*Building Serv. Eng. Res. Technol.* 9(3) 99–104 (1988) Printed in Great Britain

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**Summary** The economic viability of stand-alone heat pumps for space heating is limited by the achievable coefficient of performance. By combining the heat pump with a plate recuperator in full fresh air systems the average heating season efficiency can be increased to the equivalent of a COP of 6. The system works both as a heating and cooling system and will provide over 90% of annual energy requirement, making the use of central heating plant for back-up heating uneconomical. The resulting capital cost saving can be used to offset in part or in full the extra capital cost of the heat pump equipment.

# Plate recuperators used with air/air heat pumps in building ventilation systems

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Received 3 August 1987, in final form 24 March 1988†

## 1 Economic limitations of stand-alone heat pumps

The application of air-to-air heat pumps has increased significantly since the first energy crisis in the early 1970s. They have mainly been used in commercial buildings and have generally been reverse cycle machines, with the emphasis on air cooling, and the heating cycle regarded as a useful by-product. For heating alone the heat pump is, however, severely limited in its economic application, due to the low coefficient of performance that can be achieved. Certainly, for ambient source heat pumps the average COP during the heating season is rarely over 2.5 and often lower. Even for exhaust air heat pumps, where a constant supply of high grade (20°C) heat is available, the average COP is not normally higher than 3.5–4.0.

The heat pump has to compete with heating systems using conventional fuels, such as oil, coal and gas. Traditionally gas has been the most widely available, convenient and most economical of the conventional fuels. At a price of 37p therm$^{-1}$ and a point-of-use efficiency of 70% the nett cost of gas is approximately 1.8p kWh$^{-1}$. For electrically driven heat pumps to compete successfully with gas the price of electricity divided by the average coefficient of performance must be lower than 1.8p kWh$^{-1}$. With a price for electricity of approximately 5.0p kWh$^{-1}$ the average COP must therefore be at least 5.0/1.8 = 2.8.

The difference in nett energy cost between using conventional fuels and using a heat pump must pay for the extra capital cost often associated with heat pumps.

A COP of the order of 5.2 as in Table 1 can clearly not be achieved by a standard heat pump. It is, however, possible to obtain average efficiencies even higher than this by combining the heat pump with another heat recovery device: the plate recuperator. This paper describes the way such a system is built up, its performance and its economic benefits.

**Table 1** Example calculation of economic heat pump coefficient of performance

| | |
|---|---|
| Average heat output required | 50 kW |
| Annual running time | 2000 h |
| Annual consumption = 50 × 2000 | = 100 000 kWh |
| Extra capital cost for heat pump | £2500 |
| Simple pay-back required | 3 years |
| Cost saving required = 2500/3 | = £833 year$^{-1}$ |
| Required COP: 100 000 × 0.018 – 100 000 × 0.05/COP = 833 | |
| COP = 5.2 | |

## 2 Plate recuperator

The plate recuperator is probably the most commonly used heat recovery device in air-to-air systems. Because there are no moving parts reliability and durability are very good and systems used for normal environmental conditions are comparatively inexpensive. Under normal environmental conditions plates foul very slowly and need only be cleaned every five years or so. Some manufacturers make provision for easy cleaning, by making the recuperator removable, others include spray cleaning.

Plate recuperators are in the main only applicable to full fresh air systems, to which all the following calculations apply. Where recirculation is required, for example for night time operation, the duct system should be arranged so that the air does not pass through the plate recuperator, in order to save fan power.

In theory it is possible to design a plate recuperator with optimum performance for a given set of conditions. In practice, however, each manufacturer has a range of units available from which the most suitable is chosen.

In general, the supply air temperature efficiency obtained under normal environmental conditions will be about 60–70%, depending mainly on the amount of latent heat recoverable from the exhaust air.

The amount of energy to be recovered should be set in relation to the amount of energy used to drive the system. As a plate recuperator is a passive system, the only energy input is that associated with overcoming the pressure drop over the recuperator.

