CHAPTER 13

Energy Efficient Air Conditioning and Mechanical Ventilation

Much energy is wasted in buildings through the use of inappropriate air-conditioning and mechanical ventilation systems. This situation has arisen because building designers are often ignorant of the issues associated with air conditioning and also because designers of air-conditioning systems are more interested in minimizing first costs rather than reducing overall energy consumption. There are, however, a number of new and innovative technologies, which have the potential to reduce energy consumption greatly. This chapter discusses the issues associated with the design of air-conditioning and mechanical ventilation systems, and introduces some of these new low energy technologies.

13.1 The Impact of Air Conditioning

Over the last 40 years or so, there has been a trend towards large deep plan buildings with highly insulated envelopes. This trend, coupled with the increased use of personal computers and the use of high illumination levels, has meant that many buildings overheat and thus require cooling for large parts of the year, even in countries which experience cool, temperate climates. Over the years boilers have steadily reduced in size and the use of air conditioning and mechanical ventilation has increased. When it is also considered that most of the Earth's population live in countries which have warm or hot climates, it is not difficult to appreciate that the provision of adequate cooling and ventilation is a much greater global issue than the provision of adequate heating. Unfortunately, many building designers are not fully aware of this simple fact, with the result that a great number of poorly designed buildings are erected, relying on large air-conditioning systems in order to maintain a tolerable internal environment.

The contribution of mechanical cooling towards overall global energy consumption should not be underestimated. In the UK alone, it has been estimated that approximately 10,000 GWh of electrical energy is consumed per annum by air-conditioning equipment [1]. This represents approximately 14% of all the electrical energy consumed in the commercial and public administration sectors in the UK. Of this figure, approximately 5853 GWh is consumed solely by refrigeration plant, the rest being consumed by fans, pumps and controls [2]. In the USA, the energy consumed by air-conditioning equipment is much higher. Indeed, in many of the southern states in the USA, electrical demand increases by 30–40% during the summer months solely due to the use of air-conditioning equipment [3]. As a result of this, the utility companies in the southern states of the USA have to install excess generating capacity to meet the summer peak, even though for most of the year this plant remains idle, which is clearly a very uneconomical situation. The problems faced by the electrical utilities in the USA are typical of many companies operating in warm climates throughout the world. In some countries, electrical demand is so high during the summer months that the authorities ration electricity by restricting the capacity of power cables which enter properties. In doing so they force building owners and users to utilize low energy design solutions.

It is a common misconception that most of the energy consumed by air-conditioning plant is associated with the operation of refrigeration machines. This is not the case. In reality much more energy is consumed by air-handling plant. A recent study of typical 'standard' air-conditioned office buildings in the UK found that refrigeration plant consumed 13% of total electricity consumption, while fans, pumps and controls consumed 26.5% of all the electrical energy consumed. In this type of office building approximately 35% of the total energy costs were spent on running the air-conditioning and mechanical ventilation systems [4]. A summary of the results of this study is presented in Table 13.1.

The environmental impact of air-conditioning equipment is considerable. Air conditioning is uniquely catastrophic from an environmental point of view, since it:

- Can contribute directly to atmospheric ozone depletion through the leakage of harmful refrigerants.
- Contributes directly to global warming through the leakage of refrigerants which are powerful greenhouse gases.
- Contributes to global warming by consuming large amounts of electricity and indirectly releases large quantities of CO_2 into the atmosphere.

A full discussion of the environmental problems associated with refrigerants is beyond the scope of this book. Nevertheless, a brief discussion of environmental matters is perhaps relevant here. Until recently, both CFCs and HCFCs were extensively used as refrigerants. Although CFCs and HCFCs are known to be potent greenhouse gases, they are much

TABLE 13.1 Energy consumption in various UK office buildings [4]

	Naturally ventilated cellular		Naturally ventilated open-plan		Air-conditioned standard		Air-conditioned prestige	
	Good practice (kWh/m^2)	Typical (kWh/m^2)	Good practice (kWh/m^2)	Typical (kWh/m^2)	Good practice (kWh/m^2)	Typical (kWh/m^2)	Good practice (kWh/m^2)	Typical (kWh/m^2)
Heating and hot water (gas or oil)	79	151	79	151	97	178	107	201
Mechanical cooling	0	0	1	2	14	31	21	41
Fans, pumps and controls	2	6	4	8	30	60	36	67
Humidification	0	0	0	0	8	18	12	23
Lighting	14	23	22	38	27	54	29	60
Office equipment	12	18	20	27	23	31	23	32
Catering (gas)	0	0	0	0	0	0	7	9
Catering (electricity)	2	3	3	5	5	6	13	15
Other electricity	3	4	4	5	7	8	13	15
Computer room (where applicable)	0	0	0	0	14	18	87	105
Total gas oil	79	151	79	151	97	178	114	210
Total electricity	33	54	54	85	128	226	234	358

more infamous for being potent ozone depletors. Indeed, it was the serious threat to the ozone layer which ended production of CFCs in 1995 under the Montreal Protocol [5]. Since then there has been heavy reliance on the use of HCFC-22 as an alternative to CFCs. While HCFC-22 is far more ozone friendly than CFC-11 or -12, it is still a potent greenhouse gas, having a global warming potential (GWP) of 1700 [6]. However, under the Montreal Protocol, HCFCs are also being phased out, with production due to cease completely by 2030 [7]. Consequently, the chemical manufacturers are currently developing a new generation of refrigerants, HFCs, to replace the old CFCs and HCFCs. Unfortunately, while HFCs are ozone benign they are still strong greenhouse gases. Notwithstanding this, the relative effect which refrigerants have on global warming is often overestimated. The contribution to global warming made by escaping refrigerants is far outweighed by the indirect CO_2 emissions resulting from the electrical consumption of refrigeration machines. This is graphically illustrated by Figure 13.1 which shows the relative contribution to global warming of associated CO_2 emissions compared with that of a variety of refrigerants [8].

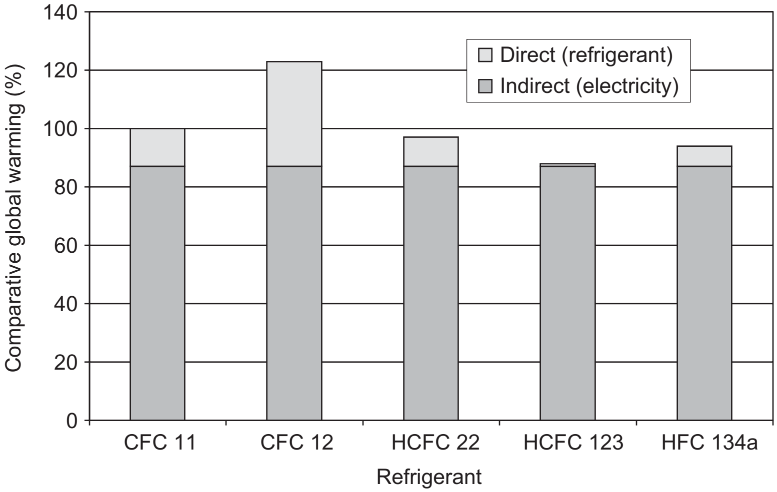

FIG 13.1 Comparison of the direct and indirect contribution of various refrigerant machine types towards global warming [8].

It can be seen from Figure 13.1 that the indirect contribution of air-conditioning equipment towards global warming is considerable. It has been estimated that in the UK alone, 4.2 million tonnes of CO_2 per annum are directly attributable to the use of air-conditioning refrigeration plant [2]. It is therefore not surprising that governments around the world are putting pressure on building designers to reduce or eliminate the need for mechanical cooling.

13.2 Air-Conditioning Systems

This chapter is not intended to be a text on the fundamentals of air-conditioning design, but rather a discussion of the application of air conditioning in buildings. Before discussing in detail the issues which affect the energy consumption of air-conditioning systems, it is necessary first to describe briefly the nature and operation of a generic air-conditioning installation. It should be noted that in this text, for ease of reference, the term *air conditioning* is used in its loosest sense to describe any system which employs refrigeration to cool air in buildings.

Figure 13.2 shows a simple air-conditioning system which illustrates many generic features. The system employs an air-handling unit (AHU) to blow air at a constant volume flow rate through ducts to a room space. Stale air is then removed from the room space via an extract duct using a return fan. In order to save energy, it is common practice to recirculate a large proportion (e.g. 70%) of the return air stream using mixing dampers located in the AHU. It is also common practice to propel air along the ducts at velocities in excess of 5 m/s. This ensures that ductwork sizes are kept to a minimum. Room space temperature is controlled by varying the temperature of the incoming supply air; in winter, air is supplied at a temperature higher than that of the room space, while in summer the air is supplied at a temperature lower than that of the room space.

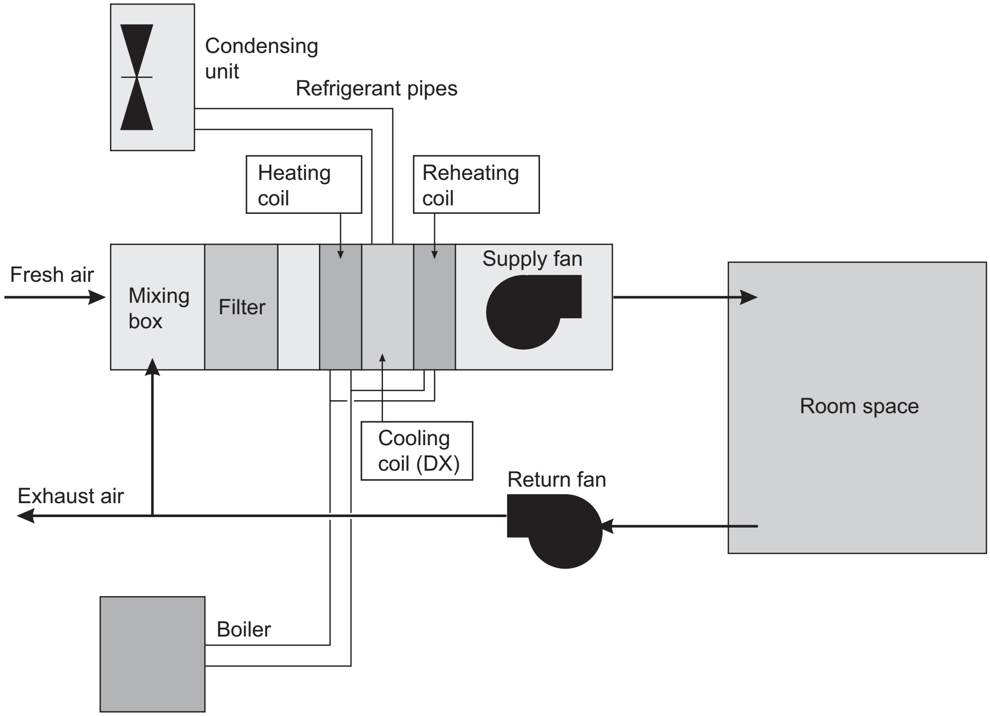

FIG 13.2 A simple ducted air-conditioning system with a direct expansion (DX) cooling coil.

In this way a comfortable environment can be maintained all year round in the room space.

The AHU in Figure 13.2 comprises:

- A mixing damper section to mix the incoming fresh air with recirculated air.
- A filter to clean the air.
- A heating coil (usually a hot water coil fed from a boiler but sometimes electric).
- A cooling coil to cool and dehumidify the air.
- A reheat coil to accurately control the air temperature and to compensate for any over-cooling by the cooling coil.
- A centrifugal fan to draw the air through the AHU and to push it through the ductwork.

In the case of the AHU shown in Figure 13.2, a direct expansion (DX) refrigeration coil is used to cool the supply air. This coil is the evaporator of a refrigeration system and contains liquid refrigerant which boils at a low temperature and pressure (e.g. at 5°C and 584 kPa) to become a low-pressure vapour. As liquid boils it draws large quantities of heat from the air stream and thus cools it. At the other end of the refrigerant pipes to the DX coil, a condensing unit is located which comprises a compressor, a condenser and a fan. The heat taken from the supply air stream by the DX coil is rejected at the condenser to the atmosphere.

