CHAPTER 11

Waste Heat Recovery

In many applications there is the potential for recovering heat energy that would otherwise go to waste. This chapter describes various waste heat recovery technologies and examines the theoretical principles behind each.

11.1 Introduction

In many applications it is possible to greatly reduce energy costs by employing some form of waste heat recovery device. However, before investing in such technology it is important to first consider some generic issues:

- Is there a suitable waste heat source? If the answer to this question is 'yes', it is important to establish that the source is capable of supplying a sufficient 'quantity' of heat, and that the heat is of a good enough 'quality' to promote good heat transfer.
- Is there a market or use for the recovered waste heat? It is important to have a use for any waste heat which may be recovered. In many applications there may be no demand for the heat that is available, with the result that a large quantity of heat energy is dumped. In other situations there may be a long time lag between waste heat production and the demand for heat. Waste heat therefore cannot be utilized unless some form of thermal storage is adopted.
- Will the insertion of a heat recovery device actually save primary energy or reduce energy costs? Often the insertion of a heat exchanger increases the resistance of

the fluid streams, resulting in fan or pump energy consumption rising. Heat energy is therefore replaced by electrical energy, which may be produced at an efficiency of less than 35%.
- Will any investment in heat recovery technology be economic? Heat recovery devices can be expensive to install. It is, therefore, essential that the economic payback period be determined prior to any investment being undertaken.

Although the questions above may appear obvious, it is not uncommon to find cases where poor planning and analysis at the design stage has resulted in an installation where the impact of the heat recovery device is either minimal, or is even increasing energy costs. Consider the case of a heat exchanger installed in a warm exhaust air stream from a building. The insertion of the device causes the resistance of the air streams to rise, resulting in greater fan energy consumption. If the unit price of electricity is four times that of gas, then in order to just break even, the heat exchanger must recover four times the increase in electrical energy consumption, arising from the increased system resistance. Also, there may well be long periods when the external air temperature is such that little or no heat can be recovered. If, however, the fans run continuously then the increased electrical energy is being expended for little or no return. Given this, it is not surprising that many so-called 'energy recovery' systems, while appearing to save energy, are in fact increasing both primary energy consumption and energy costs.

If a strategic decision is made to invest in some form of heat recovery device, then the next logical step is to select the most appropriate system. There are a wide variety of heat recovery technologies, which can be divided into the following broad categories:

- *Recuperative heat exchangers*: where the two fluids involved in the heat exchange are separated at all times by a solid barrier.
- *Run-around coils*: where an independent circulating fluid is used to transport heat between the hot and cold streams.
- *Regenerative heat exchangers*: where hot and cold fluids pass alternately across a matrix of material.
- *Heat pumps*: where a vapour compression cycle is used to transfer heat between the hot and cold streams.

In addition to these, there are a few lesser-used technologies, such as heat pipes, which are discussed in Chapter 5.

11.2 Recuperative Heat Exchangers

In a recuperative heat exchanger the two fluids involved in the heat transfer are separated at all times by a solid barrier. This means that the mechanisms which control the heat transfer are convection and conduction. The thermal resistance of a heat exchanger can therefore be expressed as follows:

$$R = \frac{1}{U} = \frac{1}{h_i} + R_w + \frac{1}{h_o} + F_i + F_o \tag{11.1}$$

where R is the thermal resistance of the heat exchanger ($m^2\,K/W$), R_w is the thermal resistance of the separating wall ($m^2\,K/W$), h_i and h_o are the heat transfer coefficient of internal and external surfaces ($W/m^2\,K$), F_i and F_o are the internal and external fouling factors, and U is the overall heat transfer coefficient (i.e. U value) ($W/m^2\,K$).

In short this can be written as:

$$\frac{1}{U_{\text{dirty}}} = \frac{1}{U_{\text{clean}}} + \text{Fouling factors} \qquad (11.2)$$

In practice heat exchangers are often oversized so that even when fouled their performance still meets design requirements. The degree of oversizing is achieved by incorporating fouling factors into the sizing calculation.

Recuperative heat exchangers are the most common type of equipment used for waste heat recovery. They can only be used in applications where the hot and cold streams can be brought into close proximity with each other. Although the precise form of a heat exchanger may change with its particular application, there are three forms which are widely used:

(a) *Shell and tube heat exchanger*: Shell and tube heat exchangers consist of a bundle of tubes inside a cylindrical shell through which two fluids flow, one through the tubes and the other through the shell (as shown in Figure 11.1). Heat is exchanged by conduction through the tube walls. Baffles are often used to direct fluid around the heat exchanger and also to provide structural support for the tubes.

(b) *Plate heat exchanger*: Plate heat exchangers consist of a large number of thin metal plates (usually stainless steel but sometimes titanium or nickel), which are clamped tightly together and sealed with gaskets (see Figure 11.2). The thin plates are profiled so that 'flow ways' are created between the plates when they are packed together, and a very large surface area is created across which heat transfer can take place. Ports located at the corners of the individual plate separate the 'hot' and 'cold' fluid flows and direct them to alternate passages so that no intermixing

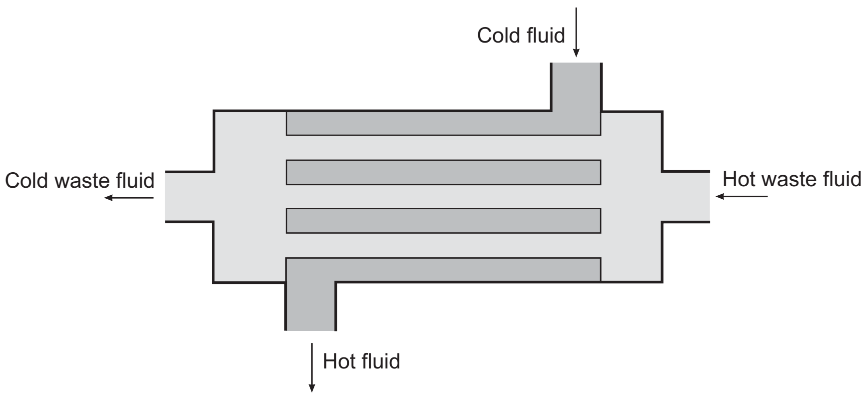

FIG 11.1 Shell and tube heat exchanger.

