Example 24.1, the enthalpy increase as measured on the chart is 7.35 kJ/kg dry air (taken at any value of humidity), giving

$68 \times 7.35 \sim 500$ kW

24.2 Mixing of airstreams

Air entering the conditioning plant will probably be a mixture of return air from the conditioned space and outside air. Since no heat or moisture is gained or lost in mixing,

Sensible heat before = sensible heat after

and

Latent heat before = latent heat after

The conditions after mixing can be calculated, but can also be shown graphically by a mix line joining the condition *A* and *B* (see Figure 24.2). The position *C* along the line will be such that

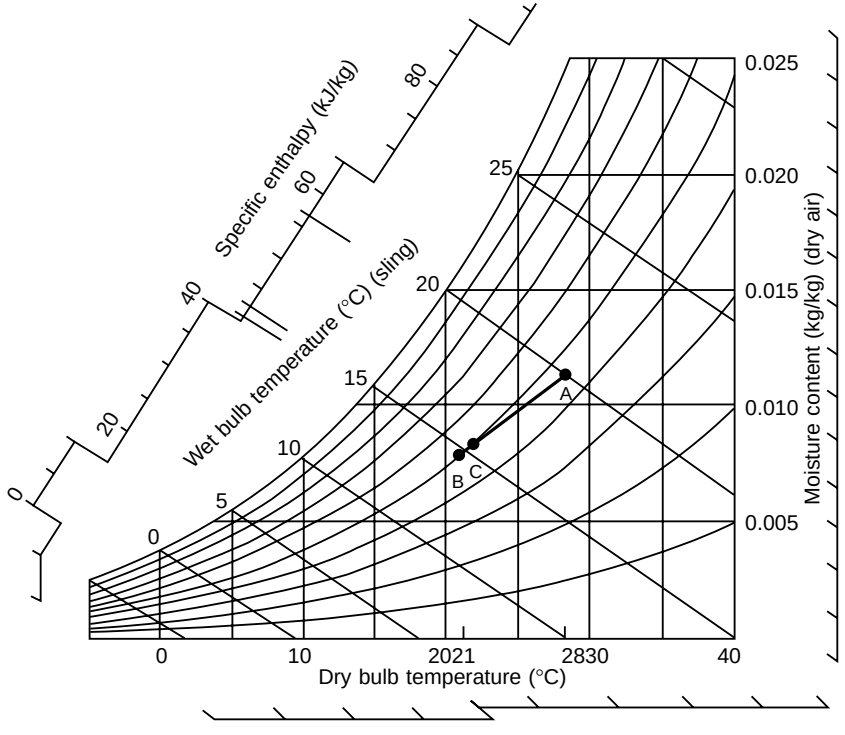


Figure 24.2 *Mixing of two airstreams*

$AC \times m_a = CB \times m_b$

This straight-line proportioning holds good to close limits of accuracy. The horizontal divisions of dry bulb temperature are almost evenly spaced, so indicating sensible heat. The vertical intervals of moisture content indicate latent heat.

Example 24.3 Return air from a conditioned space at 21°C, 50% saturation, and a mass flow of 20 kg/s, mixes with outside air at 28°C dry bulb and 20°C wet bulb, flowing at 3 kg/s. What is the condition of the mixture?

Method (*a*) Construct on the psychrometric chart as shown in Figure 24.2 and measure off:

Answer = 22°C dry bulb, 49% sat.

Method (*b*) By calculation, using dry bulb temperatures along the horizontal component, and moisture content along the vertical. For the dry bulb, using

$$AC \times m_a = CB \times m_b$$

$$(t_c - 21) \times 20 = (28 - t_c) \times 3$$

giving

$$t_c = 21.9°C$$

The moisture content figures, from the chart or from tables, are 0.0079 and 0.0111 kg/kg at the return and outside conditions, so

$$(g_c - 0.0079) \times 20 = (0.0111 - g_c) \times 3$$

giving

$$g_c = 0.0083 \text{ kg/kg}$$

If only enthalpy is required, this can be obtained from the same formula in a single equation:

$$(h_c - h_a) \times m_a = (h_b - h_c) \times m_b$$

$$(h_c - 41.8) \times 20 = (56.6 - h_c) \times 3$$

giving

$$h_c = 43.7 \text{ kJ/kg dry air}$$

Readers will recognize that the calculation methods lend themselves to computing.

24.3 Sensible cooling

If air at 21°C dry bulb, 50% saturation, is brought into contact with a surface at 12°C, it will give up some of its heat by convection. The cold surface is warmer than the dew point, so no condensation will take place, and cooling will be sensible only (Figure 24.3).

This process is shown as a horizontal line on the chart, since there is no change in the moisture content. The loss of sensible heat can be read off the chart in terms of enthalpy, or calculated from the dry bulb reduction, considering the drop in the sensible heat of both the dry air and the water vapour in it.

24.4 Water spray (adiabatic saturation)

The effect of spraying water into an airstream will be as shown in Figure 23.2, assuming that the air is not already saturated. Evaporation

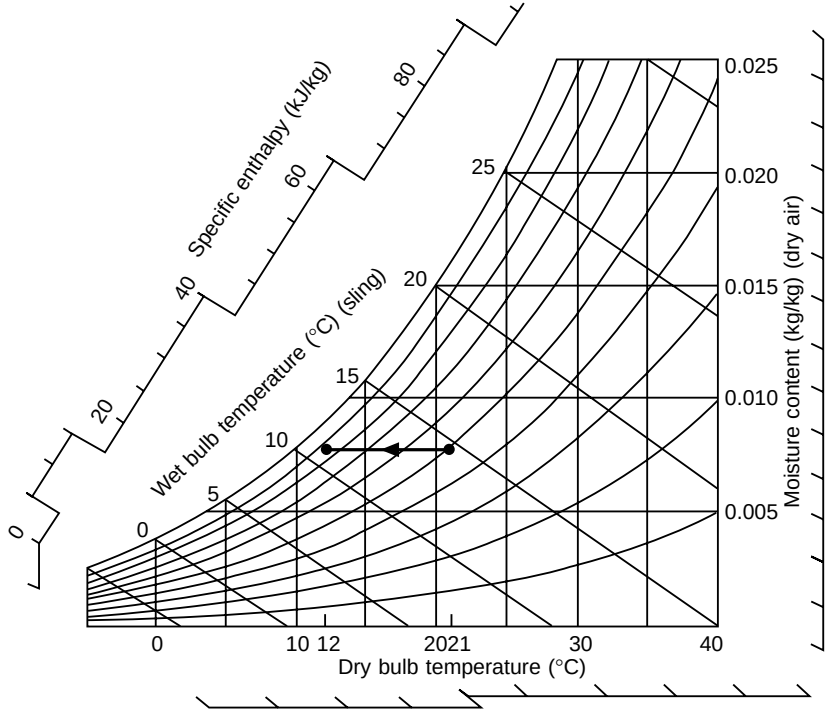


Figure 24.3 *Sensible cooling of air*

will take place and the water will draw its latent heat from the air, reducing the sensible heat and therefore the dry bulb temperature of the air (Figure 24.4).

Example 24.4 Water is sprayed into an airstream at 21°C dry bulb, 50% saturation. What would be the ultimate condition of the mixture?

No heat is being added or removed in this process, so the enthalpy must remain constant, and the process is shown as a movement along the line of constant enthalpy. Latent heat will be taken in by the water, from the sensible heat of the air, until the mixture reaches saturation, when no more water can be evaporated.

Initial enthalpy of air = 41.08 kJ/kg

Final enthalpy of air = 41.08 kJ/kg

Final condition, 14.6°C dry bulb, 14.6°C wet bulb, 14.6°C dew point, 100% saturated.

It should be noted that this ultimate condition is difficult to reach, and the final condition in a practical process would fall somewhat

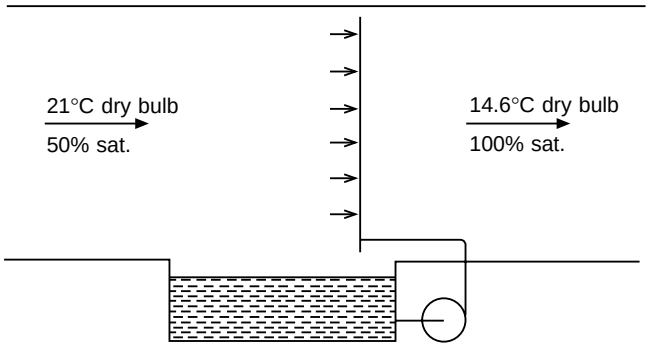


Figure 24.4 *Adiabatic saturation to ultimate condition*

short of saturation, possibly to point *C* in Figure 24.5. The proportion *AC/AB* is termed the *effectiveness* of the spray system.

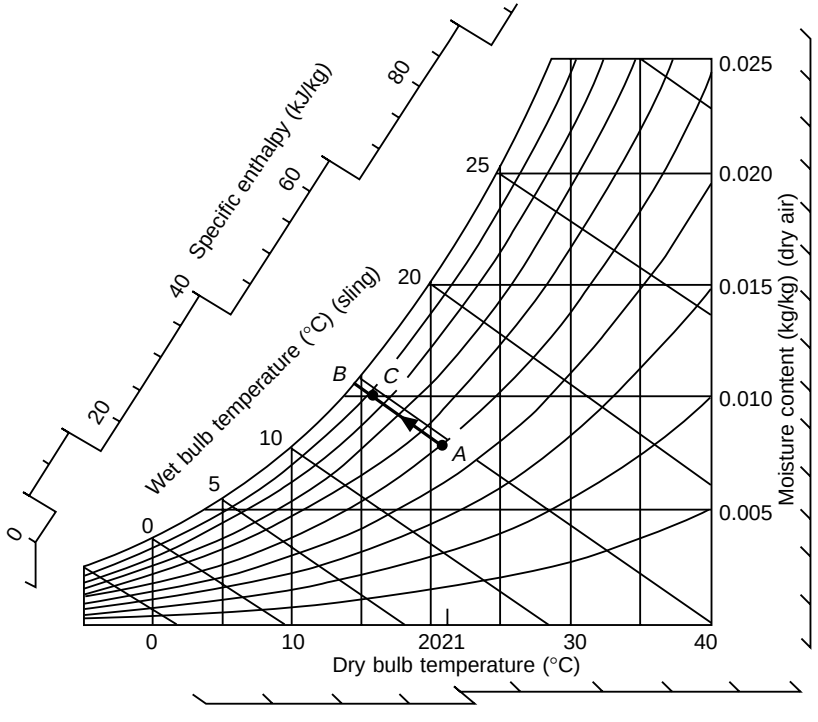


Figure 24.5 *Adiabatic saturation – process line*

The adiabatic (constant enthalpy) line *AC* is almost parallel to the line of constant wet bulb. Had the latter been used, the final error would have been about 0.2 K, and it is sometimes convenient and quicker to calculate on the basis of constant wet bulb. (This

correlation applies only to the mixture of dry air and water vapour, and not to other gas mixtures.)

24.5 Steam injection

Moisture can be added to air by injecting steam, i.e. water which is already in vapour form and does not require the addition of latent heat (Figure 24.6). Under these conditions, the air will not be cooled and will stay at about the same dry bulb temperature. The steam will be at 100°C when released to the atmosphere (or may be slightly superheated), and so raises the final temperature of the mixture.

Example 24.5 Steam at 100°C is blown into an airstream at 21°C dry bulb, 50% saturation, at the rate of 1 kg steam/150 kg dry air. What is the final condition?

Moisture content of air before = 0.0079 kg/kg

Moisture added, 1 kg/150 kg = 0.0067 kg/kg

Final moisture content = 0.0148 kg/kg

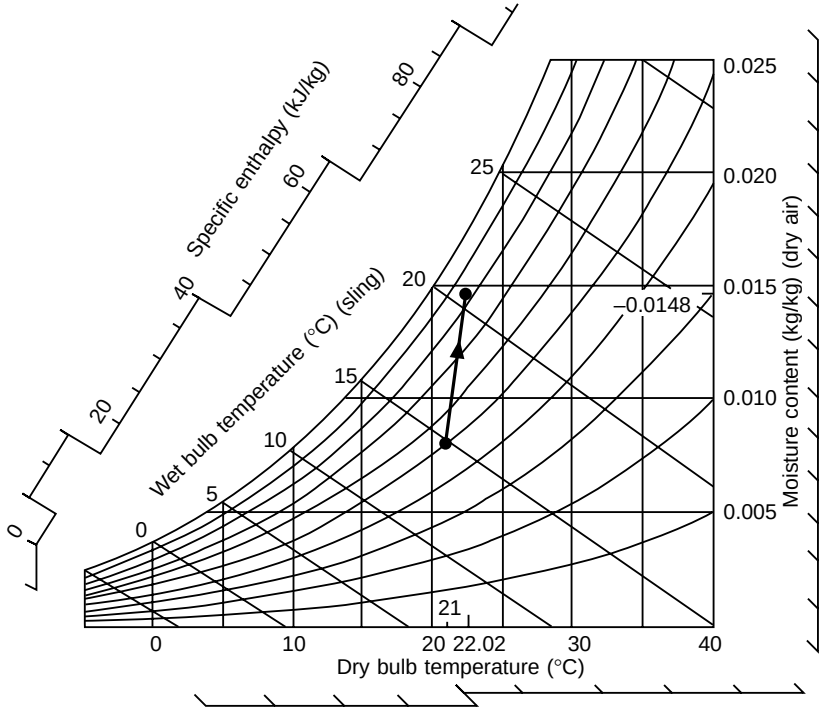


Figure 24.6 *Addition of steam to air*

An approximate figure for the final dry bulb temperature can be obtained, using the specific heat capacity of the steam through the range 20–100°C, which is about 1.972 kJ/kg. This gives

Heat lost by steam = heat gained by air

$$0.0067 \times 1.972(100 - t) = 1.006(t - 21)$$

giving

$$t = 22.02°C$$

Where steam is used to raise the humidity slightly, the increase in dry bulb temperature is small.

24.6 Air washer with chilled water

The process of adiabatic saturation in Section 24.4 assumed that the spray water temperature had no effect on the final air condition. If, however, a large mass of water is used in comparison with the mass of air, the final condition will approach the water temperature. If this water is chilled below the dew point of the entering air, moisture will condense out of the air, and it will leave the washer with a lower moisture content (see Figure 24.7).

The ultimate condition will be at the initial water temperature *B*. Practical saturation efficiencies (the ratio *AC*/*AB*) will be about 50–80% for air washers having a single bank of sprays and 80–95% for double spray banks (see Figure 24.8).

Example 24.6 Air at 23°C dry bulb, 50% saturation, enters a single-bank air washer having a saturation efficiency of 70% and is sprayed with water at 5°C. What is the final condition?

(*a*) By construction on the chart (Figure 24.7), the final condition is 10.4°C dry bulb, 82% saturation.
(*b*) By proportion:
Dry bulb is 70% of the way from 23°C down to 5°C

$$23 - [0.7(23 - 5)] = 10.4°C$$

Moisture content is 70% down from 0.008 9 to 0.005 4 kg/kg (i.e. saturated air at 5°C)

$$0.008\,9 - [0.7(0.008\,9 - 0.005\,4] = 0.006\,45 \text{ kg/kg}$$

Example 24.7 In Example 24.6, water is sprayed at the rate of 4 kg water for every 1 kg air. What is the water temperature rise?

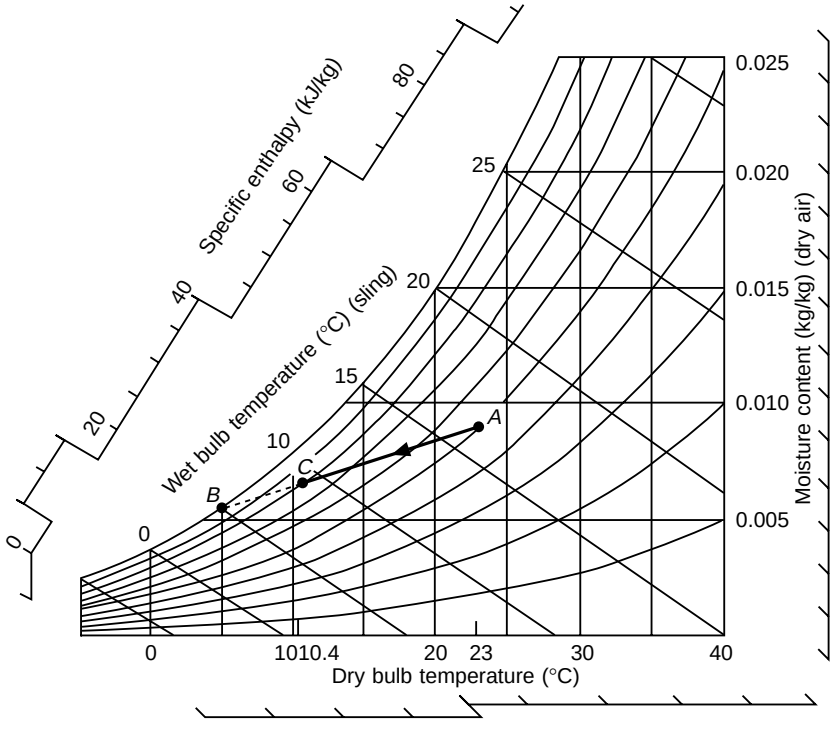


Figure 24.7 *Air washer with chilled water*

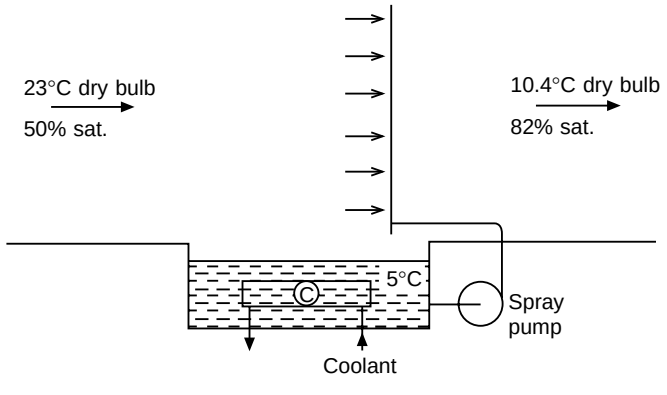


Figure 24.8 *Chilled water spray*

Enthalpy of air before = 45.79 kJ/kg

Enthalpy of air after = 26.7 kJ/kg

Heat lost per kilogram air = 19.09 kJ

$$\text{Heat gain per kilogram water} = 19.09/4$$
$$= 4.77 \text{ kJ}$$
$$\text{Temperature rise of water} = \frac{4.77}{4.187}$$
$$= 1.1 \text{ K}$$

24.7 Cooling and dehumidifying coil

In the previous process, air was cooled by close contact with a water spray. No water was evaporated, in fact some was condensed, because the water was colder than the dew point of the entering air.

A similar effect occurs if the air is brought into contact with a solid surface, maintained at a temperature below its dew point. Sensible heat will be transferred to the surface by convection and condensation of water vapour will take place at the same time. Both the sensible and latent heats must be conducted through the solid and removed. The simplest form is a metal tube, and the heat is carried away by refrigerant or a chilled fluid within the pipes. This coolant must be colder than the tube surface to transfer the heat inwards through the metal.

The process is indicated on the chart in Figure 24.9, taking point *B* as the tube temperature. Since this would be the ultimate dew point temperature of the air for an infinitely sized coil, the point *B* is termed the apparatus dew point (ADP). In practice, the cooling element will be made of tubes, probably with extended outer surface in the form of fins (see Figure 7.3). Heat transfer from the air to the coolant will vary with the fin height from the tube wall, the materials, and any changes in the coolant temperature which may not be constant. The average coolant temperature will be at some lower point *D*, and the temperature difference $B - D$ will be a function of the conductivity of the coil. As air at condition *A* enters the coil, a thin layer will come into contact with the fin surface and will be cooled to *B*. It will then mix with the remainder of the air between the fins, so that the line *AB* is a mix line.

The process line *AB* is shown here as a straight line for convenience of working. Analysis of the air as it passes through a cooling coil shows the line to be a slight curve.

The proportion *AC/AB* is termed the coil contact factor. The proportion *CB/AB* is sometimes used, and is termed the bypass factor.

Example 24.8 Air at 24°C dry bulb, 45% saturation, passes through

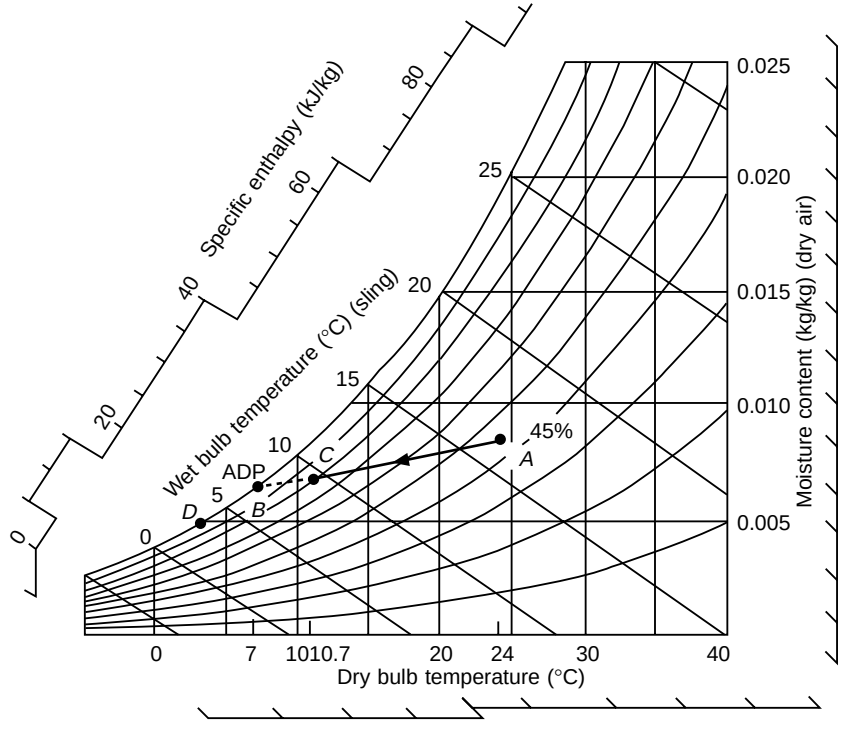


Figure 24.9 *Cooling and dehumidifying coil – process line*

a coil having an ADP of 7°C and a contact factor of 78%. What is the off-coil condition?

(*a*) By construction on the chart (Figure 24.9), 10.7°C dry bulb, 85% saturation.

(*b*) By calculation, the dry bulb will drop 78% of 24 to 7°C:

$24 - [0.78 \times (24 - 7)] = 10.7°C$

and the enthalpy will drop 78% of 45.85 to 22.72 kJ/kg:

$45.85 - [0.78 \times (45.85 - 22.72)] = 27.81$ kJ/kg

The two results obtained here can be compared with tabulated figures for saturation and give about 84% saturation.

Example 24.9 Air is to be cooled by a chilled water coil from 27°C dry bulb, 52% saturation, to 15°C dry bulb, 80% saturation. What is the ADP?

This must be done by construction on the chart, and gives an ADP of 9°C. The intersection of the process and saturation lines can also be computed. Again, it has been assumed that the process line is straight.

24.8 Sensible–latent ratio

In all cases the horizontal component of the process line is the change of sensible heat, and the vertical component gives the latent heat. It follows that the slope of the line shows the ratio between them, and the angle, if measured, can be used to give the ratio of sensible to latent to total heat. On the psychrometric chart in general use (Figure 23.5), the ratio of sensible to total heat is indicated as angles in a segment to one side of the chart. This can be used as a guide to coil and plant selection.

Example 24.10 Air enters a coil at 23°C dry bulb, 40% saturation. The sensible heat to be removed is 36 kW and the latent 14 kW. What are the ADP and the coil contact factor if air is to leave the coil at 5°C?

Plotting on the chart (Figure 24.10) from 23°C/40% and using the ratio

$$\frac{\text{Sensible heat}}{\text{Total heat}} = \frac{36}{36 + 14} = \frac{36}{50} = 0.72$$

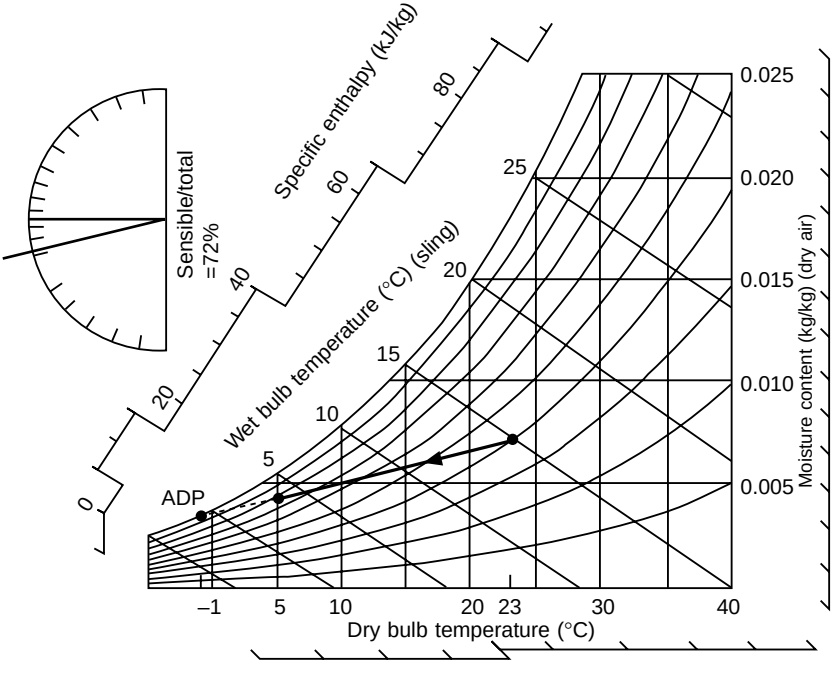


Figure 24.10 *Cooling and dehumidifying coil*

The process line meets the saturation curve at – 1°C, giving the ADP (which means that condensate will collect on the fins as frost).

Taking the 'off' condition at 5°C dry bulb and measuring the proportion along the process line gives a coil contact factor of 75%.

24.9 Multistep processes

Some air treatment processes cannot be made in a single operation, and the air must pass through two or more consecutive steps to obtain the required leaving condition.

Example 24.11 If air is to be cooled and dehumidified, it may be found that the process line joining the inlet and outlet conditions does not meet the saturation line, e.g. in cooling air from 24°C dry bulb, 45% saturation, to 19°C dry bulb, 50% saturation, the process line shows this to be impossible in one step (Figure 24.11). The air must first be cooled and dehumidified to reach the right moisture level of 0.006 9 kg/kg and then re-heated to get it back to 19°C.

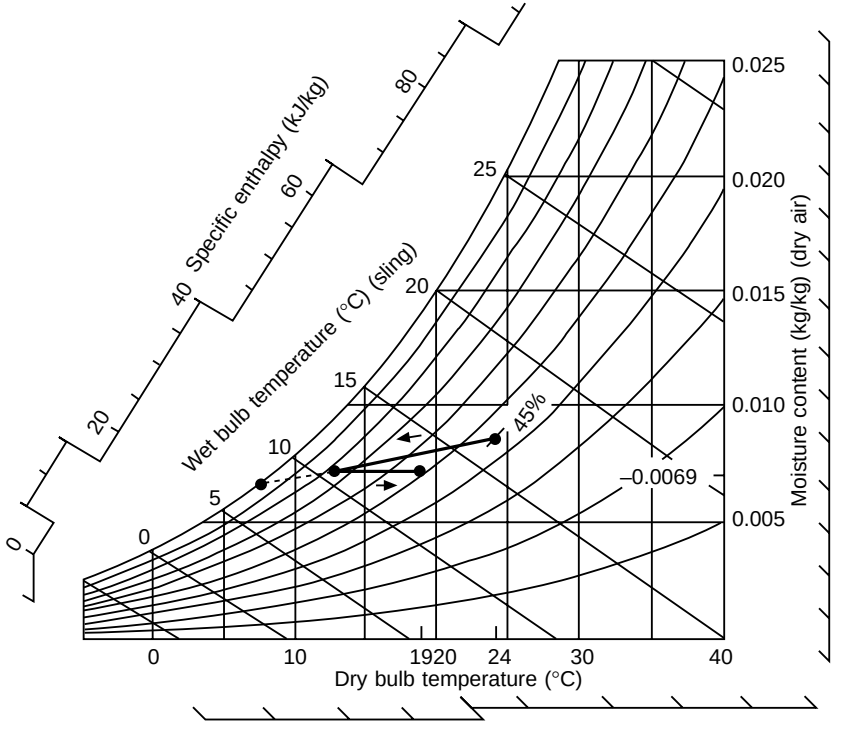


Figure 24.11 *Cooling with dehumidifying, followed by re-heat – process lines*

The first part is identical to that in Example 24.8, and the second step is the addition of sensible heat in a reheat coil.

Example 24.12 Winter outside air enters at 0°C dry bulb, 90% saturation, and is to be heated to 30°C, with a moisture content of 0.012 kg/kg.

This can be done in several ways, depending on the method of adding the moisture and final dry bulb control (see Figure 24.12). If by steam injection, the air can be pre-heated to just below 30°C and the steam blown in (line *ABC*). To give better control of the final temperature, the steam may be blown in at a lower condition, with final re-heat to get to the right point (line *ADEC*).

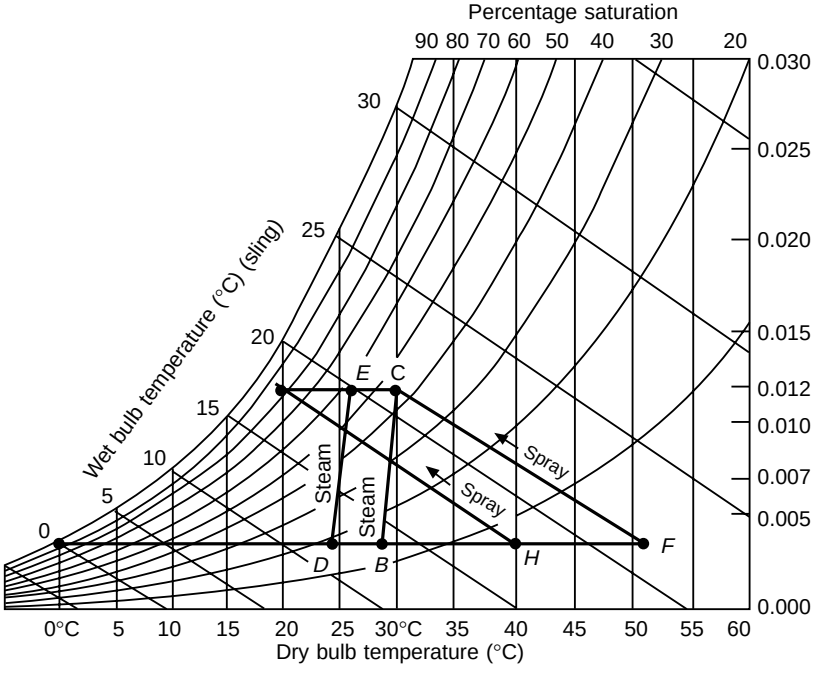


Figure 24.12 *Pre-heating and humidification in winter – process lines*

If by water spray or washer, the necessary heat must be put into the air first to provide the latent heat of evaporation. This can be done in two stages, *A* to *F* to *C*, or three stages *A* to *H* to *J* to *C*, if re-heat is required to get the exact final temperature. The latter is easier to control.

Example 24.13 Air enters a packaged dehumidifier (see Chapter 29) at 25°C dry bulb and 60% saturation. It is cooled to 10°C dry

bulb and 90% saturation, and then re-heated by its own condenser. What is the final condition?

All of the heat extracted from the air, both sensible and latent, passes to the refrigerant and is given up at the condenser to re-heat, together with the energy supplied to the compressor and the fan motor (since the latter is in the airstream). Figures for this electrical energy will have to be determined and assessed in terms of kilojoules per kilogram of air passing through the apparatus. A typical cycle is shown in Figure 24.13 and indicates a final condition of about 47°C dry bulb and 10% saturation.