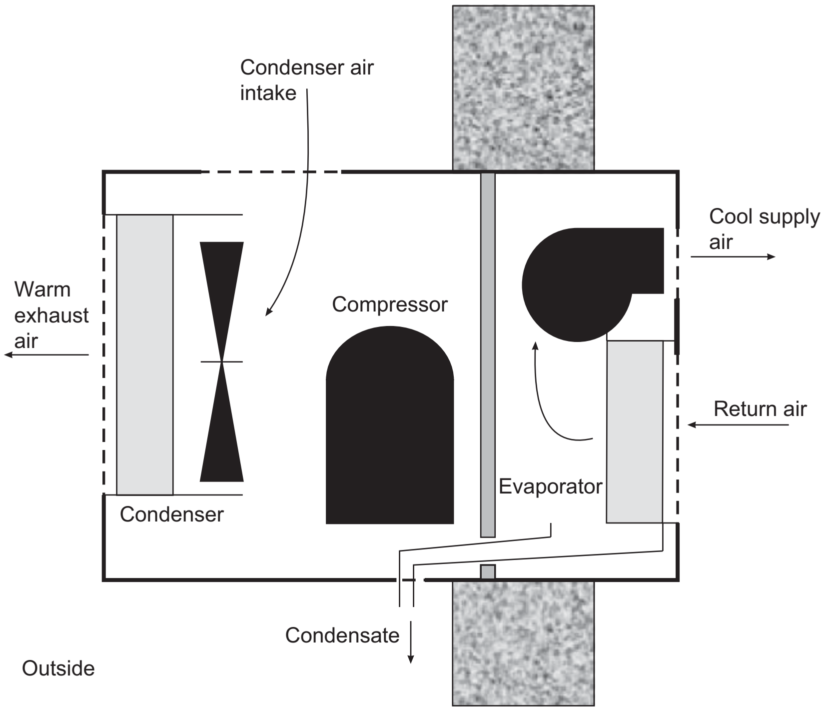

FIG 13.3 A through-the-wall air-conditioning unit.

The system shown in Figure 13.2 is generic and is typical of many systems found throughout the world. There are, however, some variations which are worthy of note. In many applications in hot countries there is no requirement for heating, and so the heating coils are removed, leaving only the DX cooling coil. Similarly there may be no requirement to supply fresh air, in which case the filter and mixing dampers can be omitted. Common examples of such simple systems are the 'through-the-wall unit' (see Figure 13.3) and the 'split unit' systems (see Figure 13.4). These systems are inexpensive and easy to maintain, and not surprisingly, are very popular in many hot countries.

One of the major disadvantages of the systems described above is that in larger installations they require many condensing units to be placed on the outside of buildings. This can be both unsightly and impractical. So in many larger buildings, a superior solution is to install a centralized refrigeration machine, known as a chiller, to produce chilled water (e.g. at 7°C) which can then be pumped to a number of remote AHUs (see Figure 13.5). In this type of system, each AHU is fitted with a chilled water cooling coil instead of a DX coil. Chilled water cooling coils are superior to DX coils, because they facilitate closer control of the supply air temperature. Centralized chillers also have an environmental advantage over remote DX systems, insomuch as there are fewer refrigeration circuits to maintain, resulting in lower risk of refrigerant leaks. Chillers can utilize either air- or water-cooled condensers. Water-cooled condensers are more efficient than the air-cooled variety, but usually require a cooling tower, and are therefore a potential *Legionella pneumophila* hazard. For this reason, air-cooled chillers have

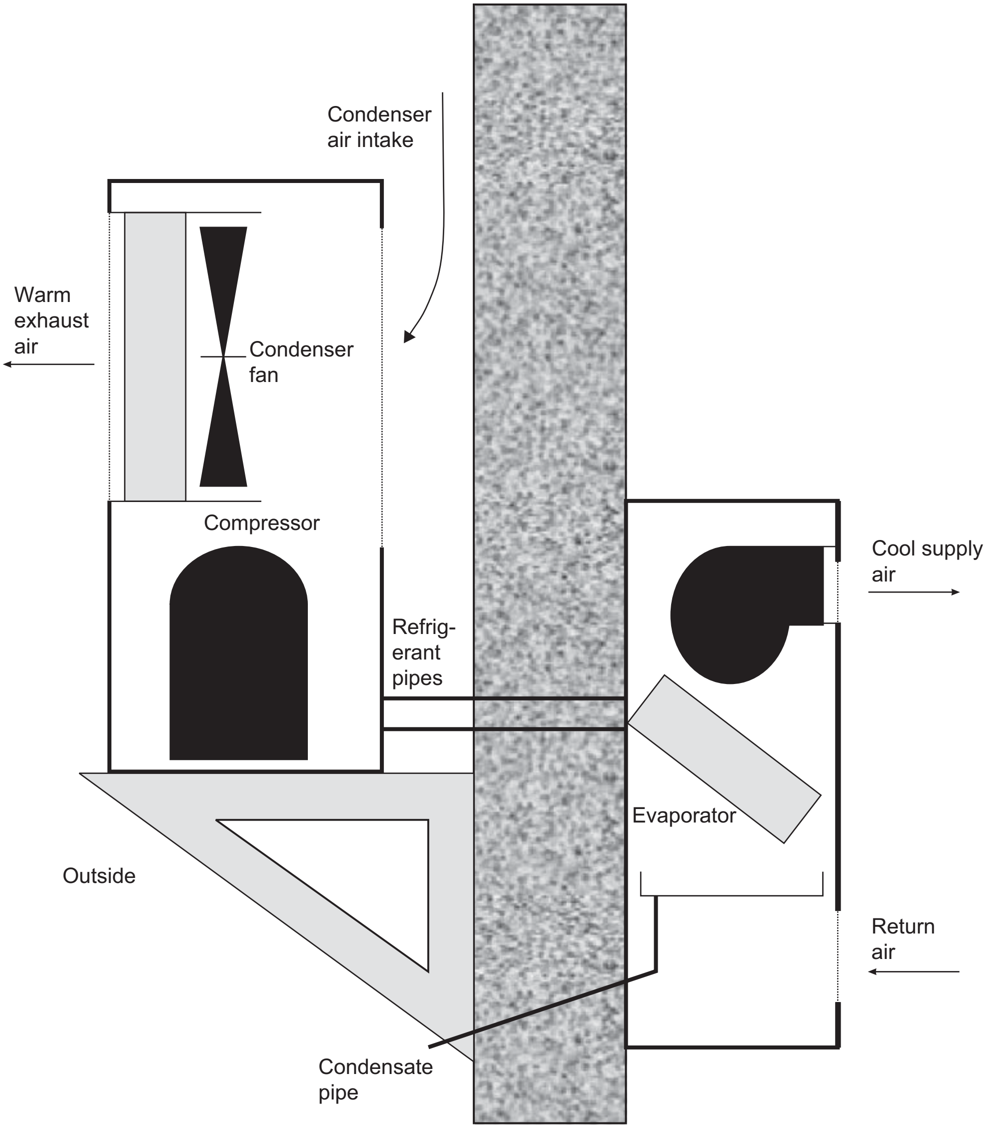

FIG 13.4 A split unit air-conditioning system.

become more popular than water-cooled chillers, because despite being less efficient they present no health hazard.

13.3 Refrigeration Systems

Most air-conditioning plant relies on some form of vapour compression refrigeration machine to remove heat from air. Figure 13.6 shows a schematic diagram of a simple,

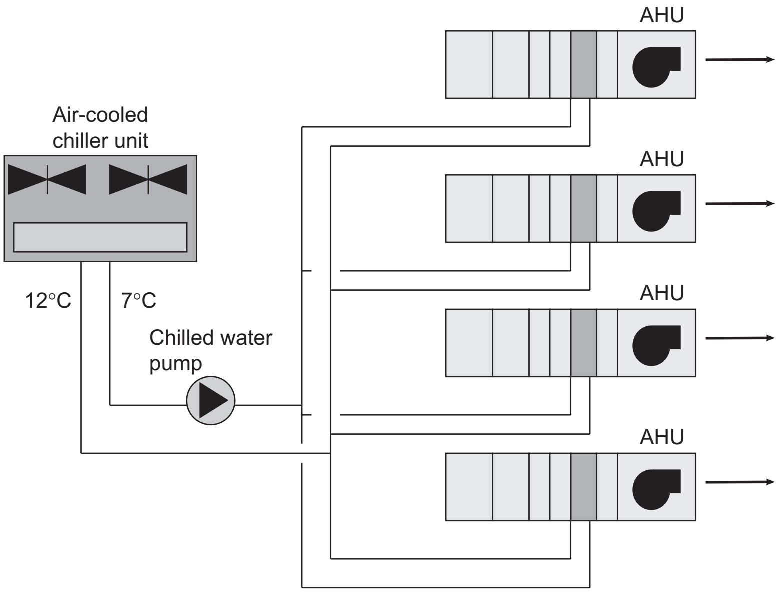

FIG 13.5 An air-cooled chiller system with multiple air-handling units.

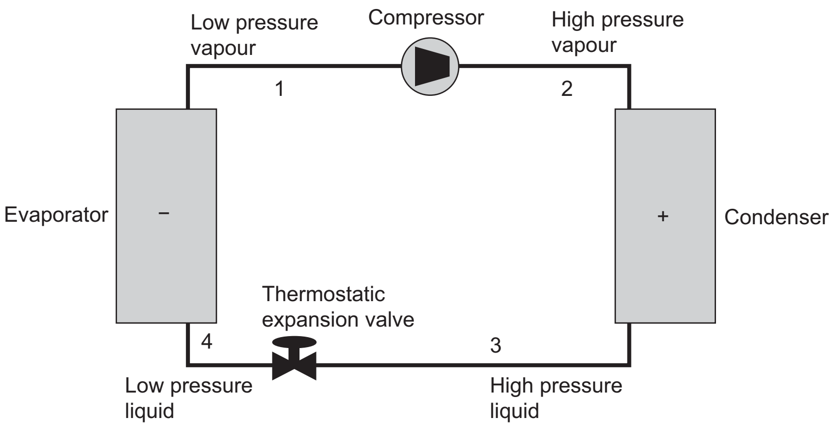

FIG 13.6 Vapour compression refrigeration cycles.

single-stage, vapour compression refrigeration system, similar to that found in many air-conditioning systems.

The vapour compression refrigeration cycle operates as follows:

1. Low-pressure liquid refrigerant in the evaporator boils to produce low-pressure vapour. The heat required to boil and vaporize the liquid within the evaporator is taken from an air or water stream passing over the outside of the evaporator.
2. After leaving the evaporator, the low-pressure refrigerant vapour enters the compressor where both its temperature and its pressure are raised by isentropic compression.

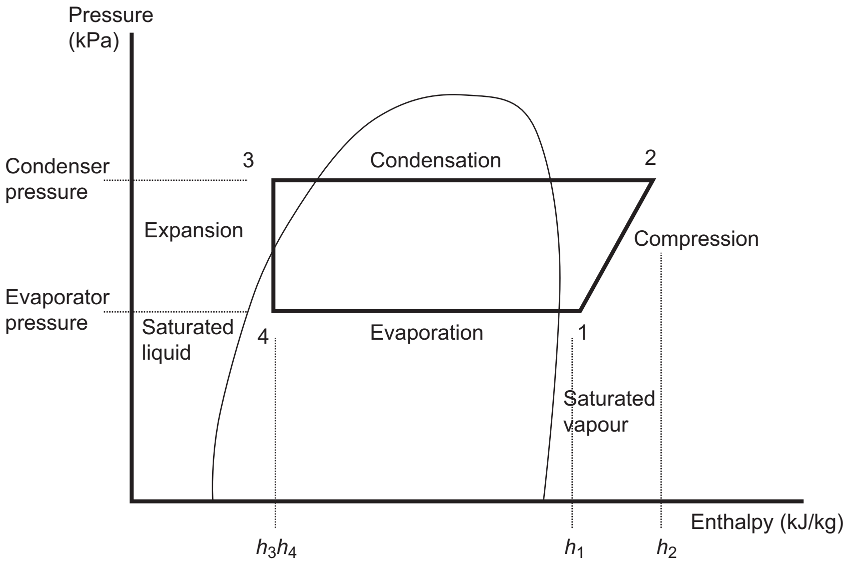

FIG 13.7 Pressure/enthalpy chart of vapour compression process.

