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DOMESTIC VENTILATION HEAT RECOVERY USING HEAT PUMPS

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SUMMARY

The application of heat pumps to ventilation heat recovery in domestic houses is considered. It is shown that the most effective system is a combination of heat pump and heat recovery unit; a plate heat exchanger is the type commonly used. Such units are now commercially available, and can provide heat at a lower cost per kilowatt hour than the Economy 7 tariff. The performance of several units is presented, and seasonal running costs have been computed for a house equivalent to the Capenhurst low energy house design. A heat pump VHR unit would be incorporated in a full house ventilation system, and could provide up to half of the heat required on a design day. It would be practical to reduce or eliminate the storage heating capacity in the house, and use direct acting heaters to provide flexible, accurate temperature control. Such a heating system would provide good temperature distribution and low running costs. The application of a heat pump VHR unit would therefore seem to be an attractive option in a low energy house. Development work is needed on the details of installation and control.

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Notation

C	specific heat of air at constant pressure	J/kgK
m_{in}	supply air mass flow rate	kg/s
m_{out}	exhaust air mass flow rate	kg/s
m_{mn}	make up flow rate, $m_{mm} = m_{in} \sim m_{out}$	kg/s
m_{adv}	adventitious ventilation	kg/s
m_{mech}	greater of m_{out} & m_{in}	kg/s
P_{fan}	power consumption of one fan	W
P_{comp}	power consumption of heat pump	W
P_m	power consumption of motors etc.	W
P_{hr}	total power consumption	W
H_{ex}	heat transferred by heat exchanger	W
H_{hp}	heat delivered by heat pump, including useful heat from compressor	W
H_{hr}	total heat supplied to incoming air, including useful fan power	W
V_{mech}	mechanical ventilation rate at house temperature	m^3/s
V_{ex}	exhaust flow rate, at house temperature	m^3/s
V_{in}	supply flow rate, at house temperature	m^3/s
ε	energy effectiveness of heat exchanger	ND
η_s	seasonal COP of system	ND
η_{hp}	COP of heat pump	ND
η_{vhr}	COP of whole system	ND
θ	temperature efficiency of heat exchanger	ND
θ_{sup}	temperature efficiency at supply temperature (includes fan)	ND
ρ_i	density of air at T_i	kg/m^3
T_i	inside air temperature	°C
T_o	outside air temperature	°C
T_{sup}	supply air temperature	°C
T_{exh}	exhaust air temperature	°C

1. INTRODUCTION

The Capenhurst low energy house successfully used mechanical ventilation with heat recovery (VHR) as part of its energy saving system. Several VHR systems are available, mostly manufactured in northern Europe. They consist of a central unit, containing intake and exhaust fans, together with the heat exchanger, which is used to transfer heat from the warm exhaust air to the incoming cool fresh air. Tempered fresh air is ducted to living and bedrooms, while warm stale air is extracted from bathroom and kitchen. The inlet and extract fans are designed to run continuously during the heating season. This provides complete ventilation, and no window opening is necessary.

The heat exchanger in such systems operates with an energy efficiency of up to 70%, and so the system saves a substantial part of the ventilation heat loss. This saving must be offset against the running cost of the fans and the capital cost of the equipment itself. The performance of a VHR unit can be improved by incorporating a heat pump, which extracts heat from the exhaust air and delivers it to the supply air stream. The performance of the heat exchanger-heat pump combination is a function of both indoor and outdoor temperature, and so its potential energy savings depend on the pattern of weather over the heating season. This Memorandum analyses the seasonal performance of heat pump ventilation heat recovery systems for a standard weather pattern.

2. HEAT RECOVERY SYSTEMS

There are several types of heat recovery system available, most of which have found application in domestic use. The systems are summarised briefly in this section. Extensive surveys are given by Reay (1979) and Shurcliff (1981).

2.1 Run around coil (Figure 2.1)

Heat exchange coils are incorporated in the intake and exhaust air streams. A pump is used to circulate the heat exchanger fluid between the coils; the fluid is commonly a water-antifreeze mixture. The liquid picks up heat from the warm exhaust gas and tansfers it to the incoming fresh air. The efficiency of this arrangement is usually 50% or below.