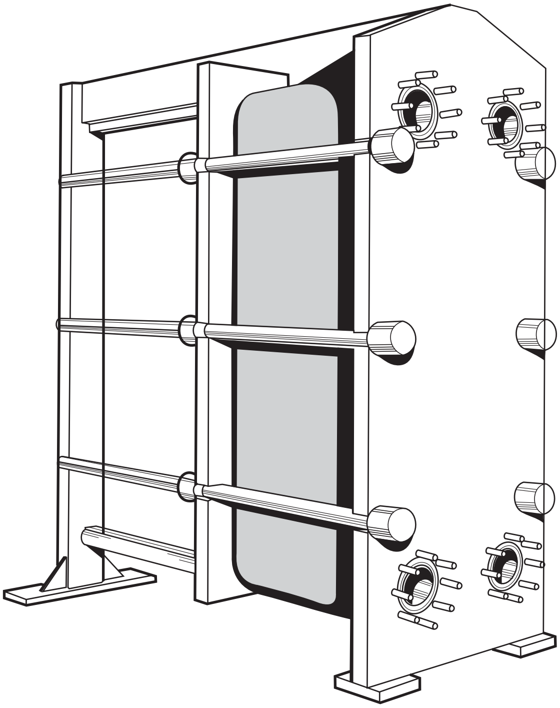

FIG 11.2 Plate heat exchanger.

of the fluids occurs. The whole exchanger experiences a counter-flow pattern. The maximum operating temperature is usually about 130°C if rubber sealing gaskets are fitted, but this can be extended to 200°C if compressed asbestos fibre seals are used [1]. Plate heat exchangers have become popular in recent years because they are extremely compact and can easily be expanded or contracted to accommodate future system modification.

(c) *Flat plate recuperator*: Flat plate recuperators consist of a series of metal (usually aluminium) plates separating 'hot' and 'cold' air or gas flows, sandwiched in a box-like structure (see Figure 11.3). The plates are sealed in order to prevent intermixing of the two fluid flows. They are often used in ducted air-conditioning installations to reclaim heat from the exhaust air stream, without any cross-contamination occurring.

11.3 Heat Exchanger Theory

The two most commonly used heat exchanger flow configurations are *counter flow* and *parallel flow*. These flow patterns are represented in Figures 11.4 and 11.5 respectively, along with their characteristic temperature profiles.

It should be noted that with the parallel flow configuration the 'hot' stream is always warmer than the 'cold' stream. With the counter-flow configuration it is possible for the

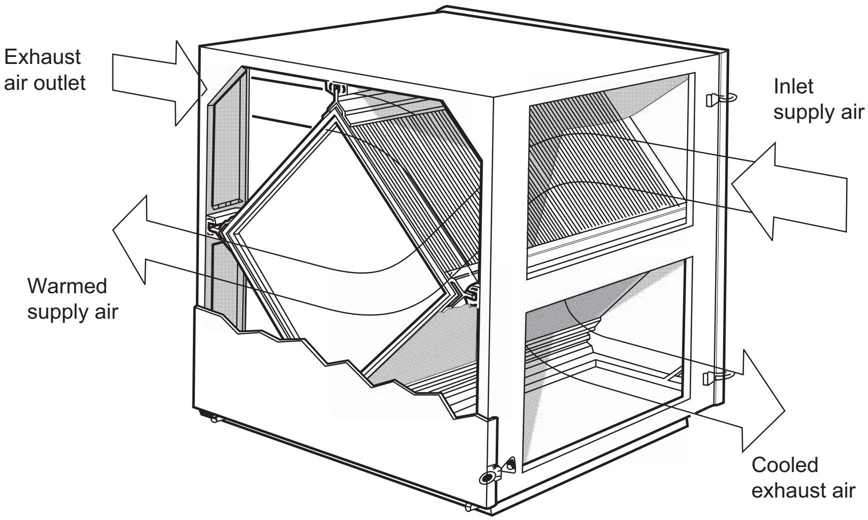

FIG 11.3 Flat plate recuperator.

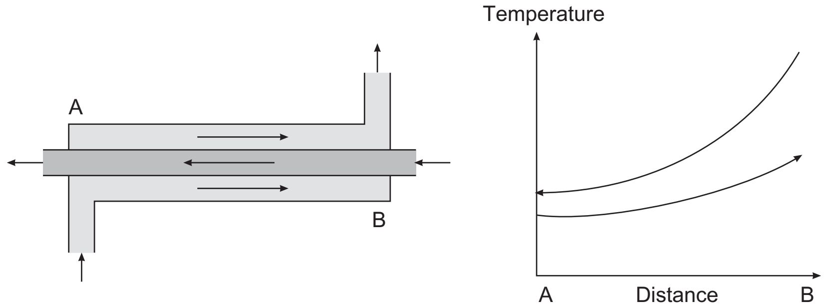

FIG 11.4 Counter-flow heat exchanger.

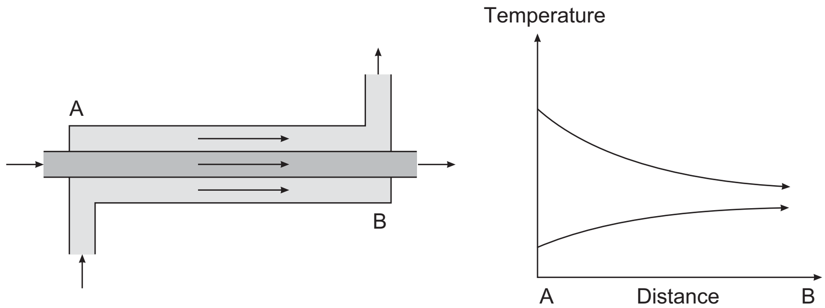

FIG 11.5 Parallel-flow heat exchanger.

outlet temperature of the cold fluid to be higher than the outlet temperature of the hot fluid.

The general equations which govern the heat transfer in recuperative heat exchangers are as follows:

$$Q = \dot{m}_h c_h (t_{h1} - t_{h2}) = \dot{m}_c c_c (t_{c1} - t_{c2}) \tag{11.3}$$

and

$$Q = UA_o(\text{LMTD})K \tag{11.4}$$

where Q is the rate of heat transfer (W), $\dot{m}_h$ is the mass flow rate of hot fluid (kg/s), $\dot{m}_c$ is the mass flow rate of cold fluid (kg/s), c_h is the specific heat of hot fluid (J/kg K), c_c is the specific heat of cold fluid (J/kg K), t_{h1} and t_{h2} are the inlet and outlet temperatures of hot fluid (°C), t_{c1} and t_{c2} are the outlet and inlet temperatures of cold fluid (°C), U is the overall heat transfer coefficient (i.e. U value) (W/m^2 K), A_o is the outside surface area of heat exchanger (m^2), LMTD is the logarithmic mean temperature difference (°C), and K is the constant which is dependent on the type of flow through the heat exchanger (e.g. $K = 1$ for counter flow and parallel flow, and is therefore often ignored).