† The paper was presented to the 1987 CIBSE Technical Conference.

**Table 2** Example plate recuperator design conditions[1]

| | |
|---|---|
| Exhaust air volume | 2.8 $m^3 s^{-1}$ |
| Supply air volume | 2.8 $m^3 s^{-1}$ |
| Exhaust air condition | 20°C/55% RH |
| Supply air condition | −1°C/90% RH |
| Energy recovered | 50.9 kW |
| Fan energy used | 2.3 kW |

Although the term COP is strictly incorrect in this context it does express the relationship between the useful energy added to the fresh air stream and the energy used to drive the system. In the example of Table 2 the COP (as defined) is 22.1.

The design conditions shown above are, of course, not a true indication of the average performance. Figure 1 shows the relationship between COP (as defined above) and ambient temperature.

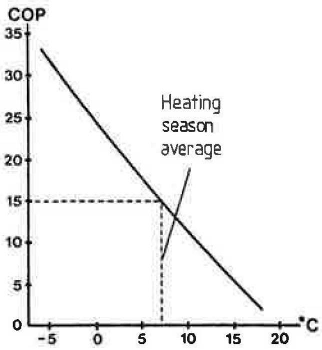


**Figure 1** Efficiency of plate recuperator

The limitation of the plate recuperator is that it can only recover a proportion of the heat available in the exhaust air, but cannot upgrade that heat. By combining the plate recuperator with an air-to-air heat pump this can, however, be achieved.

## 3 Principles of plate recuperator/heat pump combination

The heat pump is combined with the plate recuperator so that heat is recovered in the plate recuperator before the air reaches the coils of the heat pump (Figure 2).

This way the fresh supply air is preheated over the recuperator before reaching the condenser coil. Here the heat extracted by the evaporator coil from the precooled exhaust air is given off to the supply air, together with the compressor energy. The total energy input to the system consists of four parts: the pressure drop over the recuperator; compressor consumption; the pressure drop over the condenser coil; the pressure drop over the evaporator coil. The total energy output is the sum of heat recovered in plate recuperator and the condenser coil output.

For the example of Table 3 the coefficient of performance (as defined previously) is COP = 69.8/8.5 = 8.2.

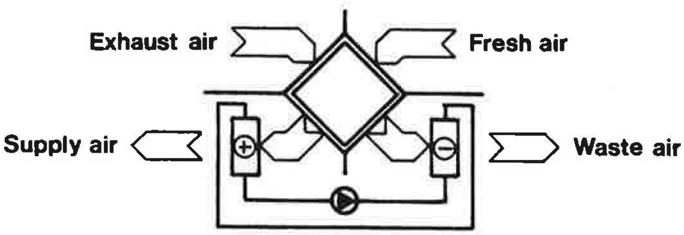


**Figure 2** Plate recuperator/heat pump layout

**Table 3** Example plate recuperator/heat pump design conditions[1]

| | |
|---|---|
| Exhaust air volume | 2.8 $m^3 s^{-1}$ |
| Supply air volume | 2.8 $m^3 s^{-1}$ |
| Exhaust air condition | 20°C/55% RH |
| Supply air condition | −1°C/90% RH |
| Energy recovered in recuperator | 50.9 kW |
| Energy output from condenser coil | 18.9 kW |
| Total output | 69.8 kW |
| Fan energy for recuperator | 2.3 kW |
| Compressor energy | 4.9 kW |
| Fan energy for condenser coil | 0.6 kW |
| Fan energy for evaporator coil | 0.7 kW |
| Total input | 8.5 kW |

This performance applies only to the design winter conditions. The relationship between COP and ambient temperature is shown in Figure 3.

The dotted line in Figure 3 shows the weighted average for the heating season in London weather conditions[2]. With an average COP of around 6.0 the real cost of heat is 5.0/6.0 = 0.83p $kWh^{-1}$, or less than half the cost of gas.