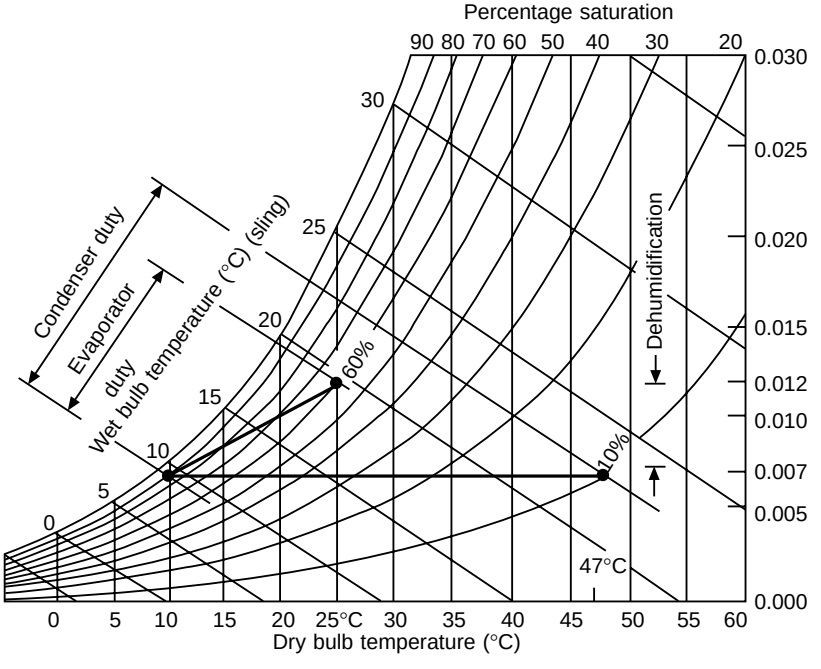


Figure 24.13 *Dehumidifier with condenser re-heat – process lines*

24.10 Cycle analysis

The last three examples indicate the importance of analysis of the required air treatment cycle on the psychrometric chart as a guide to the methods which can be adopted and those which are not possible. This analysis can also provide optimization of energy flows for a process.

Direct desk calculations would have indicated the overall energy flows between the inlet and outlet states, but may not have shown the cycles.

25 Practical air treatment cycles

25.1 Heating

The majority of air-conditioned buildings are offices or are used for similar indoor activities, and are occupied intermittently. The heating system must bring them up to comfortable working conditions by the time work is due to start, so the heating must come into operation earlier to warm up the building.

A large part of the heating load when operating in daytime will be for fresh or outside air, which is not needed before occupation, and the heat-up time will be reduced if the fresh air supply can remain inoperative for this time.

The required warm-up time will vary with ambient conditions, being longer in cold weather and least in warm. Optimum-start controllers are now in general use which are programmed for the building warm-up characteristics and sense the inside and ambient conditions. They then transduce the required start-up period and set the heating plant going only when needed. This, and the previous scheme, will save fuel.

Air-cooling systems commonly have a mass flow of 0.065 kg/(s kW) of cooling load. The normal heating load will be less than the cooling load for most of the time and, if this full air flow is maintained, the air inlet temperature will be of the order of 30–32°C. This is below body temperature and may give the effect of a cold draught, although it is heating. Where possible, the winter air flow should be reduced to give warmer inlet air. This is particularly so with packaged air-conditioners of all sizes, which may have to be located for convenience rather than for the best air-flow pattern.

The addition of moisture to the winter air in the UK is not usually necessary, except for systems using all outside air, or where persons with severe respiratory trouble are accommodated. With a winter

ambient of 0°C dry bulb, 90% saturation, outside air pre-heated to 25°C will then be 17% saturation, which could itself cause discomfort. However, this is diluted with the return air, and it is unlikely that indoor humidities will fall below 35% saturation. Humidification of this to 50% saturation would permit a slightly lower dry bulb (0.5 K less) to give a similar degree of comfort, thus slightly reducing the conduction losses from the building fabric. However, this is at the cost of the latent heat to evaporate this moisture and a higher dew point (10.4°C instead of 5°C) with increased condensation on cold building surfaces and greater deterioration (see Figure 25.1).

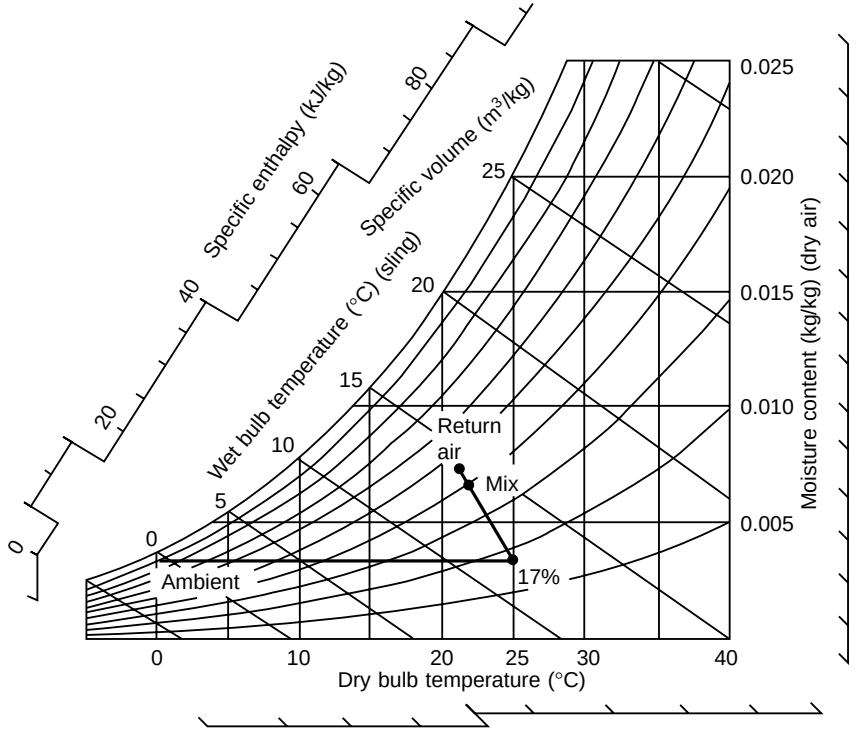


Figure 25.1 *Pre-heating of outside air and mixing with return air – process lines*

25.2 Addition of moisture

Methods of adding moisture to the airstream (see Sections 24.4 and 24.5) are difficult to control, since a lot of water remains in the apparatus at the moment of switching off humidification. For this reason, the heat–humidify–re-heat cycle as shown in Figure 24.12 is to be preferred, as the final heater control can compensate for overshoot.

Air washers require water treatment and bleed-off, since they concentrate salts in the tank. Steam will be free from such impurities, but the boiler will need attention to remove accumulations of hardness.

Mist and spray humidifiers, unless the water is pure, will leave a powder deposit of these salts in the conditioned space.

The use of standard factory-packaged air-conditioners to hold close humidities, together with a separate humidifier to correct for overdrying, is a common source of energy wastage, since both may operate at the same time. Packaged units, unless specifically built for the duty, will pull down to 45% saturation or lower under UK conditions. Humidity tolerances for process conditioning such as computer and standards rooms can often be 45–55% saturation, and this differential gap should be wide enough to prevent simultaneous operation of both humidifying and dehumidifying plant.

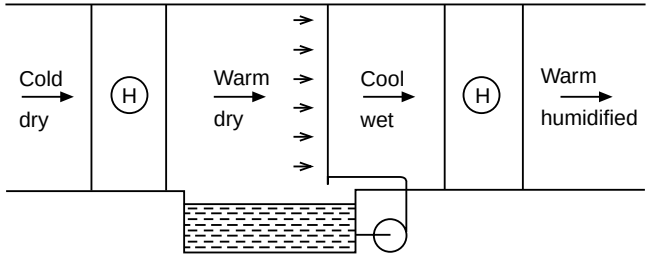


Figure 25.2 *Pre-heat, humidify, re-heat cycle – apparatus*

25.3 Outside air proportion

The high internal heat load of many modern buildings means that comfort cooling may be needed even when the ambient is down to 10°C or lower. Under these conditions, a high proportion of outside air can remove building heat and save refrigeration energy. This presupposes that:

1. The fresh air ducting and fan can provide more air.
2. This outside air can be filtered.
3. There are adequate automatic controls to admit this extra air only when wanted.
4. Surplus air in the building can be extracted.

See also Chapter 34.

25.4 Cooling and dehumidification

The cooling load will always be greatest in the early afternoon, so

no extra start-up capacity is required. The general practice of using a single coil for cooling and dehumidification without reheat, for comfort cooling, will give design balance conditions only at full load conditions. Slightly different conditions must be accepted at other times. Closer control can be obtained by variation of the coolant temperature and air mass flow over the coil, but such systems can be thrown out of calibration, and measures should be taken to avoid unauthorized persons changing the control settings or energy will be wasted with no benefit in the final conditions.

25.5 Evaporative coolers

Many of the warmer climates have a dry atmosphere (see Figure 23.8). In such areas, considerable dry bulb temperature reduction can be gained by the adiabatic saturation cycle (Section 24.4). The apparatus draws air over a wetted pad and discharges it into the conditioned space. It is termed an *evaporative* or *desert* cooler (Figure 25.3).

Example 25.1 Air at 37°C dry bulb, 24% saturation, is drawn through a desert cooler having an adiabatic saturation efficiency of 75%. What is the final dry bulb, and how much water is required?

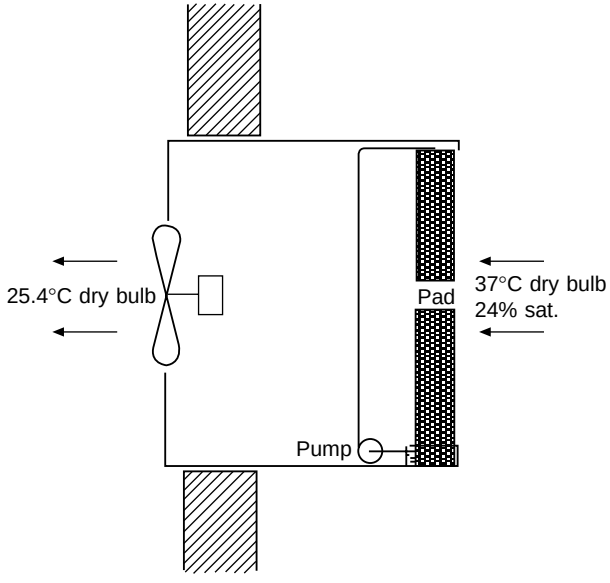
The entering enthalpy is 62.67 kJ/kg, and this remains constant through the process.

By construction on the chart, or from tables, the ultimate saturation condition would be 21.5°C, and 75% of the drop from 37°C to 21.5°C gives a final dry bulb of 25.4°C.

The water requirement can be calculated from the average latent heat of water over the working range, which is 2425 kJ/kg. The amount of water to be evaporated is $1/2425 = 0.4 \times 10^{-3}$ kg/(s kW).

This process is very much used for ambient control in textile mills and, to a lesser extent, in greenhouses for vegetable production in hot, dry climates.

A two-stage evaporative cooler (Figure 25.4) uses the cooled water from the first stage to pre-cool the air entering the second stage. The two air systems are separate. Outside air is drawn through the first stage (Figure 25.4), passing through the upper wetted pad, and so cools the water down to a temperature approaching the ambient wet bulb. This chilled water then circulates through a dry coil to cool another supply of outside air, thus reducing its wet bulb temperature. This second-stage air then passes through the lower wetted pad and into the cooled space. Water make-up is required to both circuits.



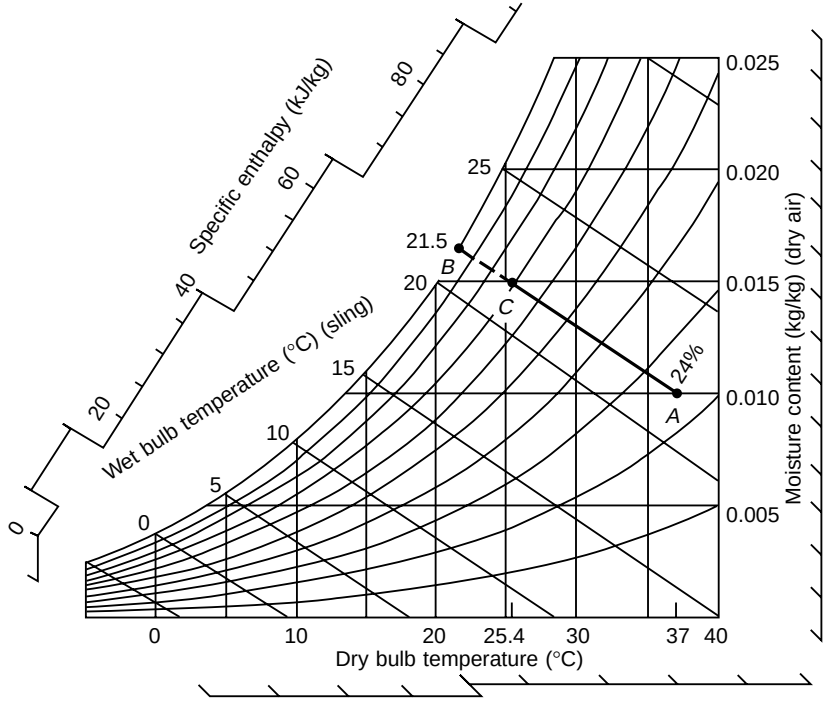
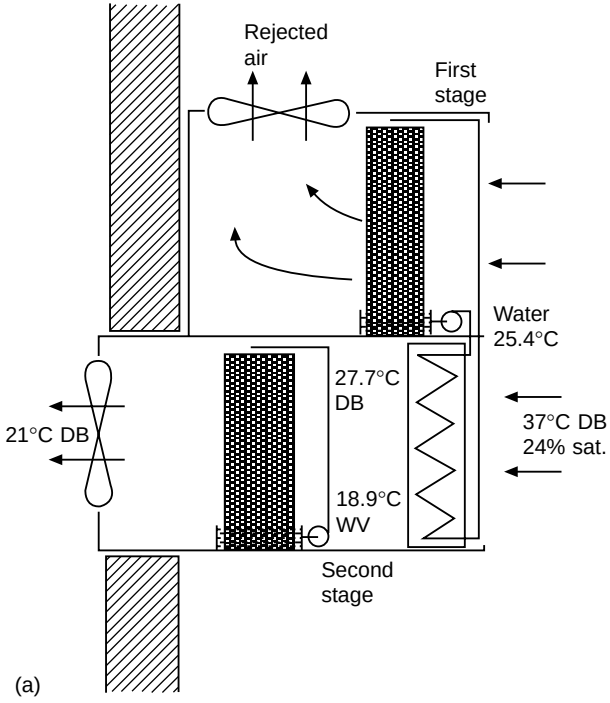


Figure 25.3 *Desert cooler. (a) Apparatus. (b) Process line*



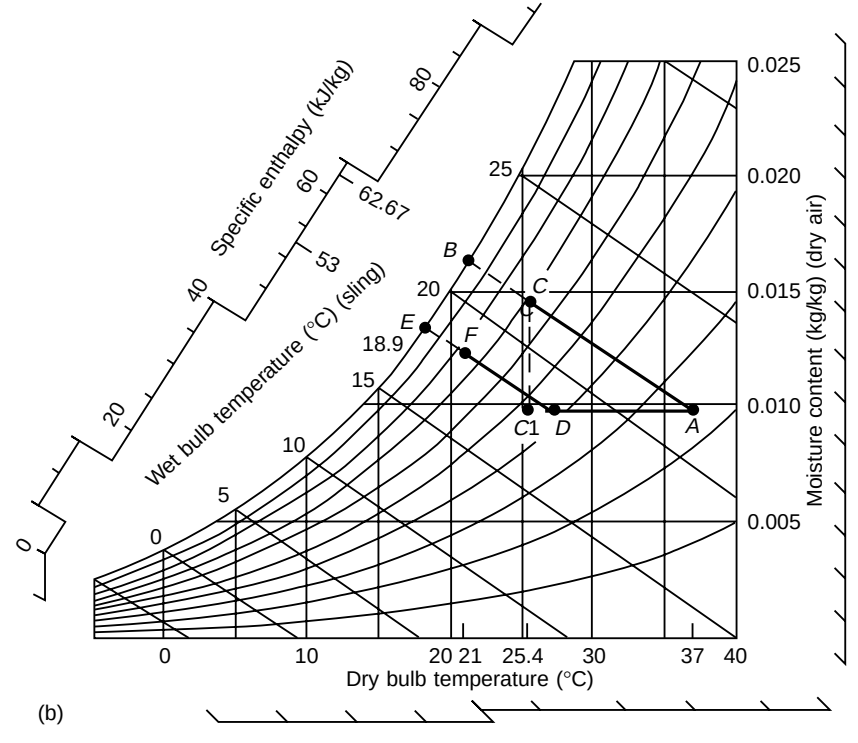


Figure 25.4 *Two-stage desert cooler. (a) Apparatus. (b) Process line*

Example 25.2 Taking the first stage as Example 25.1, the water would be cooled to 25.4°C and could be used in a coil of 80% contact factor to pre-cool outside air to

37 – 0.8(37 – 25.4) = 27.7°C (point *D*, Figure 25.4*b*)

The wet bulb is now 18.9°C and the enthalpy is 53 kJ/kg. A second-stage evaporative cooler with an efficiency of 75% will bring this down to 21°C dry bulb (point *F*).

The evaporative cooler has no refrigeration system and only requires electric power for fans and water pumps plus, of course, an adequate supply of water. No moisture can be r T0<Nq0¿0¿0¿Nu
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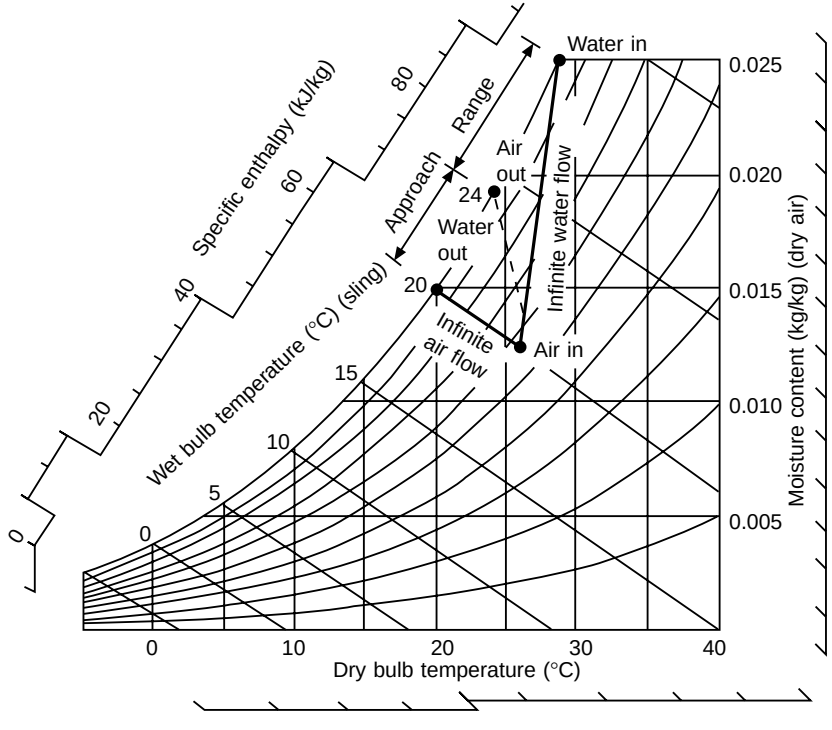


Figure 25.5 *Water tower – possible process lines under one set of atmospheric conditions*

The process is complex and cannot be simplified into an ultimate balance condition, from which to work back to a supposed operating point, so factors are used for design and application which are based on similar apparatus.

Heat from the water is transferred to the air, so the available heat gain by the air will depend on its initial enthalpy. This is usually expressed in terms of ambient wet bulb temperature, since the two are almost synonymous and the wet bulb is more easily recognized. This is used as a yardstick to describe performance in terms of the approach of the leaving water temperature to the ambient wet bulb. The two could only meet ultimately in a tower having an air flow infinitely larger than the water flow, so the term is descriptive rather than a clear indication of tower efficiency.

Assuming the mass flows of air and water to be equal, an approximate balance can be found.

Example 25.3 Air enters a cooling tower at 26°C dry bulb and 20°C wet bulb. Water at the same mass flow enters at 29°C and leaves at 24°C. If the air leaves the tower at 98% saturation, what is its final condition?

Heat from water = $4.187(29 - 24) = 21$ kJ/kg

Enthalpy of entering air = 57.1 kJ/kg

Enthalpy of leaving air = 78.1 kJ/kg

From the chart, the air leaves at about 25.7°C dry bulb.

Calculations of this sort are only of importance to the tower designer. Manufacturers' application data will give the cooling range or capacity in terms of wet bulb, inlet water temperature and mass flow [16, 19].

In the case of the evaporative condenser, the heat is input to the condenser coils, which are kept wet by the spray. The water acts both as a heat transfer medium and an evaporative coolant, and its temperature will vary through the stack of tubes. The overall process is complex and ratings are determined from practical tests on a complete condenser [16].

26 Air-conditioning load estimation

26.1 Components of load

The cooling load to maintain steady temperature and humidity in a conditioned space will have four components:

1. Heat leakage through the fabric by conduction from warmer surroundings
2. Heat gain by radiation through transparent surfaces – usually solar but occasionally by other means (radiant heat from a process, such as furnaces)
3. Heat gain by forced or natural convection – air infiltration and fresh air supply – sensible and latent heat
4. Internal heat sources – lights, people, machines, etc. – sensible and latent heat

26.2 Conduction heat gains

Conduction of heat through plain surfaces under steady-state conditions is given by the product of the area, temperature difference, and overall conductance of the surface (see Section 1.8):

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

where

$$U = \frac{1}{R_{si} + R_1 + R_2 + R_3 + \cdots + R_{so}}$$

and R_{si} is the inside surface thermal resistance, R_{so} is the outside surface thermal resistance, and R_1, R_2, etc. are the thermal resistances of the composite layers of the fabric.

Example 26.1 A building wall is made up of pre-cast concrete panels 40 mm thick, lined with 50 mm insulation and 12 mm plasterboard. The inside resistance is 0.3 $(m^2 K)/W$ and the outside resistance 0.07 $(m^2 K)/W$. What is the *U* factor?

$$U = \frac{1}{0.3 + 0.040/0.09 + 0.050/0.037 + 0.012/0.16 + 0.07}$$

$$= \frac{1}{2.24}$$

$$= 0.45 \text{ W}/(\text{m}^2 \text{ K})$$

The conductivity figures 0.09, 0.037 and 0.16 can be found in Section A3 of the CIBSE Guide [2].

Figures for the conductivity of all building materials, of the surface coefficients, and many overall conductances can be found in standard reference books [1, 2, 51].

The dominant factor in building surface conduction is the absence of steady-state conditions, since the ambient temperature, wind speed and solar radiation are not constant. It will be readily seen that the ambient will be cold in the morning, will rise during the day, and will fall again at night. As heat starts to pass inwards through the surface, some will be absorbed in warming the outer layers and there will be a time lag before the effect reaches the inner face, depending on the mass, conductivity and specific heat capacity of the materials. Some of the absorbed heat will be retained in the material and then lost to ambient at night. The effect of thermal time lag can be expressed mathematically (CIBSE Guide, A3, A5).

The rate of heat conduction is further complicated by the effect of sunshine onto the outside. Solar radiation reaches the earth's surface at a maximum intensity of about 0.9 kW/m^2. The amount of this absorbed by a plane surface will depend on the absorption coefficient and the angle at which the radiation strikes. The angle of the sun's rays to a surface (see Figure 26.1) is always changing, so this must be estimated on an hour-to-hour basis. Various methods of reaching an estimate of heat flow are used, and the sol-air temperature (see CIBSE Guide, A5) provides a simplification of the factors involved. This, also, is subject to time lag as the heat passes through the surface.

26.3 Solar heat

Solar radiation through windows has no time lag and must be estimated by finite elements (i.e. on an hour-to-hour basis), using calculated or published data for angles of incidence and taking into account the type of window glass (see Table 26.1).

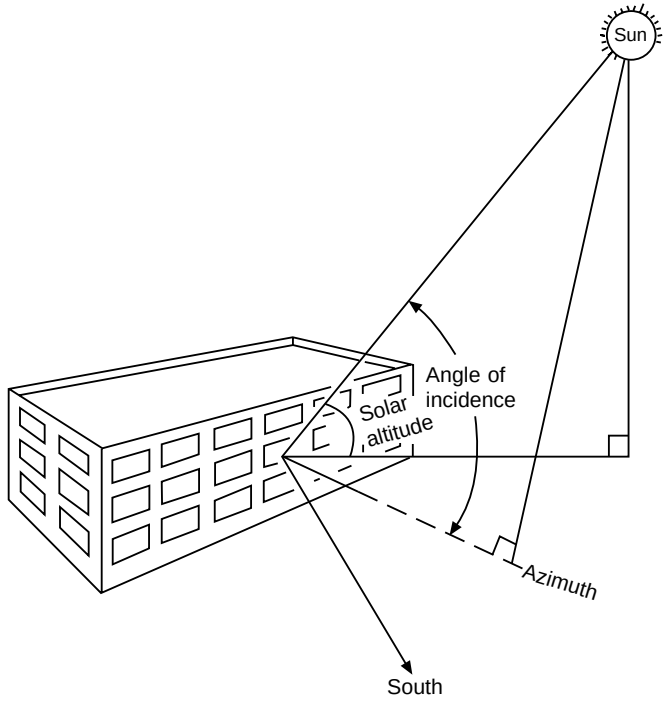


Figure 26.1 *Angle of incidence of sun's rays on window*

Since solar gain can be a large part of the building load, special glasses and window constructions have been developed, having two or more layers and with reflective and heat-absorbing surfaces. These can reduce the energy passing into the conditioned space by as much as 75%. Typical transmission figures are as follows:

Plain single glass	0.75 transmitted
Heat-absorbing glass	0.45 transmitted
Coated glass, single	0.55 transmitted
Metallized reflecting glass	0.25 transmitted

Windows may be shaded, by either internal or external blinds, or by overhangs or projections beyond the building face. The latter is much used in the tropics to reduce solar load (see Figure 26.2). Windows may also be shaded for part of the day by adjacent buildings.

All these factors need to be taken into account, and solar transmission estimates are usually calculated or computed for the hours of daylight through the hotter months, although the amount of calculation can be much reduced if the probable worst conditions can be guessed. For example, the greatest solar gain for a window facing west will obviously be after midday, so no time would be

Table 26.1 Heat gain by convection and radiation from single common window glass for 22 March and 22 September*. (W/m^2 of masonry opening) (The Trane Company, 1977, used by permission)

Time of year	*Sun time*	*Direction for North latitude (read down)*									
		N	*NE*	*E*	*SE*	*S*	*SO*	*O*	*NO*	*Horizontal*	
	6 am	0	0	0	0	0	0	0	0	0	
	7 am	17	208	359	302	45	19	16	15	70	
	8 am	26	209	507	475	158	33	30	27	220	
22 March	9 am	31	109	481	537	275	44	39	35	362	22 Sept.
and	10 am	34	51	357	524	380	54	46	40	477	and
22 Sept.	11 am	35	54	182	453	442	161	49	42	544	22 March
	12 noon	35	53	71	327	467	327	71	53	565	
North latitude	1 pm	35	42	49	161	442	453	182	54	544	South latitude
	2 pm	34	40	46	54	380	524	357	51	477	
	3 pm	31	35	39	44	275	537	481	109	362	
	4 pm	26	27	30	33	158	475	507	209	220	
	5 pm	17	15	16	19	45	302	359	208	70	
	6 pm	0	0	0	0	0	0	0	0	0	
		S	SE	E	NE	N	NO	O	SO	Horizontal	Time of year
		Direction for South latitude (read up)									

*This table is for 40 degrees North latitude. It can be used for 22 March and 22 September in the South latitude by reading up from the bottom.

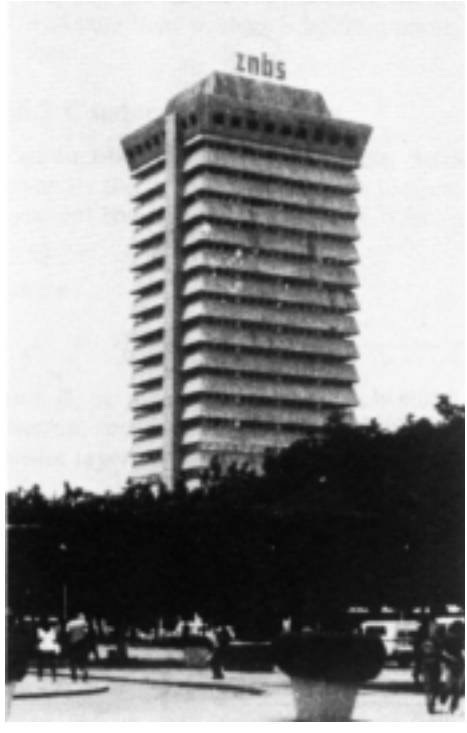


Figure 26.2 *Structural solar shading (ZNBS Building, Lusaka)*

wasted by calculating for the morning. Comprehensive data on solar radiation factors, absorption coefficients and methods of calculation can be found in reference books [1, 2, 51, 52].

There are several abbreviated methods of reaching an estimate of these varying conduction and direct solar loads, if computerized help is not readily available. One of these [53] suggests the calculation of loads for five different times in summer, to reach a possible maximum at one of these times. This maximum is used in the rest of the estimate (see Figure 26.3).

Where cooling loads are required for a large building of many separate rooms, it will be helpful to arrive at total loads for zones, floors and the complete installation, as a guide to the best method of conditioning and the overall size of plant. In such circumstances, computer programs are available which will provide the extra data as required.

26.4 Fresh air

The movement of outside air into a conditioned building will be

Air conditioning load calculation sheet

Job . . . . . Date

Summer cooling load

Outside design condition . . . . .
Inside design condition . . . . .

Table A Solar heat gains glass walls and roof sensible heat

Glass aspect	Glass area m^2 (ft^2)	Window factor Fig. 3.21	Shade factor Fig. 3.23	JUNE 10.00 h *F1* *Fig. 3.18*	*W* *(Btu/h)*	16.00 h *F2* *Fig. 3.18*	*W* *(Btu/h)*	SEPTEMBER 10.00 h *F3* *Fig. 3.18*	*W* *(Btu/h)*	14.00 h *F4* *Fig. 3.18*	*W* *(Btu/h)*	16.00 h *F5* *Fig. 3.18*	*W* *(Btu/h)*
Wall aspect	Wall m^2 (ft^2)	U		*F6* *Fig. 3.19*	*W* *(Btu/h)*	*F7* *Fig. 3.19*	*W* *(Btu/h)*	*F8* *Fig. 3.19*	*W* *(Btu/h)*	*F9* *Fig. 3.19*	*W* *(Btu/h)*	*F10* *Fig. 3.19*	*W* *(Btu/h)*
Roof	Roof m^2 (ft^2)	U		*F11* *Fig. 3.20*	*W* *(Btu/h)*	*F12* *Fig. 3.20*	*W* *(Btu/h)*	*F13* *Fig. 3.20*	*W* *(Btu/h)*	*F14* *Fig. 3.20*	*W* *(Btu/h)*	*F15* *Fig. 3.20*	*W* *(Btu/h)*
Total for each time of day				–		–		–		–		–	

Figure 26.3 *Air-conditioning load calculation sheet (part) (Courtesy of the Electricity Council)*

balanced by the loss of an equal amount at the inside condition, whether by intent (positive fresh air supply or stale air extract) or by accident (infiltration through window and door gaps, and door openings). Since a building for human occupation must have some fresh air supply and some mechanical extract from toilets and service areas, it is usual to arrange an excess of supply over extract, to maintain an internal slight pressure and so reduce accidental air movement and ingress of dirt.

The amount of heat to be removed (or supplied in winter) to treat the fresh air supply can be calculated, knowing the inside and ambient states. It must be broken into sensible and latent loads, since this affects the coil selection.

Example 26.2 A building is to be maintained at 21°C dry bulb and 45% saturation in an ambient of 27°C dry bulb, 20°C wet bulb. What are the sensible and latent air-cooling loads for a fresh air flow of 1.35 kg/s?

There are three possible calculations, which cross-check.