The LMTD can be determined by:

$$\text{LMTD} = \frac{\Delta t_1 - \Delta t_2}{\ln(\Delta t_1/\Delta t_2)} \tag{11.5}$$

The following examples illustrate how the above equations can be used to design and analyse heat exchangers.

Example 11.1

A liquid waste stream has a flow rate of 3.5 kg/s and a temperature of 70°C, with a specific heat capacity of 4190 J/kg K. Heat recovered from the hot waste stream is used to preheat boiler make-up water. The flow rate of the make-up water is 2 kg/s, its temperature is 10°C and its specific heat capacity is 4190 J/kg K. The overall heat transfer coefficient of the heat exchanger is 800 W/m^2 K. If a make-up water exit temperature of 50°C is required, and assuming that there are no heat losses from the exchanger, determine:

(i) The heat transfer rate.
(ii) The exit temperature of the effluent.
(iii) The area of the heat exchanger required.

Solution

(i) Now:

$$\begin{aligned} Q &= \dot{m}_c c_c (t_{c1} - t_{c2}) \\ &= 2 \times 4190 \times (50 - 10) \\ &= 335{,}200 \text{ W} = 335.2 \text{ kW} \end{aligned}$$

(ii) Now

$$\dot{m}_h c_h (t_{h1} - t_{h2}) = \dot{m}_c c_c (t_{c1} - t_{c2})$$
$$3.5 \times 4190 \times (70 - t_{h2}) = 2 \times 4190 \times (50 - 10)$$
$$t_{h2} = 47.14°C$$

(iii) Now, because the water outlet temperature is above the outlet temperature of the effluent, a counter-flow heat exchanger is required:

$$\text{LMTD} = \frac{\Delta t_1 - \Delta t_2}{\ln(\Delta t_1/\Delta t_2)}$$
$$= \frac{((70 - 50) - (47.14 - 10))}{\ln((70 - 50)/(47.14 - 10))}$$
$$= 27.69°C$$

Now

$$Q = UA(\text{LMTD})$$

therefore

$$A = \frac{335,200}{800 \times 27.69} = 15.13 \text{ m}^2$$

Example 11.2

Consider the *counter-flow* heat exchanger shown in Figure 11.6. Given the data below, determine the overall heat transfer rate for the heat exchanger.

Data:

Length of heat exchanger = 2 m
Internal radius of heat exchanger surface = 10 mm
External radius of heat exchanger surface = 11 mm
Thermal conductivity of heat exchanger surface = 386 W/m K

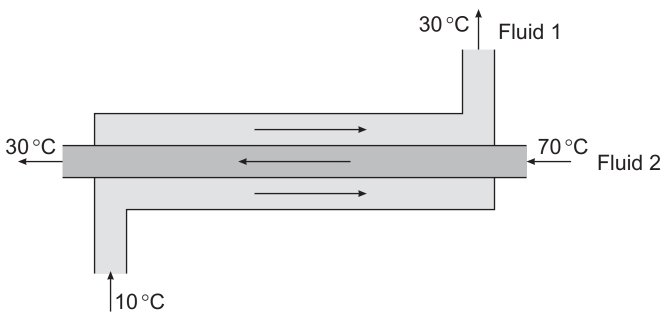

FIG 11.6 Heat exchanger.

Heat transfer coefficient of Fluid 1 = 50 W/m^2 K
Heat transfer coefficient of Fluid 2 = 90 W/m^2 K

Solution

By combining eqns (10.26) and (10.28) it can be shown that the total thermal resistance, R_t, of the heat exchanger is:

$$R_t = \frac{1}{h \cdot A_1} \times \frac{\ln(r_2/r_1)}{2\pi k \cdot l} \times \frac{1}{h \cdot A_2}$$

where k is the thermal conductivity of the pipe wall (W/m K), r_1 is the internal radius of the pipe (m), r_2 is the external radius of the pipe (m), l is the length of the pipe (m), h is the heat transfer coefficient (W/m^2 K), and A_1 and A_2 are the external and internal surface areas (m^2).

And using eqn (10.30) the total heat transfer rate can be expressed as:

$$Q = \frac{\Delta t}{R_t} \text{(W)}$$

Now:

$$A_1 = 2\pi \times 0.011 \times 2 = 0.138 \text{ m}^2$$

and

$$A_2 = 2\pi \times 0.01 \times 2 = 0.126 \text{ m}^2$$

$$R_t = \frac{1}{90 \times 0.126} \times \frac{\ln(0.011/0.01)}{2\pi \times 386 \times 2} \times \frac{1}{50 \times 0.138}$$

therefore

$$R_t = 0.233 \text{ W/K}$$

and

$$\text{LMTD} = \frac{(70 - 30) - (30 - 10)}{\ln[(70 - 30)/(30 - 10)]} = 28.85°\text{C}$$

therefore

$$Q = \frac{28.85}{0.233} = 123.8 \text{ W}$$

11.3.1 Number of Transfer Units (NTU) Concept

In some situations only the inlet temperatures and the flow rates of the *hot* and *cold* streams are known. Under these circumstances the use of the LMTD method results in a long and complex mathematical solution. To simplify such calculations the NTU method was developed [2,3].

NTU is defined as the ratio of the temperature change of one of the fluids divided by the mean driving force between the fluids, and can be expressed as:

For the hot fluid:

$$\mathrm{NTU_h} = \frac{(t_{h1} - t_{h2})}{(\mathrm{LMTD})K} = \frac{UA_o}{(\dot{m}c)_h} \tag{11.6}$$

For the cold fluid:

$$\mathrm{NTU_c} = \frac{(t_{c1} - t_{c2})}{(\mathrm{LMTD})K} = \frac{UA}{(\dot{m}c)_c} \tag{11.7}$$

(NB: For counter-flow and parallel-flow heat exchangers the K term can be ignored) Equations (11.6) and (11.7) are more commonly simplified to:

$$\mathrm{NTU} = \frac{UA_o}{(\dot{m}c)_{min}} \tag{11.8}$$

where $(\dot{m}c)_{min}$ is the minimum thermal capacity (kW/K).