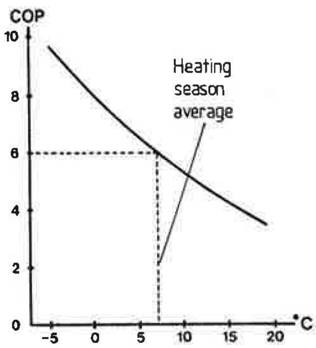


**Figure 3** Efficiency of plate recuperator/heat pump combination

The energy output in the example of Table 3 would raise the supply air temperature against the ambient temperature.

Figure 4 shows that at 0°C ambient the leaving air temperature will be 20°C, or room temperature. This means that the supply air temperature efficiency is 100%, or that the full ventilation loss for the building has been recovered.

The leaving air temperature rises with the ambient temperature. At 10°C outside the LAT is 23°C. The 3°C

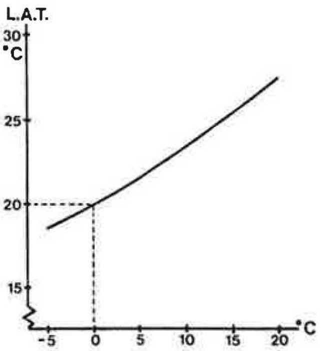


**Figure 4** Leaving air temperature for system in heating mode

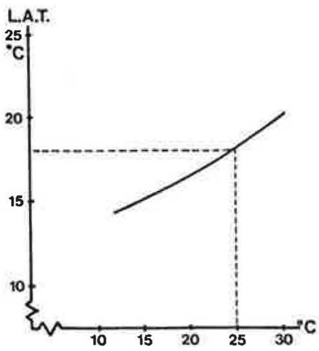


**Figure 5** Leaving air temperature for system in cooling mode

difference between room condition and LAT is used to cover a proportion of the fabric loss for the building. In this case the contribution is 8 kW.

In most cases, depending on insulation values and casual gains, the total building heating requirement will be satisfied by the plate recuperator/heat pump unit at ambient temperatures above 8–10 °C. In modern well-insulated buildings where casual gains are often in balance with the fabric loss at winter design conditions the point at which the system covers the total heating requirement is often much lower.

The plate recuperator/heat pump combination was developed mainly as a heating unit, and its economic viability is normally considered on this basis alone. It is, however, an obvious next step to reverse the heat pump cycle and use the system as a cooling unit as well.

Under reverse cycle conditions the fresh supply air is cooled by the heat pump, as occurs in a standard heat pump system. But providing the room temperature is kept below ambient then the plate recuperator will also act as a cooling unit, precooling the air before it reaches the evaporator coil of the reversed heat pump.

**Table 4** Example reverse cycle design conditions[1]

| | |
|---|---|
| Exhaust air volume | 2.8 $m^3$ s |
| Supply air volume | 2.8 $m^3$ s |
| Exhaust air condition | 20°C/55% RH |
| Supply air condition | 25 °C/50% RH |
| Supply air condition after recuperator | 21.9°C 60% RH |
| Supply air condition after evaporator | 17.7°C 71% RH |
| Cooling energy saved over recuperator | 10.8 kW (all sensible) |
| Cooling energy saved over evaporator | 22.6 kW (incl. 8.1 kW latent) |
| Energy saving, total | 33.4 kW |
| Energy input, total | 9.9 kW |

As the leaving air temperature is below room condition it means that all the ventilation gains have been covered. The difference, 2.3 °C sensible, corresponding to 7.7 kW, is used to cover casual gains in the building. If the casual gains are greater than 7.7 kW then the room temperature will rise unless additional cooling capacity is added. The effect of the rise in room temperature is shown later. Figure 5 shows the leaving air temperature versus ambient. The graph shows that at outside temperature up to 30°C the system will recover all the ventilation gains.