3. The high-pressure refrigerant vapour then passes to the condenser where it is cooled and liquefied. The heat extracted in the condenser is released to the environment either directly by forcing air over the outside of the condenser, or indirectly using a secondary fluid, usually water, and a cooling tower.
4. The high-pressure liquid refrigerant then passes from the condenser to the expansion valve, where its pressure is lowered, and approximately 10% of the liquid 'flashes' (i.e. instantly turns from a liquid to a vapour) thus cooling the remainder of the liquid. The cooled low-pressure liquid then flows into the evaporator and the cycle begins all over again.

The boiling point of refrigerants varies with pressure. At low pressures, refrigerants boil at low temperatures (e.g. 2°C), while at much higher pressures the boiling point of the refrigerant is significantly raised (e.g. 35°C). In this way refrigerants can be vaporized and condensed at different temperatures, simply by altering system pressure. Since condensing and evaporating temperatures correspond to particular pressures, they are normally 'measured' using pressure gauges located before and after the compressor.

Figure 13.7 shows a plot of the vapour compression cycle on a pressure/enthalpy diagram. The refrigeration capacity of the system is the amount of cooling which the plant can achieve and is proportional to the length of the line between points 4 and 1. The power input to the system is through the compressor drive, and is represented by the line 1 to 2. Line 2 to 3 represents the heat rejection at the condenser. Line 3 to 4 represents the passage of the refrigerant through the expansion device and is a constant enthalpy process. It should be noted that the heat rejected by the condenser is equal to the total energy input at the evaporator and at the compressor. Note also that as the condensing pressure decreases, so too does the power input from the compressor.

The overall 'efficiency' of a vapour compression machine is normally described by the COP. The COP of a refrigeration machine is the ratio of the refrigeration capacity to the power input at the compressor. It can be expressed as (referring to Figure 13.7):

$$COP_{ref} = \frac{h_1 - h_4}{h_2 - h_1} \tag{13.1}$$

where h is the specific enthalpy of refrigerant (kJ/kg).

The higher the COP, the more efficient the refrigeration process. In the UK, refrigeration machines generally exhibit COPs in the range of 2.0–3.0 [2].

13.4 The Problems of the Traditional Design Approach

Having briefly discussed the nature of air-conditioning and refrigeration systems, the overall design of 'air-conditioned' buildings must now be considered. It is generally the case that in buildings the air-conditioning design is something of an afterthought. In many buildings the form and the envelope are designed in complete isolation from the mechanical services. Usually, air-conditioning engineers are required to design and install systems which fit unobtrusively into buildings (i.e. behind suspended ceilings); often these systems are required to overcome the environmental shortcomings of poor envelope design.

The traditional approach to air conditioning is to employ a constant volume flow rate system, similar to that described in Section 13.2. However, this design approach has a great many inherent weaknesses, which can loosely be categorized as:

- Weaknesses of the building design.
- Weaknesses of the refrigeration system.
- Weaknesses of the air system.

13.4.1 Building Design Weaknesses

Buildings have to function properly in a great many harsh environments around the world. In hot desert climates they are required to keep their occupants cool, while in polar regions keeping warm is the important issue. Buildings should therefore be designed so that the external envelope is the primary climate modifier, with the internal mechanical services simply fine-tuning the shortcomings of the envelope. This may seem an obvious statement, but clearly it is not one which is fully understood by many building designers. In Texas, which has a hot arid climate, there are many glass-clad office buildings. It would be intolerable to work in these buildings were it not for the use of very large air-conditioning systems to compensate for the inappropriate building envelope. This apparently ludicrous situation comes about for three specific reasons:

1. Energy efficiency is often a low priority; minimizing the first cost is usually the prime consideration.

2. Design professionals often work in isolation from each other and have little understanding of building physics, or of how buildings function when occupied.
3. There is great incentive to maintain the status quo. Building services design engineers are usually paid a fee which is a fixed proportion of the total capital cost of the building services. Consequently, there is little incentive to reduce the capacity of the mechanical building services.

In order to avoid the creation of energy wasteful buildings, it is important that energy efficiency be at the forefront of the designer's mind. Critical decisions made at the design stage have huge ramifications on both capital and operating costs. If the envelope is a poor climate modifier then the building will experience high summer heat gains and high winter heat losses, necessitating the installation of large boilers and refrigeration chillers. These items of equipment may, however, only operate at peak load for a few hours per year, with the result that for most of the year they operate very inefficiently at part load. Conversely, if the building envelope successfully attenuates the winter and summertime peaks, then the plant sizes can be greatly reduced, resulting in the plant operating near its rated capacity for a much greater part of the year. Clearly, the latter situation is a much better utilization of capital expenditure than the former.

13.4.2 Refrigeration System Weaknesses

The strategy of installing refrigeration plant to meet the peak summertime cooling load of buildings often results in greatly oversized mechanical plant which operates inefficiently, at part load, for most of the year. It can also result in a greatly oversized electrical installation, since larger cables, transformers and switchgear must be installed to meet the peak refrigeration capacity. Not only does a system such as this have a high capital cost, it is also expensive to run since it uses peak-time electricity. In hot countries it may also incur high electrical demand related charges. Refrigeration chillers are often oversized because:

- System designers overestimate peak building heat gains to ensure that plant is not undersized.
- System designers make design assumptions which are widely inaccurate. For example, in most buildings the actual cooling load is much less than the design cooling load. This discrepancy primarily occurs because designers assume very high 'office equipment' heat gains, which in practice rarely materialize.
- Refrigeration chillers are often rated for hot climates such as that found in the USA. So when these machines are installed in a temperate climate location such as in the UK, their condensers are oversized and so they operate at part load even when under peak-load conditions.

The general oversizing of refrigeration machines results in very poor overall energy efficiencies. It has been estimated that of the refrigeration plant currently in operation in the UK, the average air-cooled chiller has a working gross COP of approximately 1.9, while water-cooled chillers exhibit an average gross COP of 3.0 [2]. These low figures are mainly due to the design of most of the refrigeration chillers used in the UK, namely machines which use thermostatic expansion valves and maintain a relatively fixed condensing pressure under part-load conditions. These machines display poor COPs under part-load conditions, which is unfortunate since for most of the year they operate in this state.

There are a number of alternative design strategies which can be used dramatically to reduce the size of refrigeration plant and improve operating costs. These are:

- The use of a solar defensive building envelope, incorporating features such as external shading and solar reflective glass to reduce the peak cooling load.
- The use of a thermally massive structure to absorb both internal and solar heat gains during peak periods.
- The use of night ventilation to purge the building structure of heat accumulated during the daytime.
- The use of ice thermal storage to shift some of the peak-time cooling load to the night-time.
- The use of a floating internal air temperature strategy, which allows internal temperatures to rise when conditions are exceptionally hot.

13.4.3 Air System Weaknesses

The strategy of using an *all-air* system to condition room air has the major disadvantage that it necessitates the transportation of large volumes of air and is thus inherently inefficient. It is generally the case that much larger volumes of supply air are required to sensibly cool room spaces than are required for the purpose of pure ventilation. Consequently, the provision of sensible cooling using an *all-air* system results in large fans and associated air-handling equipment, and also in large ceiling (or floor) voids to accommodate oversized ductwork. Although an expensive solution, resulting in increased energy and capital expenditure, all-air systems are still very popular, despite the existence of superior alternatives which use chilled water to perform the sensible cooling. One such alternative strategy is the use of chilled ceilings to perform sensible cooling, while reserving the ductwork system for ventilation purposes only. This strategy results in greatly reduced fan and ductwork sizes.

Another major drawback of the constant volume approach, described in Section 13.2, is that air duct and fan sizes are determined by the peak summertime condition which may only last for a few hours per year. For the rest of the year the fans push large volumes of air around needlessly, with the result that energy consumption on air handling is large. Many air-conditioning system designers argue that constant volume systems have the potential to provide large amounts of free cooling during the spring and autumn seasons. This unfortunately is a misconception since oversized fans consume such large quantities of electrical energy that any saving in refrigeration energy is wiped out. The evidence for this can be seen in Table 13.1, where in the air-conditioned buildings almost twice as much electrical energy is consumed by air-distribution systems compared with the refrigeration machines. One alternative strategy which overcomes this problem is to adopt a variable air volume system in which the quantity of the air handled reduces with the cooling load.

It is common practice in ducted air systems to size the main ducts assuming air velocities of 4–7 m/s. Designers use these relatively high air velocities in order to keep duct sizes to a minimum. Unfortunately, the use of such high air velocities results in large system resistances.

The fan power consumed in a ducted air system can be determined using eqn (13.2). Fan power,

$$W = \dot{v} \times \Delta P_{\text{total}} \tag{13.2}$$

where $\dot{v}$ is the volume flow rate of air discharged by the fan (m^3/s), and ΔP_{total} is the total system pressure drop or resistance (Pa).

From eqn (13.2) it can be seen that there is a linear relationship between fan power and system resistance; the higher the system resistance, the higher the energy consumed by the fan. The system resistance is the sum of the system static pressure drop and the velocity pressure drop. The velocity pressure in particular strongly influences the pressure drop across ductwork bends and fittings. From eqn (13.3) it can be seen that the pressure drop across a ductwork fitting is a function of the square of the air velocity.

Pressure drop across ductwork fitting:

$$\Delta P = k \times (0.5 \rho v^2) \tag{13.3}$$

where k is the velocity pressure loss factor for fitting, ρ is the density of air (kg/m^3), v is the velocity of air (m/s), and ($0.5\ \rho v^2$) is the velocity pressure (Pa).

Given eqns (13.2) and (13.3) it is not difficult to see that the use of high air velocities (i.e. in the region of 5 m/s) results in high fan powers and high energy consumption. However, if air velocities are reduced to approximately 1–2 m/s, as is the case in some *low energy* buildings, then fan energy consumption falls dramatically.

13.5 Alternative Approaches

The critique of the traditional approach to the design of air-conditioning systems presented in Section 13.4 highlights its many shortcomings and hints at a number of possible solutions. There are several alternative low energy strategies which may be employed to overcome the disadvantages of the conventional approach. Although interlinked, for ease of reference these alternative strategies can loosely be categorized as follows:

- Using passive solar defensive and natural ventilation measures to reduce the need for air conditioning;
- Splitting the sensible cooling and ventilation roles into two separate but complementary systems;
- Using low velocity and variable air volume flow systems;
- Using the thermal mass of buildings to absorb heat which can then be purged by a variety of ventilation techniques;
- Using thermal storage techniques to shift the peak cooling load to the night-time;
- Using displacement ventilation techniques;
- Using evaporative cooling; and
- Using desiccant cooling techniques.

Many of the energy conservation techniques listed above are discussed in detail in this chapter. Some of the techniques which relate specifically to building envelope design are specifically dealt with in Chapter 15.

13.6 Energy Efficient Refrigeration

Although a number of alternative cooling strategies are discussed in this chapter, there are still many applications which demand the use of conventional refrigeration plant. It is therefore necessary to understand the factors which influence the energy consumption of conventional vapour compression refrigeration machines. The major factors influencing energy performance are:

- The evaporating and condensing temperatures used.
- The type of refrigerant used.
- The type of compressor and condenser used.
- The defrost method used on the evaporator.
- The system controls.