The system has the advantage of simplicity, and flexibility of layout. The two gas streams do not have to be brought close together, and there is no danger of cross-contamination. The system is common in industrial use, but not found in domestic systems.

2.2 Heat wheel (rotating regenerator) (Figure 2.2)
The regenerator wheel spans two adjacent ducts, which carry the exhaust and intake gas streams in counterflow. In use, the wheel rotates slowly. The sector of the wheel in the warm gas stream absorbs heat, and its temperature rises. As this warmed sector rotates into the cooler intake stream, it gives up its heat to the incoming gas. The wheel is commonly made of a metal wire matrix. The efficiency of heat exchange is good. Some cross-contamination of the gas streams is possible, but this would not be a serious problem in domestic use. The system is occasionally found in domestic systems. It has a particular advantage, which is important in some applications, that it is possible to recover latent heat.

2.3 Plate regenerator (Figure 2.3)
The intake and exhaust air streams each pass through a regenerator unit. This is a box containing metal plates. The warm exhaust stream gives up its heat to the plates in one regenerator. When the changeover valve is operated, the incoming air is now warmed by the heated plates. The changeover of flow direction occurs every few minutes. Very high temperature efficiencies are possible. Domestic regenerators are available.

2.4 Plate heat exchanger (Figure 2.4)
The plate heat exchanger consists of a series of thin metal plates, separated by small gaps. The two air flows pass through adjacent gaps, separated only by one plate, through which heat transfer takes place by conduction. The exchanger is normally configured to give a cross flow operation, and temperature efficiencies of up to 70% are possible.

This is the most common form of heat exchanger used in domestic ventilation heat recovery.