The ratio of the thermal capacities, R, is defined as:

$$R = \frac{(\dot{m}c)_{min}}{(\dot{m}c)_{max}} \tag{11.9}$$

If both fluids in the heat exchanger have the same thermal capacity then $R = 1$. At the other extreme when one of the fluids has an infinite thermal capacity, as in the case of an evaporating vapour, then $R = 0$.

Another useful concept is the *effectiveness*, E, of a heat exchanger. Effectiveness can be defined as the actual heat transfer divided by the maximum possible heat transfer across the heat exchanger, and can be expressed as:

$$E = \frac{Q}{Q_{max}} = \frac{Q}{(\dot{m}c)_{min}(t_{h\,max} - t_{c\,min})} \tag{11.10}$$

It is possible to derive the relationship between E, NTU and R for a variety of heat exchanger applications. The mathematical expressions for some of the more common applications are given below:

(i) Parallel flow:

$$E = \frac{1 - e^{[-\mathrm{NTU}\,(1+R)]}}{1 + R} \tag{11.11}$$

(ii) Counter flow:

$$E = \frac{1 - e^{[-NTU\,(1-R)]}}{1 - Re^{[-NTU\,(1-R)]}} \tag{11.12}$$

If $R = 1$ then this expression simplifies to:

$$E = \frac{NTU}{1 + NTU} \tag{11.13}$$

(iii) Heat exchanger with condensing vapour of boiling liquid on one side:

$$E = 1 - e^{[-NTU]} \tag{11.14}$$

The NTU method for heat exchanger analysis is illustrated in Example 11.3.

Example 11.3

A contaminated water stream from a factory building has a temperature of 80°C, a flow rate of 6 kg/s and a specific heat capacity of 4.19 kJ/kg K. The incoming water supply to the manufacturing process is at 10°C and has a flow rate of 7 kg/s and a specific heat capacity of 4.19 kJ/kg K. It is proposed to install a counter-flow heat exchanger to recover the waste heat. If the heat exchanger has an overall area of 30 m^2 and an overall heat transfer coefficient of 800 W/m^2 K (assuming that there are no heat losses from the heat exchanger), determine:

(i) The effectiveness of the heat exchanger.
(ii) The heat transfer rate.
(iii) The exit temperature of the incoming water stream leaving the heat exchanger.

Solution

Now

$$(\dot{m}c)_{min} = 6 \times 4.19 = 25.14 \text{ kW/K}$$

and

$$(\dot{m}c)_{max} = 7 \times 4.19 = 29.33 \text{ kW/K}$$

therefore

$$R = \frac{25.14}{29.33} = 0.857$$

and

$$NTU = \frac{30 \times 800}{25.14 \times 1000} = 0.955$$

(i) Therefore:

$$E = \frac{1 - e^{[-0.955(1-0.857)]}}{1 - 0.857\,e^{[-0.955(1-0.857)]}}$$

$$E = 0.506$$

(ii) Now:

$$E = \frac{Q}{(\dot{m}c)_{\min}(t_{\mathrm{h\,max}} - t_{\mathrm{c\,min}})}$$

therefore:

$$Q = 0.506 \times 25.14 \times [80 - 10] = 890.46 \text{ kW}$$

(iii) Therefore:

$$890.46 = 29.33 \times [t_{\mathrm{off}} - 10]$$

therefore:

$$t_{\mathrm{off}} = 40.4°\mathrm{C}$$

11.4 Run-Around Coils

When two recuperative heat exchangers are linked together by a third fluid which transports heat between them, the system is known as a run-around coil. Run-around coils are often employed to recover waste heat from exhaust air streams and to preheat incoming supply air, thus avoiding the risk of cross-contamination between the two air streams. Such a system is shown in Figure 11.7. Run-around coils usually employ a glycol/water mixture as the working fluid which avoids the risk of freezing during the winter.

Run-around coils have the advantage that they can be used in applications where the two fluid streams are physically too far apart to use a recuperative heat exchanger.

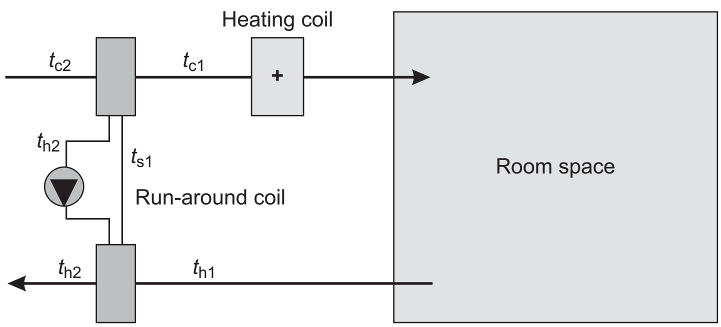

FIG 11.7 Run-around coil system.

Whilst this feature is usually considered advantageous it does result in increased energy consumption since a pump is introduced into the system, and may also result in heat loss from the secondary fluid. This makes it important to insulate the pipework circuit, otherwise the overall effectiveness of the system will become unacceptably low. Despite these drawbacks, when compared with many other methods of recovering waste heat, run-around coils are relatively inexpensive to install since they utilize standard air/water heating coils.

In the case of the system shown in Figure 11.7 the thermal capacity ($\dot{m}c$) of the *cold* fluid is equal to that of the *hot* fluid since the two heat exchangers are identical. Therefore:

$$(\dot{m}c)_h = (\dot{m}c)_c = (\dot{m}c)_s \qquad (11.15)$$

where $(\dot{m}c)_s$ is the thermal capacity of the secondary fluid (kW/K).

Consequently, it can be shown that:

$$t_{s1} = \frac{(t_{h1} + t_{c1})}{2}$$

and

$$t_{s2} = \frac{(t_{h2} + t_{c2})}{2}$$

where t_{s1} and t_{s2} are the flow and return temperatures of the secondary fluid (°C), t_{h1} and t_{h2} are the temperatures of the hot fluid stream before and after heat exchanger (°C), and t_{c2} and t_{c1} are the temperatures of the cold fluid stream before and after heat exchanger (°C).

Also, the overall heat transfer can be defined by

$$Q = (UA)_o\ (t_{h1} - t_{c1})$$

and

$$\begin{aligned} Q &= (UA)_h\ (t_{h1} - t_{s1}) \\ &= (UA)_h \left(t_{h1} - \frac{(t_{h1} + t_{c1})}{2} \right) \end{aligned}$$

where $(UA)_o$ is the product of U and A for the whole run-around coil (W/K), and $(UA)_h$ is the product of U and A for the heat exchanger in the hot stream (W/K).