Each of these factors can have a profound effect on overall energy consumption and are therefore worthy of further investigation.

13.6.1 Evaporators

The efficiency of vapour compression systems increases as the evaporating temperature increases. Generally, the higher the evaporating temperature used, the greater the system COP and the lower the energy consumption. It has been estimated that a rise in the evaporating temperature of 1°C results in an operating cost reduction of between 2% and 4% [9]. It is therefore desirable to maintain the evaporating temperatures as high as is practically possible. Maximum heat transfer across the evaporator should be achieved in order to prevent the evaporating temperature from dropping. In practice, this can be achieved by increasing the fluid flow across the evaporator, or by increasing its surface area. Also, in order to ensure high evaporating temperatures it is essential that good control of the system be maintained.

On air-cooling applications where the evaporator may be operating below 0°C the fin spacing must allow for ice build-up. In order to maintain an adequate airflow through the evaporator it is necessary to defrost the coil periodically. Defrosting techniques involve either the use of an electric heating element built into the coil or periodically reversing the refrigeration cycle so that the evaporator effectively becomes a hot condenser. While essential for the correct operation of the system, the defrost process can be a potential source of energy wastage. It is therefore important that the defrost operation only be initiated when absolutely necessary and that the defrost heat be evenly distributed over the whole of the fin block. Prolonged defrosting is energy wasteful and therefore the defrost cycle should be stopped as soon as possible. If not controlled and monitored properly defrost systems can needlessly waste large amounts of energy.

13.6.2 Condensers

Condensing temperature can have a dramatic influence on system COP, with lower condensing temperatures usually resulting in lower operating costs. It is estimated that a 1°C drop in condensing temperature reduces operating costs by approximately 2–4% [9]. However, if the condensing pressure fluctuates widely, problems can occur on machines which utilize thermostatic expansion valves. This is because such valves are unable to reliably control refrigerant flow at low pressure differentials. In order to overcome this problem, these machines often employ some form of condenser pressure control to raise the condenser pressure artificially. This results in unnecessarily high energy consumption, which could be avoided if electronic expansion devices were used instead.

There are three basic condenser systems commonly in use: air-cooled condensers, water-cooled condensers and evaporative condensers, each of which has its own peculiarities. Air-cooled condensers are by far the most popular heat rejection system. They generally comprise a fin and tube heat exchanger in which refrigerant vapour condenses. Air is forced over the heat exchanger by fans. Well-designed condensers should operate at a temperature no higher than 14°C above the ambient air temperature [10]. In larger air-cooled systems, condenser pressure is often controlled by switching off or slowing down fans. Although this practice is inefficient it does save on energy consumed by the condenser fans. One important advantage of air-cooled condensers is that they present no *Legionella pneumophila* risk.

Water-cooled condensers are much more compact than their air-cooled counterparts and comprise a shell-containing refrigerant, through which pass water-filled tubes. Secondary cooling water flows through these tubes to a cooling tower where the heat is finally rejected through an evaporative cooling process. In an efficient system, the temperature rise of the water passing through the condenser should be 5°C, with a difference of 5°C existing between the condensing temperature and the temperature of the water leaving the condenser [10].

Water-cooled systems are considerably more efficient than air-cooled systems, with the former requiring a cooling tower airflow rate of approximately 0.04–0.08 m^3/s per kW of rejected heat and the latter requiring 0.14–0.2 m^3/s to perform the same task [11]. However, there is a risk of *Legionella pneumophila* bacteria breeding in cooling towers, if they are not monitored and regularly treated with biocides. For this reason water-cooled condensers have become less popular in recent years.

Evaporative condensers are less popular than either air- or water-cooled condensers. They comprise refrigerant condensing tubes which are externally wet and over which air is forced. Evaporation of the water on the outside of the tubes increases the heat rejection rate and so this type of condenser is more efficient than its air-cooled counterpart. Evaporative condensers do, however, pose a *Legionella pneumophila* risk.

13.6.3 Compressors

The compressor is the only part of a vapour compression system which consumes energy. It is therefore important that the factors which influence compressor performance are well understood. The efficiency of a compressor can be expressed in several ways. In terms of overall energy consumption the most critical 'efficiency' is isentropic efficiency, which is defined as follows:

$$\text{Isentropic efficiency} = \frac{\text{ideal 'no loss' power input}}{\text{actual power input to shaft}} \times 100 \qquad (13.4)$$

It should be noted that isentropic efficiency does not take into account motor and drive inefficiencies, which must be allowed for when determining the overall efficiency of a compressor. With most types of compressors, particularly screw and centrifugal compressors, efficiencies fall dramatically under part-load operation. Compressor motor efficiency also decreases at part load. In general, therefore, part-load operation should be avoided if high efficiencies are to be maintained.

Part-load operation is the main reason for poor refrigeration plant efficiency. Many refrigeration machines spend less than 20% of the year operating at their nominal design condition. During the rest of the year they operate at part load, partly because of cooler ambient temperatures and partly because of reduced cooling duties. Unless these part-load conditions are properly allowed for at the design stage, it is likely that overall system COP will be poor. It is therefore important to select a compressor which exhibits good part-load efficiency. Multi-cylinder reciprocating compressors achieve reasonable part-load efficiencies because they are able to unload cylinders so that output is reduced in steps (e.g. 75%, 50% and 25%). Variable speed drives can also be used. These give good flexible control and can achieve reasonable efficiencies above 30% of full load [12].

13.6.4 Expansion Devices

In a refrigeration machine the expansion valve is used to reduce the pressure of the returning liquid refrigerant from the condensing pressure to the evaporating pressure. It also controls the flow of liquid refrigerant to the evaporator. It is therefore important that expansion valves be correctly selected and installed, since incorrect operation of the expansion valve can lead to reduced energy efficiency.

Thermostatic expansion valves are the most commonly used type of refrigerant regulation device. They regulate the flow of refrigerant through the system by opening and closing a small orifice using a 'needle' connected to a diaphragm (as shown in Figure 13.8). The diaphragm responds to pressure changes created inside a control phial which senses the temperature of the refrigerant leaving the evaporator. Both the phial and the valve contain refrigerant. As the load on the evaporator changes, so the temperature of the refrigerant leaving the evaporator also changes. The control phial senses these changes in temperature and automatically adjusts the refrigerant flow to accommodate the load changes.

The major disadvantage of thermostatic valves is that they cannot cope with large pressure differentials, such as those created when the condensing pressure is allowed to float with ambient air temperature. Thermostatic expansion valves tend to operate unsatisfactorily at less than 50% of their rated capacity [13]. Therefore, in refrigerating machines using thermostatic expansion valves it is often necessary to maintain an artificially high condensing temperature during conditions of low ambient temperature.

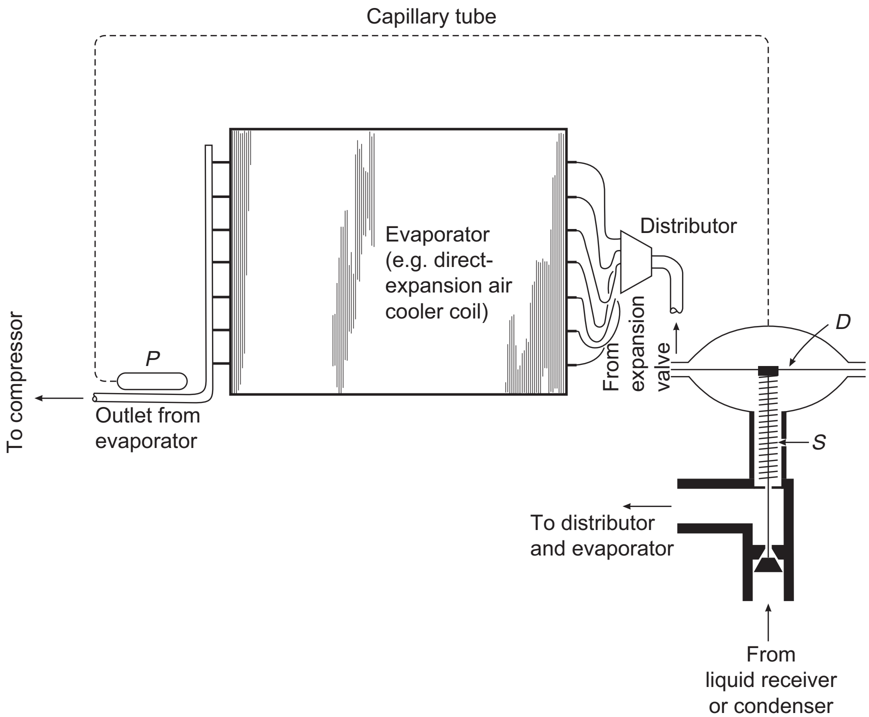

FIG 13.8 Thermostatic expansion device [11].

It can be seen from the discussion in Section 13.6.2 that if the condensing pressure is allowed to fall, the COP increases. Therefore, in theory, system efficiency should improve when ambient air temperature falls. Unfortunately, due to the operational characteristics of thermostatic expansion valves it is not possible to take advantage of this situation. Thermostatic expansion valves are therefore inherently inefficient. In recent years an alternative technology has arisen which overcomes this problem. By using electronic expansion valves it is possible to allow condensing pressures to drop while still maintaining a constant evaporating pressure. Unlike conventional thermostatic expansion valves, which operate on the degree of superheating in the evaporator, electronic expansion valves employ a microprocessor which constantly monitors the position of the valve, the temperature of the liquid in the evaporator, and the temperature of the vapour leaving the evaporator. They can therefore respond quickly to fluctuations in load and are not dependent on large differential pressures between the condenser and the evaporator. Consequently, it is possible under low ambient conditions to allow the condensing pressure to fall and the COP to improve.

13.6.5 Heat Recovery

Vapour compression refrigeration machines are simply a specific form of heat pump. This means that they reject large quantities of waste heat at the condenser. In many applications, through a little careful thought at the design stage, it is possible to utilize this waste heat profitably to reduce energy costs. In refrigeration installations heat can be recovered from:

- The compressor discharge gas, which is generally in the region 70–90°C, but can be as high as 150°C in some installations.
- The condenser, which is generally 10–30°C above ambient temperature.

High-quality waste heat can be recovered from the compressor discharge gas by using a desuperheater heat exchanger. This device recovers heat from the high temperature vapour before it reaches the condenser. At this point in the cycle the vapour is at its highest temperature and therefore the heat transfer is at its greatest. It is important to locate desuperheaters above condensers so that if any refrigerant vapour condenses, the liquid can safely drain away.

Heat may also be recovered at the condenser. However, because condensing temperatures should ideally be as low as possible, any heat recovered here will inevitably be at a relatively low temperature. The quality of this heat should, however, be sufficient to preheat domestic hot water. If higher condensing temperatures are envisaged as a result of any proposed heat recovery scheme, careful analysis should be undertaken to ensure that economic benefit will accrue. Remember, there are many so-called 'energy-saving' schemes in existence which have actually increased energy costs!

13.7 Splitting Sensible Cooling and Ventilation

When considering air conditioning, it is possible to save substantial amounts of energy by splitting up the sensible cooling and ventilation roles into two separate systems. This also enables fan and duct sizes to be greatly reduced. One commonly used method of separating these two roles is to use a fan coil system, which circulates chilled water through water/air heat exchangers (incorporating recirculation fans) located in room-mounted units. Although fan coils function well in many applications, they can take up valuable room space and also be noisy. In addition, they utilize recirculation fans which consume energy. A novel alternative approach is the use of passive chilled ceilings or beams, which comprise a cold surface mounted at high level within a room space. Chilled ceilings and beams perform room's sensible cooling and leave the ducted air system to perform the ventilation and latent cooling roles.