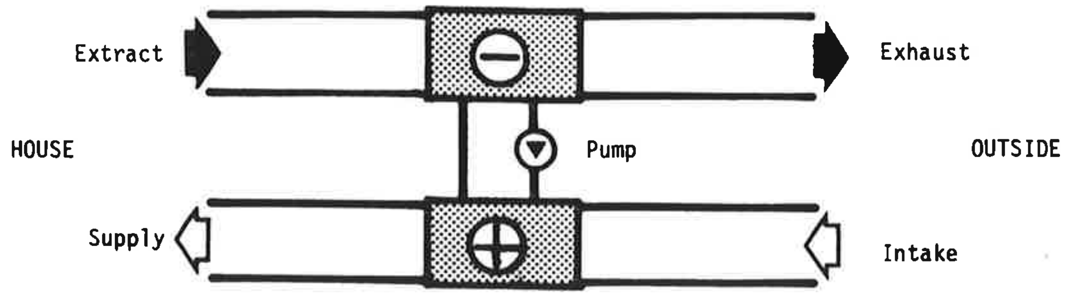


Figure 2.1 Simple run around coil

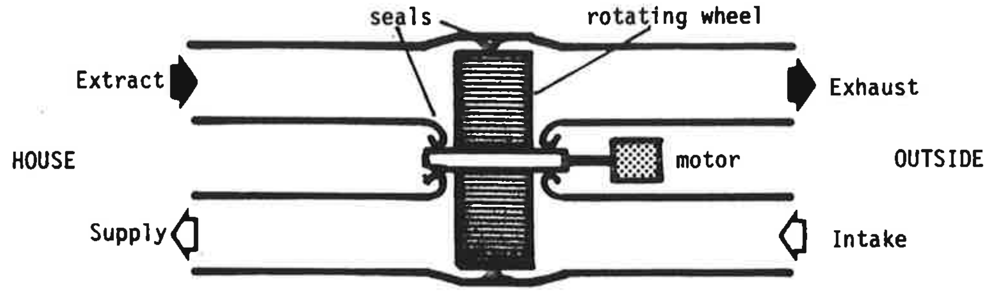


Figure 2.2 Heat wheel regenerator

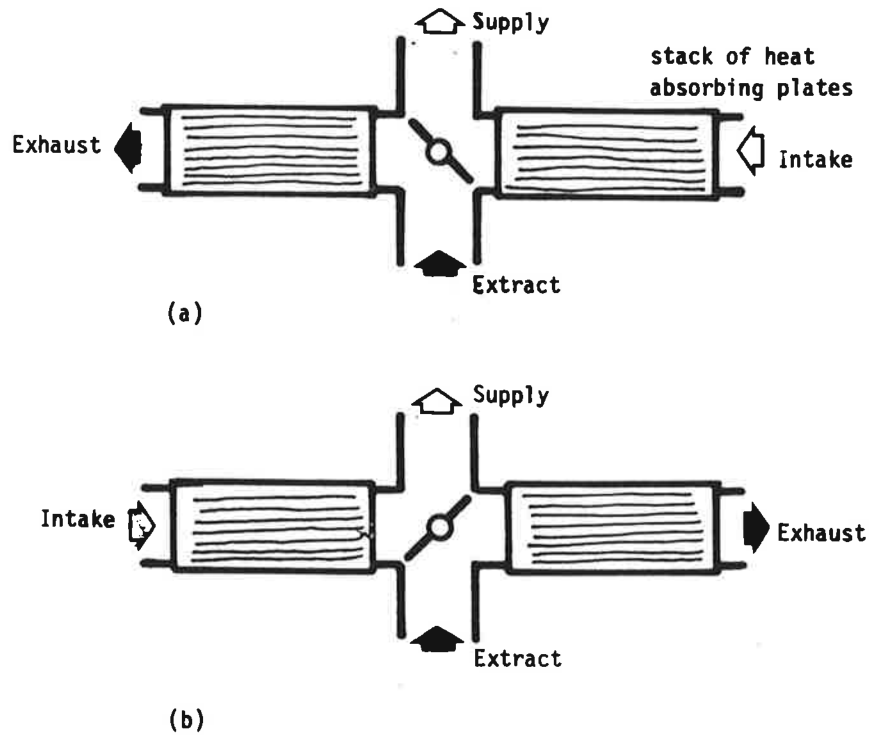


Figure 2.3 Plate regenerator. The diverter changes from position (a) to (b) every few minutes.

2.5 The heat pump (Figure 2.5)

In its simplest form, the heat pump is arranged with the evaporator in the exhaust air stream and the condenser in the intake air stream; if the compressor itself is positioned in the intake stream as well then the heat dissipated by the motor will be recovered in the intake air. The exhaust air temperature may be reduced well below the intake temperature, giving a "temperature efficiency" of over 100%. High COP's are possible, since the evaporator is working in a relatively warm air stream, and also high condensing temperatures are not required. Domestic systems are available.

2.6 Heat exchanger plus heat pump (Figure 2.6)

In this system a heat pump is used to boost the performance of a heat exchanger. A plate recuperator is shown in the diagram, but any recuperative heat exchanger could be used. The evaporator of the pump is placed in the exhaust air stream, downstream of the heat exchanger, and the compressor and condenser are placed in the supply stream. By this means very high overall COP's can be obtained. Since the heat pump only handles part of the total heat exchange, it can be relatively small, thus saving on capital cost. Such systems have been introduced by manufacturers, but are not yet common in domestic use. Olsen (1984) gives an introduction to the Dantherm systems for commercial buildings. Trumper & Hain (1984) compare the heat pumps plus heat exchanger with other domestic ventilation systems. It is possible to boost the performance of a run around coil (Section 2.1) with a heat pump. This system is described by Richarts et al. (1984).

3. PERFORMANCE OF A VENTILATION HEAT RECOVERY SYSTEM

3.1 Basis of measurement

The performance of a VHR system is often expressed in terms of percentage efficiencies of heat recovery, or coefficients of performance if a heat pump is incorporated. While these figures give an idea of how well the unit is performing, they do not necessarily give the information required to calculate the energy consumption of the house. The approach adopted in this memorandum is to consider the energy recovery of the system as a separate energy input to the house, treated independently of the