1. Total heat:

$$\begin{aligned} \text{Enthalpy at 27°C DB, 20°C WB} &= 57.00 \text{ kJ/kg} \\ \text{Enthalpy at 21°C DB, 45\% sat.} &= \underline{39.08} \text{ kJ/kg} \\ \text{Heat to be removed} &= 17.92 \\ Q_t = 17.92 \times 1.35 &= 24.2 \text{ kW} \end{aligned}$$

2. Latent heat:

$$\begin{aligned} \text{Moisture at 27°C DB, 20°C WB} &= 0.011\,7 \text{ kg/kg} \\ \text{Moisture at 21°C DB, 45\% sat.} &= \underline{0.007\,0} \text{ kg/kg} \\ \text{Moisture to be removed} &= 0.004\,7 \\ Q_l = 0.004\,7 \times 1.35 \times 2440 &= 15.5 \text{ kW} \end{aligned}$$

3. Sensible heat:

$$Q_s = [1.006 + (4.187 \times 0.011\,7)]\ (27 - 21) \times 1.35 = 8.6 \text{ kW}$$

Where there is no mechanical supply or extract, factors are used to estimate possible natural infiltration rates. Empirical values may be found in several standard references, and the CIBSE Guide ([2], A4) covers this ground adequately.

Where positive extract is provided, and this duct system is close to the supply duct, heat exchange apparatus (see Figure 26.4) can be used between them to pre-treat the incoming air. For the air flow in Example 26.2, and in Figure 26.5, it would be possible to save

5.5 kW of energy by apparatus costing some £1600 (price as at July 1988). The winter saving is somewhat higher.

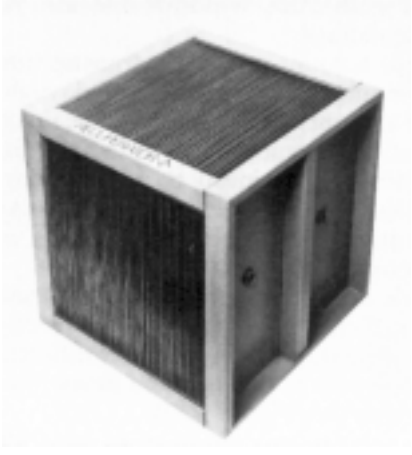


Figure 26.4 *Multi-plate air-to-air heat exchanger (Courtesy of Recuperator Ltd)*

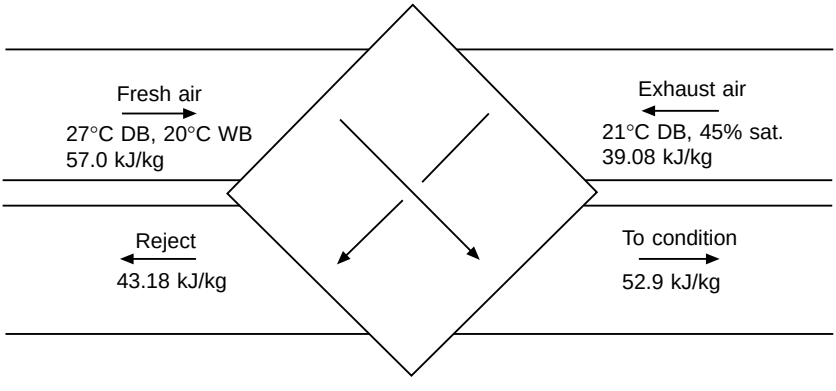


Figure 26.5 *Heat recovery to pre-cool summer fresh air*

26.5 Internal heat sources

Electric lights, office machines and other items of a direct energy-consuming nature will liberate all their heat into the conditioned space, and this load may be measured and taken as part of the total cooling load. Particular care should be taken to check the numbers of office electronic devices, and their probable proliferation within the life of the building. Recent advice on the subject is to take a liberal guess 'and then double it'.

Lighting, especially in offices, can consume a great deal of energy

and justifies the expertise of an illumination specialist to get the required light levels without wastage, on both new and existing installations. Switching should be arranged so that a minimum of the lights can be used in daylight hours. It should always be borne in mind that lighting energy requires extra capital and running cost to remove again.

Ceiling extract systems are now commonly arranged to take air through the light fittings, and a proportion of this load will be rejected with the exhausted air.

Example 26.3 Return air from an office picks up 90% of the input of 15 kW to the lighting fittings. Of this return air flow, 25% is rejected to ambient. What is the resulting heat gain from the lights?

Total lighting load = 15 kW

Picked up by return air, $15 \times 0.9 = 13.5$ kW

Rejected to ambient, $13.5 \times 0.25 = 3.375$ kW

Net room load, $15.0 - 3.375 = 11.625$ kW

The heat input from human occupants depends on their number (or an estimate of the probable number) and intensity of activity. This must be split into sensible and latent loads. The standard work of reference is CIBSE Table A7.1, an excerpt from which is shown in Table 23.2.

The energy input of part of the plant must be included in the cooling load. In all cases include fan heat, either net motor power or gross motor input, depending on whether the motors are in the conditioned space or not. Also, in the case of packaged units within the space, heat is given off from the compressors and may not be allowed for in the manufacturer's rating.

26.6 Assessment of total load estimates

Examination of the items which comprise the total cooling load may throw up peak loads which can be reduced by localized treatment such as shading, modification of lighting, removal of machines, etc. A detailed analysis of this sort can result in substantial savings in plant size and future running costs.

A careful site survey should be carried out if the building is already erected, to verify the given data and search for load factors which may not be apparent from the available information [21].

It will be seen that the total cooling load at any one time comprises a large number of elements, some of which may be known with a

degree of certainty, but many of which are transient and which can only be estimated to a reasonable closeness. Even the most sophisticated and time-consuming of calculations will contain a number of approximations, so short-cuts and empirical methods are very much in use. A simplified calculation method is given by the Electricity Council [53], and abbreviated tables are given in Refs [23], [51] and [52]. Full physical data will be found in [1] and the CIBSE Guide Book A [2].

There are about 37 computer programs available, and a full list of these with an analysis of their relative merits is given by the Construction Industry Computing Association, Cambridge, Evaluation Report No. 5.

Since the estimation will be based on a desired indoor condition at all times, it may not be readily seen how the plant size can be reduced at the expense of some temporary relaxation of the standard specified. Some of the programs available can be used to indicate possible savings both in capital cost and running energy under such conditions [54]. In a cited case where an inside temperature of 21°C was specified, it was shown that the installed plant power could be reduced by 15% and the operating energy by 8% if short-term rises to 23°C could be accepted. Since these would only occur during the very hottest weather, such transient internal peaks may not materially detract from the comfort or efficiency of the occupants of the building.

27 Air movement

27.1 Static pressure

Air at sea level exerts a static pressure, due to the weight of the atmosphere, of 1013.25 mbar. The density, or specific mass, at 20°C is 1.2 kg/m^3. Densities at other conditions of pressure and temperature can be calculated from the Gas Laws:

$$\rho = 1.2\left(\frac{p}{1013.25}\right)\left(\frac{273.15 + 20}{273.15 + t}\right)$$

where p is the new pressure, in mbar, and t is the new temperature in °C.

Example 27.1 What is the density of dry air at an altitude of 4500 m (575 mbar barometric pressure) and a temperature of – 10°C?

$$\rho = 1.2\left(\frac{575}{1013.25}\right)\left(\frac{293.15}{263.15}\right)$$

$$= 0.76 \text{ kg/m}^3$$

Air passing through a closed duct will lose pressure due to friction and turbulence in the duct.

An air-moving device such as a fan will be required to increase the static pressure in order to overcome this resistance loss (see Figure 27.1).

27.2 Velocity and total pressure

If air is in motion, it will have kinetic energy of

$0.5 \times \text{mass} \times (\text{velocity})^2$

Example 27.2 If 1 m^3 of air at 20°C dry bulb, 60% saturation, and

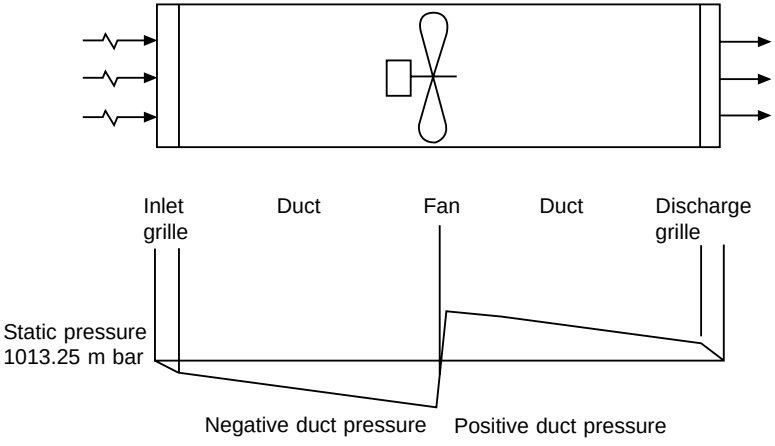


Figure 27.1 *Static pressure in ducted system*

a static pressure of 101.325 kPa is moving at 7 m/s, what is its kinetic energy?

Air at this condition, from psychrometric tables, has a specific volume of 0.8419, so 1 m^3 will weigh 1/0.8419 or 1.188 kg, giving:

$$\text{Kinetic energy} = 0.5 \times 1.188 \times (7)^2$$
$$= 29.1 \text{ kg/(m s}^2\text{)}$$

The dimensions of this kinetic energy are seen to be the dimensions of pascals. This kinetic energy can therefore be expressed as a pressure and is termed the velocity pressure.

The total pressure of the air at any point in a closed system will be the sum of the static and velocity pressures. Losses of pressure due to friction will occur throughout the system and will show as a loss of total pressure, and this energy must be supplied by the air-moving device, usually a fan.

27.3 Measuring devices

The static pressure within a duct is too small to be measured by a bourdon tube pressure gauge, and the vertical or inclined manometer is usually employed (Figure 27.2). Also, there are electromechanical anemometers. The pressure tapping into the duct must be normal to the air flow.

Instruments for measuring the velocity as a pressure effectively convert this energy into pressure. The transducer used is the Pitot tube (Figure 27.3), which faces into the airstream and is connected to a manometer. The outer tube of a standard pitot tube has side

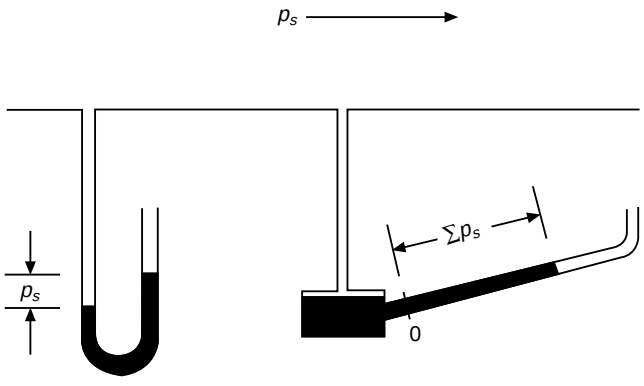


Figure 27.2 *Vertical and inclined manometers*

tappings which will be normal to the air flow, giving static pressure. By connecting the inner and outer tappings to the ends of the manometer, the difference will be the velocity pressure.

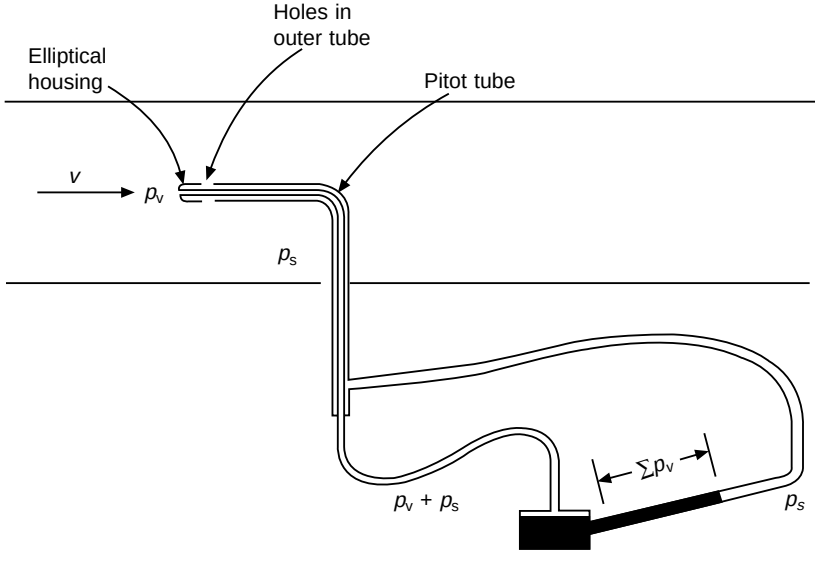


Figure 27.3 *Pitot tube*

Sensitive and accurate manometers are required to measure pressures below 15 Pa, equivalent to a duct velocity of 5 m/s, and accuracy of this method falls off below 3.5 m/s. The pitot head diameter should not be larger than 4% of the duct width, and

heads down to 2.3 mm diameter can be obtained. The manometer must be carefully levelled.

Air speed can be measured with mechanical devices, the best known of which is the vane anemometer (Figure 27.4). In this instrument, the air turns the fan-like vanes of the meter, and the rotation is counted through a gear train on indicating dials, the number of turns being taken over a finite time. Alternatively, the rotation may be detected electronically and converted to velocity on a galvanometer. The rotating vanes are subject to small frictional errors and such instruments need to be specifically calibrated if close accuracy is required. Accuracies of 3% are claimed. Moving air can be made to deflect a spring-loaded blade and so indicate velocity directly.

Figure 27.4 *Vane anemometer (Courtesy of Airflow Developments)*

A further range of instruments detects the cooling effect of the moving air over a heated wire or thermistor, and converts the signal to velocity. Air velocities down to 1 m/s can be measured with claimed accuracies of 5%, and lower velocities can be indicated.

Air flow will not be uniform across the face of a duct, the velocity being highest in the middle and lower near the duct faces, where the flow is slowed by friction. Readings must be taken at a number of positions and an average calculated. Methods of testing and positions for measurements are covered in BS1042. In particular, air flow will be very uneven after bends or changes in shape, so measurements should be taken in a long, straight section of duct.

More accurate measurement of air flow can be achieved with nozzles or orifice plates. In such cases, the measuring device imposes a considerable resistance to the air flow, so that a compensating fan is required. This method is not applicable to an installed system and is used mainly as a development tool for factory-built packages, or for fan testing. Details of these test methods will be found in BS.1042, BS.2852, and ASHRAE 16-83.

27.4 Air-moving devices

Total pressures required for air-conditioning systems and apparatus are rarely in excess of 2 kPa, and so can be obtained with dynamic air-moving machinery rather than positive-displacement pumps. The centrifugal fan (Figure 27.5) imparts a rotation to the entering air and the resulting centrifugal force is converted to pressure and velocity in a suitable outlet scroll. Air leaving the tips of the blades will have both radial and tangential velocities, so the shape of the blade will determine the fan characteristics.

The forward-curved fan blade increases the tangential velocity considerably (see Figure 27.5*b*). As a result, the power required will increase with mass flow, although the external resistance pressure is low, and oversize drive motors are required if the system resistance can change in operation. The backward-curved fan runs faster and has a flatter power curve, since the air leaves the blade at less than the tip speed (see Figure 27.5*c*).

Since centrifugal force varies as the square of the speed, it can be expected that the centrifugal fans, within certain limits, will have the same characteristics. These can be summed up in the General Fan Laws:

Volume varies as speed.
Pressure varies as $(\text{speed})^2$.
Power varies as $(\text{speed})^3$.

Where a centrifugal fan is belt driven and some modification of performance may be required, these laws may be applied to determine a revised speed and the resulting power for the new duty. Since the resistance to air flow will also vary as the square of the speed of the

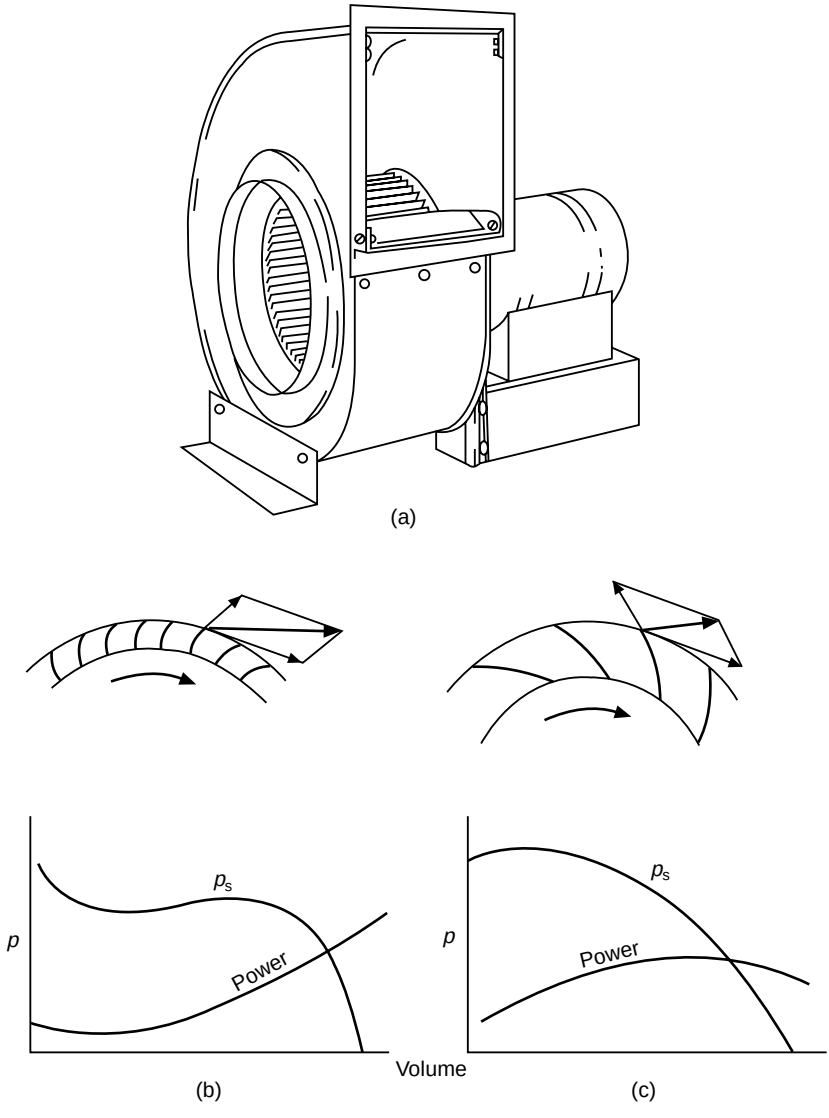


Figure 27.5 *Centrifugal fan. (a) Construction. (b) Forward-curved blades and typical performance curves. (c) Backward-curved blades and typical performance curves*

air within the duct (see Section 27.6), it follows that a change of fan speed proportional to the required change in volume should give a close approximation for the new duty. Two-speed motors and electronic speed controls are in use.

At no-flow (stall) conditions, these fans will not generate any

velocity pressure and the absorbed power will be a minimum, used only in internal turbulence.

Large volumes of air at low pressures can be moved by the propeller fan (Figure 27.6). The imparted energy is mainly in an axial direction and any large external resistance will cause a high proportion of slip over the blades.

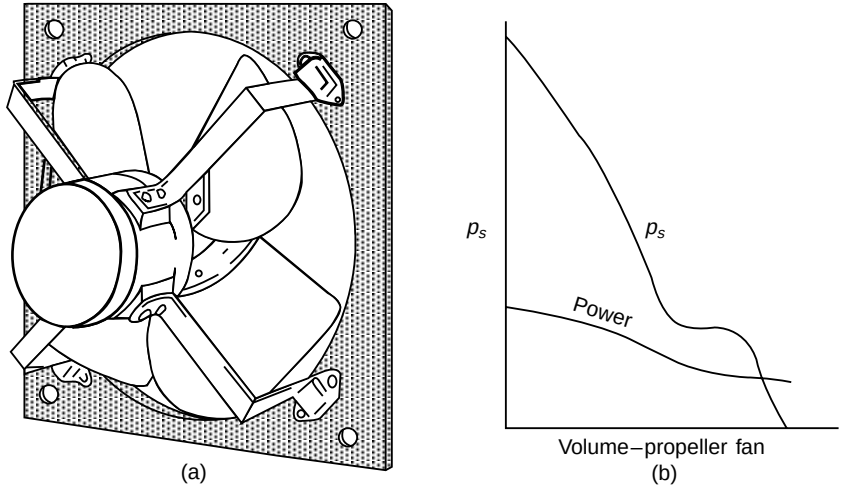
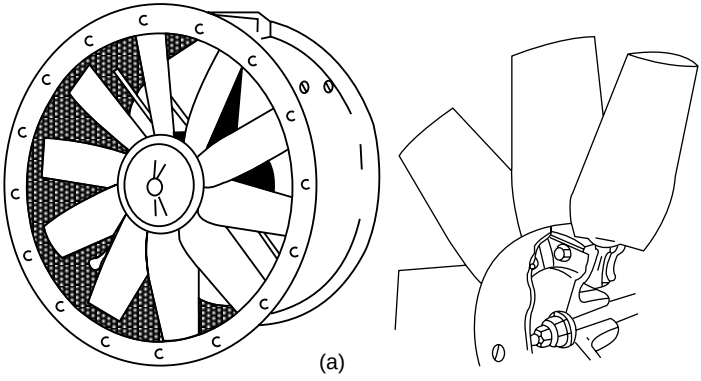


Figure 27.6 *Propeller fan. (a) Construction. (b) Typical performance curves*

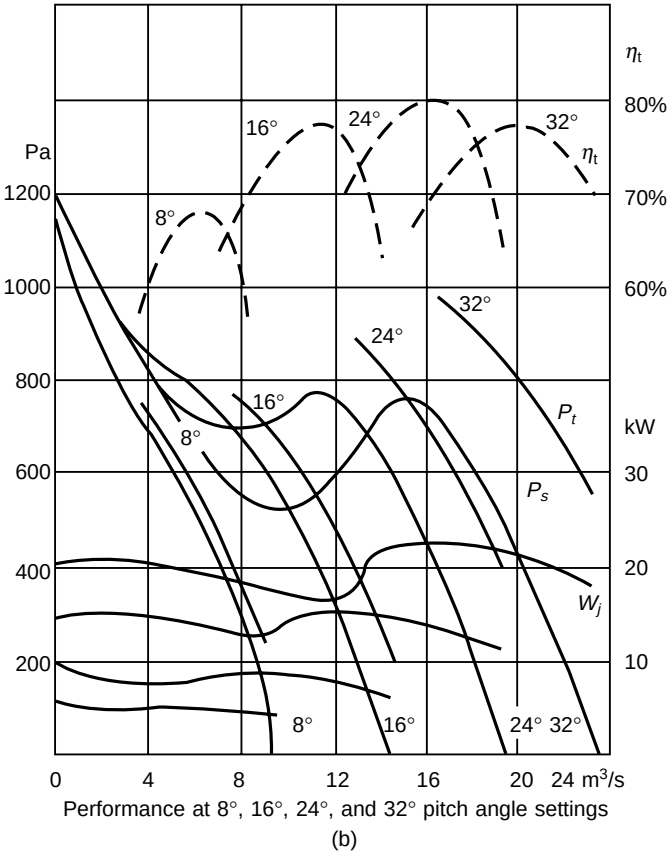
The working pressure limits of the propeller fan, depending on its diameter, are of the order of 150 Pa. The characteristic curve has a pronounced 'trough', which should be avoided in application if at all possible, since wide variations in air flow can occur for a small change in pressure. Performance varies with aperture shape, clearance and position.

Peak efficiency and pressure capability can be achieved with axial-flow fans by using blades of correct aerofoil shape and ensuring a low tip clearance. Such fans are termed aerofoil, axial flow (Figure 27.7), or tube axial, to differentiate them from propeller fans. The pitch angle of the blades will determine the working characteristics and best working efficiency. Commercially available fans are commonly made so that the angle of pitch can be selected for its application and pre-set at the factory or on installation. Some large axial-flow fans can be obtained with blades which can be varied in pitch while running, similar to variable-pitch aircraft propellers, so that the fan performance can be varied as required by the system load.

Air leaves the blades of an axial-flow fan with some turning motion,



(a)



Performance at 8°, 16°, 24°, and 32° pitch angle settings

(b)

Figure 27.7 *Axial flow fans. (a) Construction. (b) Typical performance curves (Reproduced, with permission, from Wood's Practical Guide to Fan Engineering [55])*

and the provision of straightening vanes after the rotor will recover some of this energy, adding to the performance and efficiency of the fan; pre-rotational vanes also help slightly.

Higher pressures can be obtained by putting two axial-flow fans in series. If they are placed close together and contrarotated, the spin imparted by the first can be recovered by the second, and more than twice the pressure capability can be gained.

The best efficiency of the axial-flow fan is to the right of the trough seen in the pressure curve, and the optimum band of performance will be indicated by the manufacturer. In particular, the air flow should not be less than the given minimum figure, since the fan motor relies on air flow for cooling.

It is possible to readjust the blade angles on site but, if so, great care must be taken to get them all at the same angle. The procedure is not to be recommended. Most such fans are direct drive, so the speed cannot be changed except electronically.

It will be seen that there is no change in velocity through an axial-flow fan, and the blade energy is used in increasing the static pressure of the air flow. Since the velocity through the fan casing will probably be higher than adjacent duct velocities, these fans commonly have inlet and outlet cones, which must be properly designed and constructed to minimize energy losses.

The mixed-flow fan combines the geometry of the axial-flow and centrifugal fans and can give a very high efficiency at a predetermined operating load, but it less flexible in operation outside that point in its curve. It requires an accurately fitting housing, and is not in general use on commercial applications because of the close working tolerances.

The cross-flow or tangential fan sets up an eccentric vortex within the fan runner, the air coming inwards through the blades on one side and leaving outwards through the blades on the other. It can, within mechanical limits, be made as long as necessary for the particular duty.

The cross-flow fan generates only very slight pressure and its use is limited to appliances where the air pressure drop is low and predetermined. Its particular shape is very suitable for many kinds of air-handling devices such as fan coil units and fan convectors.

The fans used in air-conditioning duct systems are centrifugal or axial flow. Since both types are available in a wide range of sizes, speeds and manufacture, the final choice for a particular application is often reduced to a suitable shape – the centrifugal having its inlet and outlet ducts at 90° while the axial flow is in-line.

The centrifugal fan may be direct-coupled, i.e. having the fan runner on an extension of the motor shaft, or belt driven. In the

latter case the motor must be mounted with the fan, to withstand belt tension. This arrangement has the advantage that the speed can be selected for the exact load, and can be changed if required. The axial-flow fan usually has the motor integral, and so is restricted to induction motor speeds of 2900, 1450 or 960 rev/min, and cannot be altered. Precise application and possible future duty changes may be accommodated within the range of blade angles. Much use is now made of electronic fan speed control on small air conditioners.

27.5 Noise and vibration

All manufacturers now publish sound pressure levels for their products and such figures should be scrutinized and compared as part of a fan-selection decision. Fans are statically, and sometimes dynamically, balanced by the manufacturer. If it is necessary to dismantle a fan for transport, it should be rebalanced on commissioning, imposing a load close to that ultimately required.

Fans are balanced in a clean condition, but will tend to collect dirt in operation, which will adhere unevenly to the blades. It is therefore essential to provide antivibration mountings for all fan assemblies including their drive motors. Since the fan will then be free to move relative to the ductwork, which is fixed, flexible connections will be needed to allow for this movement. With belt-driven fans, care must be taken that the antivibration mountings are suitable for the rotational speeds of both fan and motor. Where motors may be electronically speed controlled, the antivibration mountings must be suitable for the expected working range of speeds. Fans with high tip speeds will generate noise levels which may need attenuation. The normal treatment of this problem is to fit an acoustically lined section of ductwork on the outlet or on both sides of the fan. Such treatment needs to be selected for the particular application regarding frequency of the generated noise and the degree of attenuation required, and competent suppliers will have this information. The attenuators will be fixed, and located after the flexible connectors, so these latter will also need acoustic insulation to prevent noise breaking out here [56].

The reduction of cost of electronic speed control for fan motors has led to a much wider use of this method. The general circuit is to *invert* the supply by first rectifying it to *direct current* and then pass this through a chopper to produce a new *alternating current* with the frequency for the new motor speed.

Most large fans need to be cleaned thoroughly every year to remove deposits of dirt and so limit vibration.

27.6 Flow of air in ducts

General laws for the flow of fluids were determined by Reynolds, who recognized two flow patterns, laminar and turbulent. In laminar flow the fluid can be considered as a series of parallel strata, each moving at its own speed, and not mixing. Strata adjacent to walls of the duct will be slowed by friction and will move slowest, while those remote from the walls will move fastest. In turbulent flow there is a general forward movement together with irregular transfer between strata.

In air-conditioning systems, all flow is turbulent, and formulas and charts show the resistance to air flow of ducting of various materials, together with fittings and changes of shape to be met in practice. The reader is referred to the tables and charts in CIBSE Guide C4 [4] and in [55] (Chapter 6).

High duct velocities show an economy in duct cost, but require more power which will generate more noise. Velocities in common use are as follows:

High-velocity system, main ducts	20 m/s
High-velocity system, branch ducts	15 m/s
Low-velocity system, main ducts	10 m/s
Low-velocity system, branch ducts	6 m/s
Ducts in quiet areas	3–4 m/s

Ducting construction must be stiff enough to retain its shape, be free from air-induced vibration (panting) and strong enough to allow air-tight joints along its length. Such construction is adequately covered by HVCA [57] Specification No. DW.141 for sheet metal, No. DW.151 for plastics, and No. DW.181 for grp.

The frictional resistance to air flow within a duct system follows the general law

$$H = a\frac{v^2}{d}$$

where a is a coefficient based on the roughness of the duct surface and the density of the air. Duct-sizing charts are based on this law. Since such charts cannot cater for all shapes, they give resistances for circular ducts, and a subsidiary chart shows how to convert rectangular shapes to an equivalent resistance round duct.

Example 27.3 What is the resistance pressure drop in a duct measuring 700 × 400 mm, if the air flow through it is 2 m^3/s? What is the velocity?

From the chart ([4], Figure C4.4), reading down the 700 × 400 line until it meets the horizontal line through 2 m/s gives

Pressure drop = 1.0 Pa/m

Velocity = 7.1 m/s

It should be noted that the energy for this pressure drop must come from static pressure, since the velocity, and hence the velocity pressure, remains constant.

Frictional resistance to air flow of fittings such as bends, branches and other changes of shape or direction will depend on the shape of the fitting and the velocity, and such figures are tabulated with factors to be multiplied by the velocity pressure. Tables of such factors can be found in standard works of reference [1, 4, 55].

Example 27.4 The duct specified above has in it two bends, for which a pressure loss factor of 0.28 is shown in the tables ([4], Table C4). What is the total pressure loss?

Pressure loss per bend = $p_v \times 0.28$

$$p_v = 0.5 \times 1.2 \times v^2$$

where

$$v = 7.1$$

$$p_v = 30.25 \text{ Pa}$$

$$\text{Pressure loss} = 2 \times 0.28 \times 30.25$$

$$= 16.94 \text{ Pa}$$

The sizing of ductwork for a system will commence with an assumption of an average pressure-loss figure, based on a working compromise between small ducts with a high pressure drop and large ducts with a small pressure drop. An intitial figure for a commercial air-conditioning plant will be 0.8–1.0 Pa/m. This will permit higher velocities in the larger ducts with lower velocity in the branches within the conditioned spaces, where noise may be more noticeable.

Pressure drops for proprietary items such as grilles and filters can be obtained from manufacturers.

An approximate total system resistance can be estimated from the design average duct loss and the maximum duct length, adding the major fittings. However, this may lead to errors outside the fan power and it is safer to calculate each item and tabulate as shown in Table 27.1 for the system shown in Figure 27.8. Only the longest branch need be taken for fan pressure.

It will be seen that where there are a number of branches from a main duct, there will be an excess of available pressure in these

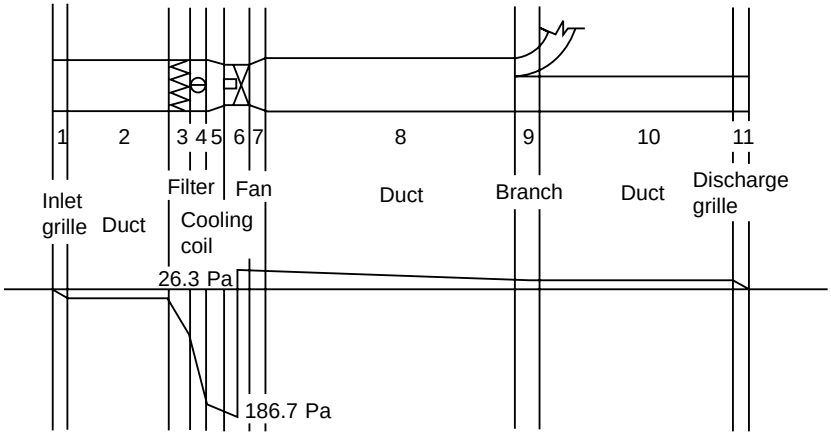


Figure 27.8 *Ducted system with fittings and fans, showing static pressure*

branches. In order to adjust the air flows on commissioning, dampers will be required in the branch ducts or, as is more usually provided, in the necks of the outlet grilles. The latter arrangement may be noisy if some of these dampers have to be closed very far to balance the air flow, with a resulting high velocity over the grille blades.