Chilled ceilings have a very slim profile and usually comprise a metal pipe coil bonded to a flat metal plate (as shown in Figure 13.9). They can be fixed directly to the soffit of a structural floor slab, thus eliminating the need for an expensive suspended ceiling. They are usually designed to have an average surface temperature of approximately 17°C, which can be achieved in practice by supplying low-grade chilled water at about 13°C. Because relatively high water temperatures are used it means that chiller evaporating temperatures can be high, resulting in very good refrigeration COPs being achieved.

The cooling power of chilled ceilings varies with the individual design used and the extent to which air turbulence occurs across the heat exchanger surface. In general an

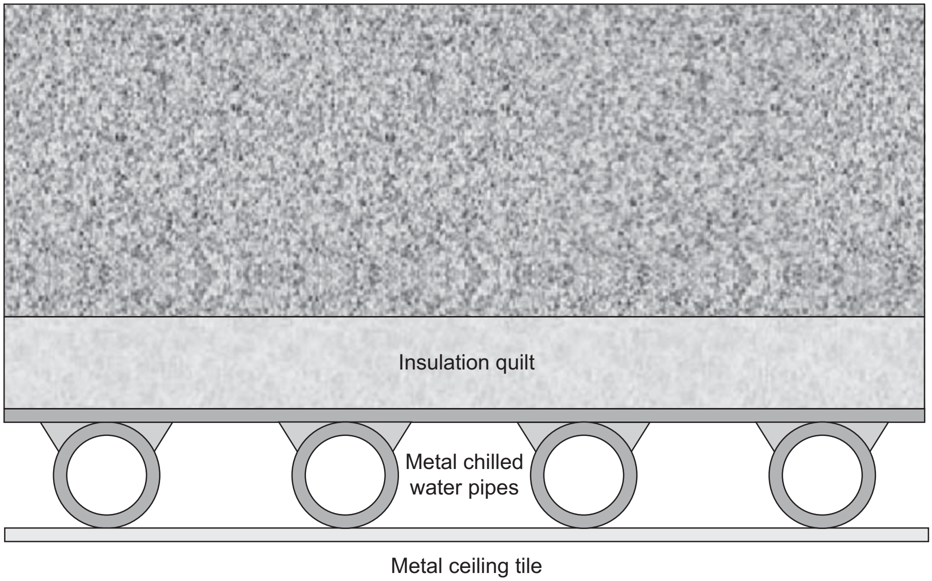

FIG 13.9 Typical chilled ceiling.

output of approximately 50 W/m^2 is considered to be the maximum that can be obtained. This means that chilled ceilings are not suitable for applications which experience very high internal heat gains. Heat is transferred to the chilled ceiling by natural convection and radiation. Approximately 50% of the heat transfer occurs when warm air at the top of the room space comes into contact with the cool surface. The remaining heat transfer is by radiation from room occupants and other warm surfaces within the room space.

The radiative cooling capability of chilled ceilings is of particular importance and is worthy of further comment. At relative humidities below 70%, the thermal comfort of building occupants is primarily governed by room air temperature and room surface temperature. In order to assess and quantify relative thermal comfort, a number of thermal comfort indices have been developed. Although these comfort indices vary slightly from each other, they all seek to quantify the convective and radiative heat transfer to and from an occupant within a room space. The most widely used thermal comfort index in the UK is *dry resultant temperature*. Provided room air velocities are less than 0.1 m/s (which is usually the case), dry resultant temperature can be expressed as:

$$t_{res} = 0.5t_a + 0.5t_r \tag{13.5}$$

where t_r the mean radiant temperature (°C), and t_a is the air temperature (°C).

Mean radiant temperature is the average surface temperature of all the surfaces 'seen' in a room space. It can be either measured indirectly using a globe thermometer or calculated from surface temperatures. For most cuboid-shaped rooms, the mean radiant temperature in the centre of the room can be expressed as:

$$t_r = \frac{\Sigma(A_s \cdot t_s)}{\Sigma A_s} \tag{13.6}$$

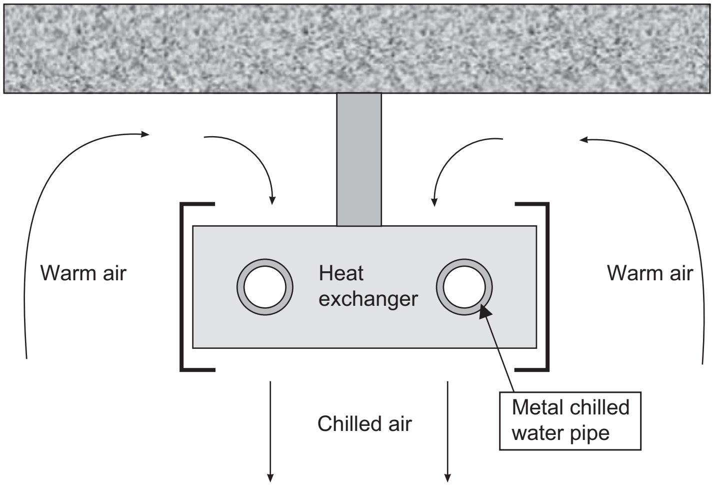

FIG 13.10 Typical chilled beam.

where A_s is the area of each component surface (m^2), and t_s is the temperature of each component surface (°C).

By installing a large surface area of chilled ceiling at 17°C, it is possible to substantially reduce room mean radiant temperature. Consequently, air temperatures can be allowed to rise to, say, 23°C or 24°C without any deterioration in perceived comfort, with the result that energy can be saved.

A variation on the chilled ceiling theme is the passive chilled beam system (see Figure 13.10). Passive chilled beams work on a similar principle to chilled ceilings, but they achieve a much greater cooling output (e.g. 185 W/m^2) and exhibit a much higher convective cooling component than chilled ceilings (e.g. approximately 85%). However, this can cause problems, since uncomfortable downdraughts can be created.

13.7.1 Ventilation

By utilizing chilled ceilings or beams, it is possible to free up ductwork systems to concentrate solely upon the ventilation and latent cooling tasks. Normally the fresh air requirement of room occupants is in the region 8–12 l/s per person. However, when using chilled ceilings or beams the ducted air ventilation system also has to perform all of the room latent cooling. In order to do this, it is common for the ventilation rate to be increased slightly to 18 l/s per person [14].

With chilled ceilings and beams it is important to ensure that the room air moisture content be maintained at a low level, otherwise condensation may occur on the cool surface and ultimately 'internal rain' may be formed. This makes it important to ensure that the incoming ventilation air is dehumidified, so that the room dew point temperature remains a few degrees below the surface temperature of the chilled ceilings. An ideal air condition for a room incorporating a chilled ceiling is 24°C and 40% relative humidity.

13.8 Fabric Thermal Storage

Although a full discussion of the role of fabric thermal storage is contained in Chapter 15, a few words on the subject are perhaps relevant here. The widespread use of suspended ceilings and carpets in buildings means that otherwise thermally heavyweight structures are converted into thermally lightweight ones. These low-admittance buildings are not able to absorb much heat and so surface temperatures tend to rise, with the result that there is a great need to get rid of the heat gains as they occur. This is one of the main reasons why air conditioning has become such an essential requirement of so many office buildings. By contrast, if the mass of the building structure is exposed, then a high-admittance environment is formed, and the thermal capacity of the structure can be utilized successfully to combat overheating.

The creation of a high-admittance environment by exposing thermal mass has implications on the comfort of occupants. It can be seen from the discussion in Section 13.7 that it is the dry resultant temperature and not the air temperature which is critical when establishing a comfortable environment. By exposing the 'mass' of a building it is possible to reduce the mean radiant temperature within the space, and thus the dry resultant temperature. So if an office building with openable windows and exposed concrete floor soffits has a mean radiant temperature of, for example, 20°C and the air temperature in the space is 28°C, then the perceived temperature (i.e. the dry resultant temperature) in the space will be only 24°C. While an internal air temperature of 28°C is generally considered unacceptable, a dry resultant temperature of 24°C will be perceived as tolerable on hot summer days.

In buildings which employ 'thermal mass' to control internal temperatures, it is common practice to expose concrete floor soffits to create a high-admittance environment, as can be seen in examples such as the Queens Building at De Montfort University [15] and the Elizabeth Fry Building at the University of East Anglia [16]. While it is possible to create a high-admittance environment by exposing concrete floor soffits, the structure needs to be purged periodically of heat absorbed over time, otherwise the mean radiant temperature of the room spaces will steadily rise until conditions become unacceptable. One effective method which can be employed to purge heat from the structure of buildings is night venting. Night venting involves passing cool outside air over or under the exposed surface of a concrete floor slab so that it is purged of the heat accumulated during the daytime. This can be done either by natural or mechanical means. At its most rudimentary, night venting may simply entail the opening of windows at night-time to induce cross-ventilation, while at its most sophisticated it may involve a dedicated mechanical night ventilation system and the use of floor voids.

When creating a night venting scheme it is important to ensure that good thermal coupling occurs between the air and the mass of the concrete floor, whilst at the same time ensuring that fan powers are kept to a minimum. One system which manages to achieve this is the Swedish Termodeck hollow concrete floor slab system (see Chapter 15). The Termodeck system has been used effectively in many locations, throughout northern Europe and in the UK [16,17], to produce buildings which are both thermally stable and energy efficient.

13.9 Ice Thermal Storage

In the exposed concrete soffit system described in Section 13.8, the cooling process is effectively 'load shifted' to the night-time by using a passive thermal storage technique. While this can be very effective, the system is limited to cool, temperate countries where night-time temperatures are low enough to purge accumulated heat. In hotter climates this strategy is impossible and so an alternative approach to load shifting is required. One alternative technology is ice thermal storage, which utilizes low cost night-time electricity to produce a 'cold store' for use during the daytime. The technique involves running refrigeration chillers during 'off-peak' hours to produce an ice store. During the daytime when electricity prices are high, the ice is melted to overcome building or process heat gains. The principal advantages of the system are as follows:

- Refrigeration energy costs can be significantly reduced, as a substantial portion of the cooling is undertaken using off-peak electricity.
- The capital cost of the refrigeration plant can be significantly reduced, if both the chillers and the store combine to satisfy the peak cooling load requirement.
- If an ice store is coupled with a conventional refrigeration plant, then it is possible to run the chiller constantly at 100% of its rated capacity and thus operate it in an efficient manner.
- If an ice store is coupled with an electronically controlled refrigeration plant, then it is possible to minimize the refrigeration energy expended. This is because the refrigeration plant will be running at night-time when ambient temperatures are low and so operating COPs will be high.
- Any electricity maximum demand charges incurred by the system will be significantly lower than those incurred by conventional refrigeration plant.
- By installing additional ice stores it is possible to increase the overall capacity of existing air-conditioning installations without purchasing new chillers or upgrading electrical systems.
- Ice storage systems enable CO_2 emissions to be reduced through load shifting [18] and can also reduce the quantity of refrigerant used.

Ice storage systems are generally associated with air-conditioned commercial or public buildings. However, ice storage systems have also been used successfully in process industries which experience large and predictable cooling loads. In this type of application it is often the case that a relatively small refrigeration machine can, over a long period of time, generate a large ice store. The ice store can then be melted over a relatively short period of time, to satisfy the peak cooling load. In this way small refrigeration machines can be used to satisfy very large cooling loads, with the result large capital savings can be made on refrigeration plant. In addition, capital cost savings can be made on electrical cables and switch gear, which is of particular importance for applications in remote locations.