Therefore:

$$(UA)_o\ (t_{h1} - t_{c1}) = (UA)_h\ \frac{(t_{h1} - t_{c1})}{2}$$

so

$$Q = \frac{(UA)_h\ (t_{h1} - t_{c1})}{2}$$

and since

$$Q = (\dot{m}c)_c \times (t_{c1} - t_{c2})$$

it can be shown that

$$Q = \frac{(UA)_h\ (t_{h1} - t_{c2})}{2 + [(UA)_h/(\dot{m}c)_c]} \tag{11.16}$$

Example 11.4

A run-around coil is applied to a heating and ventilation system as shown in Figure 11.7. Air is supplied to the room space at 28°C and leaves at 20°C. The outside air temperature is −1°C. The supply air to the space has a mass flow rate of 3 kg/s and a mean specific heat capacity of 1.012 kJ/kg K. The specific heat of the secondary fluid is 3.6 kJ/kg K, and:

$$(UA)_c = (UA)_h = 5\ \text{kW/K}$$

Given this information, determine:

(i) The required mass flow rate of the secondary fluid.
(ii) The temperature of the air entering the supply air heating coil.
(iii) The percentage energy saving achieved by using the run-around coil.

Solution

(i) From eqn (11.15) it can be seen that:

$$(\dot{m}c)_c = (\dot{m}c)_s$$

therefore:

$$\dot{m}_s = \frac{3 \times 1.012}{3.6} = 0.843\ \text{kg/s}$$

(ii) Now:

$$Q = \frac{(UA)_h\ (t_{h1} - t_{c2})}{2 + [(UA)_h/(\dot{m}c)_c]}$$

therefore:

$$Q = \frac{5 \times [20 - (-1)]}{2 + [5/(3 \times 1.012)]} = 28.79\ \text{kW}$$

and since

$$t_{c1} = t_{c2} + \frac{Q}{(\dot{m}c)_c}$$

therefore

$$t_{c1} = -1 + \frac{28.79}{(3 \times 1.012)} = 8.48°\text{C}$$

(iii) With run-around coil

$$Q = 3 \times 1.012 \times [28 - 8.48] = 59.263 \text{ kW}$$

Without run-around coil

$$Q = 3 \times 1.012 \times [28 - (-1)] = 88.044 \text{ kW}$$

therefore

$$\text{Percentage saving} = \frac{(88.044 - 59.263)}{88.044} \times 100 = 32.7\%$$

While it is relatively simple to derive an expression for the heat transfer of a run-around coil when the thermal capacities of the fluids are equal, it becomes much more complex when the thermal capacities of the two fluids are different, and the heat exchangers are also different. However, this problem can be overcome by using the NTU method.

It can be shown that for a run-around coil

$$\frac{1}{(UA)_o} = \frac{1}{(UA)_h} + \frac{1}{(UA)_c} \tag{11.17}$$

and from eqn (11.8)

$$\text{NTU} = \frac{UA_o}{(\dot{m}c)_{min}}$$

therefore

$$\text{NTU} = \frac{UA_h \times UA_c}{(\dot{m}c)_{min}(UA_h + UA_c)} \tag{11.18}$$

Example 11.5 illustrates how the NTU method can be applied to a run-around coil problem.

Example 11.5

It is intended that a run-around coil be installed to recover waste heat from a flue gas stream at 250°C, and to preheat a water stream at 10°C. The flue gas has a mass flow rate of 4 kg/s and that of the water is 2 kg/s. The individual heat exchangers used in the system are of a counter flow type. Given the following data determine:

(i) The overall effectiveness of the run-around coil.
(ii) The exit temperature of the water stream.

Data:

Specific heat capacity of flue gas = 1.2 kJ/kg K
Specific heat capacity of water = 4.19 kJ/kg K
UA for the flue gas heat exchanger = 5 kW/K
UA for the water heat exchanger = 18 kW/K

Solution

(i) Now

$$(\dot{m}c)_{min} = 4 \times 1.2 = 4.8 \text{ kW/K}$$

and

$$(\dot{m}c)_{max} = 2 \times 4.19 = 8.38 \text{ kW/K}$$

therefore

$$R = \frac{4.8}{8.38} = 0.573$$

and

$$\text{NTU} = \frac{5 \times 18}{4.8 \times (5+18)} = 0.815$$

and from eqn (11.13)

$$E = \frac{1 - e^{[-0.815(1-0.573)]}}{1 - 0.573e^{[-0.815(1-0.573)]}} = 0.494$$

(ii) Now

$$E = \frac{Q}{(\dot{m}c)_{min}(t_{hmax} - t_{cmin})}$$

therefore

$$Q = 0.494 \times 4.8 \times [250 - 10] = 569.1 \text{ kW}$$

therefore

$$569.1 = 8.38 \times [t_{off} - 10]$$

therefore

$$t_{off} = 77.9°C$$

11.5 Regenerative Heat Exchangers

In a regenerative heat exchanger a matrix of material is alternately passed from a hot fluid to a cold fluid, so that heat is transferred between the two in a cyclical process. The most commonly used type of regenerative heat exchanger is the thermal wheel, which has a matrix of material mounted on a wheel, which slowly rotates at approximately 10 revolutions per minute, through hot and cold fluid streams (as shown in Figure 11.8). The major advantage of a thermal wheel is that there is a large surface area to volume ratio resulting in a relatively low cost per unit surface area.

The matrix material in a thermal wheel is usually an open-structured metal, such as knitted stainless steel or aluminium wire, or corrugated sheet aluminium or steel [1]. For use at higher temperatures honeycomb ceramic materials are used. Although thermal wheels are usually employed solely to recover sensible heat, it is possible to reclaim the enthalpy of vaporization of the moisture in the 'hot' stream passing through a thermal wheel. This is achieved by coating a non-metallic matrix with a hygroscopic or desiccant material such as lithium chloride [1].