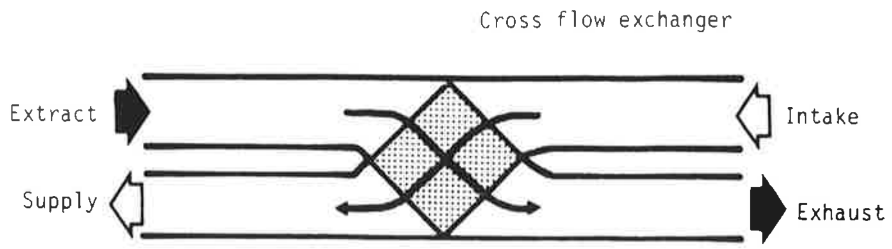


Figure 2.4 Plate heat exchanger

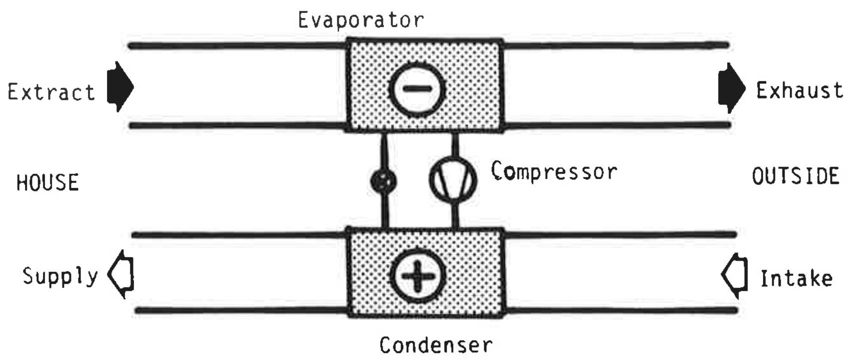


Figure 2.5 Heat pump heat recovery

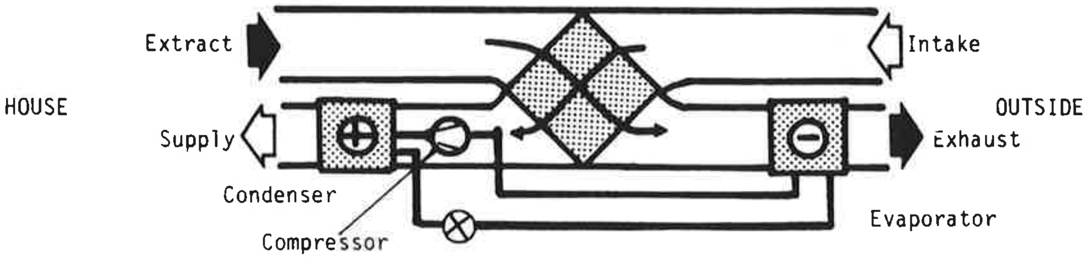


Figure 2.6 Heat exchanger plus heat pump

ventilation loss. This approach avoids confusion, and is particularly helpful where the incoming and outgoing air flows in the VHR system are not equal.

This memorandum considers the energy exchanges entirely in terms of sensible heat. A full treatment would use the enthalpy of the ventilation air. The exhaust air in general will have a higher humidity content than the incoming air, and so the enthalpy difference between the two air streams is higher than the sensible energy difference. Since most heat exchangers only transfer sensible heat, the efficiency of enthalpy reclaim is lower than the sensible heat energy reclaim efficiency. Enthalpy may be reclaimed in two ways: by transfer of moisture from the outgoing to incoming air stream, or by the recovery of latent heat by condensation in the exhaust air stream. Reclaim of moisture will produce a practical energy saving only for conditioned spaces where humidification is required, e.g. some offices or industrial processes. There is no point whatever in reclaiming moisture in the domestic situation, since one of the primary functions of ventilation is the removal of moisture. Reclaim of latent heat by condensation would be advantageous, and can indeed happen in cold weather with a heat exchanger. However, it would be misleading to consider this as part of the general efficiency of the system, since moisture production generally occurs from cooking or human perspiration, and so is not a charge on the space heating system of the house. Table 1 sets out the physical constants of air used in this report.

Table 1. Properties of air

		Temperature (°C) 0	20
Specific heat C	J/KgK	1004	1004
Density	kg/m^3	1.286	1.203
Volumetric specific heat ρC	J/m^3K	1291	1208

The above figures are for dry air at standard atmospheric pressure. The values vary slightly with humidity. For simplicity and consistency throughout the report we use the following values:

Specific heat $C = 1000$ J/kgK

Volumetric specific heat at room temperature $\rho C = 1200\ J/m^3K$

3.2 Performance parameters

The following sections derive the parameters which will be used to describe the performance of the various heat exchange systems.