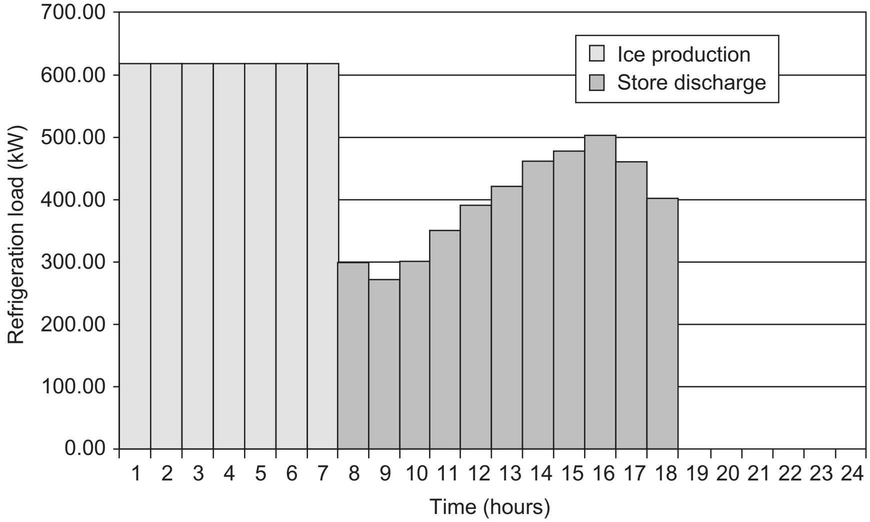

FIG 13.11 Full storage strategy.

13.9.1 Control Strategies

Ice thermal storage systems can be operated in a variety of ways, with the major control strategies being *full storage, partial storage* and *demand-limited storage*.

(a) *Full storage*: Under a *full storage* control strategy the total daytime cooling load is shifted to the night-time, with the chillers producing an ice store during the period when off-peak electricity charges apply. During the daytime the ice store is discharged to meet the building or process cooling load, as shown in Figure 13.11. While being the most effective of all the control strategies in terms of energy costs, *full storage* has the major drawback that the ice store and chiller plant required are much larger than for the other control strategies. Due to its prohibitively high capital cost *full storage* is rarely used.

(b) *Partial storage*: *Partial storage* is the collective term given to those ice storage control strategies which require both the chiller plant and the ice store to operate together to satisfy the daytime cooling load. During periods in which the building or industrial process experiences a cooling load, the ice store and the chiller plant work simultaneously to satisfy the cooling load. The advantage of *partial storage* is that both the store and the chiller plant are substantially smaller than would be the case for a *full storage* installation and thus the capital cost is lower. This makes *partial storage* a very popular option. The umbrella term partial storage can be sub-divided into two separate and distinct sub-strategies, namely *chiller priority* and *store priority*.

Under a *chiller priority* control strategy, the refrigeration plant runs continuously through both the ice production and the store discharge periods. During the daytime the refrigeration plant carries out the base-load cooling and the ice store is used to top up the refrigeration capacity of the chiller plant (see Figure 13.12), which would otherwise be unable to cope with the peak demand.

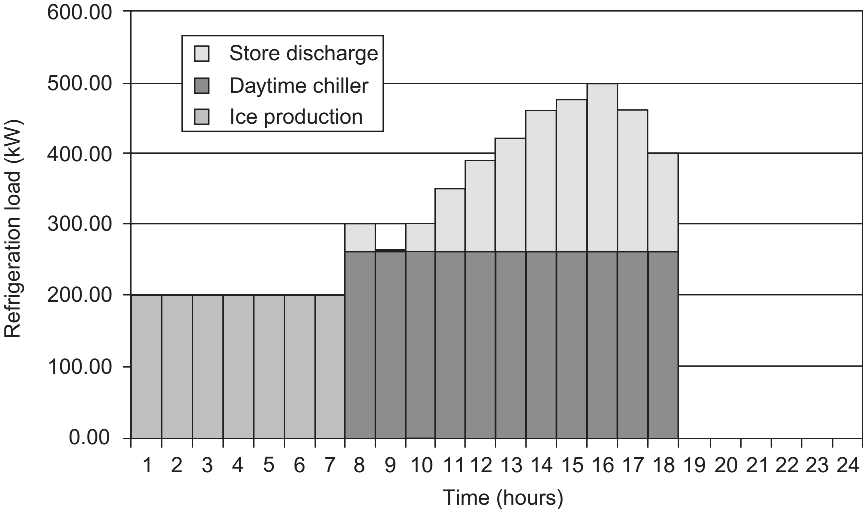

FIG 13.12 Chiller priority strategy.

Under a *chiller priority* strategy it is possible to achieve reductions in the region of 50% in chiller capacity when compared with a conventional refrigeration installation. The capital cost of installing an ice store can therefore be offset against the capital cost saving arising from the reduction in chiller capacity.

The philosophy behind the *store priority* control strategy is the opposite of the *chiller priority* strategy. Under a *store priority* strategy the ice store is given priority over the chiller during the daytime (see Figure 13.13). The objective of this strategy is to minimize the operation of the refrigeration plant during periods when electricity prices are high. The refrigeration chiller is only used to top up the refrigerating energy released by the ice store.

(c) *Demand-limited storage*: The object of a *demand-limited* control strategy is to limit peak electrical demand by shifting the cooling load out of periods in which the peak demand naturally occurs (see Figure 13.14). This greatly reduces the overall maximum demand of the installation and improves the overall load factor of the building, putting the operators in a stronger position when it comes to negotiating electricity supply contracts with the utility companies.

A *demand-limited* control strategy is particularly useful in situations where a utility company offers an electricity tariff which has either high unit charges or high demand charges for part of the daytime (e.g. from 12 am to 6 pm), as is often the case in hot countries during the summertime. Under these circumstances, during the period for which peak charges apply, the cooling load should be entirely satisfied by the refrigeration energy released from the ice store.

13.9.2 Ice Thermal Storage Systems

In broad terms, ice storage systems fall into two main categories, static systems and dynamic systems. Static systems have the general characteristic that ice is melted in the same location as it is generated. Unlike static systems where the ice remains in one

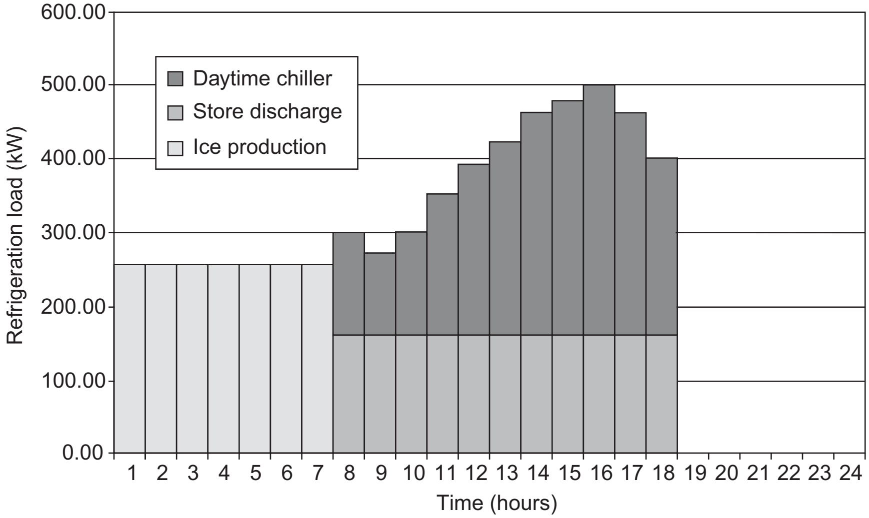

FIG 13.13 Store priority strategy.

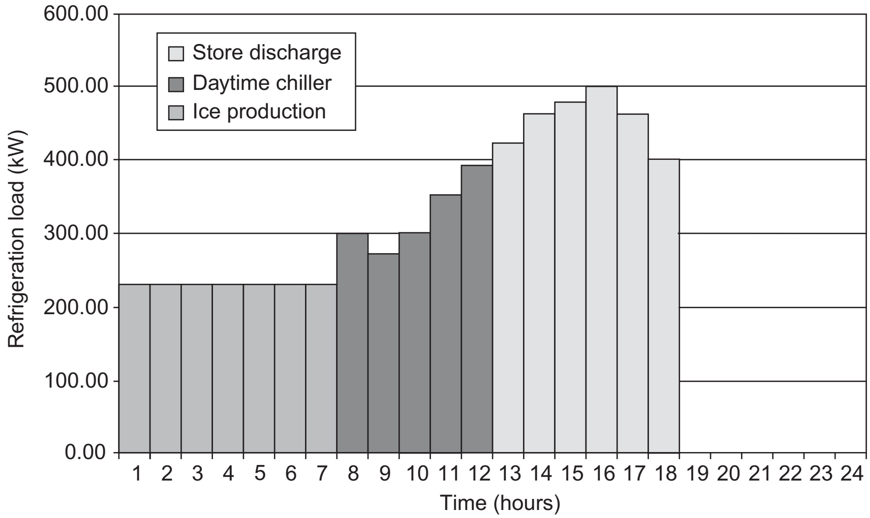

FIG 13.14 Demand-limiting strategy.

location throughout the entire operation of the installation, in dynamic systems the ice, once formed, is transported by some means to another location where it comes into contact with the working fluid, which is usually water.

The ice bank system shown in Figure 13.15 is typical of a static ice system. It consists of an insulated water storage tank, which contains a submerged bundle of small tubes. These tubes are evenly spaced within the tank volume, often in a spiral or serpentine form. During ice production a glycol/water solution at a sub-zero temperature is circulated through the tubes. This causes the water in the tank to freeze solid. During the discharge cycle, the ice is melted by the same glycol/water solution, this time circulating at a temperature above 0°C.

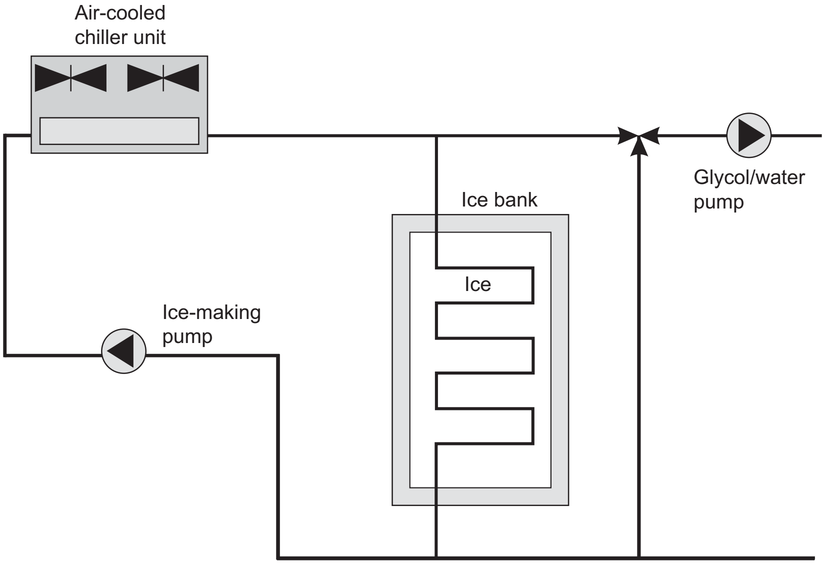

FIG 13.15 Ice bank system.

The most widely used dynamic system is the ice harvester. Ice harvesters have been used in the dairy industry for many years. They consist of an open insulated tank, above which a number of vertical refrigerant evaporator plates are located. Water is trickled over the surface of the plates so that it becomes frozen. Typically, within about 20 minutes an 8–10 mm thick layer of ice can be built up. The ice is harvested by removing it from the evaporator and allowing it to fall into the tank below. This process is achieved by interrupting the flow of liquid refrigerant through the evaporator plates and diverting hot discharge gas through them so that their surface temperature reaches approximately 5°C. A photoelectric switch can be used to stop ice production when the ice in the sump reaches the required level. To discharge the ice store, system water is circulated through the ice sump. A typical ice harvester installation is shown in Figure 13.16.