The previous calculations have concentrated on the system performance at the extreme design conditions, both summer and winter. Of equal importance is, of course, the total performance over a full year, both as a heating unit and as a cooling unit. This is best evaluated by looking in detail at an example.

A London building, size 60 m L × 15 m W × 3.3 m H = 3000 $m^3$, with an overall U-value of 0.6 W $m^{-2}$ $K^{-1}$, requires 2.9 $m^3$ $s^{-1}$ of ventilation.

Room temperature is 20°C, fabric loss (at −1°C outside) 30 kW, casual gains 15 kW. Using a simplified rectilinear expression for the energy demand, including ventilation loss, the picture shown in Figure 6 emerges.

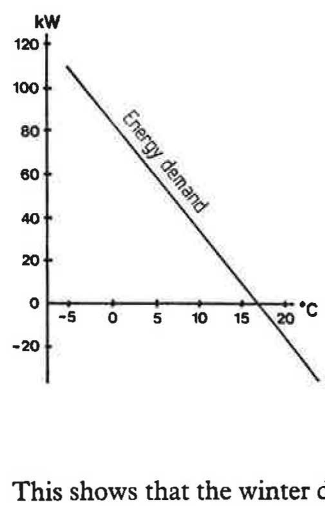


**Figure 6** Building energy demand

This shows that the winter design demand is 89 kW and the summer design cooling demand (at 27 °C) is 50 kW. Entering the output curves for the plate recuperator and heat pump, for both heating and cooling, gives Figure 7.

The middle curve of Figure 7 is the plate recuperator output. The top and bottom curves are the total system outputs for heating and cooling respectively. The curves provide the following information:

(*a*) At temperatures up to 6°C outside additional heating is required. At −1°C the additional heating needed is 16 kW. The heat pump is working continuously.

(*b*) At this point, 6°C outside, the heating demand is fully covered by the system, with no back-up heating required.

(*c*) In the temperature interval 6°C to 15°C the heat pump works intermittently.