27.7 Flow of air under kinetic energy

Any static pressure at the outlet of a duct will be lost as the air expands to atmospheric pressure. This expansion, which is very small, will be in all directions, with no perceptible gain of forward velocity. Static pressure can be converted to velocity at the outlet by means of a converging nozzle or by a grille. In both cases the air outlet area is less than the duct area, and extra forward velocity is generated from the static pressure. The leaving air will form a jet, the centre of which will continue to move at its original velocity, the edges being slowed by friction and by entrainment of the surrounding air. (See Figure 27.9.) The effect is to form a cone, the edges of which will form an included angle of 20–25°, depending on the initial velocity and the shape of the outlet. Since the total energy of the moving air cannot increase, the velocity will fall as the mass is increased by entrained air, and the jet will lose all appreciable forward velocity when this has fallen to 0.25–0.5 m/s.

If the air in a horizontal jet is warmer or cooler than the surrounding air, it will tend to rise or fall. This effect will lessen as the jet entrains air, but may be important if wide temperature differences have to be used or in large rooms [58, 59].

Table 27.1 System pressure loss

Item	*Type*	*Size* (mm)	*Length* (m)	*Air flow* (m^3/s)	*Velocity* (m/s)	p_v (Pa)	*Resistance factor*	*Pressure loss* (Pa)	p_t(Pa)	p_s(Pa)
1	Inlet louvres	900 × 600	–	1.3	2.41	3.5	0.40	2.1	–2.1	–5.6
2	Duct	900 × 600	2	1.3	2.41	3.5	0.1	0.2	–2.3	–5.8
3	Filter	900 × 600	–	1.3	2.41	3.5	60 Pa	60*	–62.3	–65.8
4	Cooling coil	900 × 600	–	1.3	2.41	3.5	97 Pa	97*	–159.3	–162.8
5	Reduce	900 × 600 to 500 diameter	–	1.3	6.62	26.3	0.04	1.1	–160.4	–186.7
6	Fan	500 diameter	–	1.3						
7	Enlarge	500 diameter to 600 × 600	–	1.3	3.61	7.8	0.4	3.1	34.1	26.3
8	Duct	600 × 600	8	1.3	3.61	7.8	0.2	1.6	31.0	23.2
9	Branch, straight		–	1.3	3.61	7.8	0.04	0.3	29.4	21.6
10	Duct	600 × 400	6	0.65	2.7	4.4	0.18	1.1	29.1	24.7
11	Outlet grille	600 × 400	–	0.65		4.4		28*	28.0	23.6

*Typical catalogue figures.
Required fan pressure = 186.7 + 26.3 = 213 Pa.

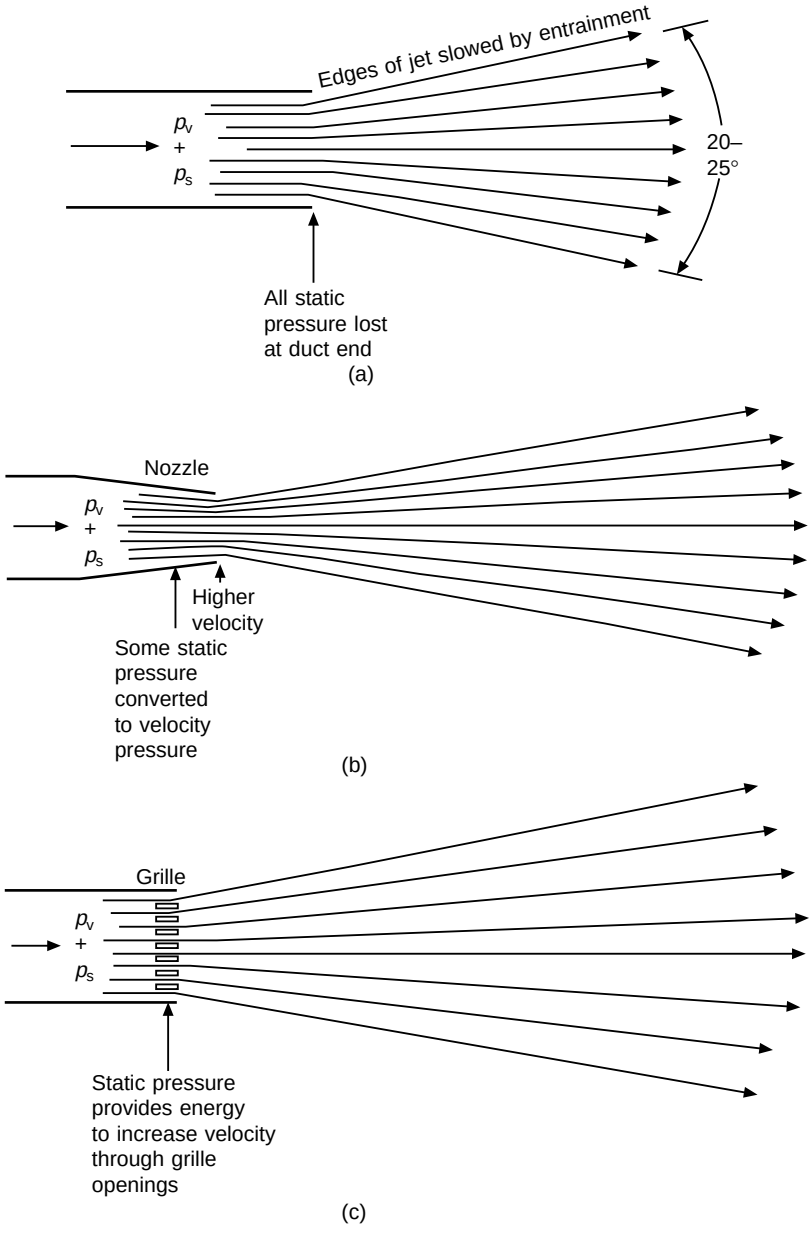


Figure 27.9 *(a) Air leaving open-ended duct. (b) Air leaving nozzle. (c) Air leaving grille*

If an air jet is released close to a plane surface (ceiling or wall usually), the layer of air closest to the surface will be retarded by

friction and the jet will tend to cling to the surface. Use of this effect is made to distribute air across a ceiling from ceiling slots or from grilles high on the walls. (See Figure 27.10.) Air is entrained on one side only and the cone angle is about half of that with a free jet. This produces a more coherent flow of input air with a longer throw.

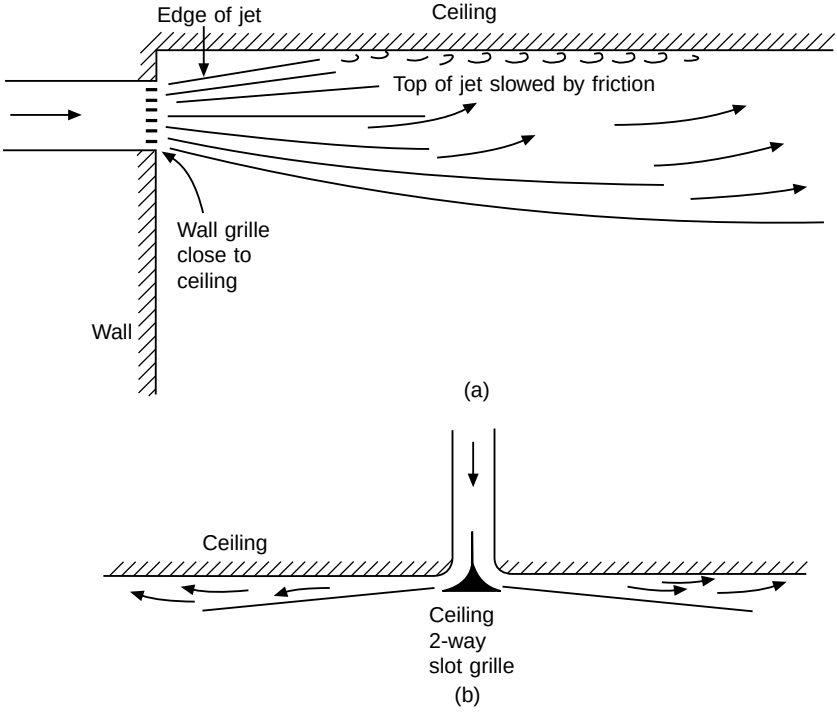


Figure 27.10 *Restriction of jet angle by adjacent surface. (a) Wall grille close to ceiling. (b) Ceiling slots*

If the air jet is held within a duct expansion having an included angle less than 20°, only duct friction losses will occur. Since there is no entrained air to take up some of the kinetic energy of the jet, a large proportion of the drop in kinetic energy will be regained as static pressure, i.e. the static pressure within the duct after the expansion will be greater than it was before the expansion (see Figure 27.11).

The optimum angle for such a duct expansion will depend on the air velocity, since the air must flow smoothly through the transition and not 'break away' from the duct side with consequent turbulence and loss of energy. This included angle is about 14°. With such an expansion, between 50 and 90% of the loss of velocity pressure will be regained as static pressure [51, 52].

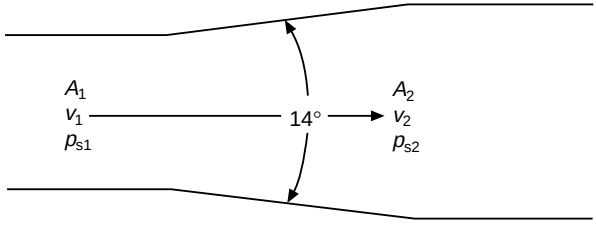


Figure 27.11 *Duct expansion with static pressure regain*

Example 27.5 Air moving in a duct at 8 m/s is gently expanded to a velocity of 5.5 m/s. If the friction losses are 20% of the available velocity pressure change, what is the amount of static regain?

$$\text{Velocity pressure entering expansion} = 0.5 \times 1.2 \times 8^2$$
$$= 38.4 \text{ Pa}$$
$$\text{Velocity pressure leaving expansion} = 0.5 \times 1.2 \times 5.5^2$$
$$= 18.15 \text{ Pa}$$
$$\text{Friction losses} = 0.2(38.4 - 18.15) = 4.05 \text{ Pa}$$
$$\text{Static regain} = 0.8(38.4 - 18.15) = 16.2 \text{ Pa}$$

27.8 Flow of air in a room

Since incoming air may be as much as 11 K colder or 25 K warmer than the conditioned space, the object of the duct and grille system must be to distribute this air and mix it with the room air with the least discomfort to the occupants. The subjective feeling of discomfort will depend on the final temperature difference, the velocity, and the degree of activity, cold air being less acceptable than warm. Velocities at head level should be between 0.1 m/s and 0.45 m/s and comprehensive factors will be found in the CIBSE Guide, Table B3.1.

Figure 27.12*a* shows a typical office or hotel room with supply duct in the central corridor ceiling space and a wall grille blowing air towards the window, which will usually be the greatest source of heat gain or loss. High-level discharges of this type work best when cooling, since the incoming air jet will fall as it crosses the room. On heating it will tend to rise, and so must have enough velocity to set up a forced circulation in the pattern shown.

Figure 27.12*b* shows perimeter units under the window and discharging upwards to absorb the heat flow through the window. The angle and velocity of discharge should be enough for the air to

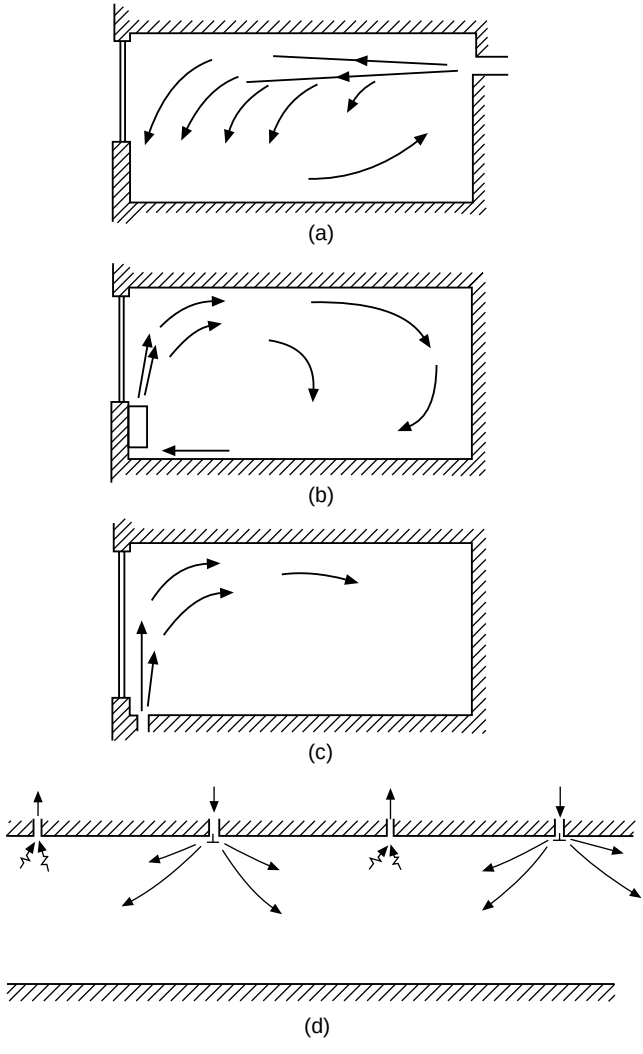


Figure 27.12 *Room air circulation patterns. (a) Grilles on corridor wall. (b) Sill outlets. (c) Floor outlets. (d) Ceiling slots*

set up a circulation within the room to reach the far wall or, in an open-plan room, to the area covered by an adjacent grille. In such units, air returns to the face or underside of the unit.

Figure 27.12*c* shows floor slots, setting up a pattern similar to the perimeter units. This arrangement has been adopted in some buildings having all glass walls. The position of the return grille varies with room layout and stagnant zones can occur.

Figure 27.12*d* shows ceiling grilles or slots, requiring all ducting

to be within the ceiling void. This system is generally adopted for open plan rooms, since the area can be divided into strips with alternate supply and extract slots, or into squares (or near-squares) for supply and extract by grilles.

It will be seen that all of these patterns require some consideration and planning, since the best equipment cannot cope with an impossible air circuit [59].

27.9 Grilles

The air inlet grille will be recognized as a device for converting static pressure to outlet velocity, having the required speed and direction to take the conditioned air across the room and entrain the surrounding air so as to reach the occupants at a suitable temperature.

Wall Grilles will have directional vanes in one or both planes, which can be set on commissioning. These need to be set by a competent person who is aware of the required room flow pattern. It is advisable that such a setting adjustment should be operated by special tools to prevent subsequent tampering.

Perimeter unit grilles must direct the air upwards and slightly away from the window to start the circulatory pattern. Fixed angles are preferable on these units, rather than the adjustable segments supplied with many packaged products. A common fault with such an installation is obstruction of inlet and outlet grilles by office equipment. It will be a definite advantage if the upper surface is sloped, to discourage its use as a shelf.

The geometry of ceiling grilles and sots can be fixed or adjustable. In the former case the flow pattern is set by the spacing and volume (= velocity), so site adjustment cannot always compensate for a faulty layout. Slots have a limited throw, of the order of 5 m at full volume, and the layout of supply and return slots must take into account any operating variation in this volume. The setting on commissioning of all these, especially those of adjustable geometry, should be left to competent hands and then locked. Figure 27.13 shows two kinds of grille.

For very large areas, such as assembly halls or sports arenas, jets of air will be required to obtain the large throw distances. Localized draughts may be unavoidable in such installations.

Many air moving devices now have discharge grilles which move during operation, to distribute the air in a varying pattern, to avoid stagnation.

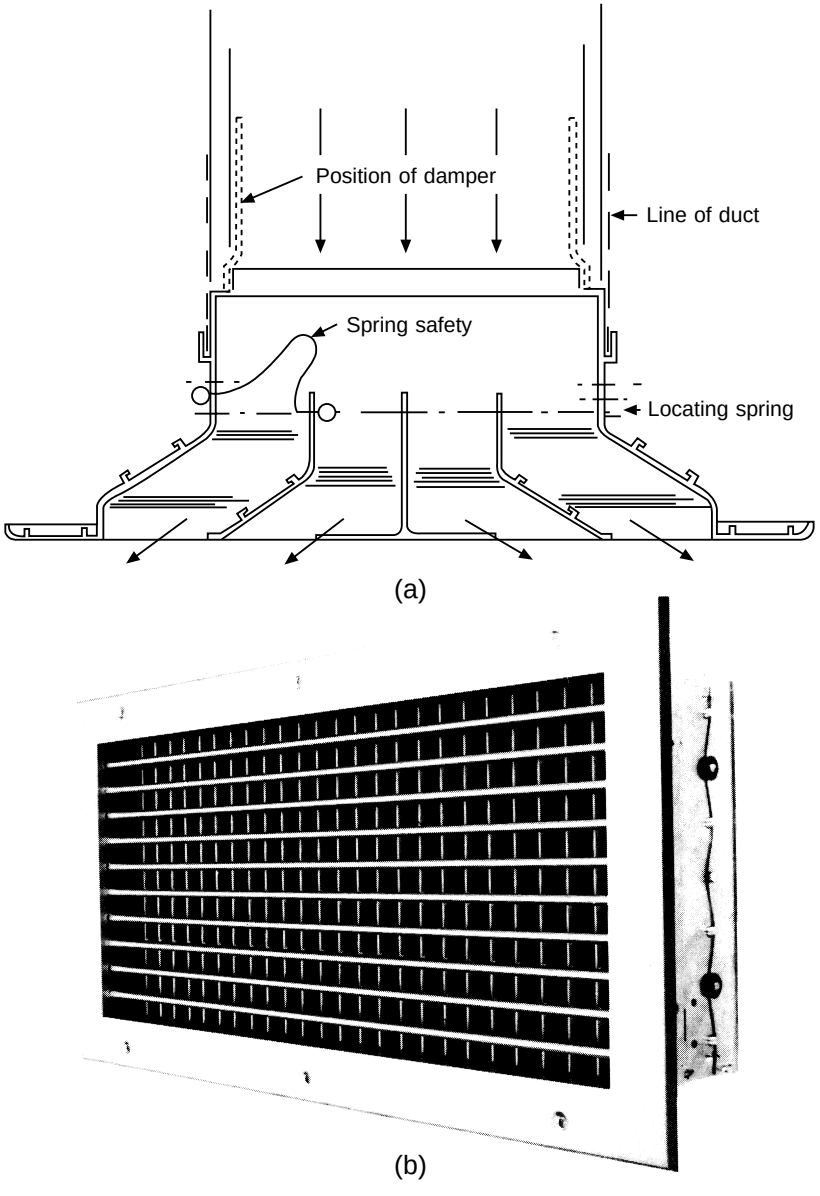


Figure 27.13 *Air discharge grilles. (a) Ceiling (Courtesy of Myson RCM Ltd). (b) Wall*

27.10 Return air

Air entering a return duct will be moved by the difference in pressure, the duct being at a lower static head than the room. Such movement

will be radially towards the inlet and non-directional. At a distance of only 1 m from the grille this pressure gradient will be quite low, so return grilles can be located close to supply grilles, providing the overall circulation pattern ensures coverage of the space. In Figure 27.12*a*, the return air grilles can also be in the corridor wall, if far enough from the inlets. (See Figure 27.14.)

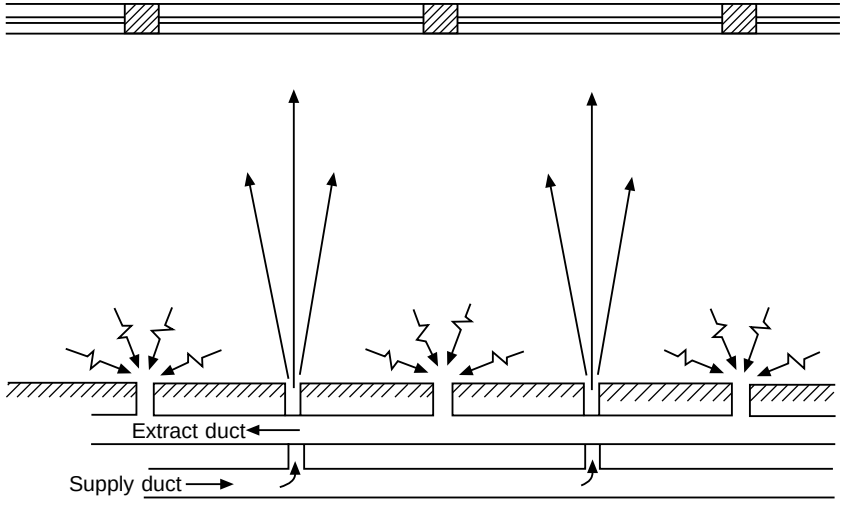


Figure 27.14 *Discharge and return grilles on same wall (plan view)*

With ceiling inlet and extract systems, the opportunity is presented to remove heat from light troughs. This can reject a proportion of the cooling load, possibly as high as 20%, in the exhaust air. The recirculated air is also warmer, improving heat transfer at the cooling coil. (See also Example 26.3.)

27.11 Air filtration

Ambient air contains many solid impurities, ranging from visible grit down to fine dusts, smokes and fumes [10]. An air-conditioning system will aim to remove a proportion of these, depending on the application. There are three reasons for air filtration:

1. To remove impurities which may be harmful to a process, e.g. fine dust in a computer room, bacteria in a pharmaceutical packing room
2. For the comfort of occupants and the cleanliness of papers and furnishings
3. To keep the inside of the air-conditioning apparatus and ductwork clean

Impurities may be classified by size:

Pollens	9–80 μm
Mould spores	3–50 μm
Fine ash	0.7–60 μm
Bacteria	1–10 μm
Tobacco smoke	0.1–7 μm
Viruses	up to 0.1 μm

Filtration apparatus is available to remove any size, but the very fine particles require a deep, bulky and expensive filter, which itself sets up a high resistance to air flow and therefore requires high fan power. A practical balance must be reached to satisfy the requirements:

1. To remove a high proportion of impurities in the air
2. To hold a large weight of dust before having to be cleaned or replaced, so as to reduce the frequency of maintenance to an acceptable level (i.e. if maintenance is required too frequently, it may be neglected)
3. The filter must be cleanable or reasonably cheap to replace

A high proportion of the weight of dust and fluff in the air is in large particles and so is fairly easy to trap. Filters for general air-conditioning duty comprise a felt of glass or other fibres, used in a dry state and termed 'impingement filters'. Air passage through the fibres is turbulent, and dust particles strike the fibres and adhere to them. The filter material may be flat, but is more usually corrugated, so as to present a large surface area within a given face area. A typical filter in a comfort air-conditioning system is 50 mm deep and may collect 95% or more of the impurities in the air, down to a size of 1 μm.

Increased dust-holding capacity can be obtained by making the filter material in a series of bags, which are normally about 400 mm deep, but also made up to 900 mm where maximum retaining capacity is required. Some bag filters are shown in Figure 27.15.

Finer filtration is possible, down to 0.01 μm. Such filter elements are only used when the process demands this high standard. These fine filters would clog quickly with normal-size impurities, so they usually have a coarser filter upstream, to take out the larger dusts. They are about 300 mm deep, and require special mounting frames so that dirty air cannot escape around the edges.

Very fine particles such as smokes can be caught by electrostatic precipitation. A high voltage is applied to plates or wires within the filter bank, to impart a static charge to dirt particles. These will then be attracted to earthed plates, and adhere to them. Impurities are generally cleaned off the plates by removing the stack and washing.

Figure 27.15 *Bag filters (Courtesy of Camfil Ltd)*

Electrostatic filters will not arrest large particles, and need to be backed up by coarser impingement filters for this purpose.

As a filter element collects dust, the air resistance through it will rise, to a point where the system air flow is impaired. Users need to have an objective indication of this limit, and all filters except those on small package units should be fitted with manometers (see Figure 27.2). On installation, marks should be set to indicate 'clean' and 'dirty' resistance pressure levels.

Dry impingement filters cannot be effectively cleaned and will usually be replaced when dirty. Thin filters of this type are used on some package air-conditioners and much of the dirt can be dislodged by shaking, or with a vacuum cleaner. The problem of air filtration on small packaged units is the low fan power available and the possible neglect of maintenance. Since users will be reluctant to buy new filters when needed, some form of cleanable filter is employed. One such type is a plastic foam. Where replaceable filters are used, it is good practice to always have a complete spare set ready to insert, and to order another set when these are used. This avoids the inevitable delay which will occur if new filters are not ordered until the need is urgent.

Air filters are not used on cold store coolers, since the air should be a lot cleaner and small amounts of dust will be washed off the fins by condensate or by melted frost. Air-cooled condensers are not fitted with filters, since experience shows that they would never be maintained properly. In dusty areas, condensers should be selected with wide fin spacing, so that they can be cleaned easily.

27.12 Cleanliness and cleaning of ducting

Filters in air-conditioning systems do not remove all the dirt from the air, and this will settle on duct walls. There is an increasing awareness that ducting systems can harbour a great deal of dirt, and that this dirt will hold bacteria, condensed oils such as cooking fats and nicotine, fungi and other contaminants.

Where ducting cannot be stripped down for cleaning, it is strongly advisable to leave frequent access holes for inspection and cleaning. Some guidance on this subject will be available from HVCA [57] in 1989.

28 Air-conditioning methods

28.1 Requirement

The cooling load of an air-conditioned space comprises estimates of the sensible and latent heat gains, and is $Q_S + Q_L$. This rate of heat flow is to be removed by a cooling medium which may be air, water, brine or refrigerant, or a combination of two of these. (See Figure 28.1.)

Example 28.1: All air A space is to be held at 21°C dry bulb and 50% saturation, and has an internal load of 14 kW sensible and 1.5 kW latent heat gain. The inlet grille system is suitable for an inlet air temperature of 12°C. What are the inlet air conditions and the mass air flow?

$$\text{Inlet air temperature} = 12.0°\text{C}$$

$$\text{Air temperature rise through room, } 21 - 12 = 9.0 \text{ K}$$

$$\text{Air flow for sensible heat, } \frac{14}{9 \times 1.02} = 1.525 \text{ kg/s}$$

$$\text{Moisture content of room air, } 21°\text{C, } 50\% = 0.007\,857 \text{ kg/kg}$$

$$\text{Moisture to pick up, } \frac{1.5}{2440 \times 1.525} = 0.000\,403$$

$$\text{Moisture content of entering air} = 0.007\,454$$

From tables [4], this gives about 85% saturation.

Note that the figure of 1.02 in the third line is a general figure for the specific heat capacity of moist air, commonly used in such calculations. (The true figure for this particular example is slightly higher.) The figure of 2440 for the latent heat is, again, a general quantity in common use, and is near enough for these calculations.

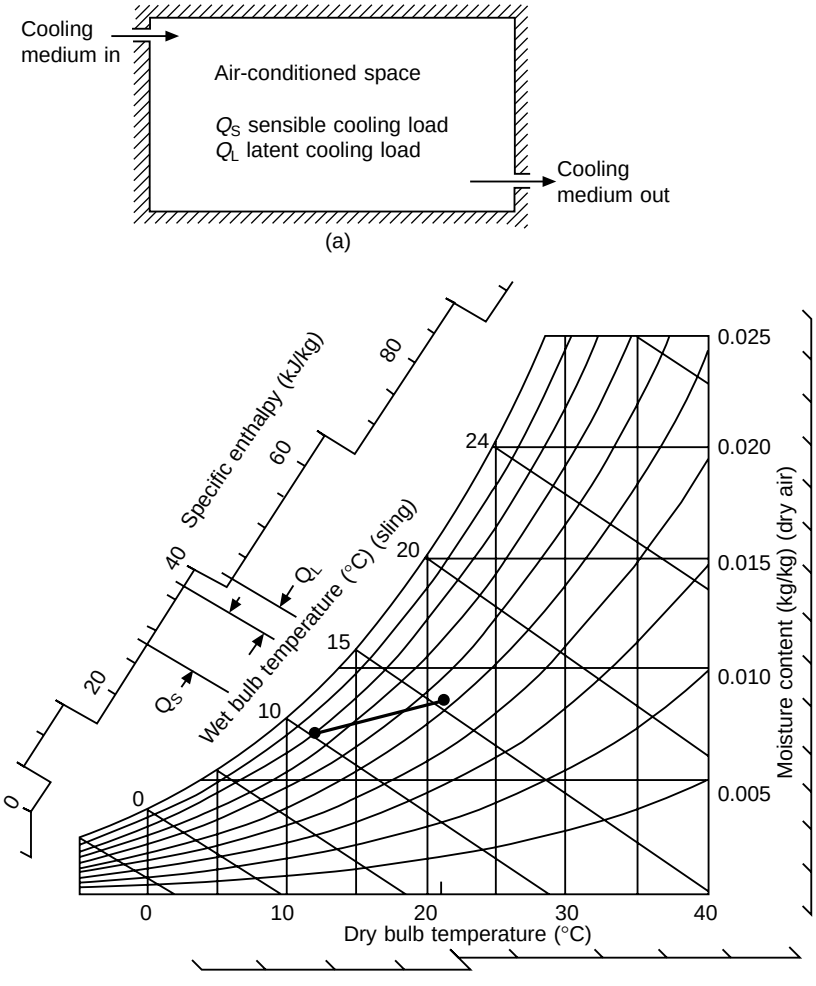


Figure 28.1 *Removal of sensible and latent heat from conditioned space. (a) Flow of cooling medium. (b) Process line*

Example 28.2: Chilled water For the same duty, a chilled water fan coil unit is fitted within the space. Water enters at 5°C and leaves at 10.5°C. The fan motor takes 0.9 kW. What water flow is required?

Total cooling load, 14.0 + 1.5 + 0.9 = 16.4 kW

$$\text{Mass water flow, } \frac{16.4}{4.19 \times (10.5 - 5)} = 0.71 \text{ kg/s}$$

Example 28.3: Refrigerant For the same duty, liquid R.22 enters the expansion valve at 33°C, evaporates at 5°C, and leaves the cooler at 9°C. Fan power is 0.9 kW. What mass flow of refrigerant is required?

Total load, as Example 27.2 = 16.4 kW

Enthalpy of R.22, evaporated at 5°C,
superheated to 9°C = 309.39 kJ/kg

Enthalpy of liquid R.22 at 33°C = 139.84 kJ/kg

Refrigerating effect = 169.55 kJ/kg

Required refrigerant mass flow, $\frac{16.4}{169.55}$ = 0.097 kg/s

Example 28.4: Primary air and chilled water For the same application, primary air reaches induction units at the rate of 0.4 kg/s and at conditions of 13°C dry bulb and 72% saturation. Chilled water enters the coils at 12°C and leaves at 16°C. What will be the room condition and how much water will be used?

The chilled water enters higher than the room dew point temperature, so any latent heat must be removed by the primary air, and this may result in a higher indoor condition to remove the design latent load:

Moisture in primary air, 13°C DB, 72% sat. = 0.006 744 kg/kg

Moisture removed, $\frac{1.5}{2440 \times 0.4}$ = 0.001 537 kg/kg

Moisture in room air will rise to = 0.008 281 kg/kg

which corresponds to a room condition of 21°C dry bulb, 53% saturation.

Sensible heat removed by primary air,
$0.4 \times 1.02 \times (21 - 13)$ = 3.26 kW

Heat to be removed by water, 14.0 – 3.26 = 10.74 kW

Mass water flow, $\frac{10.74}{4.19 \times (16 - 12)}$ = 0.64 kg/s

28.2 Air-conditioning and comfort cooling

The removal of heat within an enclosed space must be considered as a multi-step heat transfer process. Heat passes from the occupants or equipment to the air within the space, and from there to the refrigerant or chilled water. It follows that the temperature differences at each step are a reciprocal function of the air mass flow. Where there is a high latent heat load within the space, the relative humidity will also vary with the air flow – the variation being higher with low air flow.