13.9.3 Sizing of Ice Storage Systems

The design calculations used to size ice storage systems depend on the precise control strategy which is adopted [19]. If a *chiller priority* control strategy is adopted then eqns (13.7)–(13.13) should be used. For a *store priority* strategy eqns (13.14)–(13.16) should be used:

$$Q_{st} + Q_{ch} = Q_j \tag{13.7}$$

where Q_{st} is the refrigeration energy contained within the ice store (kWh), Q_{ch} is the refrigeration energy produced by chiller plant when operating in the daytime (kWh), and Q_j is the daily cooling load (energy) under design condition (kWh).

Under a *chiller priority* control strategy it is intended that the chiller plant should operate at full capacity throughout the daytime period. However, it is not always possible to

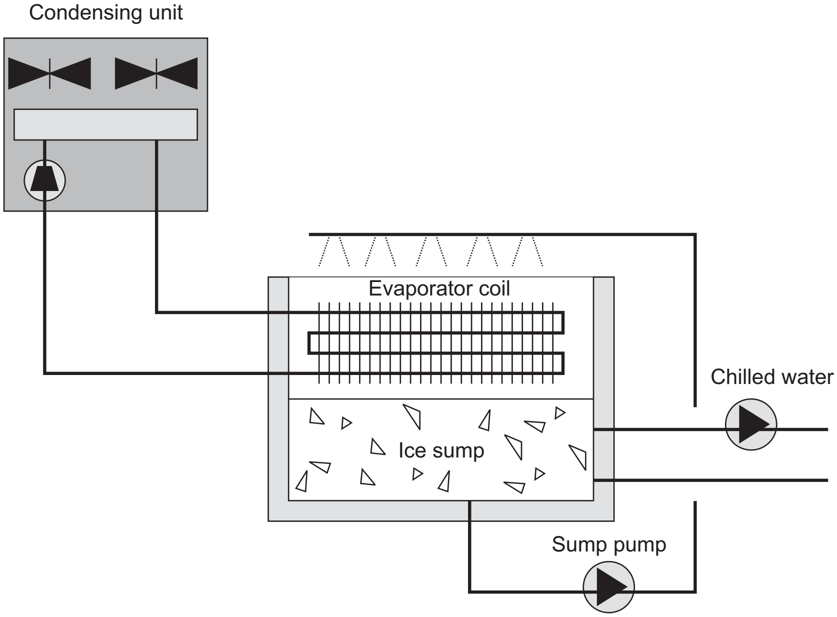

FIG 13.16 Ice harvester system.

achieve this. A chiller plant will often operate at below its rated capacity for part of the daytime. Consequently, eqn (13.7) must be modified to accommodate this:

$$Q_{st} + Q_{ch} = Q_j + Q_u \tag{13.8}$$

where Q_u is the unused chiller refrigeration energy (kWh).

The evaporating temperatures experienced by the refrigeration plant are much lower during the ice production than those experienced during daytime operation. Consequently, during the store-charging period the chiller plant will experience reduced refrigerating capacity. It can therefore be stated that:

$$Q_{ch} = P_r \cdot H \tag{13.9}$$

and

$$Q_{st} = P_r \cdot k_r \cdot h \tag{13.10}$$

where P_r is the rated duty of chiller under daytime operation (kW), k_r is the reduction factor for chiller producing ice, H is the duration of daytime chiller operation (hours), and h is the duration of ice production period (hours).

Therefore

$$Q_{st} + Q_{ch} = P_r \cdot (H + k_r \cdot h) \tag{13.11}$$

By combining eqns (13.8) and (13.11) it can be shown that:

$$P_r = \frac{Q_j + Q_u}{H + k_r \cdot h} \tag{13.12}$$

By combining eqns (13.8) and (13.9) it can be shown that:

$$Q_{st} = Q_j + Q_u - H \cdot P_r \tag{13.13}$$

In order to derive the plant-sizing equations for a *store priority* control strategy, a slightly different approach is taken to that for the *chiller priority* equations. The concept of peak cooling load (P_m) is introduced. It can therefore be stated that:

$$Q_{st} + H \cdot P_r = H \cdot P_m - Q_v \tag{13.14}$$

where P_m is the peak cooling load experienced by building (kW), and Q_v is the unused ice storage capacity (kWh).

Therefore

$$Q_{st} = H \cdot P_m - Q_v - H \cdot P_r \tag{13.15}$$

By combining eqns (13.10) and (13.14) the following is produced:

$$P_r = \frac{H \cdot P_m - Q_v}{H + k_r \cdot h} \tag{13.16}$$

The process involved in sizing ice thermal storage systems is illustrated in Example 13.1.

Example 13.1

An office building has a peak daily cooling load of 5210 kWh, with a maximum instantaneous cooling duty of 620 kW. Given the following data:

(i) Determine the size of the ice store and chiller plant required for chiller priority, store priority and full storage control strategies, and for a conventional chiller-only system.
(ii) Determine the daily costs for the options outlined in (i) above.
(iii) Determine the system capital costs for the options outlined in (i) above.

Data:
Off-peak electricity period = 00.00–07.00 hours
Peak electricity period = 07.00–24.00 hours
Air-conditioning operation period = 08.00–18.00 hours
Daytime average COP = 3.00
Ice production COP = 2.25
Chiller capacity reduction factor for ice production = 0.75
Peak unit charge = 5.50p/kWh
Off-peak unit charge = 2.57p/kWh

Capital cost of ice storage = 25 £/kWh
Capital cost of refrigeration chiller = 240 £/kW
NB: Assume that there is no unused refrigeration energy or ice store capacity in the process.

Solution

(i) *Conventional chiller-only system*: There is no ice store so the chiller must have a refrigeration capacity of 620 kW.

Full storage control strategy: The chiller is not in operation during the daytime and so the ice store is required to satisfy all the daytime cooling load. Thus a large ice store is required to be built up over the 7-hour off-peak period. Therefore:

$$\text{Ice store capacity} = 5210\ \text{kWh}$$

and

$$\text{Nominal chiller duty} = \frac{5210}{7 \times 0.75} = 992.4\ \text{kW}$$

Chiller priority control strategy: The office air-conditioning system is operational for 10 hours.

Therefore:

$$\text{Nominal chiller duty} = \frac{5210}{10 + (0.75 \times 7)} = 341.6\ \text{kW}$$

and

$$\text{Store capacity} = 5210 - (10 \times 341.6) = 1793.6\ \text{kWh}$$

Store priority control strategy: The peak summertime cooling duty is 620 kW. Therefore:

$$\text{Nominal chiller duty} = \frac{10 \times 620}{10 + (0.75 \times 7)} = 406.6\ \text{kW}$$

and

$$\text{Store capacity} = (10 \times 620) - (10 \times 406.6) = 2134.4\ \text{kWh}$$

(ii) *Conventional chiller-only system*:

$$\text{Daily energy cost} = \frac{5210 \times 5.50}{3.0 \times 100} = £95.52$$

Full storage control strategy:

$$\text{Daily energy cost} = \frac{5210 \times 2.57}{2.25 \times 100} = £59.51$$

Chiller priority control strategy:

$$\text{Daily energy cost} = \frac{1793.6 \times 2.57}{2.25 \times 100} + \frac{(10 \times 341.6) \times 5.50}{3.0 \times 100}$$
$$= £83.11$$

Store priority control strategy:

$$\text{Daily energy cost} = \frac{2134.4 \times 2.57}{2.25 \times 100} + \frac{(5210 - 2134.4) \times 5.50}{3.0 \times 100}$$
$$= £80.77$$

(iii) *Conventional chiller-only system*:

$$\text{Capital cost of installation} = 620 \times 240 = £148,800.00$$

Full storage control strategy:

$$\text{Capital cost of installation} = (992.4 \times 240) + (5210 \times 25) = £368,426.00$$

Chiller priority control strategy:

$$\text{Capital cost of installation} = (341.6 \times 240) + (1793.6 \times 25) = £126,824.00$$

Store priority control strategy:

$$\text{Capital cost of installation} = (406.6 \times 240) + (2134.4 \times 25) = £150,944.00$$

Results summary:

	Conventional chiller-only system	**Chiller priority**	**Store priority**	**Full storage**
Chiller duty	620 kW	341.6 kW	406.6 kW	992.4 kW
Store capacity	n.a.	1793.6 kWh	2134.4 kWh	5210 kWh
Capital cost	£148,800	£126,824	£150,944	£368,426
Peak daily cost	£95.52	£83.11	£80.77	£59.51
Payback period*	n.a.	–4.85 years	0.40 years	16.71 years

*Indicative payback periods only, since calculated daily costs only apply to the peak summertime day.

From the results shown in Example 13.1 it can be seen that the initial choice of control strategy has a huge impact on the economic viability of any ice storage scheme. Given the inordinately large capital cost associated with *full storage* systems it is not surprising that these systems are rarely installed. The most popular strategy used is the *chiller priority* strategy which, as in the case in Example 13.1, can result in capital cost savings and a negative payback period.

13.10 Evaporative Cooling

So far, most of the cooling techniques discussed in this chapter have utilized the vapour compression refrigeration cycle to perform air cooling. There are, however, a number of alternative technologies, such as evaporative cooling and desiccant cooling, which can be utilized to perform air cooling. Direct evaporative cooling is perhaps the simplest of all the air-cooling techniques and is extremely energy efficient. It relies on an adiabatic heat exchange between air and water in which the air is both sensibly cooled and humidified. Most direct evaporative cooling systems comprise an open porous matrix over which water is trickled and through which air can pass (see Figure 13.17). As air passes over the wetted media, water evaporates and so the air becomes more humid. In order to evaporate, the water needs a 'package' of latent heat energy; this it takes from the air stream, with the result that the air is sensibly cooled. Figure 13.18 shows the evaporative cooling process on a psychrometric chart. It should be noted that the whole process is adiabatic and it follows the line of constant enthalpy on the psychrometric chart, which approximates to the line of constant wet-bulb temperature.

While direct evaporative coolers generally exhibit efficiencies of about 85% [20], their sensible cooling effectiveness depends very much on the dryness of the air entering the cooler. If the air is very dry, then a large amount of sensible cooling will be

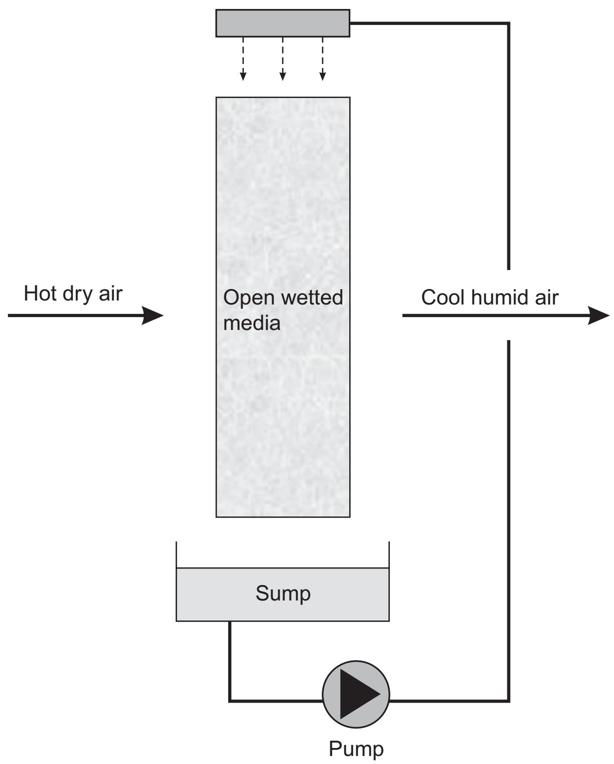

FIG 13.17 A direct evaporative cooler.