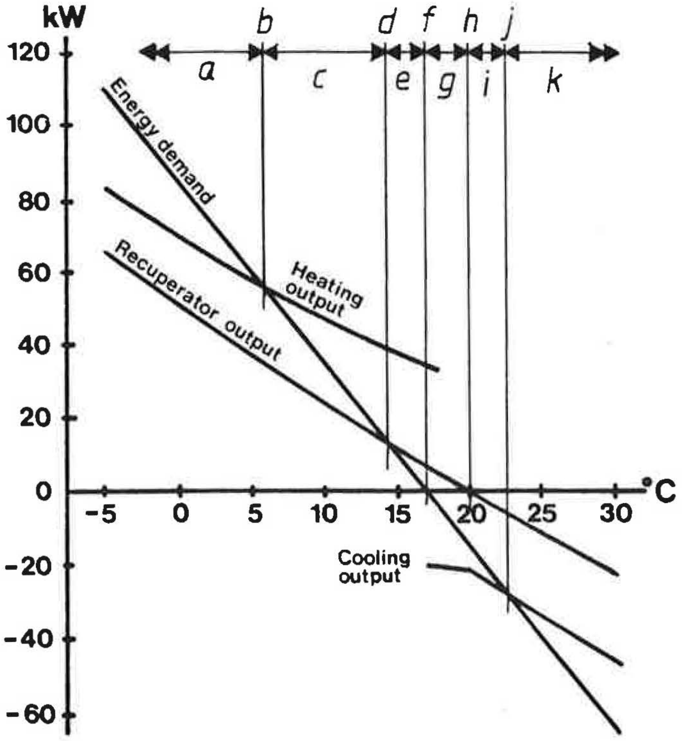


**Figure 7** System performance related to demand

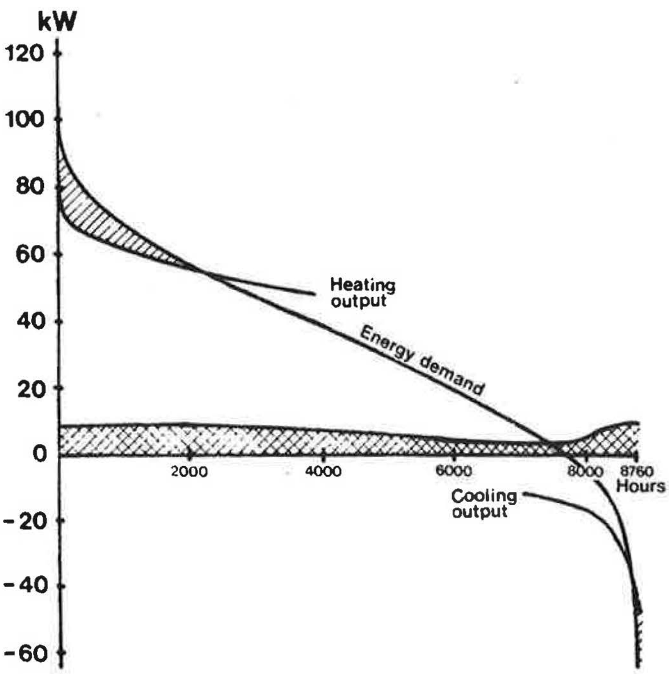


**Figure 8** Heating analysis graph

(*d*) At this point, 15°C, the heating requirement is fully covered by the plate recuperator alone. The heat pump is off.

(*e*) Provision of a bypass system for the plate recuperator allows a portion of the fresh supply air to bypass the recuperator, reducing the effective output. The heat pump is off. If a bypass is not used the room temperature will start to rise and the heat pump will switch to reverse cycle cooling.

(*f*) At this point, 17°C outside, the energy demand switches from heating to cooling. If a bypass system is used this will now be fully open.

(*g*) In the temperature interval 17–20°C the system uses free cooling by the fresh air, providing a bypass system is used. The heat pump works intermittently in cooling mode.

(*h*) At this point, 20°C outside, the plate recuperator will start to provide a cooling output, providing the room temperature is maintained at 20°C. The bypass damper will close.

(*i*) In the temperature interval, 20°C to 23°C outside, the plate recuperator provides a cooling output, and the heat pump works intermittently.

(*j*) At 23°C the demand reaches the output from the system.

(*k*) If the room temperature of 20°C is to be maintained, additional cooling capacity is now required otherwise the room temperature will rise.

This description of how the system should operate under varying outside conditions is also the basis for the design of the control sequence. This is discussed later.

To establish the energy consumption over the year it is necessary to relate the system consumption to temperature frequency. This is illustrated by using meteorological frequency data[2] to construct an 'S' curve for the energy demand and draw in the output lines for the plate recuperator/heat pump system. The curves are shown in Figure 8.

Based on 24-hour operation the graph shows that for approximately 1800 hours of the year some additional heating is required. Similarly, for approximately 300 hours of the year additional cooling is needed to maintain 20°C room temperature. The fourth curve on the graph is the electrical consumption, including compressor, recuperator and coil pressure drops. The hatched areas express the amount of energy required as back-up (kWh). The cross-hatched area under the electrical consumption curve expresses the total energy consumption of the system.

In the example the total consumption figures are: compressor and fans 60 500 kWh; back-up heating 10 700 kWh; back-up cooling 1400 kWh. This should be compared with the total energy output from the system: heating 275 500 kWh and cooling 21 700 kWh. If this consumption is related to the building size it shows a total annual consumption under 24 hour operation of 80 kWh per $m^2$ floor area. If the operating period is reduced to 0800 to 1800, five days per week, the consumption and output figures will instead be: compressor and fans 18 000 kWh; back-up heating 2 600 kWh; back-up cooling 700 kWh; energy output, heating 73 000 kWh; energy output, cooling 10 000 kWh. This corresponds to an annual consumption of 24 kWh $m^{-2}$, which is considerably less than conventional systems can achieve, even if they use far less fresh air ventilation.