Thermal wheels do have the major disadvantage that there is the possibility of cross-contamination between the air streams. This can be reduced considerably by ensuring that the cleaner of the two fluids is maintained at the highest pressure, and by using a purging device. Most thermal wheels incorporate a purge unit which allows a small

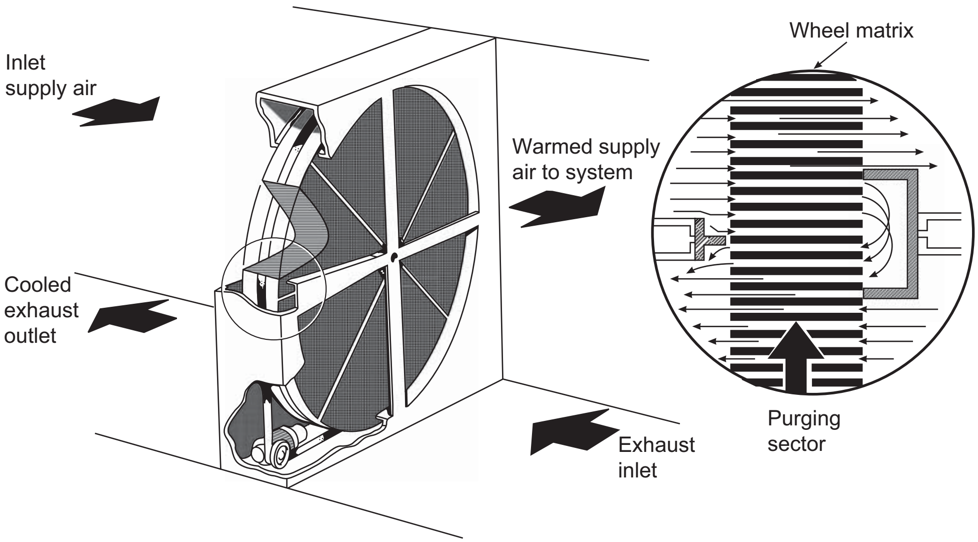

FIG 11.8 Thermal wheel.

proportion of the supply air to flush the contaminants from the wheel, thus keeping cross-contamination to a minimum (e.g. less than 0.04%) [1].

Thermal wheels are often used to recover heat from room ventilation systems such as that shown in Figure 11.9. In this type of application the thermal efficiency, η_t, can be defined by:

$$\eta_t = \frac{t_2 + t_1}{t_3 + t_1} \tag{11.19}$$

Similarly the overall (total energy) efficiency, η_x, can be expressed as:

$$\eta_x = \frac{h_2 + h_1}{h_3 + h_1} \tag{11.20}$$

where t_1, t_2 and t_3 are the air temperatures (°C), and h_1, h_2 and h_3 are the air enthalpies (°C).

In a similar manner to a recuperative heat exchanger it can be shown that for a thermal wheel the relationship between (UA) and (hA) is:

$$\frac{1}{(UA)_o} = \frac{1}{(hA)_h} + \frac{1}{(hA)_c} \tag{11.21}$$

where $(UA)_o$ is the product of overall heat transfer coefficient and surface area of matrix, $(hA)_h$ is the product of heat transfer coefficient between the hot fluid and surface area of matrix, and $(hA)_c$ is the product of heat transfer coefficient between the cold fluid and surface area of matrix. Since the matrix area is constant, therefore:

$$U = \frac{h}{2} \tag{11.22}$$

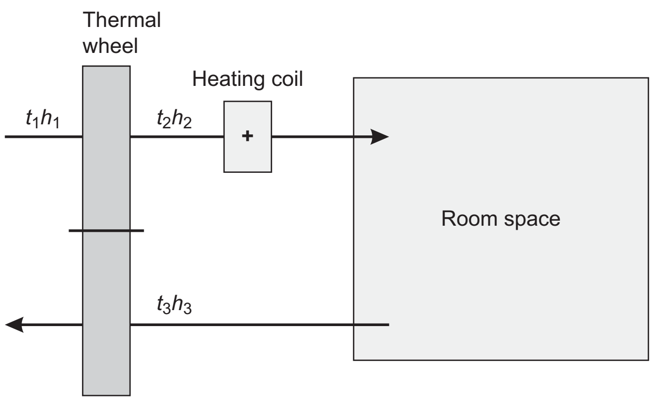

FIG 11.9 Thermal wheel application.

As with recuperative heat exchangers and run-around coils it is possible to use the NTU method to simplify the analysis of thermal wheels. Kays and London [4] derived the following empirical formula to describe the effectiveness of thermal wheels:

$$E = E_c \times \left(1 - \frac{1}{9[(\dot{m}c)_M/(\dot{m}c)_{min}]^{1.93}}\right) \tag{11.23}$$

where

$$(\dot{m}c)_M = N \cdot M \cdot c_M \tag{11.24}$$

N is the wheel revolutions per second, M is the mass of the matrix (kg), c_M is the specific heat capacity of matrix material (kJ/kg K), and

$$E_c = \frac{1 - e^{[-NTU(1-R)]}}{1 - Re^{[-NTU(1-R)]}} \quad \text{where} \quad R = (\dot{m}c)_{min}/(\dot{m}c)_{max}$$

or

$$E_c = \frac{NTU}{1 + NTU} \quad \text{when} \quad R = 1$$

Example 11.6

The exhaust air from a factory building is at a temperature of 35°C and has a flow rate of 6 kg/s and a specific heat capacity of 1.025 kJ/kg K. The incoming fresh air to the building is at −1°C and has a flow rate of 7 kg/s and a specific heat capacity of 1.025 kJ/kg K. It is proposed to insert a thermal wheel between the air streams to recover the sensible waste heat. Given the following information, determine:

(i) The effectiveness of the thermal wheel.
(ii) The heat transfer rate.
(iii) The exit temperature of the fresh air leaving the thermal wheel.
(iv) The exit temperature of the fresh air leaving the thermal wheel if its rotational speed is doubled.

Data:

Wheel diameter = 1.2 m
Wheel depth = 0.4 m
Mass of wheel = 140 kg
Surface area to volume ratio = 2500 m^2/m^3
Specific heat of matrix material = 1.3 kJ/kg K
Wheel speed = 8 rev/min
Heat transfer coefficient for each air stream = 35 W/m^2 K

Solution

$$\text{Face area of wheel} = \frac{\pi \times 1.2^2}{4} = 1.12 \text{ m}^2$$
$$\text{Volume of wheel} = 1.13 \times 0.4 = 0.452 \text{ m}^3$$