Further unintended variations will occur with the flow of the primary cooling medium. With two-step (on–off) control of the compressor within an air-conditioning unit, the temperature will slowly rise while the compressor is 'off' until the compressor re-starts.

The design engineer must consider the effect of such variations on the load within the space. This governs the selection of the cooling apparatus and method of control. A wide variation of equipment is available and the engineer needs to be aware of the characteristics and correct application of each.

Close control of conditions may require diversion of the main air flow, see Figure 28.10, or moving human operatives outside the sensitive area. Coolant flow control should be modulating or infinitely variable, where possible.

Where conditions can be allowed to drift, within the general limits of human comfort, see Figure 23.8, or any similar zone which is acceptable to a majority of the occupants. Such standards of air-conditioning are generally termed *comfort cooling*.

28.3 Central station system. All air

The centralization of all plant away from the conditioned space, originating from considerations of safety, also ensures the best access for operation and maintenance and the least transmission of noise. Since all air passes through the plantroom, it is possible to arrange for any proportion of outside air up to 100%. This may be required for some applications, and the option of more outside air for other duties will reduce the refrigeration load in cold weather. For example, in the systems considered in Section 28.1, there may still be a cooling load required when the ambient is down to 12°C dry bulb, but this is the design supply air temperature, so all cooling can be done with ambient air and no mechanical refrigeration.

The distribution of air over a zone presupposes that the sensible and latent heat loads are reasonably constant over the zone (see Figure 28.2). As soon as large variations exist, it is necessary to provide air cold enough to satisfy the greatest load, and re-heat the air for other areas. Where a central plant serves a number of separate rooms and floors, this resolves into a system with re-heat coils in each zone branch duct (see Figure 28.3). It will be recognized that this is wasteful of energy and can, in the extreme, require a re-heat load almost as high as the cooling load.

To make the central air system more economical for multizone installations, the quantity of cooled air to the individual zones can be made variable, and reduced when the cooling load is less. This

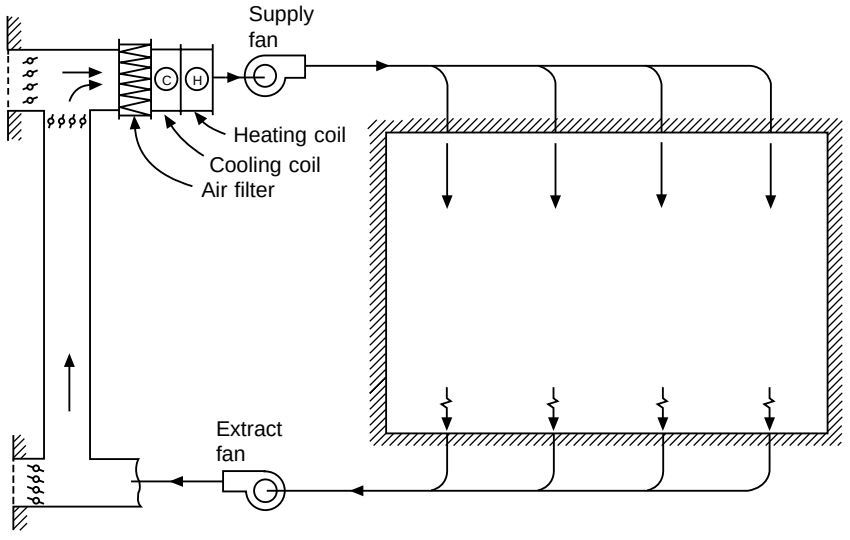


Figure 28.2 *All-air system*

will also reduce the amount of re-heat needed. This re-heat can be by means of a coil, as before, or by blending with a variable quantity of warmed air, supplied through a second duct system (see Figures 28.4 and 28.6).

In the first of these methods, the reduction in air mass flow is limited by considerations of distribution velocities within the rooms, so at light load more air may need to be used, together with more re-heat, to keep air speeds up. Within this constraint, any proportion of sensible and latent heat can be satisfied, to attain correct room conditions. However, full humidity control would be very wasteful in energy and a simple thermostatic control is preferred.

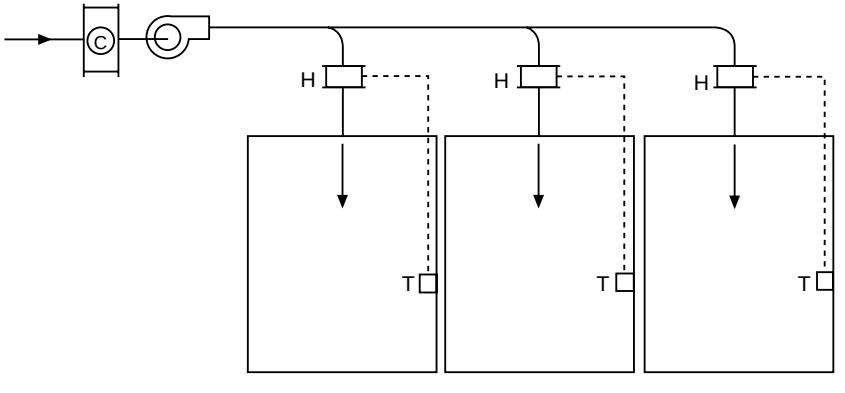


Figure 28.3 *Re-heat for individual zones*

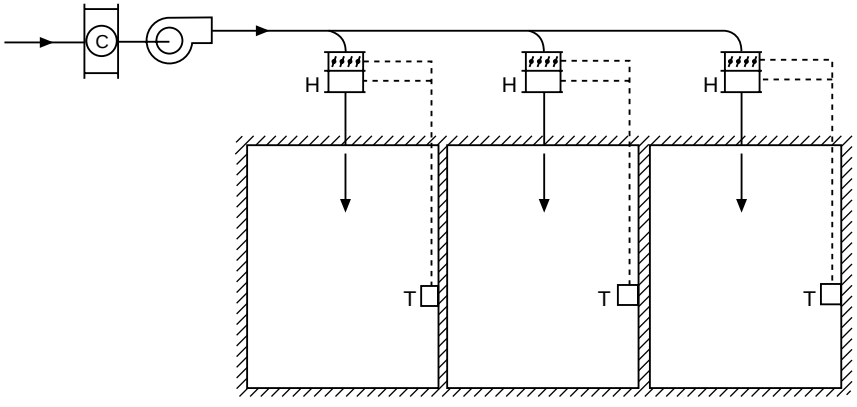


Figure 28.4 *Variable air flow with re-heat to individual zones*

Example 28.5 A room is to be maintained at 21°C, with a preferred 50% saturation, using air at 13°C dry bulb, 78% saturation and re-heat. The load is 0.7 sensible/total ratio. (See Figure 28.5.)

Air at the supply condition can be re-heated to about 18°C and will rise from 18°C to 21°C in the room, picking up the quantity of

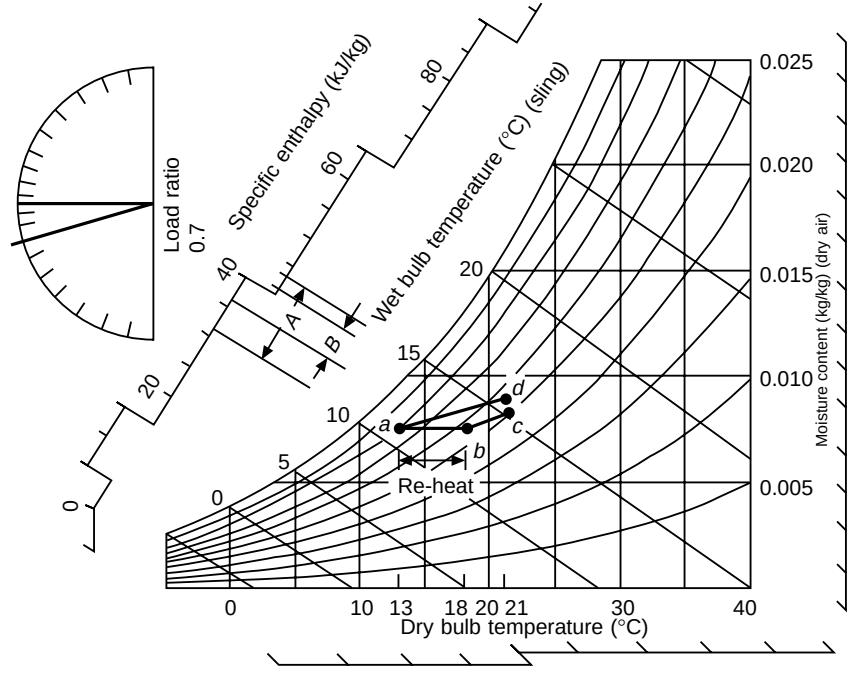


Figure 28.5 *Zone differences with re-heat*

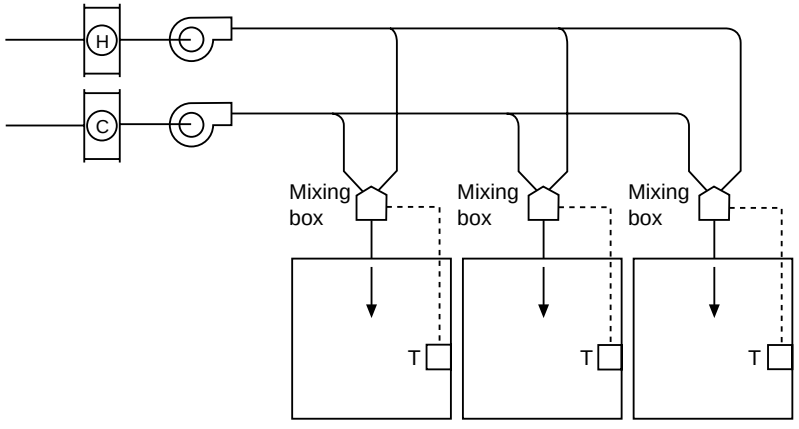


Figure 28.6 *Dual duct supplying separate zones*

heat 'B' as shown. The final condition will be 50% saturation, as required (line abc).

Alternatively, supply air is used directly, without re-heat. It now picks up the quantity of heat 'A' (about three times as much) and only one-third the amount of air is needed. The final condition will be about 55% saturation. This is still well within comfort conditions, and should be acceptable (line ad).

With this variable volume method, the cold-air supply system will be required to deliver less air into the building during colder weather and must be capable of this degree of 'turn-down'. Below 30% of design flow it may be necessary to spill air back to the return duct, with loss of energy and, in some types, cold air in the ceiling void when trying to heat the room. If the final throttling is at the inlet grille, the reduction in grille area will give a higher outlet velocity, which will help to keep up the room circulation, even at lower mass flow. One type releases the room air in pulses, to stimulate room circulation.

The *dual-duct* system, having the second method of heating by blending cold and warm air, has reached a considerable degree of sophistication, normally being accommodated within the false ceiling and having cold and warm air ducts supplying a mixing chamber and thence through ceiling grilles or slots into the zone (see Figure 28.6).

The blending of cold and warm air will be thermostatically controlled, so that the humidity in each zone must be allowed to float, being lowest in the zones with the highest sensible heat ratio.

Example 28.6 A dual-duct system supplies air at 14°C dry bulb, 75% saturation through one duct, and at 25°C dry bulb, 45%

saturation through the other. Two zones are to be maintained at 21°C and in both cases air leaves the mixing boxes at 17°C. Room A has no latent load. Room B has a sensible/total heat ratio of 0.7. What room conditions will result? (See Figure 28.7.)

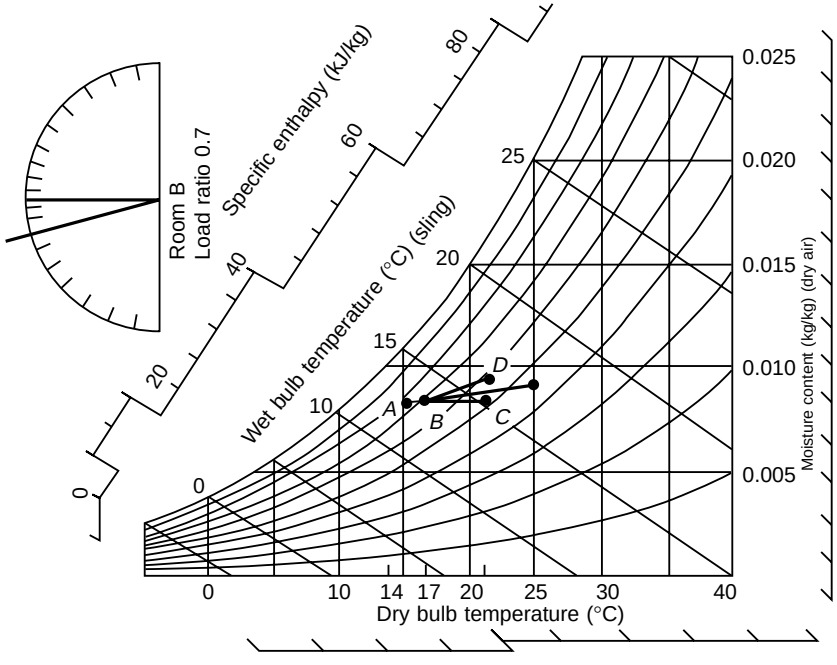


Figure 28.7 *Dual-duct differences*

Air leaving the mixing boxes will lie along the line *HC*. For these two zones it will be at *M* (17°C dry bulb). For room A, air will enter at *M* and leave at *A*, the process line being horizontal, since there is no latent heat load. The final condition is about 50% saturation. For room B, air enters at *M* and the slope of the line *MB* is from the sensible/total angle indicator. Condition B falls at about 56% saturation.

This example gives an indication of the small and usually acceptable variations found with a well-designed dual-duct system. Since a constant total flow is required with the basic dual-duct circuit, a single fan may be used, blowing into cooling and heating branches. Where variation of volume is employed, one or two fans may be used, as convenient for the circuit. In all cases an independent extract fan and duct system will be required, so that the proportion of outside/recirculated air can be controlled.

Since about 0.1 m^3/s of air flow is required for each kilowatt of cooling, the mass air flow for a large central station system will be

large and the ductwork to take this very bulky. This represents a loss of available building space, both in terms of vertical feed ducts and the extra ceiling space to accommodate branches on each floor. For a tall building, it may be necessary to have a number of plantrooms for air-handling equipment (fans, coils, filters) with the refrigeration machinery central. Instances will be seen in major cities of tall buildings having 'blank' floors to accommodate air-handling plant.

Reduction of duct size can be achieved by increasing the velocity from a low velocity of 3–6.5 m/s to a high velocity of 12–30 m/s. Such velocities cause much higher pressure losses, requiring pressures in excess of 1 kPa, for which ductwork must be carefully designed and installed, to conserve energy and avoid leakage. The use of high velocity is restricted to the supply ducts and is not practical for return air ducting.

With a supply system pressure of 1 kPa and another 250 Pa for the return air duct, the total fan energy of a central all-air system may amount to 12.5% of the maximum installed cooling load, and a much greater proportion of the average operating load. This power loss can only be reduced by careful attention to design factors.

A comprehensive and detailed analysis of all-air systems can be found in [19] (Chapter 3).

28.4 Zone, all-air systems

It will be seen that the limitation of the central station all-air system is the large ductwork and the need to arrange dual ducts or re-heat to each branch. If the conditioned space can be broken down into a number of zones or areas in which the load is fairly constant, then a single-zone air-handling unit with localized ductwork may be able to satisfy conditions without re-heat in its branches. The success of such a system will depend on the selection of the zones. Large open offices can be considered as one zone, unless windows on adjacent or opposite walls cause a diurnal change in solar load. In such cases, it will be better to split the floor into arbitrary areas, depending on the aspect of the windows. Some local variations will occur and there may be 'hot spots' close to the windows, but conditions should generally be acceptable by comfort standards.

The air-handling unit for the zone may be one of several types:

1. Direct expansion, supplied with refrigerant from the central plantroom
2. Chilled water air-handling unit taking chilled water from a package or the central plantroom

3. Water-cooled packaged direct expansion unit, using condenser water from an external tower
4. Remote condenser (split) air-cooled direct expansion unit; condenser remote, possibly on roof
5. Air-cooled direct expansion unit, mounted adjacent to an outside wall, or through the roof

28.5 Central station, combined air and chilled water

The chilled air of the central station system serves the purpose of providing the proportion of fresh air needed, and carrying heat energy away from the space. These functions can be separated, using a more convenient fluid for the latter. Since the heat is at a temperature well above 0°C the obvious choice of fluid is water, although brines are used for some applications.

The central plant is now required to supply chilled water through flow and return pipes, plus a much smaller quantity of fresh air. No air return duct may be needed.

The chilled water will be fed to a number of air-handling units, each sized for a suitable zone, where the conditions throughout the zone can be satisfied by the outlet air from the unit. This constraint has led to an increasing tendency to reduce the size of the zones in order to offer the widest range of comfort conditions within the space, until the units now serve a single room, or part of a room. Such units are made in wall-mounted form for perimeters or ceiling-mounted form to cover open areas. (See Figure 28.8.) Larger units may be free-standing.

Two methods are used to circulate the room air over the chilled water coil. In the first, an electric fan draws in the air, through a filter, and then passes it over the coil before returning it to the space. The fan may be before or after the coil. The fresh air from the plantroom may be introduced through this unit, or elsewhere. The coil is normally operated with a fin temperature (ADP) below room dew point, so that some latent heat is removed by the coil, which requires a condensate drain. Multispeed fans are usual, so that the noise level can be reduced at times of light load.

The second method makes use of the pressure energy of the primary (fresh) air supply to induce room (secondary) air circulation. This air, at a pressure of 150–500 Pa, is released through nozzles within the coil assembly, and the resulting outlet velocity of 16–30 m/s entrains or induces room air to give a total circulation four or five times as much as the primary supply. This extra air passes over the chilled water coil. Most induction units are wall mounted for perimeter cooling, but they have been adapted for ceiling mounting.

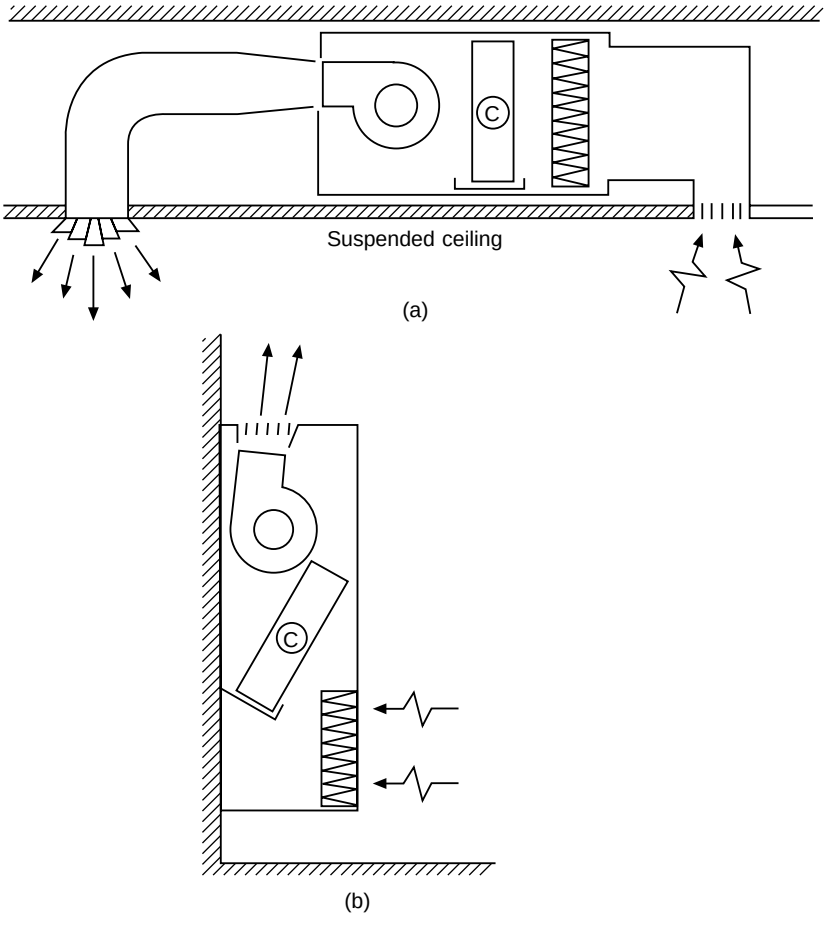


Figure 28.8 *Fan coil units. (a) Ceiling. (b) Perimeter*

With induction units, latent heat extraction can usually be handled by the primary air and they run with dry coils. Some systems have been installed having high latent loads which remove condensate at the coil.

In climates which have a well-defined summer and winter, heating when required can be obtained with fan coil or induction units, by supplying warm water to the coil instead of cold. Some variation of this is possible with induction systems which can, at times, have cold primary air with warm water, or vice versa, giving a degree of heating–cooling selection.

Most climates, however, have mid-seasons of uncertain weather so that heating and cooling may be required on the same day, and this is accentuated by buildings with large windows which may need

cooling on winter days. For these applications, units need to have a continuous supply of both chilled and warm water and a suitable control to choose one or the other without wastage. This usually implies two separate coils and four pipes, with separate chilled and warm water circuits. (See Figure 28.9.)

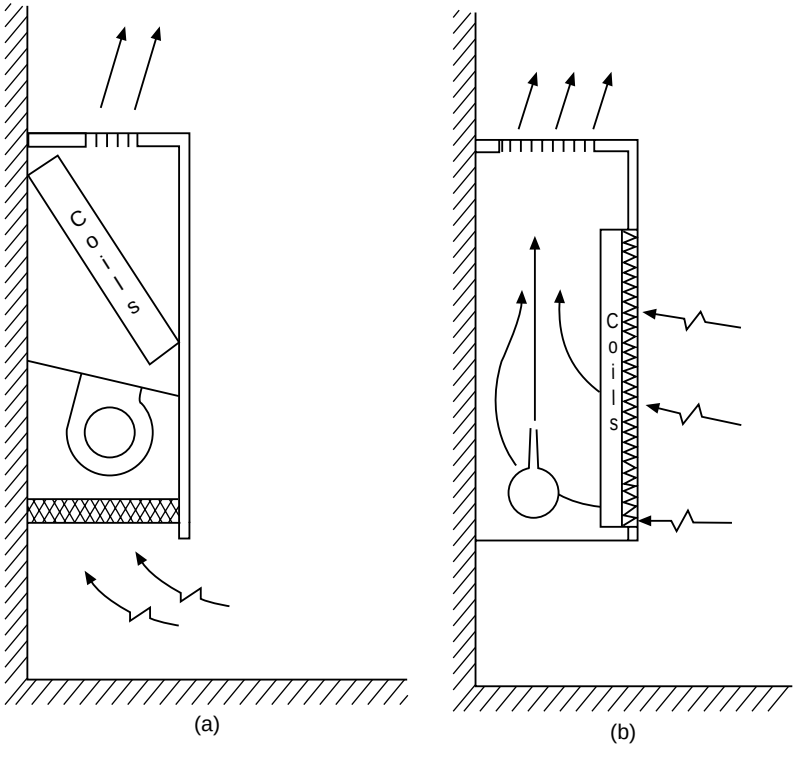


Figure 28.9 *Air-handling units. (a) Fan coil. (b) Induction*

An alternative system, lower in first cost, is the three-pipe system. Chilled and warm water are piped to the coil unit and chosen by the room thermostatic valve for cooling or heating duty as required. Water leaving the coil passes through a common third return pipe back to the plantroom. At times of peak cooling load, very little warm water is used and there may be little or no wastage of energy in this mixing of the water streams.

28.6 Underfloor systems

A room with a lot of heat-generating apparatus such as computers will have a high cooling load, and require a high air flow to carry

this heat away. If this amount of air was circulated in the usual way it would be unpleasantly draughty for the occupants.

Since computer cabinets stand on the floor, the general solution is to blow the cold air up from a false floor directly into the cabinets, with a lesser volume being blown into the room to deal with other heat loads. The air-conditioning unit will now stand on the floor, taking warm air from the upper part of the room and blowing it down into the false floor (see Figure 28.10).

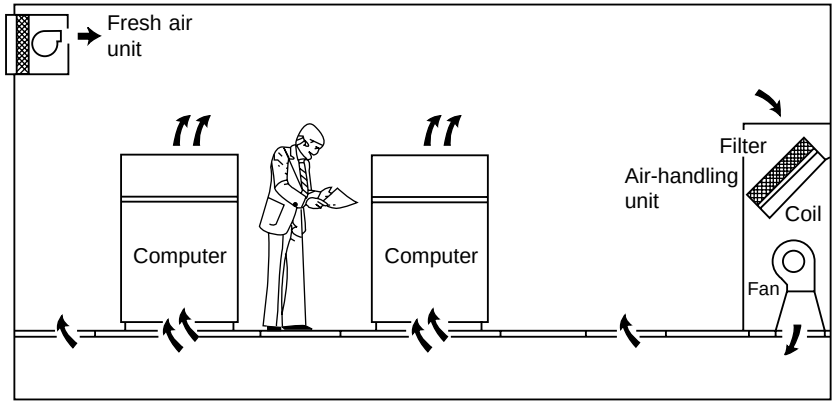


Figure 28.10 *Raised floor computer room system*

Such units may use chilled water or direct-expansion refrigerant, and will have the air filter at the top. It may not be possible to introduce outside air through it, so the room will have a pressurized fresh air supply, which will be filtered to remove fine dusts which may affect the computers. Computer room units work with a very high sensible heat ratio of 0.95 or more, so they have large coils to keep the ADP up near the dew point of the room air. Most will have an inbuilt steam humidifier to replace any moisture which is removed on the coil.

28.7 Packaged air-cooling units

This no clear demarcation between a zone, served by a unit package, and a single room or part of a room. A zone is an arbitrary selection by the design engineer, and air-handling packages are available in a very wide range of sizes to cope with such a range of loads. By definition, all such units are room air-conditioners, and fall into three classifications:

1. Self-contained, where all of the refrigeration circuit components

are integral parts of the unit. If not specifically stated, it is assumed to be air cooled, i.e. with an integral air-cooled condenser.
2. Water cooled, having an integral water-cooled condenser.
3. Split, having the condenser remote from the air-handling section, and connected to it with refrigeration piping. The compressor may be in either section.

The size of a packaged unit will be limited by installation restrictions, both in handling items into place and in the quantity of air required by the condenser. Units up to 50 kW cooling capacity are in common use.

Control of the indoor cooled condition will be by thermostat in the return airstream, and thus based on room dry bulb temperature. The resulting humidity level will depend on coil characteristics and air flow. Packaged air-conditioners for tropical applications commonly have a design coil sensible/total ratio in the order of 0.7 with entering air at 50% saturation, and will give indoor conditions nearer 45% saturation if used in temperature climates with less latent load (see Chapter 35).

Winter heating items fitted within room air-conditioners may be electric resistance elements, hot water or steam coils, or reverse cycle (heat pump). One model of water-cooled unit operates with a condenser water temperature high enough to be used also in the heating coil.

The *heat reclaim* packaged unit system comprises water-cooled room units with reverse cycle valves in the refrigeration circuits. The water circuit is maintained at 21–26°C, and may be used as a heat source or sink, depending on whether the individual unit is heating or cooling. (See Figure 28.11.)

If the water circuit temperature rises above about 26°C, the cooling tower comes into operation to reject the surplus. If the circuit drops below 21°C, heat is taken from a boiler or other heat source to make up the deficiency. During mid-season operation within a large installation, many units may be cooling and many heating, so that energy rejected by the former can be used to the latter. With correct system adjustment, use of the boiler and tower can be minimized.

Example 28.7 A large office building is to be fitted with a packaged unit in each room. During mid-season, it is estimated that 350 rooms will require cooling at an average rate of 3.5 kW and another 120 rooms will require 2 kW of heating. Three alternative systems are proposed. Calculate running costs at this time of year.

(*a*) Air-cooled units, COP 2.8, with electric heaters

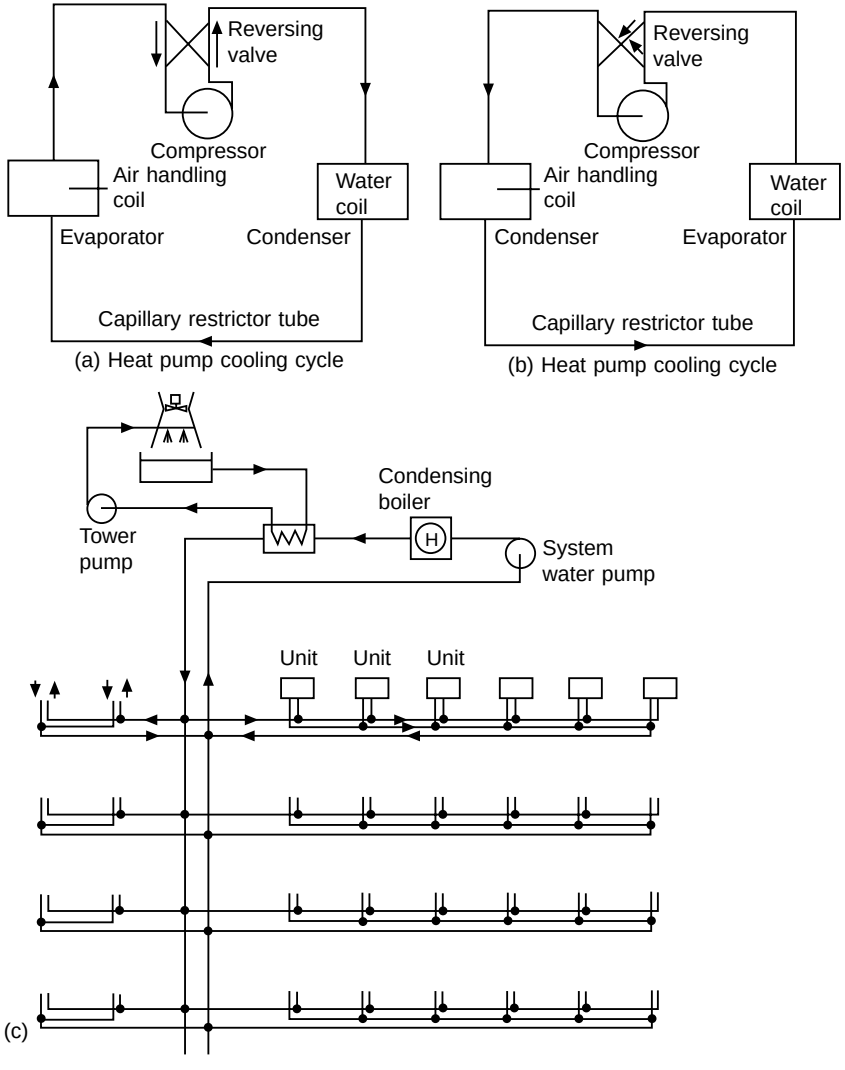


Figure 28.11 *Heat reclaim system. (a) Unit cooling. (b) Unit heating. (c) System*

(*b*) Air–air heat pump units, having a cooling COP of 2.7 and a heating COP of 2.2

(*c*) Heat reclaim units, having a cooling COP of 3.1, a heating COP of 2.6, and requiring an average of 25 kW for pumps and the tower fan

Electricity costs 5.3p/(kW h) and gas for the condensing boiler in system (c) costs 36p/therm, and is burnt at an efficiency of 92% (giving an overall gas cost of 1.34p/kW h).

	Hourly running costs (£)		
	(a)	(b)	(c)
350 × 3.5/2.8 × 0.053	23.2		
120 × 2 × 0.053	12.7		
350 × 3.5/2.7 × 0.053		24.0	
120 × 2.0/2.2 × 0.053		5.8	
350 × 3.5/3.1 × 0.053			20.9
120 × 2.0/2.6 × 0.053			4.9
25 × 0.053			1.3
	35.9	29.8	27.1

It should be noted that this example is general, and indicates the type and method of cost analysis which should be made before the selection of an air-conditioning system for any building.

28.8 Multisplits

It is possible to run two or more indoor units from a single condensing unit, with economies in the number and costs of components. Such systems are referred to as *multisplits*, and several different types of circuit will be encountered.

The usual split package air-conditioner comprises one condensing unit connected by pipes to one evaporator unit (Figure 13.4). Twin condensing units are made to save on outdoor casings and reduce the number of pieces on a roof or wall. Such twins will be connected in the usual way to two separate indoor units.