It is also significant that the plate recuperator/heat pump system will cover the total energy demand for the building from 5°C to 23°C ambient. Without any back-up heating for daytime operation this corresponds to 96% of the total requirement, leaving only 450 h when back-up heating is required and 150 h when back-up cooling is needed. The pie-charts in Figure 9 show the amounts of back-up heating and cooling required.

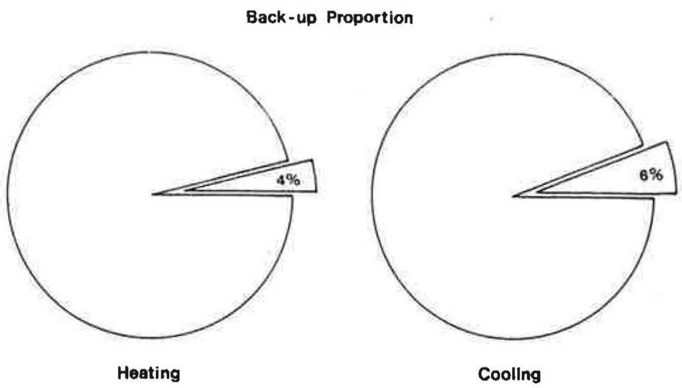


**Figure 9** System back-up requirements

At this point it is appropriate to consider the performance of the system if the proposed new building regulation $U$-values were used. This would reduce the fabric loss from 30 to 15 kW. If at the same time the room temperature is allowed to drift upwards to, say, 23°C, at 27°C ambient the output versus consumption curves shown in Figure 10 result.

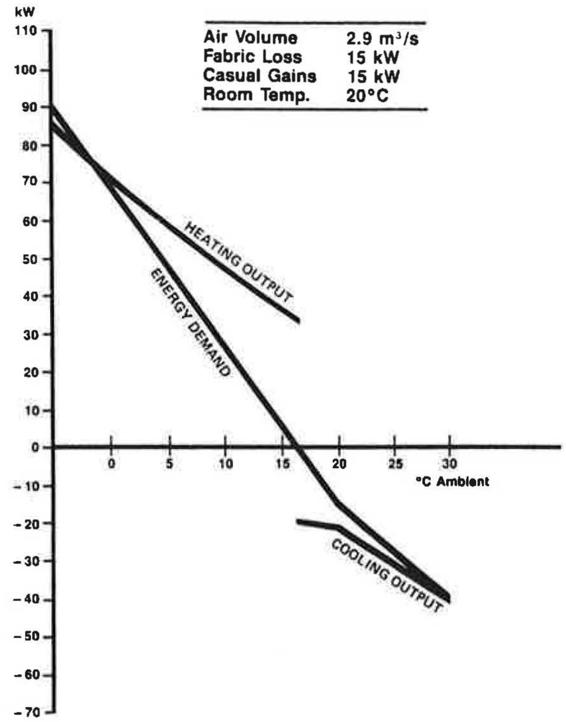


**Figure 10** System performance with room temperature drift

Because the room temperature is increased at ambient temperatures above 20°C, the fabric gains decrease. Although the system output also decreases this happens more slowly, with the result that the output and consumption curves converge. The graph shows that the system under these conditions will cover the cooling demand up to 30°C ambient.

Equally important is the situation at low ambient temperatures. The graphs show that even at temperatures below −1°C the system provides the total energy demand for the building.

Under these conditions the system will therefore cover heating, ventilation and cooling throughout the entire ambient temperature range. Central heating plant has, in other words, become obsolete.

Even under present building regulation $U$-values the back-up requirement is so low and the effective heating season so short, that central plant is uneconomical. The required back-up is more economically provided by duct mounted resistance heaters or panel heaters in the space.