(i) Now

$$\text{A} = 0.452 - 2500 = 1130 \text{ m}^2$$
$$(\dot{m}c)_{\text{min}} = 6 \times 1.025 = 6.15 \text{ kW/K}$$
$$(\dot{m}c)_{\text{max}} = 7 \times 1.025 = 7.175 \text{ kW/K}$$

therefore

$$R = \frac{6.15}{7.175} = 0.857$$

and from eqn (11.21)

$$U = \frac{h}{2} = \frac{35}{2} = 17.5 \text{ W/m}^2\text{K}$$

therefore

$$\text{NTU} = \frac{UA_{\text{o}}}{(\dot{m}c)_{\text{min}}} = \frac{1130 \times 17.5}{6.15 \times 1000} = 3.215$$

therefore

$$E_{\text{c}} = \frac{1 - \text{e}^{[-3.215\,(1-0.857)]}}{1 - 0.857\,\text{e}^{[-3.215\,(1-0.857)]}} = 0.803$$

and

$$(\dot{m}c)_{\text{M}} = N \cdot M \cdot c_{\text{M}} = \frac{8}{60} \times 140 \times 1.3 = 24.27 \text{ kW/K}$$

therefore

$$E = 0.803 \times \left(1 - \frac{1}{9[24.27/6.15]^{1.93}}\right)$$
$$= 0.797$$

(ii) Now

$$E = \frac{Q}{(\dot{m}c)_{\text{min}}\,(t_{\text{h max}} - t_{\text{c min}})}$$

therefore

$$Q = 0.797 \times 6.15 \times [35 - (-1)] = 176.46 \text{ kW}$$

(iii) Therefore

$$176.46 = 7.175 \times [t_{\text{off}} - (-1)]$$

therefore

$$t_{\text{off}} = 23.6°\text{C}$$

(iv) If $N = 2 \times 8 = 16$ rev/min, then

$$(\dot{m}c)_{\text{M}} = N \cdot M \cdot C_{\text{M}} = \frac{16}{60} \times 140 \times 1.3 = 48.53 \text{ kW/K}$$

therefore

$$\begin{aligned} E &= 0.803 \times \left(1 - \frac{1}{9[48.53/6.15]^{1.93}}\right) \\ &= 0.801 \end{aligned}$$

therefore

$$Q = 0.801 \times 6.15 \times [35 - (-1)] = 177.43 \text{ kW}$$

therefore

$$177.34 = 7.175 \times [t_{\text{off}} - (-1)]$$

therefore

$$t_{\text{off}} = 23.7°\text{C}$$

From this it can be seen that there is very little benefit to be gained from doubling the rotational speed of the thermal wheel.

11.6 Heat Pumps

A heat pump is essentially a vapour compression refrigeration machine which takes heat from a low temperature source such as air or water and upgrades it to be used at a higher temperature. Unlike a conventional refrigeration machine, the heat produced at the condenser is utilized and not wasted to the atmosphere. Figure 11.10 shows a simple vapour compression heat pump, together with the relevant pressure/enthalpy diagram.

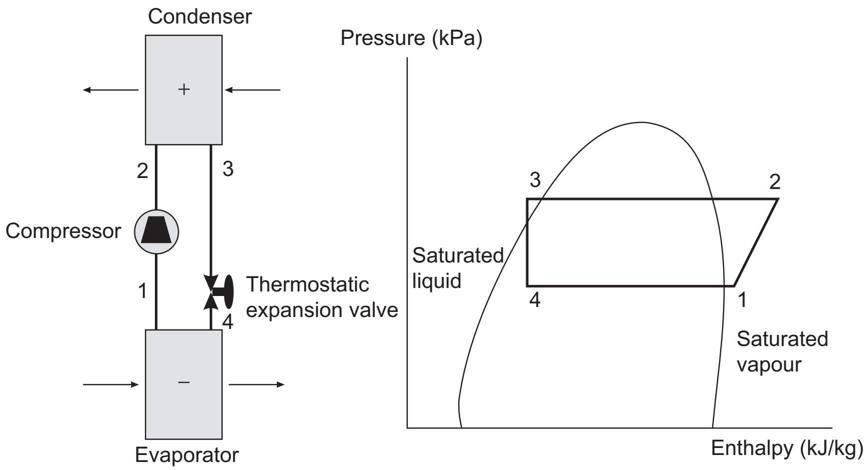

FIG 11.10 Vapour compression heat pump.

The performance of the vapour compression refrigeration cycle is quantified by the *coefficient of performance* (COP), which can be expressed as:

for a refrigeration machine:

$$COP_{ref} = \frac{\text{useful refrigeration output}}{\text{net work input}}$$

for a heat pump:

$$COP_{hp} = \frac{\text{useful heat rejected from cycle}}{\text{net work input}}$$

The COP of the vapour compression cycle is usually expressed in terms of a ratio of enthalpy differences; hence the COP of a refrigeration machine can be expressed as follows (referring to Figure 11.10):

$$COP_{ref} = \frac{h_1 - h_4}{h_2 - h_1} \tag{11.25}$$

where

$h =$ specific enthalpy of refrigerant (kJ/kg)

So, for a heat pump:

$$COP_{hp} = \frac{h_2 - h_3}{h_2 - h_1} \tag{11.26}$$

From this it can be shown that:

$$\mathrm{COP_{hp}} = \mathrm{COP_{ref}} + 1 \tag{11.27}$$

For an ideal heat pump the maximum possible COP is given by the Carnot cycle expression:

$$\mathrm{COP_{hp}} = \frac{T_c}{T_c - T_e} \tag{11.28}$$

where T_e is the evaporating absolute temperature (K), and T_c is the condensing absolute temperature (K).

In practice the Carnot COP shown above can never be achieved, but the Carnot equation shows that the greater the difference between T_c and T_e the lower the COP of the heat pump. Heat pumps are therefore well suited to applications where the evaporating and condensing temperatures are close together, which is the case when recovering heat from exhaust air in heating and air-conditioning applications. As a result, heat pumps are often used in air-conditioning applications. They are also popular in applications where there is a need for both dehumidification and heating, such as in warehouses where the occurrence of a high humidity may cause condensation problems and result in the destruction of valuable stock.

Swimming pool buildings are particularly well suited to the application of heat pumps. In swimming pools the air leaving the pool hall is very humid and contains large amounts of latent heat bound up in the water vapour. Heat pumps are particularly well suited to recovering the enthalpy of vaporization from the moisture in the exhaust air. A typical example of the heat pump used in combination with a flat plate heat recuperator is shown in Figure 11.11. In this application sensible heat is taken from the swimming pool exhaust air by the flat plate recuperator and used to preheat the supply air stream. The evaporator of the heat pump then dehumidifies the exhaust air stream and recovers the latent heat bound up in the water vapour. The heat pump then rejects this heat (plus the 'work' input by the compressor) through the condenser, and thus heats the supply air to the pool.