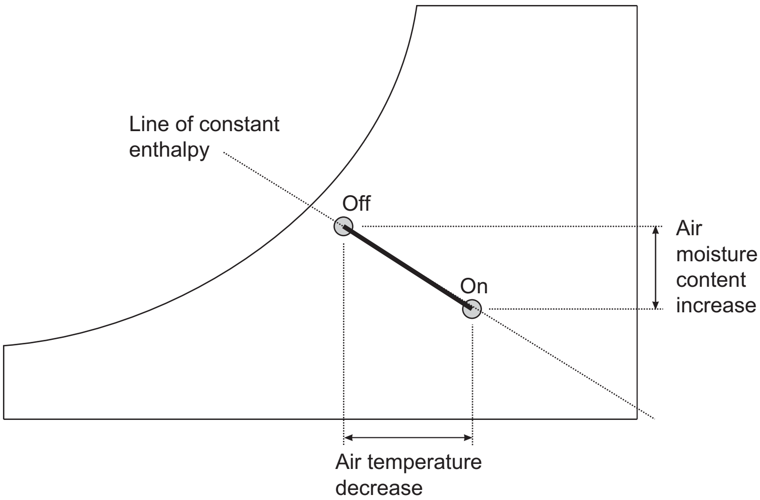

FIG 13.18 The direct evaporative cooling process.

achieved. Conversely, if the air has a high relative humidity, very little sensible cooling will be achieved. Not surprisingly, therefore, evaporative coolers have been used extensively for many years in hot arid countries, where they are often referred to as *desert coolers*. Direct evaporative cooling is very cost-effective and eliminates the need for any environmentally unfriendly refrigerants.

One major disadvantage of direct evaporative cooling is that it greatly raises the relative humidity and moisture content of the air entering the room space, which may ultimately cause discomfort to room occupants. This problem can be overcome by introducing a heat exchanger to create an *indirect* evaporative cooling system. With indirect systems it is standard practice to place an evaporative cooler in the room exhaust air stream coupled to a flat plate heat recuperator. By using this arrangement the cool but humid exhaust air stream can be used to sensibly cool the incoming fresh air supply stream. It should be noted that there is no moisture exchange between the two air streams and so the supply air remains relatively dry. Figure 13.19 shows the indirect evaporative cooling process on a psychrometric chart. Indirect evaporative coolers will usually achieve an effectiveness of at least 60% and can achieve effectiveness ratings as high as 85% [20].

13.11 Desiccant Cooling

Another alternative to the conventional vapour compression refrigeration cycle is to use a heat-driven cycle. It has long been understood that desiccant materials such as silicon can be used to dehumidify air. Such systems pass moist air over surfaces which are coated with a desiccant substance. As the moist air passes across these surfaces the desiccant material absorbs moisture from the air, thus dehumidifying the air stream. In order to drive off the moisture absorbed by the desiccant surface, the desiccant has to

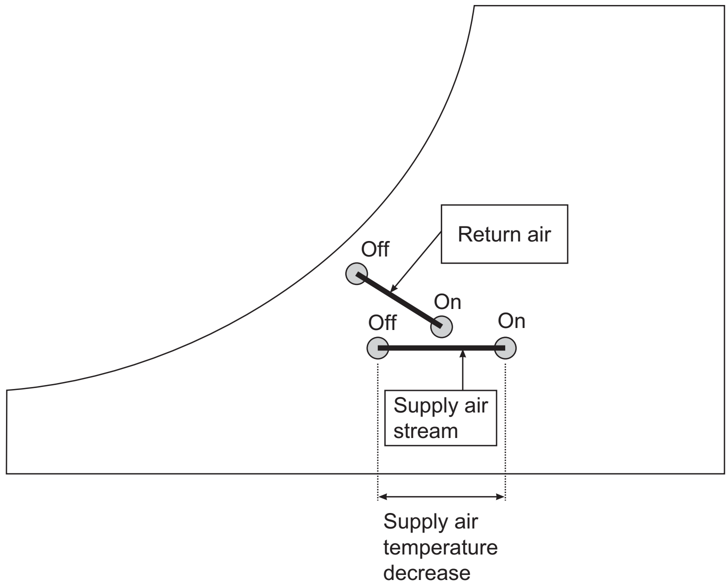

FIG 13.19 The indirect evaporative cooling process.

be physically moved into a hot dry air stream. In the case of the desiccant wheel system (one of the most commonly used desiccant systems) the moisture-laden section of the wheel rotates slowly at approximately 16 revolutions per hour from the moist air stream to the hot dry air stream where it is regenerated.

Recently desiccant cooling systems have been developed which combine a desiccant wheel with a thermal wheel in a single AHU to produce a system which is capable of heating, cooling and dehumidifying air with little or no need for conventional refrigeration [21]. Such systems have the potential to reduce both energy costs and environmental pollution when compared with conventional refrigeration systems. From an environmental point of view the desiccant cooling system has the advantage that electrical energy consumption is replaced by heat consumption, which produces much less CO_2.

A typical desiccant cooling AHU is shown in Figure 13.20. It comprises a thermal wheel and a desiccant wheel located in series. On the supply side, after the thermal wheel, a supplementary cooling coil or an evaporative cooler may be located if so required. A heating coil may also be located after the thermal wheel for use in winter if required. An evaporative cooler is located in the return air stream before the thermal wheel so that the heat transfer across the thermal wheel is increased. The desiccant cycle is an open heat-driven cycle; the 'driver' for the cycle is the regeneration heating coil located in the return air stream after the thermal wheel and before the desiccant wheel.

The psychrometric chart shown in Figure 13.21 illustrates the desiccant cooling and dehumidification process. During the summertime warm moist air at, for example, 26°C and 10.7 g/kg moisture content is drawn through the desiccant wheel so that it comes off at, say, 39°C and 7.3 g/kg moisture content. The psychrometric process line for the air passing

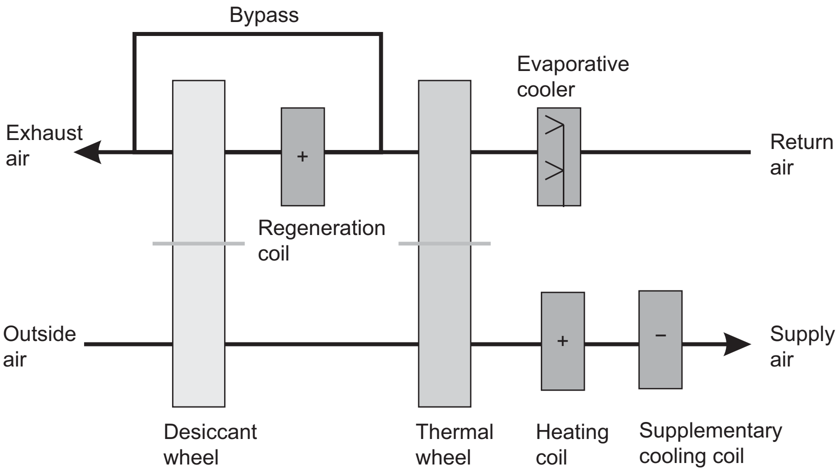

FIG 13.20 A typical desiccant cooling air-handling unit.

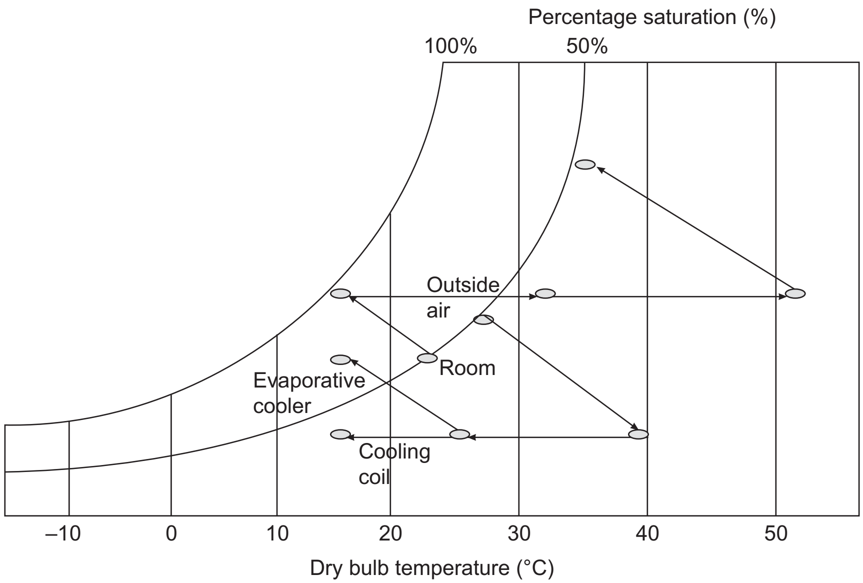

FIG 13.21 Desiccant system in cooling/dehumidification mode.

through the desiccant wheel on the supply side has a gradient approximately equal to that of a wintertime room ratio line of 0.6 on the psychrometric chart. The supply air stream then passes through the thermal wheel where it is sensibly cooled to, say, 23°C. The air then passes through a small DX or chilled water cooling coil where it is sensibly cooled to a supply condition of, say, 17°C and 7.3 g/kg moisture content. It should be noted that if humidity control is not required in the space, then the cooling coil may be replaced by an evaporative cooler with an adiabatic efficiency of approximately 85%, in which case air may be supplied to the room space at, say, 16.2°C and 10.2 g/kg moisture content.

On the return side, air from the room space at, for example, 22°C and 8.6 g/kg moisture content passes through an evaporative cooler so that it enters the thermal wheel at approximately 16.7°C and 10.8 g/kg moisture content. As the return air stream passes through the thermal wheel it is sensibly heated to approximately 33°C. The air stream is then heated up to approximately 55°C in order to regenerate the desiccant coil. It should be noted that in order to save energy, approximately 20% of the return air stream bypasses the regenerating coil and the desiccant wheel.

During the wintertime much of the heat for the supply air stream comes from recovered heat from the thermal wheel. Although the desiccant wheel could in theory be used as an additional heat exchanger, in practice it is not particularly effective due to its low rotational speed, and is therefore not normally used. Should further sensible heating be required this can be achieved either by locating a heating coil in the supply air stream after the thermal wheel, or by using radiators within the room space. In addition, an evaporative cooler on the supply side may be utilized to humidify the incoming air stream if so required.

It has been shown that the use of desiccant cooling can result in energy cost savings ranging from 14% to 50% depending on the application and cooling load [14]. Surprisingly, unlike conventional refrigeration systems, operating costs are at their lowest when desiccant cooling systems are operating under part load [14]. It is also worth noting that desiccant cooling systems are not well suited to applications in which low supply air temperatures are required. Desiccant cooling is best suited to those applications such as displacement ventilation, where supply air temperatures are close to the room air temperature. Although it is possible to make energy cost savings in 'all-air' applications, desiccant cooling systems are best applied to installations in which the bulk of the sensible cooling is performed by a water-based system, such as a chilled ceiling [14].

13.11.1 Solar Application of Desiccant Cooling

Being a heat-driven cycle, desiccant cooling affords an opportunity to utilize heat which might otherwise be wasted. It can therefore be coupled to solar collectors to produce a cooling system which, in theory, should be extremely environmentally friendly. However, the use of solar energy puts constraints on the application of desiccant cooling. For example, if the ratio of solar collectors to building floor area is 1:10, then the available heat (in a northern European application) to power the cycle will be in the region of 25–50 W/m^2, depending on the climate, type and orientation of the solar collectors [22]. Therefore, if this 'solar' heat is to be harnessed effectively, the desiccant cooling system must be applied in the correct fashion. The desiccant cooling cycle is an open cycle and as such it rejects moist air at a high temperature, which is unsuitable for recirculation. In fact, the greater the air volume flow rate supplied to the room space, the greater the fan power required and the heat energy consumed. Therefore, if desiccant cooling is used in an *all-air* application, the regeneration heat load is going to be very large, many times greater than the available solar energy. The bulk of the room's sensible cooling should therefore be carried out using a water-based system such as a chilled ceiling, with the desiccant AHU dehumidifying and 'tempering' the incoming fresh air. This strategy reduces the size of the AHU and its associated ductwork and enables the solar energy to make a significant contribution [22].

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