Units having single-speed compressors will require some automatic method of shedding the excess cooling capacity when some of the fan-coil units do not call for cooling. Liquid from the condenser coil passes directly through an expansion valve, and the resulting mixture of cold liquid and flash gas is distributed to each of the fan-coil units on the circuit. On–off control of the cold liquid to each room is effected by a solenoid valve within each indoor unit, which will be switched by the room thermostat. Returned refrigerant gas, sometimes with unevaporated liquid present, is caught in a suction trap before entering the compressor, or the liquid is boiled off with a suction/liquid heat exchanger. Both the outgoing and return refrigerant pipes to each fan-coil unit must be carefully insulated.

If any of the rooms does not require cooling, then the excess compressor capacity is taken up by injecting hot gas directly from the compressor discharge into the return. Under conditions of light load the head pressure will fall, and this pressure must be maintained by slowing the condenser fan. It may also be necessary to inject

some liquid into the return pipe, if the bypass gas makes it too hot. (See Figure 9.7.) This system is for cooling only. The COP will be lower with less than all indoor units in action. Typical values with three indoor units are:

with 3 units cooling COP = 2.2
with 2 units cooling COP = 1.6
with 1 unit cooling COP = 0.9

Better balance between cooling capacity and load can be obtained by capacity control of the compressor(s). Large systems will have a number of compressors, or built-in capacity control on the cylinders. A central condensing unit of this sort may be coupled to several fan-coils, each with its own thermostat and liquid solenoid value. The COP is good at all but the lowest load levels.

The usual method for smaller units is thyristor speed control of the compressor, and such compressors can be slowed to 40% of full speed or less. Some can also be run at higher speeds than normal, to deal with peak loads, but the COP will fall off at this condition. As many as eight fan-coil units can be connected to one condensing unit, and the installed number can be nominally greater than the compressor capacity, on the basis that not all will be cooling at the same time.

28.9 Three-pipe splits

To provide either cooling or heating at any of the fan-coil units in a system requires a supply of hot gas. This will normally come from a third pipe, taken from the compressor discharge with a feed through a valve to the top of the indoor coil, and a drain back to the main liquid line. A non-return valve is needed to prevent the hot gas flowing back to the compressor suction. Speed control of the condensor fan and if available, of the compressor is needed to keep the discharge pressure high enough to supply the required hot gas.

The use of a number of components connected in this way implies that they are integrated into a coherent circuit with compressors, fans, solenoid valves etc. under a common control system. A few major manufacturers in the world are capable of engineering a complex system of this sort and supplying matching components and training the staff to instal and maintain it.

28.10 Two-pipe splits

It is possible, with the correct selection of component packages, to arrange a two-pipe circuit which will heat in one indoor unit and

cool in another (see Figure 28.12). The outdoor unit is connected to a distribution device local to a group of fan-coils, and the direction of gas and liquid flow will be determined by the overall balance of load, whether cooling or heating. If most indoor units call for cooling, the flow will be as follows. Discharge gas from the compressor passes through a four-port valve into the outdoor coil, and is partly condensed to a high-pressure mixture of liquid and gas. The liquid is separated in the distributor unit, and passed through two stages of pressure reduction, evaporates in the indoor unit and is returned as a low-pressure gas, through a controlling solenoid valve to the compressor. If any room calls for heating, the solenoid valves change over – the 'cooling' solenoid closing and the 'heating' solenoid opening. This admits hot gas from the top of the separator to the coil, where it gives up its heat to the room air, and condenses. This liquid flows into the liquid header in the distributor, and can then pass directly to a 'cooling' unit.

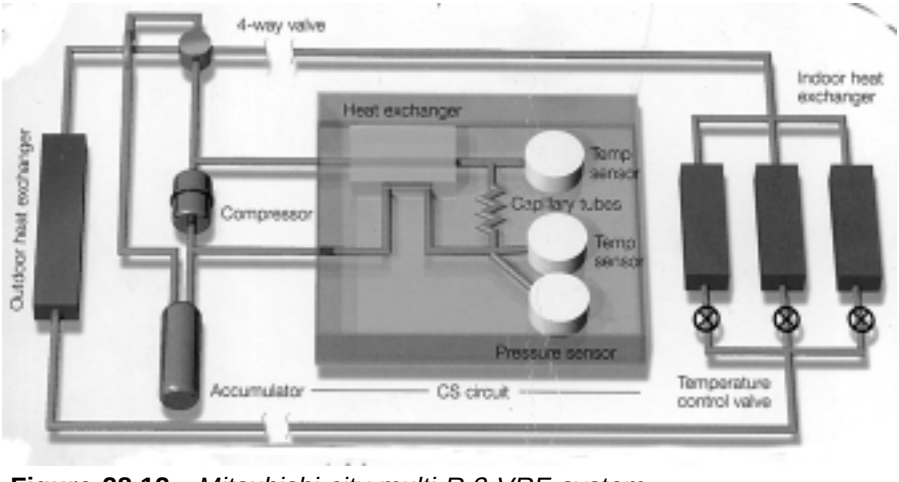


Figure 28.12 *Mitsubishi city multi R.2 VRF system*

If the greater demand is for heating, the flows are reversed. The hot gas from the compressor goes directly to the solenoid valves, and so to each unit, to give up its heat, liquefy and pass back to the compressor unit, to be boiled off in the outdoor coil. Now if any unit wishes to cool, the solenoid valves on this item changes over and liquid will be drawn from the heater in the distributor, and pass as a gas, through the separator and back to the compressor.

It will be seen that the operation of this two-pipe cooling and heat pump circuit is delicately balanced, and requires electronic control of the compressor and outdoor fan speeds in order to pump the right amount of gas, and still maintain working pressures.

28.11 Noise levels

All air systems have a noise level made up of the following:

1. Noise of central station machinery transmitted by air, building conduction, and duct-borne
2. Noise from air flow within ducts
3. Grille outlet noise

The first of these can be reduced by suitable siting of the plantroom, anti-vibration mounting and possible enclosure of the machinery. Air flow noise is a function of velocity and smooth flow. High-velocity ducts usually need some acoustic treatment.

Grille noise will only be serious if long throws are used, or if poor duct design requires severe throttling on outlet dampers.

Apart from machinery noise, these noises are mostly 'white', i.e. with no discrete frequencies, and they are comparatively easy to attenuate.

Where machinery of any type is mounted within or close to the conditioned area, discrete frequencies will be set up and some knowledge of their pattern will be required before acoustic treatment can be specified. Manufacturers are now well aware of problems to the user and should be able to supply this basic data and offer technical assistance towards a solution.

Where several units of the same type are mounted within a space, discrete frequencies will be amplified and 'beat' notes will be apparent. Special treatment is usually called for, in the way of indirect air paths and mass-loaded panels [10, 19, 56, 60].

Useful practical guidance can be gained by visiting existing installations before taking major decisions on new plant.

29 Dehumidifiers and air drying

29.1 Psychrometrics

Moisture can be removed from any material which is to be dried, by passing air over it which has a lower water vapour pressure. Also, in removing this moisture, the latent heat of evaporation must be supplied, either directly by heating, or by taking sensible heat from the airstream which is carrying out the drying process.

Moisture may be removed from air by passing it over a surface which is colder than its dew point (see Figure 24.9). In air-conditioning systems this is a continuous process, providing that the moisture condenses out as water and can be drained away. If the apparatus dew point is below 0°C, the moisture will condense as frost, and the process must be interrupted from time to time to defrost the evaporator.

Air will leave the evaporator section with a reduced moisture content, but at a low temperature and high percentage saturation. In this state, it may not be effective in removing moisture from any subsequent process, as it will be too cold.

In the unit dehumidifier process, all or part of the condenser heat is used to re-heat the air leaving the evaporator (see Figure 29.1*a*). Since the moisture in the air has given up its latent heat in condensing, this heat is reclaimed and put back into the outlet air. In a typical application, air at 25°C dry bulb and 60% saturation can be dried and re-heated to a condition of 46°C dry bulb and 10% saturation (see Figure 24.13). In this state, it is hot enough to provide the necessary latent heat to dry out the load product. The entire system is in one unit, requiring only an electrical supply and a water drain, so there is no constraint on location.

The efficiency of a unit dehumidifier can be improved by a heat exchanger which pre-cools the incoming air by using the cold air leaving the evaporator (see Figure 29.1*b*).

The performance of a dehumidifier in terms of moisture removal

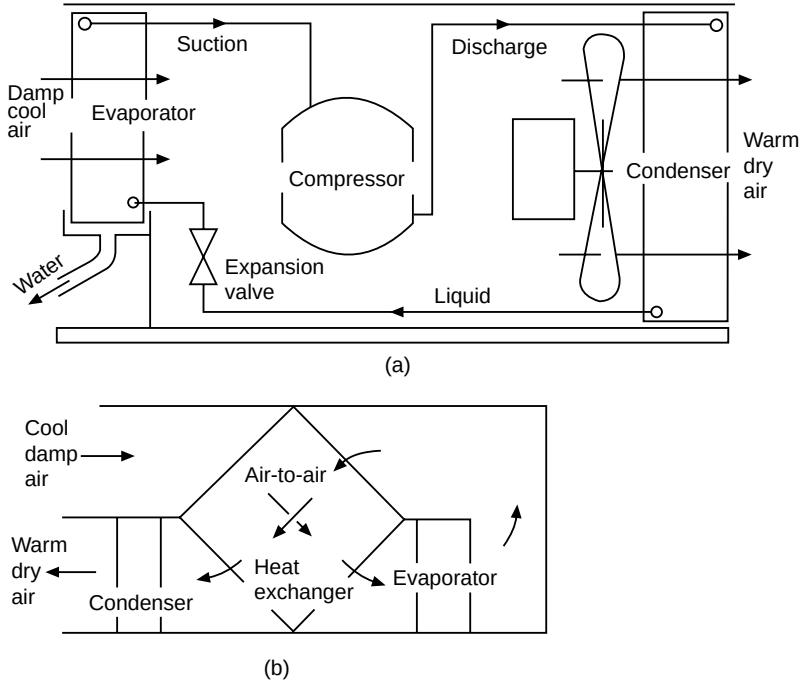


Figure 29.1 *(a) Dehumidifier. (b) Unit dehumidifier with heat recovery*

will vary very considerably with the condition of the incoming air. Typical capacity figures are shown in Figure 29.2.

The refrigeration method of drying air is the most energy efficient, down to a lower limit of about 0.005 kg/kg moisture content at atmospheric pressure. Equipment to work at frosting conditions can be duplicated, one evaporator defrosting while the other is operating. Below this limit, chemical or adsorption drying must be used [61].

29.2 Compressed air drying

If the air pressure is increased, the partial pressure of the moisture goes up in the same proportion, and more moisture can be removed without frosting the cooling surface. Air-drying evaporators for pressures above atmospheric will be designed as pressure vessels, and will take the form of shell-and-tube, shell-and-coil or plates. Such driers will be found on compressed air installations, to remove moisture from the air which would otherwise settle in distribution piping, valves and pneumatic machines, causing corrosion which is accelerated by the high partial pressure of oxygen. By means of

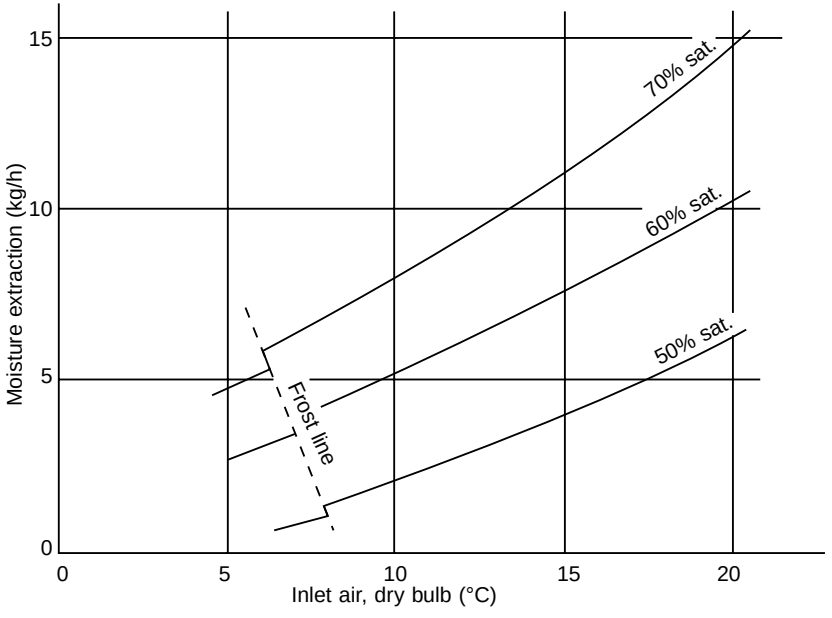


Figure 29.2 *Performance of dehumidifier*

refrigerated driers, compressed air at 7 bar can be dried to a moisture content of less than 0.001 kg/kg.

Depending on the end use of the compressed air, some or all of the condenser heat can be used to re-heat the cold air. This may be necessary in winter, when distribution piping could be colder than the evaporator. When the air is released through a power tool, the final condition may be less than 5% saturation.

Unit driers for small compressed air systems need to have capacity control, so as to maintain a steady working dew point when there is a variation in air demand.

29.3 Applications

Packaged one-piece dehumidifiers are used for:

1. Maintaining a dry atmosphere for the storage of metals, cardboard, books, timber, etc., that is, any material which is better preserved in low humidity
2. Removal of moisture from newly constructed or plastered buildings, to expedite final decoration and occupation
3. Drying out buildings which have been left unoccupied for some time, or have a condensation problem
4. Removal of excess moisture from indoor swimming pools
5. Some crop drying

The drying load in a swimming pool will vary throughout the year, and dehumidifiers built for this application may have an air-cooled condenser for re-heating the air, and also a water-cooled condenser, so that some proportion of the heat may be used to warm the pool water. Automatic controls will use the condenser heat to the best advantage.

29.4 High-temperature dehumidifiers

The kilning, or accelerated drying, of newly cut timber requires higher temperatures than will usually be found in refrigeration and air-conditioning systems. Typically, the air will be above 50°C and may be up to 80°C. Condensing temperatures of 85–90°C require R134a in open compressors, or the use of a more specialized refrigerant.

Similar high-temperature dehumidification has been used in the drying of other fibrous materials and ceramics.

30 Heat pumps. Heat recovery

30.1 The heat pump

If the flow of refrigerant in a cooling system is reversed, the heat exchanger which was the evaporator becomes the condenser and vice versa (see Figures 9.4*b* and 28.11) and the flow of energy is also reversed. Reversing valves may be fitted to cooling circuits in order to divert hot gas for defrosting the evaporator or to provide heat in winter to a conditioned space.

Where a refrigeration system is to provide cooling in summer and heating in winter, the design of the system will need to be a compromise of the two functions. In particular, the expansion valve cannot work in both directions, so two valves are required, with the necessary extra piping and non-return valves (see Figure 30.1).

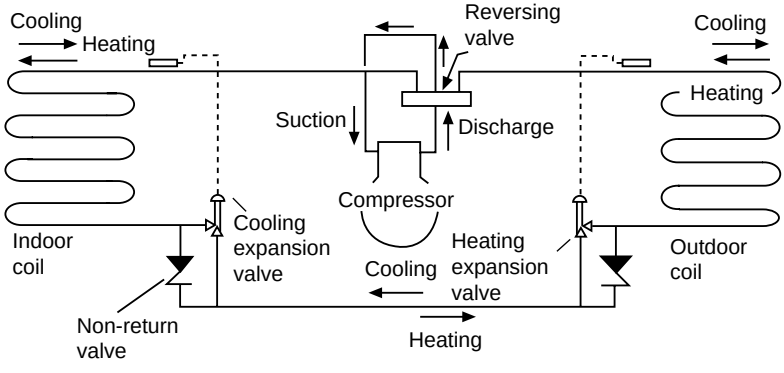


Figure 30.1 *Piping arrangement for reversible flow circuit*

The most advantageous application of the heat pump in the UK at the present time is where air cooling is required in the summer and air heating from the same piece of equipment in the winter. The alternatives will be electric resistance heating, with a higher fuel cost, or the provision of a hot water coil, at extra capital cost.

Purpose-built heat pumps will have no cooling function, but operate only to draw heat from a low-temperature source and reject this to a load at a higher temperature. There is no reversal of the refrigerant flow, so the selection and design of the system can be optimized for the duty.

Some heat pumps, such as the dehumidifier, serve the double purpose of a useful load both on the evaporator and condenser side. Again, there is no reversal of the refrigerant flow in operation so the components may be selected without compromise.

Applications can be classified according to the low-temperature heat source and the load. Typical examples are shown in Table 30.1.

Table 30.1

Heat source	*Load*
Ambient air	Heating a building
Warm air from a plantroom	Pre-heating fresh air
Extract air	Pre-heating water
Waste warm water	Pre-heating a product in an industrial process
Waste steam condensate	Drying malt
River, lake or sea	Swimming pool heating
Ice rink	
Ground	
Solar	

In all cases the source heat must be available to provide the end product heat at a cost lower than alternative methods of heating. In some instances, such as the use of ambient air in winter, the least amount of heat is available when the load is likely to be greatest. This will result in a lower COP and the need to defrost the evaporator, with a requirement for alternative heating while this is taking place. Water sources, also, may be cold enough in winter for ice to form. If this happens, the refrigerant circuit must be stopped until the flow of source water melts the ice off again. Solar heat is unreliable in most areas, and will not be available at night. The source heat needs careful examination, and a study should be made of possible extreme conditions during the working period.

Since the system may fall short of intended performance under extreme conditions, heat pumps should not be designed to provide all the heat required. They should give a base load, to the best of their ability, as conditions permit, and the balance provided with top-up heat from a conventional fuel.

Evaporators for heat pumps are designed for a low temperature difference, to get the best COP from a given heat source, so they

are larger than usual. If the refrigerant is one of the halocarbons, the circuitry of the evaporator will need special attention, to ensure good oil return and avoid very large refrigerant charges. There is a very real risk of the refrigerant migrating to the wrong part of the circuit when the heat pump is shut down, causing danger on restarting. The design of the evaporator may be different to conventional types, using bare pipe if under water for ease of cleaning, or for strength and corrosion resistance if buried in the ground. As ground and solar source heat exchangers have to be very large, the collector itself may be plastic tube and a secondary fluid pumped through this to a shell-and-tube evaporator.

Low source temperatures may require high compression ratios with subsequent low COPs. Where the system is large enough, two-stage compression may give improved working. In any case, the refrigerant system needs careful examination and design, taking into account the running conditions.

Large systems working at temperatures above freezing may be able to use centrifugal compressors, but the pressure ratio is not favourable. Most heat pumps will have reciprocating or screw compressors. Refrigerants are those in general use.

Since the end product is to be useful heat, there is an opportunity with heat pump compressors to use a drive motor which will contribute to this. The internal combustion engine is used for some drives, adding its radiator heat to the load, and at a higher temperature than the refrigerant condenser can provide. As the installation is static, and required to run for comparatively long times without attention, the ideal fuel is natural gas. Compressor and drive assemblies must be robust to withstand the extra vibration, and should be separate from the rest of the circuit. Engine combustion air intakes must be from outside the plantroom and possible refrigerant leaks. Steam drives have also been used.

The economics of the heat pump are dependent on the relative costs of the drive power and heat energy from other sources. In the UK, where electricity is currently about three times the price of natural gas, the electric drive heat pump needs to have a COP of 3 to break even, and 4 or more to be competitive. This price ratio will change as fossil fuels run out in the next century.

30.2 Heat recovery

The aim of heat recovery is to avoid wastage of any quantity of heat or cold energy which has been generated within a system. Methods of recovery may be passive or active. Mechanical heat recovery systems will generally be found under the description of Heat Pumps.

The first step is energy conservation, which is the subject of Chapter 34. Recovery of rejected or wasted heat requires a careful analysis of the heat flow within the systems under survey. Points to examine are:

1. Where cooled or warmed air is exhausted from a building or process, this might be used in heat exchangers to pre-cool or pre-warm fresh air brought in to replace it.
2. Where cooled or warmed liquids leave a process, heat exchangers can provide the means of pre-cooling or pre-warming fresh liquids entering. One of the main instances of this is the warm waste water from showers in changing rooms.
3. Hot discharge gas from a refrigeration circuit can be used to heat water (see Figure 6.5).
4. Condenser heat can be diverted into a building, for heating in winter (see Figure 20.3).

30.3 Apparatus and methods

Passive heat exchanger equipment for air heat exchange is described in Section 26.4 and shown in Figure 26.5. This can only be used where the ducts are adjacent. Other methods are:

1. The rotating heat exchanger wheel. The wheel has a rotating matrix, the mass of which picks up heat from one duct air flow and transfers it to the other. If the matrix is coated in a hygroscopic material, there may also be some transfer of moisture.
2. Coils in the two ducts, with a fluid circulated between them. It is important to get the fluid in counterflow in the two airstreams (see also Figure 1.4*c*). If fresh air is to be heated in winter, the fluid must be a non-freeze solution.
3. Heat pipes between the two ducts. These comprise a coil made with closed pipes, filled with a volatile liquid. This liquid will condense in one coil and evaporate in the other, at the same pressure and therefore at the same temperature.

All these methods will transfer heat in either direction, so providing heat recovery in summer and winter. All devices using air should be protected by filters, or they will choke with dirt and become ineffective.

Heat exchangers for liquids will be double pipe, shell-and-tube and plates (see Figures 6.3, 6.4 and 17.1). Waste fluids may be contaminated by the process, and heat exchangers for such fluids must be cleanable, and kept clean.

All heat exchange equipment should be fitted with indicating thermometers at the inlet and outlet of both fluids, so that operation can be monitored.

31 Control systems

31.1 Function

The purpose of a control system on a refrigeration or air-conditioning plant is to:

1. Provide automatic operation, i.e. avoid the cost of attendant labour or for where control is too complex for manual operation
2. Maintain the controlled conditions closer than could be achieved by manual operation
3. Provide maximum efficiency and economy of operation
4. Ensure safe operation at all times

The control system will consist of a loop, with detector (sensor), controller and controlled device. The communication between these parts of the loop will be electric, pneumatic or mechanical (see Figure 31.1) [62].

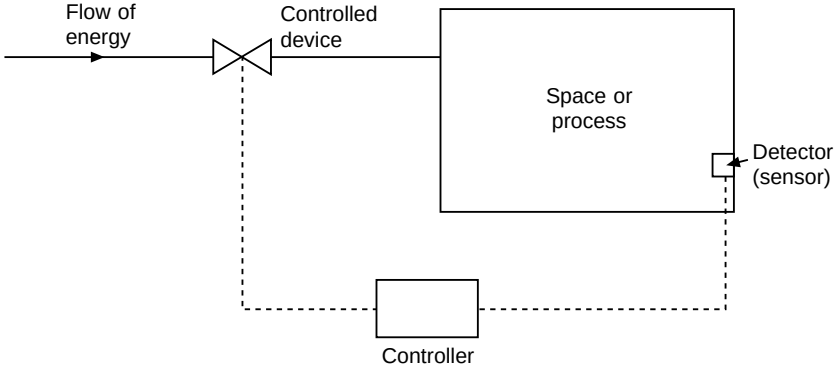


Figure 31.1 *Basic control loop*

31.2 Detectors (sensors)

Types of detector are two-position (on–off) and proportional. The

two-position will be set to actuate at upper and lower limits, and will respond when its sensitive element reaches these set limits. Since all devices have some time lag in operation, the controlled condition will overshoot to some extent, depending on the time lag of the detector and the extent to which the rate of supply of energy to the process exceeds the load. (See Figure 31.2.) The range of the control will therefore be the differential of the detector plus the upper and lower overshoots under load.

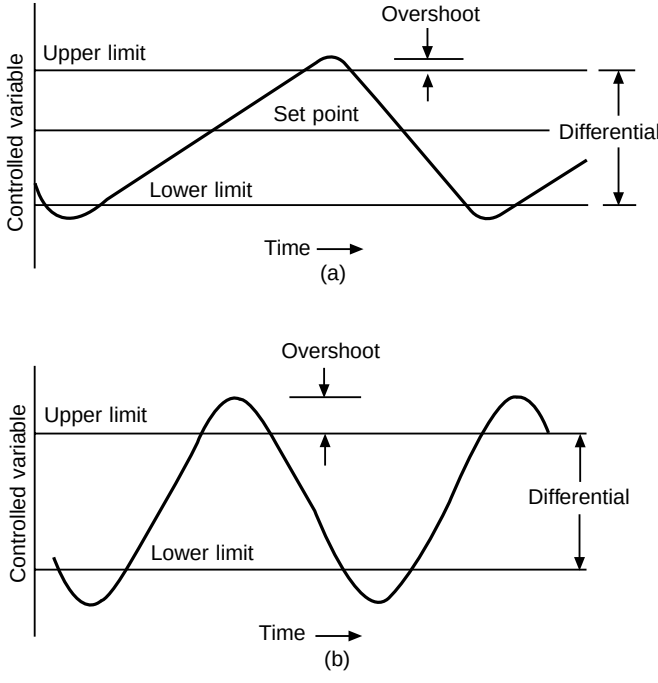


Figure 31.2 *Limits of controlled variable with two-position control. (a) Capacity closely matched to load. (b) Capacity much greater than load*

Two-position detectors such as thermostats can be fitted with an anticipatory bias to reduce the amount of overshoot. In such instruments, a small bias heater accelerates the action of the control. An alternative method to reduce overshoot is to introduce a timing device so that it acts intermittently. Where the two-position sensor is also the controller it provides only two plant outputs, maximum or zero.

A two-position detector can be used to operate a floating control. At the upper limit it will operate the control in one direction and if it reaches the lower limit it will operate the control in the other

direction. Between the two limits the control is not actuated. (See also Section 31.4.)

Two-position detectors can be classified according to the purpose:

Thermostatic	bimetal liquid expansion solid expansion vapour pressure
Pressure	diaphragm bellows bourdon tube
Fluid flow	moving vane
Time	clock bimetal and heater dashpot
Humidity	dimensional change of hygroscopic element
Level	float

Many of these devices are direct acting on the controlled device and do not require a controller to process the signal.

Proportional detectors measure the process condition, which can then be compared by the controller with the required value. They are not direct acting, and need a controller to convert the signal to a working instruction to the controlled device. Proportional detectors include:

Temperature	those above, plus electrical resistance of a metal or a semiconductor thermocouple infrared radiation
Pressure	those above, plus piezo-electric
Fluid flow	that above, plus electronic, Doppler effect orifice plate and manometer
Time	those above, plus electronic timing devices
Humidity	that above, plus resistance of a hygroscopic salt
Level	float with impedance coil

31.3 Controllers

If a controller is used with an on–off detector, it functions only as an amplifier to transmit the detector signal to the controlled device. It can modify the speed of this action by a bias or by a slow-speed operating motor, as in the floating control.

The floating control normally takes the form of a slowly rotating reversible motor moving a valve or operating a sequence of cams which control, in turn, steps of plant capacity. As the detector reaches its upper or lower limit it energizes the motor to advance or reduce the valve opening or the steps of plant capacity. When the condition has been satisfied and the detector moves away from the limit, the motor stops. The motor may be a solid-state timing or pulsing device.

Some proportional detectors are combined in the same instrument with a suitable transducer which can perform some of the functions of a controller. For example, for pneumatic systems the primary sensing element actuates a variable air jet, thus modulating an air pressure which is transmitted to a further controller or direct to the controlled device. Electric and electronic detectors such as the infrared detector include the sensing and amplifying circuits of the instrument.

Controllers generally for use with proportional detectors will measure the displacement of the signal from a pre-set value and transmit a proportional signal to the controlling device. They may also be able to measure the rate of change of that signal (derivative) or be able to modify the rate of change of the output signal (integral). The effect of these capabilities is to anticipate the deviation and so give better response to changes of load. A controller having proportional, derivative and integral actions is known as a three-term controller.

A controller may be arranged to accept input signals from more than one detector, e.g. the flow temperature of a hot water heating system may be raised at the request of an outdoor detector if the ambient falls, or may start the heating earlier in the morning to pre-heat the building before it is occupied; a servo back pressure regulation valve (Figure 9.5) can respond both to evaporator pressure and load temperature. With the advent of microcomputer devices almost any combination of signals can be processed by an electronic controller, providing the output signals can be made coherent and not conflicting.

Pneumatic controllers, which may include part of the sensing instrument, are supplied with compressed air at 1 bar gauge which is allowed to escape from an orifice controlled by a detector. The resulting pressure modulates about 0.4 bar and is used in a servo

piston, diaphragm or bellows to actuate the controlled device. (See Figure 31.3.)

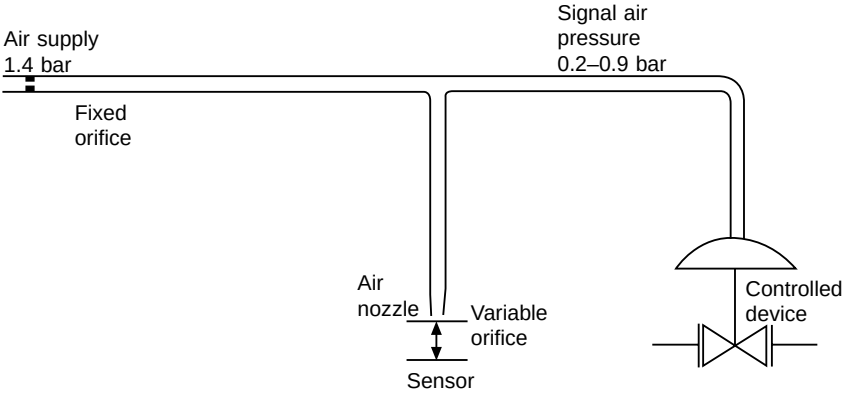


Figure 31.3 *Pneumatic operation of controlled device*

31.4 Controlled devices

Controlled devices commonly consist of an actuator, which accepts the signal from the controller and works the final element. Typical examples are as follows:

1.	Electric relay	operating	contactor motor motorized valve dampers
2.	Electric solenoid	operating	solenoid valve
3.	Modulated electronic signal	operating	magnetically positioned valve thyristor power control
4.	Pneumatic pressure (and hydraulic)	operating	pneumatic relay valve positioner damper positioner

The effect of a controlled device may not be proportional to its movement. In particular, the shape of valve plugs and the angle of opening of dampers will not give a linear result, and the signal from the controller must take this into account [10, 62].

31.5 Controls communications

Communications between the component parts of a controls system may be by:

Mechanical means – rods, levers and cables
Fluid pressure
Mains voltage electricity, usually 230 V but sometimes 115 V
Low voltage, mainly 24 V a.c.
Low voltage, electronic circuits
Low voltage, thermocouple (microvolts)
Telephone cable
Pneumatic
Optical fibre
Optical – infrared
Radio

Mechanical devices need careful installation to ensure that there is no distortion of the parts. This is especially the case with damper mechanisms, which need maintenance and periodic inspection to ensure they are working correctly.

Where fluid pressure is carried by a capillary tube, such as with the thermostatic expansion valve or pressure switches, the tube should be installed with due attention to the risk of it chafing against metal edges and wearing through. Tubes to manometers are usually in plastic, but may be copper. These must be carefully tested for leaks, as they are transmitting very low pressures.

Mains voltage communications must be run according to IEE Wiring or the appropriate safety regulations. In particular, these may cause interference with telephonic, computer and other electronic signals carried in or near the same conduit. In the same way, electronic control signals may suffer interference. Thermocouple signals are very low voltage d.c. and should be run as far as possible with unbroken conductors. Terminal boxes should be compact and insulated from sudden temperature changes. Terminals must be tight.

Pneumatic controls are used widely in hazardous situations such as chemical plants and oil refineries. The same risk of chafing applies as with capillary tubes. Pneumatic tubing is more usually in copper and is correctly secured.

Optical fibres are not yet very much in use, but there is no interference between them and electrical signals of any sort. For this reason their use will probably become more widespread. Line-of-sight optical signals require that no obstruction is inserted at any time. Such points are easily noticed when installing and commissioning, but are not so obvious if a malfunction occurs at a later date.

Remote plant is sometimes controlled or monitored by radio link. This is subject to interference and should only be considered if the cost of a permanent or telephonic connection is uneconomic.

31.6 Control system planning

Control systems can quickly escalate into unmanageable complexity, and the initial approach to the design of a suitable control system should examine the purpose of each item, and the effect on others, to eliminate those which are not essential.

The action of a control may combine two or more of the purposes, as set out in Section 31.1, which may then be interdependent. It is more informative to consider the action of a control and examine what purpose it may serve in the circuit.

Controls for economic operation should ensure that functions are shut down when not needed (the boiler in summer and the chiller in winter). Optimum start controls now complement the starting clock, to advance or retard the starting time according to the ambient.