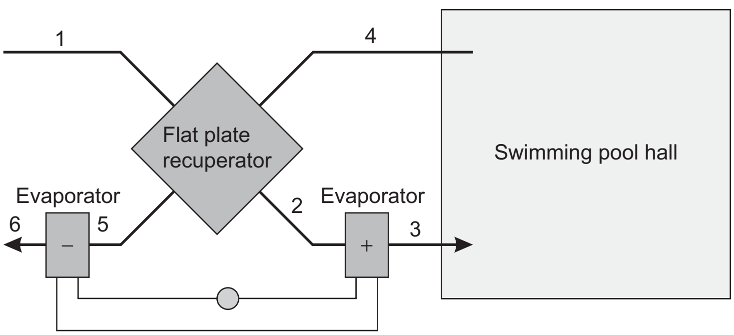

FIG 11.11 Heat recovery system for a swimming pool building.

Example 11.7

The heat pump, shown in Figure 11.11, operates on refrigerant HCFC 22. Given the following data, calculate:

(i) The COP of the heat pump.
(ii) The electrical energy consumed for each kW of heat produced.

Data:

Condensing temperature = 50°C
Evaporating temperature = 10°C
Vapour temperature (leaving compressor) = 80°C
Liquid temperature (leaving condenser) = 40°C
Combined electrical and mechanical efficiency of motor = 90%

Solution

Using a pressure enthalpy chart (see Appendix 2), or by using thermodynamic tables for HCFC 22, it is possible to plot the refrigeration process as follows:

(i) Now

$$\begin{aligned} COP_{hp} &= \frac{h_2 - h_3}{h_2 - h_1} \\ &= \frac{346 - 150}{346 - 315} = 6.323 \end{aligned}$$

(ii) Electricity consumption per kW of heat produced $= \dfrac{1}{6.323 \times 0.9} = 0.176$ kW

Example 11.8

For the heat pump installation shown in Figure 11.11, calculate:

(i) The heat output of the heat pump.
(ii) The mass flow rate of refrigerant required in the heat pump circuit.
(iii) The power input required to the electric motor.
(iv) The specific enthalpy of the air leaving the evaporator coil.

Data:

The mass flow rate of supply air = 6 kg/s
Condition of air leaving pool hall = 29°C and 70% saturation
Temperature of air supplied to pool hall = 34°C
Outside air condition = −3°C and 100% saturation
Effectiveness of flat plate recuperator = 0.7

Solution

Consider first the fresh outside air entering the system and passing through the flat plate recuperator. It enters the system at −3°C and 100% saturation; from a psychrometric chart (see Appendix 3) or from psychrometric tables, it can be determined that the moisture content of the incoming air stream is 0.0029 kg/kg (dry air) and its specific enthalpy is 4.2 kJ/kg.

Now

$$\text{The effectiveness of a flat plate recuperator} = \frac{\text{Heat transferred}}{\text{Max. theoretical heat transferred}}$$

Therefore, for the supply air stream, if the maximum theoretical heat transfer occurred, then it would be heated from −1°C to 29°C at a constant moisture content of 0.0029 kg/kg. From a psychrometric chart or tables, the specific enthalpy of air at 29°C and 0.0029 kg/kg is 36.6 kJ/kg.

Therefore

The maximum theoretical heat transfer = 36.6 − 4.2.

Therefore

$$\text{Effectiveness} = \frac{\text{Heat transferred}}{(37.2 - 4.2)}$$

therefore

$$\text{Heat transferred } (h_2 - h_1) = 0.7\ (36.6 - 4.2) = 22.68 \text{ kJ/kg}$$

therefore

$$h_2 - 4.2 = 22.68 \text{ kJ/kg}$$
$$h_2 = 22.68 + 4.2 = 26.88 \text{ kJ/kg}$$

At a moisture content of 0.0029 kg/kg, h_2 equates to an air temperature of 19.3°C. The heat pump condenser therefore has to raise the supply air temperature from 19.3°C to 34°C, at which temperature the specific enthalpy is 41.6 kJ/kg.

(i) Therefore

$$Q_{cond} = \dot{m}_{air} \times (h_3 - h_2)$$
$$= 6 \times (41.6 - 26.88) = 88.32 \text{ kW}$$

(ii) From Example 11.7 it can be seen that for the condenser

$$Q_{cond} = \dot{m}_{ref} \times (346 - 150)$$

therefore

$$\dot{m}_{ref} = \frac{88.32}{346 - 150} = 0.451 \text{ kg/s}$$

(iii) Therefore

$$\text{Electric power input to compressor} = \frac{0.451\,(346 - 315)}{0.9} = 15.534\ \text{kW}$$

(iv) Considering now the exhaust air stream through the flat plate recuperator

$$Q_{fpr} = \dot{m}_{air} \times (h_2 - h_1) = \dot{m}_{air} \times (h_4 - h_5)$$

therefore

$$Q_{fpr} = 6 \times 22.68 = 136.08\ \text{kW}$$

Now, the air leaving the pool hall has a moisture content of 0.018 kg/kg and specific enthalpy (h_4) of 75.1 kJ/kg. Therefore:

$$h_5 = h_4 - \frac{Q_{fpr}}{\dot{m}_{air}}$$

therefore

$$h_5 = 75.1 - \frac{136.08}{6} = 52.42\ \text{kJ/kg}$$

and from Example 11.7:

$$Q_{evap} = 0.451 \times (315 - 150) = 74.415\ \text{kW}$$

and

$$Q_{evap} = \dot{m}_{air} \times (h_5 - h_6)$$

therefore

$$h_6 = 52.42 - \frac{74.414}{6} = 40.02\ \text{kJ/kg}$$

Many manufacturers produce machines which have the dual ability to act as both a refrigeration machine and a heat pump. These machines have twin condensers; an air cooled one for normal operation and a water cooled one for the heat pump mode. They are often installed in buildings and act as air-conditioning chillers. When operating in the heat pump mode the waste heat from the condenser is recovered and used to produce the domestic hot water for the building. This at first sight would appear to be a classic energy conservation measure. However, such 'energy-saving' measures should be treated with caution since in order to produce the domestic hot water it may be necessary to raise the condensing pressure considerably, with the result that the COP may be significantly reduced. When it is also considered that the unit price of electricity is usually 3 to 4 times that of gas, then the adoption of such a dual purpose machine may not be quite as advantageous as it appeared originally.

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