## 4 Controlling the recuperator/heat pump system

The combination of a plate recuperator with an air-to-air heat pump has, as described above, a large potential for saving energy in building ventilation systems. In order to achieve such savings it is, however, necessary to employ the right type of control system. Although control would appear fairly simple, based as it is on dry bulb temperature, there are a number of factors that should be carefully considered.

Both the plate recuperator and the heat pump have the capacity to work both as heating and as cooling units. To this must be added back-up heating and in some cases also back-up cooling. It follows that unless the control system is correctly designed one part of the system could cool the air while another heats it again, leading to a waste of energy. Furthermore, the energy output from the individual elements in the system is obtained at different prices. It is therefore necessary that the control system should prioritise the use of different elements in such a way that the lowest cost energy is used first, and only when the output from this source is fully utilised is the next lowest brought in. Figure 11 describes this order of priority.

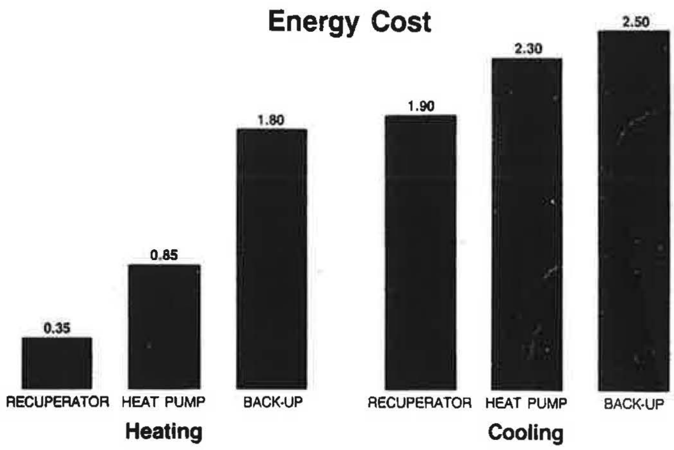


**Figure 11** Energy cost order of priority

Figure 12 shows the system configuration.

Sequence control for the heat pump and the back-up heating and cooling can be carried out using a step controller, whereas the bypass damper which controls the plate recuperator output needs a special logic controller. The damper position depends on the relative ambient and exhaust air temperatures, combined with the requirement for either heating or cooling.

## 5 Applications

The plate recuperator/heat pump combination was developed in the 1970s in Denmark, as a result of the increase in

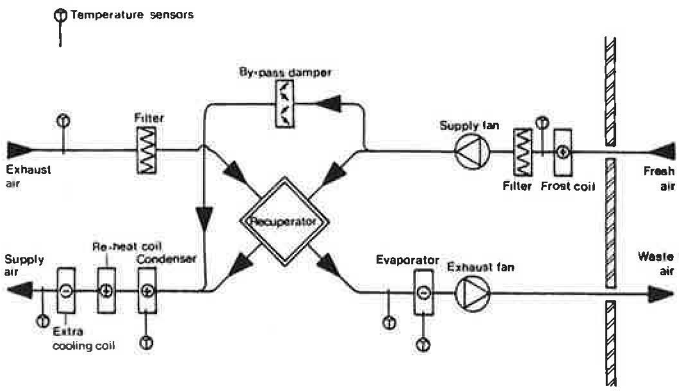


**Figure 12** System control configuration

the cost of energy. Climatic conditions in Denmark[3] are very similar to those in the UK[4], with an annual mean temperature of 8°C, corresponding to the midlands and north of England, and only a degree or so lower than the south of England. Danish experience can therefore be related directly to UK conditions (Figure 13).

In principle the system can be used wherever a conventional fresh-air ventilation system is needed. The degree of economic benefit depends mainly on operating time, air change rate versus building fabric loss, and insulation values. Generally, the longer the system is required to operate, the greater the air change rate, and the lower the $U$-values, the better the system performs.