In planning a control system, a flow diagram is needed to indicate what may influence each item of plant. In many diagrams it will be seen that complexity arises and two items work in conflict. A typical instance is the cooling and dehumidifying of air, to a room condition lower than design, with concurrent operation of a humidifier.

Since most controls will be electrical and largely of the two-position type, it is a convenient notation to set out the initial control scheme as an electrical circuit and in 'book page' form, i.e. from left to right and line by line, to indicate the sequence of operation, with the controlled device always in the right-hand column. This analysis should indicate the different items which might act to produce a final effect and bring errors to light. Figure 31.4 is a simplified control circuit for a small air-conditioning system. Non-electrical items can be shown on the same initial scheme, possibly with dotted lines to indicate a non-electrical part of the system. The possibilities of abnormal operation should be examined, and grouped as system not working, system unsafe and system dangerous, and protected accordingly. The last category requires two independent safety controls or one control and an alarm.

Complex timing and logic controlling, monitoring and indication can now be carried out with programmable computer-type devices, using algorithms stored in RAM or EPROM. These save the former complex arrangements of sequencing and interlock relays and timers but still require the same attention to planning and design of the circuit. In all cases, a copy of the basic control diagram should be left with the device, to inform users and service staff of the plan of the control system, and any subsequent modifications updated on this diagram.

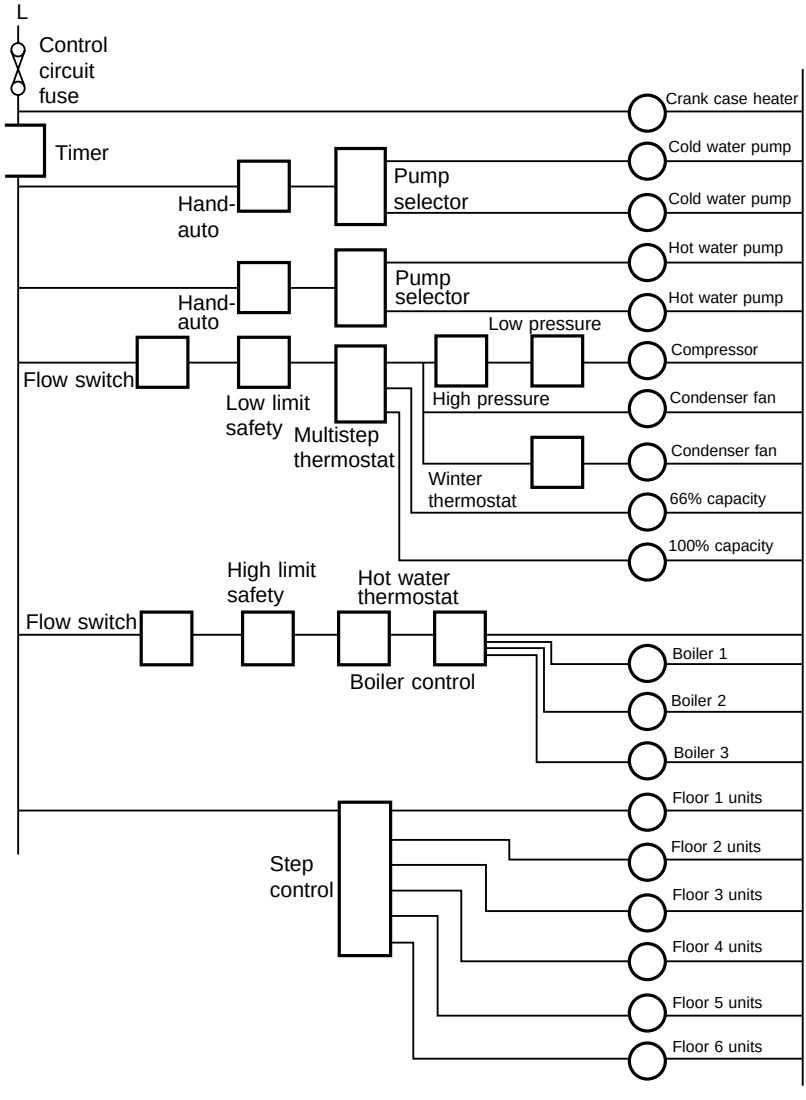


Figure 31.4 *Electrical control diagram for small air-conditioning system*

31.7 Commissioning of control systems

The setting up, testing and recording of all control functions of a refrigeration or air-conditioning system must be seen as part of the commissioning procedure. It requires that all items of equipment within the system are in working order and that the function of each item of control is checked, initially set at the design value (if

this is known), readjusted as necessary during the testing stages, and finally placed on record as part of the commissioning documentation.

Most controllers have adjustments, not only to the set points but to differentials, time delays and response rates. It is of paramount importance that these are set up by an engineer who completely understands their function. Such settings should be marked on the instrument itself and recorded separately, since unauthorized persons may later upset these adjustments [63].

32 Commissioning

32.1 Specification

The commissioning of a refrigeration or air-conditioning plant starts from the stage of static completion and progresses through the setting-to-work procedure and regulation to a state of full working order to specified requirements.

Commissioning is the completion stage of a contract, when the contractor considers that the plant is in a correct state to hand over and the purchaser considers that it is in a correct state to accept and pay for.

Since the final object of commissioning is to ensure that the equipment meets with a specified set of conditions, this specification must be clearly stated and, hopefully, would have been clearly stated when the contract was placed.

A contract should state the following:

1. The medium or product to be cooled, or the area to be cooled
2. The total required cooling capacity, or mass throughput of product with ingoing and outgoing temperatures
3. The required limits of control
4. A realistic ambient condition for condenser water or air, and for fresh air supply

Example 32.1 The equipment is to maintain a temperature of – 10°C in a coldroom measuring $X \times Y \times Z$ and insulated with 100 mm expanded polystyrene, and freeze 20 t/day of chilled beef entering at 0°C, assuming an ambient air temperature of 26°C.

Example 32.2 The plant is to cool water at the rate of 120 litre/s from 18°C to 4°C. The ambient wet bulb temperature is 19°C.

Example 32.3 The plant is to have a capacity of 325 kW when

cooling a 30% aqueous solution of propylene glycol from – 4°C to – 7.2°C. Water is available from the main cooling tower at 23°C.

Example 32.4 The direct expansion coil is to cool 6.7 m^3/s air from 21°C dry bulb, 50% saturation, to 10°C dry bulb, 85% saturation, when evaporating at 4°C.

Example 32.5 The air-conditioner is to maintain 23°C dry bulb plus or minus 0.75°C and 50% saturation plus or minus 4%, in the room shown on drawing *XYZ*, assuming an internal load of 28 kW, including the four occupants. The maximum ambient conditions are 28°C dry bulb, 19.5°C wet bulb.

If no such specification exists at the time of commissioning, some basis of acceptance must be agreed between the parties concerned.

Basic flow diagrams should be available and, if not, the commissioning engineer must draw them up, against which actual plant performance can be checked. (See Figure 32.1.)

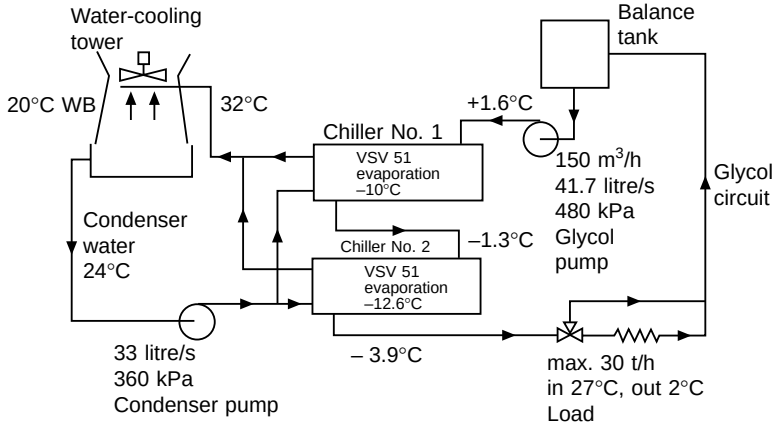


Figure 32.1 *Basic flow diagram for liquid chilling process plant*

The commissioning engineer will require details and ratings of all major items of the plant and copies of any manufacturer's instructions on setting to work and operating their products. If this information is not to hand, the work will be delayed.

32.2 Authority

The work of commissioning must be under the control of a single competent authority, whether it be the main contractor, a consultant or the user. Since this authority must accept the installation, it should be so stated in the original contract. Other specialists may

be required during the course of the work and they submit their test figures and other data to this central authority.

32.3 Setting to work

The setting-to-work procedure needs to be carried out in a logical sequence, since subsystems are interdependent. The following order will be typical:

1. A comprehensive mechanical inspection to confirm that all components are to the agreed specification.
2. Check all wiring and electrical controls, for correct circuitry, security of terminals, continuity, insulation, compliance with safety regulations such as IEE, marking of terminals and cables, etc.
3. Check action of all controls as far as may be possible without running any item [63].
4. Check all water systems filled. Start pumps and check rotation, flows, and pressures [64].
5. Start fans; check rotation, flows, and pressures [65].
6. Balance duct and grille flows [65].
7. Start main refrigeration system [34]. Allow to run on load until steady conditions are reached.
8. Set automatic controls to their approximate values, so the system will run without attention.

The services of specialist personnel and plant mechanics will be required during this period to operate the equipment and carry out any day-to-day attention. Care should be taken that this work does not come into conflict with the recommendations of suppliers, or invalidate their warranties. Where major items have not yet been accepted from suppliers, it will be advisable to retain their own commissioning engineer or other attendant until the project is complete. All necessary maintenance must be carried out, since any premature failure of a component may be blamed on such an omission.

The whole system is now left to run for a shake-down period, which may be from a few hours to several days, depending on the size and complexity. During this time, all components will be checked for vibration, leaks or other malfunction, and remedial action taken.

Low-temperature systems and cold stores should be brought down slowly, to allow for shrinkage in the structure. A fall of 5 K per day is reasonable, moving more slowly through the band + 2°C to – 2°C.

At the end of the shake-down period all strainers and filters are cleaned ready for the final test. If compressor oil is seen to be contaminated, this should be changed (see Section 11.10).

32.4 Calibration and final commissioning

Final adjustments should now be made to the following, and any other items of this sort:

1. Air flows, by setting of dampers. This entails measurement at various points and comparison with the design figures.
2. Concentration of any brines present.
3. Water and other liquid flows.
4. Starter overloads and the settings of safety controls such as pressure cut-outs and safety thermostats.

In the final commissioning stage, readings are taken and recorded of all measurable quantities in the system, and compared with the specification and design figures. The following, as applicable, should be considered as the absolute minimum to be taken and recorded:

1. Ambient conditions, dry and wet bulbs
2. All fluid flows, temperatures, and pump, fan and filter pressures
3. Refrigerant pressures and temperatures at expansion valve inlet, evaporator outlet, and compressor suction and discharge
4. Settings of all adjustable controls
5. Electric motor currents

It is probable that a full load cannot be obtained during the final test, for reasons of low ambient or lack of completion of other equipment for the process. In such circumstances, the commissioning engineer must establish the load which prevailed at the time and make an estimate of the system performance, on the basis of time run, or otherwise interpret the figures obtained. In such cases it may be advisable to agree to a tentative acceptance of the plant and carry out a full-load test at a later date.

Errors may come to light during this work and be possible to correct. If not, the acceptance or otherwise will be matter for negotiation between contractor and customer. If no agreement can be reached, it may be necessary to refer the commissioning procedure to arbitration.

32.5 Commissioning records

The commissioning log, which is the basis of the handover and acceptance of the system, should have as much of the original design data as is available, cross-checked against readings taken during the final tests. Line flow diagrams, if not already supplied, should be prepared and kept with the final drawings and wiring and other control diagrams. Details of the information which should be gathered

at this time are in the Commissioning Codes of the CIBS [34, 63, 64, 65].

On acceptance, the following should be handed over to the customer:

1. A copy of the commissioning log.
2. Flow, control, electrical and layout diagram and drawings.
3. Operating instructions. It is usual to instruct the user's operators as part of these final works.
4. Copies of instructions and manuals for all proprietary items of equipment.
5. Maintenance instructions.

It must be particularly noted that details should be entered at this time of ambient and load conditions, and any other factors which have an interface outside the plant itself. In this way, the relationship between ambient, load and plant can be established as a guide to future seasonal and load variations. It is helpful to the user if some indication can be given of operating conditions under light load, since the plant may well work most of its life at less than full load and the operators might not be able to interpret the readings taken.

Once this initial data has been recorded, a running log will indicate the performance under service conditions. As the load and ambient conditions change, the plant operators will be able to monitor the day-to-day conditions. This establishes normal running. Only by a clear understanding of what is normal can the abnormal be detected.

33 Operation. Maintenance. Service. Fault-finding. Training

33.1 Operation

A large proportion of refrigerating and air-conditioning equipment is now fully automatic in operation, but that does not absolve the user from the responsibility of understanding how it operates and being able to observe this operation. If this is not done, or if there is no other way of monitoring performance, the plant may run abnormally for some time before a fault is noticed, by which time considerable damage may have occurred.

The initial requirement is for the equipment to be fitted with sufficient pressure gauges and other monitoring devices to indicate the conditions under which it is working. It is helpful to mark these with the normal working limits when commissioning the plant.

Persons operating the plant should understand the meaning of any indicator or warning lights fitted to the control panels. It is important that the operator should be aware of the temperature gradients to be expected with the system so as to be able to compare actual working conditions with the design figures. Any changes should be interpreted as changes of ambient or load. A running log should be kept, as far as possible, to monitor working conditions.

Where the switching of plant is purely manual, the plant instructions should specify the limits of control, and not leave these to the shift operator, who may not be sufficiently skilled to take the correct decisions. Standby plant is often fitted, and it is part of the operation discipline to change over machines to ensure that they get even wear and keep all sets in running order. All operation staff should be aware of the method of bringing standby plant into use in an emergency. Where refrigerant valves need to be opened or

closed as part of plant operation, this should be carried out only by competent and responsible staff. If an open compressor is shut down for any length of time, it should be pumped down and valved off, to prevent possible loss of gas.

It is reasonable to expect a senior member of the staff to take a close interest in the operation of the system, and not delegate all the responsibility to semi-skilled personnel. This implies a good knowledge of the purpose, working and characteristics of the system and its control.

33.2 User maintenance

Where the user undertakes the day-to-day running of the plant, including most cases where the equipment is fully automatic in ordinary operation, basic maintenance will be assumed as part of this responsibility.

User maintenance includes operation where not automatic, cleaning of filters and strainers, attention to oil and lubricant levels, belt tensioning, general cleaning, running standby equipment, and verification of control operation.

The correct and efficient operation of any plant requires full flow at all times through the heat exchangers. Air and water filters need to be kept clean. Finned coils, especially outdoor condenser coils, must be cleaned frequently. The water side of heat exchanger coils should be cleaned of any accumulations of scale or algae as soon as a change in working conditions shows that they are getting dirty.

Accumulated dirt on air filters will increase the resistance and lead to reduced air flow. This is by far the most frequent cause of malfunction of air-conditioning equipment.

Example 33.1 An R.22 direct expansion coil evaporates at 3°C when cooling air from 20°C to 11°C. Condensing is at 35°C. If the air flow is reduced by 15% because of a dirty filter, what is the approximate increase in running cost?

Ignoring second-order corrections:

$$\text{Air entering coil} = 20°\text{C}$$

$$\text{Air off coil at full air flow} = 11°\text{C}$$

$$\text{Evaporating temperature at full air flow} = 3°\text{C}$$

$$\ln \text{MTD at full air flow}, \frac{17 - 8}{\ln (17/8)} = 11.94 \text{ K}$$

Air off coil at 85% air flow, $20 - \dfrac{20 - 11}{0.85} = 9.41°C$

Coil performance at 85% air flow, $(0.85)^{0.8} = 0.88$

ln MTD at 85% air flow, $\dfrac{11.94}{0.88} = 13.6$ K

See the cooling curves in Figure 33.1. The evaporating temperature will now fall to about 0.2°C. Compressor manufacturers' tables show 10.3% loss in duty for 1.5% less power at the new condition – an overall power increase of 9%. A more accurate estimate can be obtained by calculating a new basic rating for the reduced air flow. This shows about 8% extra power.

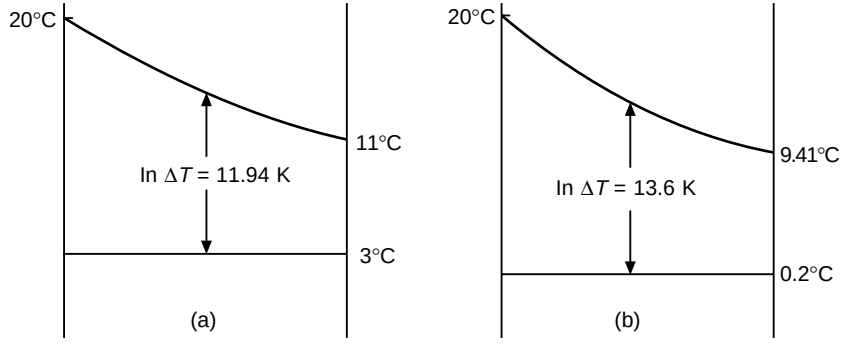


Figure 33.1 *Effect of air flow reduction. (a) Clean filters. (b) Dirty filters*

It is the responsibility of the supplying contractor to ensure that the user is aware of the need to clean or replace *air filters* and knows how to carry this out and when. Spare filters should be available so that the change of clean for dirty can be made in the one operation and the dirty filters taken away in closed bags for cleaning or disposal, to prevent release of dirt in the conditioned area. It is an advantage if the person changing the filters has a hand vacuum cleaner to pick up dirt which may become dislodged, and to clean the filter frames.

Changing of large filters will need to be left until the plant can be shut down for the time required to carry out the work. Under no circumstances should fans run without filters in place, or dirt will be deposited in inaccessible parts of the plant.

The provision of a manometer across the filter to indicate the pressure drop will give a positive indication of the need to clean or replace. Such resistances can be estimated from the filter manufacturer's data and should be recorded at the time of commissioning and also marked at the filter.

Filters of the automatic roller type need to have an independent manometer, which will give warning in the event of malfunction of the winder.

Water strainers are of the cleanable type, either a single-mesh basket, which must be removed after isolating the water flow, or a twin construction which permits the cleaning of one while the other is working. Indication of a dirty strainer will be an increase in pump pressure, and it is essential to have a pressure gauge on the pump discharge.

Strainers should be located where they can be cleaned easily, from the point of view of accessibility and isolation of the water pipe, and where the small amount of escaping water can be tolerated. Strainers in closed water systems will need cleaning soon after first starting up the circuit, but little attention once the pipe dirt is flushed out. Open systems, such as water-cooling towers, continuously wash dirt from the air and the frequency of cleaning must be judged from operating conditions, with a tendency to do so often rather than too seldom. Water tower strainers will not remove all the dirt. The larger particles will fall to the bottom of the sump and must be flushed out, possibly twice a year.

In many plants, the day-to-day operation is manually controlled and this requires a knowledge of, and familiarity with, the system which must be given by the installing contractor. A great deal of malfunction and inefficiency, many errors, and a few serious accidents arise from operation by untrained persons.

It is not sufficient that one person only has this knowledge. A clear set of operating instructions should be posted in the main plantroom, enabling any authorized person to start, run and shut down the system in a correct, safe and efficient manner. All staff who may be required to operate the plant need to be instructed and have some practice.

It is usual to mark the grade of lubricant on each item which might need periodic attention. Most equipment is designed to run for long periods without addition of lubricant and the dangers of adding too much should be noted.

The user will not normally add oil to a refrigeration system, apart from an industrial R.717 plant which will have a routine for the draining of parts of the circuit and replenishing the compressor sump.

Drive-belt tensioning and the replacement of broken or worn belts is a normal maintenance procedure, but may be missed if equipment is out of sight. A routine check will find these out.

The general cleanliness of plant is an indication of the care and interest taken by the maintenance staff, and is an encouragement

to others working on it. There is no reason or excuse for accumulations of dirt and refuse on or around any system.

Standby plant needs to be run frequently, both to ensure that it is in working order, and also to keep items such as shaft seals oiled and run-in, and thus gas-tight. The location and function of any changeover valves which must be operated in conjunction with standby plant should be clearly marked.

Many malfunctions, and some dangerous situations, arise from incorrect setting of control and safety instruments. It is assumed that these are all set and the correct settings recorded at the time of commissioning, but such settings may afterwards be tampered with by uninformed or unauthorized persons. The correct adjustment of any instruments normally set by the user remains his or her responsibility as a matter of routine operation. It is good practice to arrange that instruments are locked, sealed, or otherwise guarded from tampering – even to the extent of putting dummy controls in a conspicuous place.

The function of safety controls should be checked at least once a year.

Water treatment and corrosion inhibition systems require periodic attention, and full instructions should be left on site by the supplier or installer of the apparatus, whether or not they will be responsible for later attention.

Where water is evaporated from a circuit, such as a cooling tower, evaporative condenser, or humidifier, it must be remembered that there is no way of avoiding a steady bleed-off or frequent flushing, to restrict the concentration of dissolved solids. Much trouble arises from the efforts of well-wishing but misguided persons who stop the flow of bleed-off to 'save water'.

Many systems are shut down for periods of the year, either for process closure or if not required in winter. The advice of the supplier should be sought as to the correct procedure. In the case of refrigerant circuits, it is advisable to pump down into the receiver or condenser to minimize leakage losses. Water towers should be drained in winter in this climate, if not in use, and the tank heater disconnected.

33.3 Major maintenance work

The average user, unless of an industrial nature, will tackle only the simpler of these maintenance jobs and will entrust the major work to a specialist firm. Large concerns will have their own trained and skilled personnel and will do all work themselves.

The services of an outside maintenance contractor are usually on an annual contract basis and should clearly define the work which

is to be carried out by the two parties. Day-to-day operation and the simpler tasks such as air filter cleaning are nearly always excluded from a contract of this sort.

Various types of contract are offered and it is recommended that the original suppliers are approached during the commissioning period (or before) for their suggestions. In particular, there may be friction if a rival firm undertakes maintenance while the plant is under guarantee and it is usual to let the first year's contract to the installer.

Ammonia systems, even those with the most sophisticated separators, have continuous oil migration and require attention every week or, in some cases, every day to ensure maximum working efficiency. Plant operators need to be well trained and practised in this simple maintenance task, to minimize loss of refrigerant. In larger plants, the oil removed can be filtered and used again. It is useful to enter in the running log the quantities of oil removed and put in, since it has been known for large amounts of oil to accumulate in an evaporator without operators being aware of it.

Migration of oil over a long period in a dry expansion circuit should be treated as a design fault, and some action taken to put it right.

Air filters and water strainers have been mentioned in Section 33.2. Major work on those components and associated systems will be the cleaning and refilling of the oil sump of a viscous air filter and the periodic draining, cleaning and flushing of water tanks.

Leaks of R.717 usually make themselves apparent and motivate staff to search out the leak and repair it. In the case of the halocarbons, a regular leak test should be part of the general maintenance schedule.

Under no circumstances should refrigerant be added to a leaking circuit without first making a repair. The one exception to this rule may be a continuous process plant, where the cost of a shut-down may override the cost and inherent danger of a small continuous leak.

Where gas is detected at the shaft gland of an open compressor which is not turning, the compressor should be run for a short time to re-lubricate the gland. The leak may then cease.

Staff should be forbidden to smoke while leak testing or repairing. Many operatives are ignorant of the danger to their health if smoking in the presence of traces of the halocarbons.

Moisture in halocarbon circuits will be indicated by the colour trace on the sight glass, where this is fitted. Immediate action is required, especially with a hermetic or semi-hermetic compressor, before damage is caused. The drier should be changed and the

sight glass watched for reversal of the colour to 'dry'. Bad cases of contamination may need a second change of drier. If the liquid line leaving the drier or strainer (if separate) is colder than the inlet, there is a severe pressure drop within, indicating dirt. A new drier, or cleaning of the strainer, will cure this.

Heat exchanger surfaces need to be kept clean. Aqueous circuits (evaporator or condenser) can be cleaned with a chemical such as sulphamic acid, brushed or subjected to high-pressure water jets. In each case, all traces of dirt and chemical need to be removed from the circuit before it is put back to work. In cases of doubt, the manufacturer's advice should be sought. A layer of scale 2 mm thick on a condenser tube can cause a power increase of 16%, and the need to clean a condenser can usually be deduced from the condensing pressure. Persons using high-pressure water jets should wear face masks to avoid inhaling aerosol droplets.

The checking and readjustment as necessary of all safety controls is an essential part of periodic maintenance – possibly annually. A time should be chosen when temporary stoppage of the plant will not cause inconvenience. Unsafe conditions can be set up by throttling valves, stopping pumps, or removing the load. In each case the relevant safety control should function at the pre-set conditions. Safety checks on specialized items such as fire dampers may be required from time to time by local authorities, and these checks, together with the expert advice available from the testing officers, should be welcomed as proof of the inherent safety of the installation.

It is essential that all major maintenance work and findings are recorded in the plant running log as a guide to the reliability of components, the need for cleaning, and other indicators to future work.

Refrigerant compressors, air compressors and some other items of mechanical equipment might be subjected to a periodic part strip-down inspection and overhaul, as may be recommended by the manufacturer and indicated by running time meters or estimated running hours from the plant log sheet. Such planned maintenance entails, as its name suggests, some planning. Manuals, diagrams and drawings should be obtained beforehand, and sources of possible spare parts located. Special tools or instruments may be required. The manufacturer should be able and willing to advise and guide in this work. Failed, worn and other replaced parts should be retained for later examination and a post-mortem held in cases of doubt; records should be kept. In the case of a shut-down of a process plant, the major components should be tackled in rotation, lest a serious fault or shortage of spares prevents the process re-starting at the end of the closure.

Planned or preventative maintenance is not necessarily the best for all installations. If the service is not essential, and spares are known to be available within a reasonable time, nothing is done beyond obvious routine servicing work. Then, when a breakdown occurs, it is repaired. This approach, although not widely accepted, is an option which should be considered.

33.4 Guarantee period

Most equipment will be guaranteed by the supplier or contractor for 12 months from the date of supply, installation or commissioning, and these precise details should be agreed and noted to avoid arguments later. In particular, some items may have left the factory several months before commissioning and, if the manufacturer is advised, the guarantee may be extended from that date. Such a guarantee will probably cover the cost of repair or replacement of the item, but not labour charges in removal or refitting. Again, these details should be noted.

The situation often arises where equipment is brought into service before it has been commissioned and accepted by the purchaser. Such operation can only be on the instructions of the purchaser and with the agreement of the supplier. A maintenance contract must come into force at this time.

Many disputes arise in this first year if the installation is not maintained to the satisfaction of the supplier, and a split responsibility of this sort is to be avoided. As already stated, where possible, maintenance for the first 12 months should be by the original contractors or a firm recommended by them. There is a growing tendency to extend the guarantee period on small packaged plant to 3 years.

33.5 Fault-finding

System faults fall into two general classes: the sudden catastrophe of a mechanical breakdown and the slow fall-off of performance which can be detected as a malfunction in its early stages but will also lead to a breakdown if not rectified. Identification of the first will be obvious. To track down the cause of a malfunction will be more complicated.

Fault-tracing is seen as a multistep process of deduction, ending in normal operation again and a record of the incident to inform other operatives. The steps are as follows:

1. Detection, i.e. detection of abnormal operation

2. Knowledge of the system to track down the cause
3. Observation of exact operating conditions
4. Identification of the fault
5. Decision: what to do? how? when? can it be left?
6. Action to rectify the fault
7. Test: is it now normal?
8. Record note in log, for future information

A lot of help in fault-tracing may be had from charts for specific pieces of apparatus, prepared by the manufacturer.

Detailed examination of a sophisticated item may be beyond the skills of the plant operators and require the assistance of a specialist, such as an electronics engineer. Where such complications are part of the system, it is an advantage to know beforehand where such specialist help can be reached.

It should be accepted that fault analysis can be a slow process and that it usually defies prior estimates of the time it will take, regardless of the pressures of persons who are affected by the interruption of the service. In any case, hasty decisions and random efforts to get the plant working again are to be shunned, since more damage may result.

Training courses are available in analytical methods of fault-tracing. Computers are also in use which monitor a number of parameters and draw attention to any observed abnormality. The control/monitoring device may then make a judgement as to the cause, or this may rely on the interpretation of the operator. Considerable use is now made of logic control/monitoring devices which can oversee the operation of a large number of installations from a central computer/observation terminal.

33.6 Spare parts

Except in the case of a planned overhaul, spare parts will only be wanted in an emergency, and then in a hurry. Most manufacturers can guarantee supplies from their own stocks and will despatch quickly, providing they are given enough information to correctly identify the parts required.

There are two predominant problems. First, there is the necessity to keep the plant in operation, which may vary from a non-essential service such as the comfort cooling of an infrequently used room, to the precise temperature control of an expensive or potentially dangerous process. The second problem is the time taken in transit, which might be as long as a year in parts of the world subject to excessive docks and custom delays.

The scale of spares to be held under these varying conditions must be judged by the user, taking into account the problems which might be met and seeking advice from the supplier. For remote installations, the latter may be asked to recommend 'Spares for one year's operation – a classification which is subject to the vagaries of mechanical breakdown'. A suggested scale of spares is given in Table 33.1.

Table 33.1

Type of installation	*Availability of spares at site*			
	Very good	*Fair*	*Poor*	*Very bad*
Air-conditioning				
comfort, small	Nil	Nil	A	B
comfort, large	Nil	R	A	B
essential process	D	D	D + A	D + B
Cold stores				
small, above 0°C	Nil	R	D + A	D + B
large, above 0°C	R	D	D + A	D + B
freezer, small	R	D	D + A	D + B
freezer, large	D	D	D + A	D + B
Industrial process	D	D	D + A	D + B

Code: R, refrigerant; A, refrigerant, drier, solenoid valve coil, compressor suction and discharge valves and gaskets, electrical controls, solenoids and overloads, any other items specifically recommended by the manufacturer; B as A, together with expansion valve; D, duplicate equipment throughout plant to comply with Lloyd's Rules which say, in effect, that the process shall not be at risk in the event of the failure of any one component.

33.7 Training

The nature of refrigeration and air-conditioning equipment requires specialized training for operating and service personnel. The basic skills of those entering the industry should be the ability to read an engineering drawing and to read and understand flow, circuit and electrical diagrams. When recruiting labour it will be an advantage if the candidate already has some knowledge of electrical circuits.

A contractor supplying new equipment should be required to instruct the staff who will operate it, and to advise where further instruction may be had. If the senior operator is allocated to work alongside the contractor during the final erection and commissioning stages, he should pick up enough knowledge of the system to give him the confidence and skill to run it.

Contractors generally select school-leavers, who then alternate

sandwich technical courses with workshop or field training and experience under supervision. The more able students progress through the firm and can reach technician or engineer status. One of the larger contractors in the UK [66] runs short practical courses in various aspects of operation and maintenance, which are open to outside students. At least one major manufacturer [67] also runs courses on operation and maintenance. The HVCA [57] publish a handbook entitled 'Which College', detailing college courses and their location.

Courses with and without practical content, part- and full-time, and of various standards, are run by technical colleges and colleges of higher education. Training organizations set up short courses, mainly for plant operators and mechanics who need to have a working knowledge of refrigeration and air-conditioning.

Several other manufacturers and distributors run short courses and training schemes, open to persons outside their own organization. A recent innovation is the 'Open Tech' programme, a course in private study for personnel who have the initiative to do so in their own time.

Degree and some higher academic courses are held for full-time and part-time students at the Institute of Environmental Engineering, South Bank Polytechnic.

33.8 The running log

The detection of abnormal operation can only occur if normal operation is monitored. Since refrigeration is a thermal cycle, the obvious readings to be taken will be temperature and the related refrigerant pressure.

The skilled operator or the visiting service mechanic will have a working knowledge of the pressures and temperatures to be expected, but will not be able to make an accurate assessment of the actual conditions without plant measurements for comparison. The commissioning log (see Section 32.5) will show readings taken at that time, but only at one set of running conditions.