The main applications are those where full air conditioning is considered too expensive, but a certain amount of cooling is desirable. In such cases the system is not significantly more expensive than conventional systems, particularly when considering savings in pipework and boiler installation. The pay-back period can be very short. The system has chiefly been used in commercial buildings such as offices, hospitals, kitchens/canteens, laboratories, hotels, conference rooms and sports facilities. There are many more specialised applications.

## 6 Conclusion

The plate recuperator and heat pump are both well documented and proven energy saving devices. The combination of the two is therefore merely an extension of their application.

Controlled correctly, the system produces significant savings against conventional equipment, with pay-back periods which can be very short, particularly where connection to central heating plant cannot be justified on the grounds of low utilisation. Conversely, if compared with systems which provide a minimum of fresh air ventilation, the plate recuperator/heat pump combination gives much improved environmental quality at a cost comparable to or lower than that of minimum fresh air systems.

The unit is only applicable to full fresh air ventilation systems and cannot be used for recirculated air, except in specialist applications such as swimming pools, where the unit functions as a dehumidifier, not as an exhaust air heat pump. It does not provide full air conditioning but it approaches a similar result at significantly lower capital and operational costs.

## References


1 *Dantherm Design Guide for XVV* pp 24–27, 33–38 (Clevedon: Dantherm) (1986)

2 *Frequency Analysis for London Weather Centre, 1.10.74 to 31.12.82* (London: Meteorological Office) (1983)

3 *Reference Year Frequency Data* (Copenhagen: Meteorologisk Institut) (1981)

4 Letherman K M and Dewsbury J The Bin Method—A procedure for predicting seasonal energy requirements in buildings *Building Serv. Eng. Res. Technol.* 7(2) 55–64 (1986)


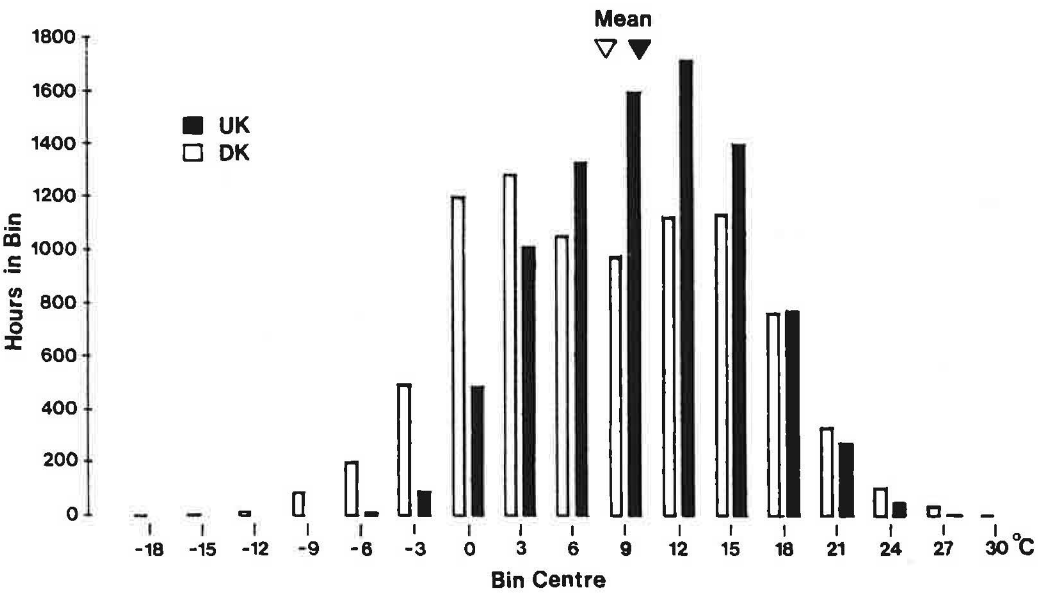


**Figure 13** Temperature frequency analysis for UK and Denmark