It is therefore essential on a plant of any size to maintain some kind of running record, so that performance can be monitored with a view to detecting inefficiency and incipient troubles. The degree of complexity of this running log must be a matter of judgement, and a small amount of useful information is to be preferred to a mass of data which would be confusing. The following would seem to be basic:

1. Compressor suction and discharge pressures and corresponding temperatures.

2. Oil pressure gauge. It would be helpful to add a column so that true oil pressure can be entered (i.e. oil – suction).
3. The load temperature (room, water, brine, etc.).
4. Load flow rate or pump pressure.
5. Ambient temperature, dry bulb *and* wet bulb if possible.
6. Condenser water flow rate or pump pressure.
7. Any motor currents where ammeters are fitted.

These, together with space for comments, date, and time, should be set out as shown below.

Sarsaparilla Brewing & Bottling plc Running log.
Plant *Borough Road* Line No. *3* Date *8 Sept. 88*

	Pressures				*Temperatures*		*Brine*			*Ambient*				
Time	S	D	OG	O	S	D	In	Out	P	DB	WB	A	*Comments*	O
08.00	2.8	10	4.7	1.9	– 8	27	+ 4	– 1	3.1	11	9	190	–	*H*
10.00	2.1	10.5	3.9	1.8	– 14	29	– 3	– 8	3.1	15	12	175	–	*H*
12.00	2.1	10.5	3.9	1.8	– 14	29	– 2	– 7.5	3.1	19	13	175	–	*H*
14.00	2.1	11	3.9	1.8	– 14	30	– 3	– 8	3.1	20	14	170	–	*H*

33.9 Exercises

Exercise 33.1 The motor driving an open compressor is switched on. Ten minutes later the compressor is not turning. List six possible reasons.

Answer
No mains electric supply
Fuse blown
One phase blown, out on single-phasing trip
Belts broken or slipping
Out on high-pressure cut-out (various)
Out on low-pressure cut-out (this may have reset, but compressor contactor held out by restart delay timer).
Short of oil
Out on thermostat
Flow switch open, in load or condenser water

Exercise 33.2 A discharge gauge reads 0.6 bar higher than the reading a week ago. List four possible reasons.

Answer
Higher ambient, dry or wet bulb
Higher load temperature or more flow
Dirty condenser

Condenser fan stopped
Non-condensible gas in system
Pump strainer dirty (condenser water)

Exercise 33.3 An automatic plant on a frozen food store uses R.717 and has three compressors working on five flooded evaporators. Draft a brief job specification for the senior operating mechanic.

Answer
Experience with ammonia
Knowledge of electrical controls and interlocks
House close by, prepared to work 'on call'
Reliable
Safety conscious
Able to instruct staff

Another factor when taking on new staff must be their attitude to a changing technology. Advances are being made in the many aspects of engineering and application in refrigeration and air-conditioning. This situation requires adaptable personnel who are always ready to assimilate new ideas.

34 Efficiency and economy in operation

34.1 Assessment, identification, planning and targets

An effective programme for energy economy must commence with an assessment of the problem areas, planning of the campaign and the setting of realistic targets [68].

The assessment stage requires a breakdown of all the costs involved in running the plant, i.e. not only all fuels, but also the costs of operation, maintenance and spares. At this stage it may be helpful to fit separate electricity meters to sections of the plant under survey, so as to identify their individual fuel costs. Many of the overall expenses will be mixed, but an effort must be made to apportion these to the areas under examination. At the same time, the running conditions and methods are noted.

These costs are now scrutinized and compared with other figures from fuel suppliers, trade associations or allied companies to identify suspected problems. Typical histories can be obtained from the Case Studies Index of the Energy Information Centre in the UK, or any similar body in other countries.

Considerable help can be obtained from, for example, the Energy Efficiency Office under their scheme for grants for short energy efficiency surveys, or from independent organizations who specialize in this work. The advantage of using an outside investigator is that interdepartmental frictions will be less, since any weakness in the existing energy programme may be taken as a criticism of the department or manager concerned and this comes easier from persons outside the company.

The planning stage decides where the effort for energy economy should be concentrated and sets target figures for the amount of savings and the necessary implementation costs with payback periods. These assessments should be realistic and as accurate as possible.

There are several fields of investigation.

34.2 Reduction of load

The first step in ensuring economy of operation of any equipment is to check that the load is an absolute minimum. This requires an energy survey of the process, the cold store operation or the building to be air-conditioned. The purpose of this survey is to determine the elements of the cooling load, as well as possible, and to consider ways of reducing each item to a minimum, consistent with the cost of doing so. Such an evaluation may not be exact, but the need at this stage is to put these cooling loads into some order of magnitude so as to find what proportion of the total load may be reduced by any form of treatment. The load elements are:

1. Heat conduction from warmer areas through building structure, insulated or otherwise. This includes excess heat from any source, such as an abutting boilerhouse wall or solar radiation on an outer surface.
2. Direct solar radiation into the cooled space.
3. Convection heat gains from infiltration, fresh air and the normal opening of cold store doors.
4. Heat input from auxiliary apparatus – fan motors, pumps, defrosting.
5. Internal electric lighting. Illumination is a specialist field and requires correct planning, installation and switching schedules. Remember that this energy is paid for twice – to put it in and then to take it out again.
6. Poor discipline of loaders and fork-lift truck drivers, leaving cold store doors and sealing gaskets open, with gain of heat and more icing in store and on coolers.

34.3 Plant running conditions

The plant operating conditions should be compared with design or commissioning figures, where these are available, or analysed in terms of expected values, to see if there are any discrepancies. Obvious points, mentioned in earlier pages, are:

1. Incorrect adjustment of controls
2. Dirt on filters, coils, fans, ducting, etc.
3. Ice on coils
4. Other obstructions to full air flow such as badly stacked produce in cold stores, desks and partitions in offices
5. Shortage of refrigerant or excess of oil in evaporator
6. Incorrect adjustment of expansion valve

7. Incorrect operation of pumps or valves, causing feed tanks to overflow with the loss of chilled or hot water

In most of these instances the overall reduction in heat flow may be small, but the cumulative effect within a system may be enough to reduce the evaporating temperature by as much as 2 K, resulting in a loss in COP of some 6%.

34.4 Operating techniques

The design operating conditions for a comfort air-conditioning system give opportunities for considerable savings. It is generally specified that the ideal comfort condition is 21°C and 50% saturation, but variations within the accepted comfort band can reduce energy costs, both for building heat gain or loss and for the treatment of the proportion of fresh air which is needed. For example, summer fresh air reduced to 21°C, 60% saturation, imposes 4 kJ/kg less cooling load, and winter air raised to 40% saturation requires 4 kJ/kg less heating than at 50% saturation. These changes in humidity cannot be detected without instruments and have a small effect on personal comfort.

A further consideration is to allow the indoor condition to drift slightly above the design figure in extremely hot weather. For example, the design maximum for the London area may be taken as 27°C dry bulb, but exceeds this for an average of 25 h/year. The increased cooling load to try to hold 21°C under such conditions is considerable, and experience has shown that short-term rises to 23°C might be permitted. While this is a design factor, operators should be discouraged from trying to squeeze extra capacity out of the plant, at a poor COP and high power requirement. Automatic controls can have a bias set point imposed by a high ambient.

34.5 Condition of plant

A careful mechanical inspection is made for:

1. Badly worn machinery, tight packing glands, badly adjusted drive belts, etc., leading to wastage of drive power
2. Leaking ductwork, leading to loss of chilled/warm air
3. Damaged insulation and vapour barriers, leading to heat gain/loss and condensation or ice on cold surfaces.

34.6 Excess condenser pressure

This is the greatest cause of excess power and loss of duty in a refrigeration system. Causes are:

1. Incorrect setting of head pressure controls
2. Dirty or choked spray nozzles in water tower or evaporative condenser, so that the surface is not fully wetted
3. Non-condensible gas in circuit
4. Bad location of condensers, so that air recirculates
5. Undersized condensers
6. Dirty fins on air-cooled condensers
7. Fans not working or broken
8. Water strainers blocked
9. Undersize pumps fitted
10. Air in water circuit

While all these factors affect the good running of positive-displacement compressors, the effect is far worse with centrifugal machines, which can approach stall condition and so give a much reduced cooling duty.

34.7 Maintenance

The good running order of equipment depends on the standards of maintenance. This is a running cost to be assessed with all the others. If it is found to be faulty, the investigation must consider what this is costing in terms of plant inefficiency and the expenditure to reach acceptable standards. This might be in the replacement or extra training of staff, or in contracting the work out. If the latter, the cost must include supervisory expenses.

34.8 Remedial action on existing equipment

The faults described above are largely self-revealing and most of them can be corrected or improved without a great deal of expenditure. The presence of separate metering devices should give an immediate indication of the savings made.

34.9 Improved controls and equipment on existing plant

Deficiencies on the original plant might be corrected by comparatively minor improvements, changes and additions. Each should be assessed for its individual contribution to energy economy and how it may improve the performance of other parts of the system.

1. Optimum-start controls.
2. Ambient-biased set point controls.

3. Modifications to give improved air and water flows, where these were shown to be deficient, i.e. increase fan speeds, change fans, change pumps, improve ductwork or piping to reduce pressure losses.
4. Improved defrost control, to defrost coils only when and for as long as necessary.
5. Improved cold store door-operating mechanisms (see Figure 15.8).
6. Improved condenser pressure control. If the expansion valve is too tightly rated to accept lower condenser pressure, change the expansion valve, possibly for the electronic type.
7. Automatically switch off plant which is not in use (boiler in summer, tower in winter, lights at night, etc.).
8. Switch off some of the cold store fans and coolers at night and weekends.
9. Fit an automatic load-shedding maximum demand limiter.
10. Resite condensers for better air flow.

More drastic items may be:

1. Replace worn, obsolete or undersize compressors, evaporators or condensers.
2. Add new compressors, evaporators or condensers if these can be shown to be economical.

34.10 Design of systems for energy economy

Previous chapters have outlined the methods of estimating loads, choosing methods to achieve the required conditions, and how to select and balance plant for correct operation. They have also mentioned the factors which will give economy in running costs.

The maximum use should be made of energy-saving methods, where these may be applicable. Some of these are:

1. Use of all fresh air for air-conditioning, if required in cold weather
2. Provide mid-season heating from condenser heat or heat pump (reverse-cycle) operation
3. Run plant at night on low-cost electricity and make ice, to use for chilled water when load comes on (ice-bank)
4. Switch plant off for periods when electricity is at a premium tariff
5. Two-speed or electronically speed-controlled motors for lower compressor, fan and pump speeds at low load
6. Arrange the coolers within a cold store so that they will give adequate air circulation at night when half of them are switched off.

Much attention has been given in recent years to the power consumed in the refrigeration process and the development of more efficient compressors. A few points to consider are:

1. Avoid high compression ratios on piston compressors.
2. Avoid single-stage compression for very low temperatures.
3. Avoid machines which are working at the upper or lower limits of their range.
4. Always ask the running power required at load conditions.

The resulting system design will not be the lowest in first cost.

34.11 Commitment to energy savings

A positive energy policy needs to be a company decision, taken at boardroom level and backed by boardroom authority, since it cuts across departmental boundaries and may conflict with the opinions of senior staff. Typical objections are:

1. The capital, operating, maintenance and fuel costs come from four separate budgets, possibly accounted for by four different managers, so these budgets need to be adjusted. Separate fuel meters are needed to prove the savings, which might otherwise be held in question.
2. There may be some disruption to normal working while the schemes are being carried out. This may affect departments not concerned directly with the programme.
3. Staff may need to be released for training schemes.
4. The improvements may need changes in operating techniques which are thought to be adequate already.

Some of these, such as the tightening of discipline of fork-lift truck drivers, may provoke open conflict, which must be foreseen and headed off.

It is important to be able to quantify the results of the energy programme and make these known to all concerned. A conservation programme of this sort is an ongoing process and should keep all staff concerned alert to the possibilities of further improvements.

35 Catalogue selection

35.1 General

Manufacturers will publish rating and application data for their products, based on standard test conditions and for the more usual range of uses. They cannot be expected to have accurate figures for every possible combination of conditions for an individual purpose, although most will produce estimates if asked.

The widespread use of packaged units of all sizes requires interpretation of catalogue data by applications engineers, sales engineers, and others, and by the end user.

The first step is to be certain of the basis of the published data and consider in what ways this will be affected by different conditions. Revised figures can then usually be determined. For extensive interpretation work, simple mathematical models of performance can be constructed [69].

35.2 Compressors

Refrigeration compressors which will probably be used on flooded evaporators (R.717 and the larger machines generally) will be rated with the suction at saturated conditions, since there will be little or no superheat in practice. Compressors for dry expansion systems may be rated at a stated amount of superheat, commonly 8 K.

There will be a pressure drop and heat gain in the suction line, and these are frequently ignored if the pipe run is short. In other cases, some allowance must be made. Both these factors will increase the specific volume.

Example 35.1 An ammonia compressor is rated at 312 kW with saturated suction at −15°C. It is installed with a very long suction line, causing a pressure drop of 0.4 bar, and picks up 6 K superheat from its evaporator condition. Estimate the capacity loss.

Evaporator pressure at – 15°C = 2.36 bar abs.

Suction pressure, 2.36 – 0.4 = 1.96 bar abs.

Rating suction temperature = – 15.0°C

Actual suction temperature, – 15 + 6 K = – 9.0°C

The absolute gas pressures must be used in this calculation (see Section 1.4).

The volume pumped by the compressor will remain about the same, but the density of the gas is reduced, and thence the mass flow.

Using the General Gas Laws:

$$\frac{m_2}{m_1} = \frac{p_2 T_1}{p_1 T_2} = \frac{1.96 \times 258.15}{2.36 \times 264.15} = 0.81$$

So the capacity loss is of the order of 19%, or 59 kW. There may also be a slight drop due to the higher compression ratio, ignored here as the condensing pressure is not known.

Halocarbon systems are almost invariably controlled by mechanical or electronic thermostatic expansion valves, requiring a superheat signal to operate the control. The superheating of the suction gas into the compressor will cause it to expand, resulting in a lower mass flow for a given swept volume. Reduction of the superheat setting of the expansion valve will therefore result in better use of the compressor. The limit will be reached when there is insufficient signal to work the expansion valve.

Example 35.2 An R.22 compressor is rated at 15.9 kW when evaporating at – 5°C, with 8 K superheat. Estimate the gain in capacity if it can be run safely with half the superheat.

Rating suction temperature, – 5 + 8 = 3°C

= 276.15 K

Working suction temperature, – 5 + 4 = – 1°C

= 272.15 K

$$\text{Ratio of mass pumped} = \frac{m_2}{m_1} = \frac{T_1}{T_2} + \frac{276.15}{272.15} = 1.015$$

This gives a gain in capacity of about 1.5%, or 0.24 kW.

There will also be a gain in usage of the evaporator coil and a corresponding rise in the evaporator temperature, giving a further increase in compressor capacity. This would need to be evaluated from the compressor curves, but might be a further 1%.

Example 35.3 A hermetic compressor is rated at 18.2 kW for R.22 when evaporating at 7°C, suction superheated to 35°C, condensing at 54°C, and with 8 K subcooling of the liquid. Assuming that the inlet gas picks up another 30 K as it passes over the compressor motor, estimate the change in capacity if the suction is superheated to 12°C.

(*a*) Change in mass flow:

Compressor inlet temperature, rating, 35 + 30 = 65°C

= 338.15 K

actual, 12 + 30 = 42°C

= 315.15 K

$$\frac{m_2}{m_1} = \frac{T_1}{T_2} = \frac{338.15}{315.15} = 1.073$$

(*b*) Change in enthalpy (kJ/kg):

Enthalpy of suction gas at 35°C	= 329.8	
Enthalpy of suction gas at 12°C	=	311.7
Enthalpy of liquid at (54 – 8) 46°C	= 157.0	157.0
Refrigerating effect (kJ/kg)	= 172.8	154.7

Change in enthalpy, $\frac{154.7}{172.8} = 0.895$

Overall change in capacity, $1.073 \times 0.895 = 0.96$

Corrected working capacity, $18.2 \times 0.96 = 17.5$ kW

35.3 Condensing units

Rating curves for condensing units (see also Section 13.2) will be for stated entering temperatures of the condensing medium – air or water. These may not go as high as the particular application may demand, and figures must be extrapolated.

The main effects of a higher condensing temperature will be a drop in the refrigerating effect, since the liquid enters the expansion valve hotter, and a decrease in volume pumped due to the lower volumetric efficiency. There will also be an increase in the drive motor power.

Example 35.4 An air-cooled condensing unit is rated at 13.7 kW on R.22 when evaporating at 5°C and with ambient air at 43°C. Estimate the duty with ambient air at 52°C.

Some assumptions must be made regarding the condenser coil performance, and this may have a ΔT of 14 K between the entering air and condensing refrigerant and subcooling the liquid 5 K, with suction gas entering the compressor with 6 K superheat.

	Rating	*Working*
Rating condensing temperature, 43 + 14 =	57 °C	
Working condensing temperature, 52 + 14 =		66 °C
Enthalpy of suction gas at (5 + 6) = 11°C =	312.1	312.1
Enthalpy of liquid at (57 – 5) = 52°C =	165.3	
Enthalpy of liquid at (66 – 5) = 61°C =		178.5
Refrigerating effect (kJ/kg) =	146.8	133.6

In addition, the compression ratio has increased considerably and there must be a correction for loss of volumetric efficiency.

	Rating	*Working*
Suction pressure (bar abs) at 5°C =	5.82	5.82
Discharge pressure at 57°C =	22.84	
Discharge pressure at 66°C =		27.76
Compression ratio =	3.92	4.77
Volumetric efficiency (from Figure 2.8) =	0.75	0.68

$$\text{Estimated new duty} = 13.7 \times \frac{133.6}{146.8} \times \frac{0.68}{0.75} = \quad 11.3 \text{ kW}$$

This is approximate, but probably within 0.2 kW.

35.4 Evaporators

The rating of an evaporator will be proportional to the temperature difference between the refrigerant and the cooled medium. Since the latter is changing in temperature as it passes over the cooler surface (see Section 1.8), an accurate calculation for a particular load is tedious and subject error.

To simplify the matching of air-cooling evaporators to condensing units, evaporator duties are commonly expressed in basic ratings (see Figure 35.1), in units of kilowatts per kelvin (formerly in British thermal units per hour per degree Fahrenheit). This rating factor is multiplied by the ΔT between the entering air and the refrigerant.

Example 35.5 An air-cooling evaporator has a mass air flow of 8.4 kg/s and a published 'rating' of 3.8 kW/K. What will be its rated

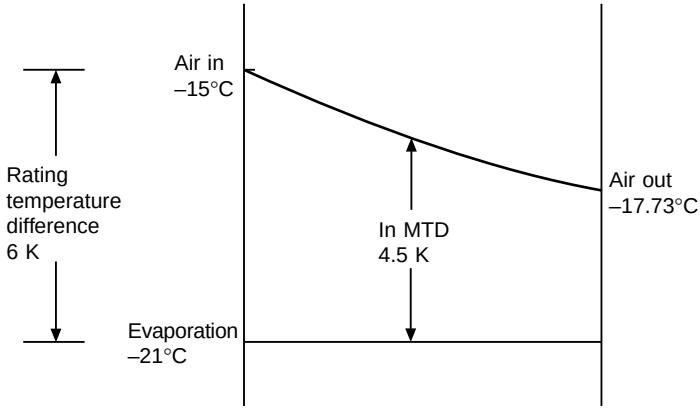


Figure 35.1 *Basic rating and ln MTD*

duty at – 15°C coldroom temperature with refrigerant at –21°C? What is the true ln MTD?

$$\text{Entering air temperature} = -15°\text{C}$$

$$\text{Refrigerant temperature} = -21°\text{C}$$

$$\text{'Rating' } \Delta T = 6 \text{ K}$$

$$\text{Rated duty} = 3.8 \times 6 = 22.8 \text{ kW}$$

$$\text{Reduction in air temperature} = \frac{22.8}{1.006 \times 8.4} = 2.73 \text{ K}$$

$$\text{Air leaving temperature} = -15 - 2.73 = -17.73°\text{C}$$

$$\text{ln MTD} = \frac{6 - 3.27}{\ln (6/3.27)} = 4.5 \text{ K}$$

It follows that there would be an error at other conditions and the basic rating is only accurate at one point, so this short-cut factor must only be used within the range specified by the manufacturer.

The method of balancing such an evaporator with a condensing unit is graphical. The condensing unit capacity is shown as cooling duty against evaporator temperature, line *CD* in Figure 35.2. The coil rating is plotted as the line *AB*, with *A* at the required coldroom (or 'air-on') temperature, and the slope of the line *AB* corresponding to the basic rating. The intersection of this line with the condensing unit curve *CD* gives the graphical solution of the system balance point. Similar constructions for higher condenser air conditions (*EF*, *GH*) or different room temperatures (A_1B_1) will show balance points for these conditions.

The graph also indicates the change in evaporating temperature

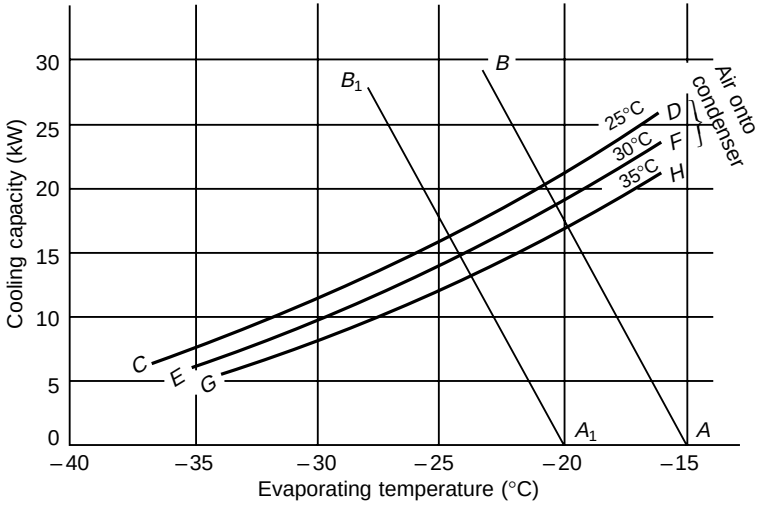


Figure 35.2 *Graphical balance of evaporator with condensing unit*

and coil duty when the ambient is lower or higher than the design figure. This will show if there is any necessity to control the evaporating temperature in order to keep the correct plant operation. (See also Sections 9.8 and 9.11.)

35.5 Reduction of air flow

Frequently, coil data will be available for a design air flow, but the system resistance reduces this flow to a lower value. There is a double effect: the lowering of the ln MTD and the lower heat transfer from the coil by convection.

The outer surface coefficient is the greatest thermal resistivity (compared with conduction through the coil material and the inside coefficient), and a rough estimate of the total sensible heat flow change can be made on the basis of [5] and [6]:

$$h = \text{constant} \times (V)^{0.8}$$

Example 35.6 An air cooling coil extracts 45 kW sensible heat with air entering at 24°C and leaving at 18°C, with the refrigerant evaporating at 11°C. Estimate the cooling capacity at 95, 90 and 85% mass air flow.

$$\text{Design mass air flow} = \frac{45}{1.02 \times (24 - 18)} = 7.35 \text{ kg/s}$$

An approximate analysis comes out:

	Air flow (%)			
	100	*95*	*90*	*85*
Mass air flow (kg/s)	7.35	6.99	6.62	6.25
Air temperature on coil (°C)	24	24	24	24
ΔT for 45 kW (K)	6	6.3	6.7	7.1
Air temperature off coil (°C)	18	17.7	17.3	16.9
In MTD, refrigerant at 11°C (K)	9.7	9.5	9.2	9.0
h, in terms of design (from $V^{0.8}$) (%)	100	96	92	88
Capacity, $(45 \times h \times \ln \text{MTD})/9.7$ (kW)	45	42.3	39.3	36.7

This first estimate for the evaporator coil performance must now be corrected for the change in compressor duty if it is a direct expansion coil, or of water temperature change if using chilled water. Another method is to re-calculate the basic rating figures at the new air flows and plot these against compressor curves.

With all calculations involving convective heat transfer, it must be remembered that the figures are predictions based on previous test data, and not precise.

35.6 Room air-conditioners

The catalogue-rated cooling capacity of a room air-conditioner, if not qualified, will be based on ASHRAE Standard 16-1983. This specifies test conditions of air onto the evaporator at 80°F dry bulb, 50% relative humidity (26.7°C, 49.1% saturation), and air onto the condenser at 95°F dry bulb, 75°F wet bulb (35°C and 23.9°C). The original basis for this specification was the ambient condition prevailing in the mass-market area of the USA.

For these units, British Standard 2582: Part 1, 1982 gives three sets of alternative rating conditions, corresponding to the ASHRAE Standard, for tropical, arid and temperate ambients. They are:

	Room air temperature		*Outside air temperature*	
	DB	*WB*	*DB*	*WB*
Condition A	27	19	35	24
Condition B	29	19	46	24
Condition C	21	15	27	19

and catalogue ratings quoting BS.2852 will be qualified with the appropriate conditions letter.

The International Document ISO R 859 evolved from existing national standards and does not specify any test conditions, only

test methods. Any catalogue ratings quoting this ISO must be qualified with test conditions.

Performance of the average commercial room air-conditioner at BS.2852, condition *C*, will be some 10–15% lower than at condition *A*, since it will evaporate some 5 K lower. This reduction factor should be applied to any unqualified unit rating if used under UK ambient conditions.

A further complication arises with the application to temperate conditions of room air-conditioners which have been designed primarily for tropical markets. These units typically work with a sensible/total heat ratio of 0.7. Plotting this process line on the psychrometric chart (see Figure 35.3) shows that the ADP will be about 9°C.

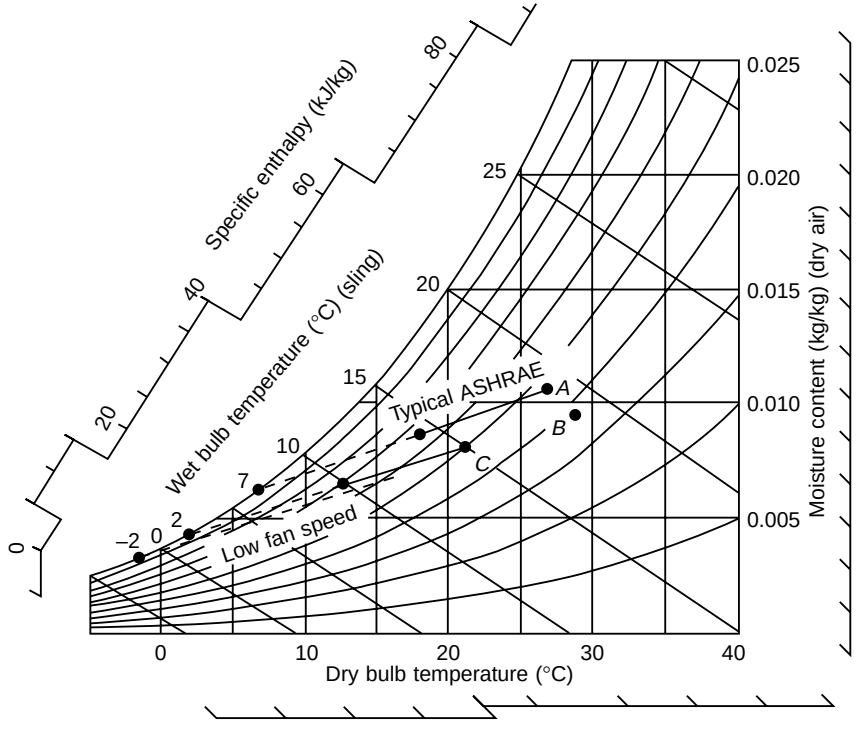


Figure 35.3 *Typical process lines for room air-conditioners*

For a room condition to BS.2852.C., and at full air flow, the ADP will be just above freezing point. If the unit is fitted with a low fan speed control, the ADP can fall below freezing and the coil frost over. Such units need to be fitted with a defrosting control and an allowance made for the time that the compressor will not be running.

35.7 Product quality

All equipment should comply with the relevant British and other Standards regarding dimensions, methods of determining ratings, compliance with safety regulations, robustness and general quality of manufacture [70]. BS.5750, Quality Systems, concentrates on the subject of product quality as it affects design, manufacture and installation. In addition to Standards, there are various Codes of Practice [71, 72].

Most catalogues give insufficient information for comparisons of quality, and an objective assessment may be difficult. For major items of equipment and in cases of doubt, it will be helpful to visit an existing installation or the factory.

Where the standard is for compliance with a safety requirement, a certificate to this effect should be provided, and may be demanded by insurers.

35.8 Analytical catalogue selection

Since a large proportion of refrigeration and air-conditioning equipment will be bought on the basis of catalogue data, an analytical approach should be adopted to ensure correct selection. The principles to be applied are those of *value analysis* – to start with the basic need and no preconceived method, to consider all the different methods of satisfying the need, and to evaluate each of these objectively before moving towards a choice.

The details of such an approach will vary considerably, and the following guidelines should be taken as an indication of the factors to be considered, rather than as an exhaustive list:

1. What is the basic need?
   To cool something, a dry product, in air: temperature?
   humidity?
   maximum air speed?
   other solid product?
   a liquid: what liquid?
   temperature range?
   viscosity?

   To keep something cool, a solid product / an enclosed space } conditions?

2. What is the load?
   Temperature?
   If at ambient, can it be done without mechanical refrigeration?

Product cooling load?
Heat leakage, sensible and latent?
Convection heat gains, sensible and latent?
Internal heat gains?
Time required?

3. Constraints
   Degree of reliability?
   Position of plant?
   Automatic/manned?
   Refrigerant?
   Same type of equipment as existing?
4. Possible methods
   Direct expansion?
   Indirect – what medium?
   Part by tower water or ambient air?
   Thermal storage?
   Existing plant spare capacity?
5. Location
   Plantroom?
   Adjacent space?
   Within cooled space?
   Maintenance access?
6. Condenser
   Inbuilt: water?
   air?
   Remote?
   Availability of cooling medium?
   Maintenance access?
7. Economy of first cost and running costs?
8. Options?

If these steps have been carried through in an objective manner, there will be at least three options for most projects, and possibly as many as five.

Enquiries can now go out for equipment to satisfy the need, based on the options presented. No attempt should be made to reach a decision until these have been evaluated.

Appendix Units of measurement

The *International System of Units* (SI) provides a coherent system of measurement units, and all the physical quantities required for refrigeration and air-conditioning can be derived from the basic standards:

Length	metre	m
Mass	kilogram	kg
Time	second	s
Electric current	ampere	A
Temperature	kelvin	K
Electric potential	volt	V

From these basic units are derived:

Area	square metre	m^2
Volume	cubic metre	m^3
Liquid volume	litre	$m^3 \times 10^{-3}$
Power	watt	W (ampere volt)
Force	newton	N (kg m/s^2)
Energy (Work)	joule	J (N m or W s)
Pressure	pascal	Pa (N/m^2)
also	bar	bar ($Pa \times 10^5$)
Temperature	degree Celsius	°C (K – 237.15)

From these, in turn, can be derived other units for use in the calculation of refrigeration and air-conditioning loads:

Specific heat capacity	J/(kg K) or kJ/(kg K)
Specific enthalpy	J/kg or kJ/kg
Thermal conductivity	W/(m K) ((W m)/(m^2 K))
Thermal conductance	W/(m^2 K)

In addition to SI, there are a number of expressions which remain in common use, since much available data is still recorded in these units, and practising engineers should be familiar with them:

Thermal energy	British thermal unit	Btu	= 1.055 kJ
	therm (Btu × 10^5)	therm	= 105.5 MJ
	kilocalorie	kcal	= 4.187 kJ
Thermal work	British thermal units per hour	Btu/h	= 0.293 W
	kilocalories per hour	kcal/h	= 1.163 W
	ton refrigeration	TR or t.r.	3.517 kW
Electrical energy	'unit of electricity'	kWh	= 3.6 MJ
Volume	Imperial gallon	Imp gal	= 4.546 litre
	US gallon	US gal	= 3.785 litre
Mass	pound	lb	= 0.4537 kg
	Imperial ton (2240 lb)	ton	= 1016 kg
	US ton (2000 lb)	US ton	= 907 kg
Length	foot	ft	= 0.305 m
Temperature	degree Fahrenheit	°F	= (1.8 × °C) + 32
Force	pound-force	lbf	= 4.448 N
Pressure	pound-force per square inch	lbf/in^2	= 6.895 kPa
	kilogram-force per square centimetre	kgf/cm^2	= 98.07 kPa
	inch water gauge	in w.g.	= 249 Pa
	bar	bar	= 100 kPa

Other terms not given here may be encountered from time to time and will be found in standard reference works [1, 2, 4, 10].

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