3.2.1 Recuperative heat exchanger

Figure 3.1 shows a generalised heat exchanger, which transfers heat from exhaust to incoming air stream. The placing of the fans is important in the analysis, since this determines whether the power consumption of the fan will be utilised or not. The fan position shown in the diagram is common to most domestic systems; the intake fan is placed entirely within the air stream, so that its energy dissipation is delivered to the house. The energy of the exhaust fan is, however, wasted.

The temperature efficiency of the heat exchanger is defined as

$$\theta = \left(T_2 - T_1\right) \Big/ \left(T_3 - T_1\right) \tag{3.1}$$

and so a 100% efficient exchanger would deliver incoming air at a temperature $T_2 = T_3$ i.e. at a temperature equal to the internal house temperature. The efficiency θ is a function of flow rate, with the

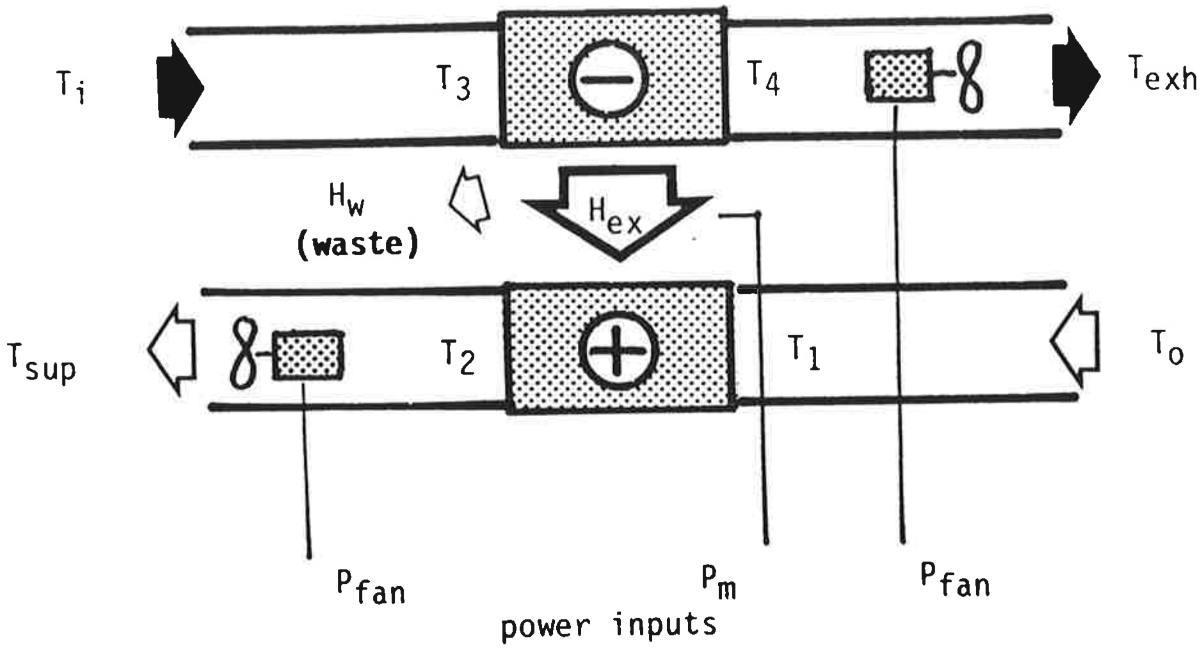


Figure 3.1 General heat recovery system, showing power inputs P and heat transfers H

efficiency generally rising as the flow rate through the heat exchanger is reduced.

The temperature efficiency is sometimes defined implicitly in manufacturers' literature as

$$\theta_{sup} = \left(T_{sup}-T_1\right)\Big/\left(T_3-T_1\right) \qquad (3.2)$$

While in some senses this is a more useful definition, allowing the supply temperature to be calculated from knowledge of the inside and outside temperatures, it is somewhat misleading to refer to it as an efficiency, since the less efficient the intake fan motor, the more heat it will dissipate, and so increase the apparent heat exchange efficiency. The value θ_{sup} is a function of outside temperature.

The mass flows through the two halves of the heat exchanger are not necessarily equal, and so the temperature change does not always reflect the energy transfer. This is expressed by the energy effectiveness of the heat exchanger, defined as the amount of heat transferred to the incoming air stream divided by the maximum heat transfer possible. Maximum heat transfer would occur in an exchanger of infinite area, and would reduce the outgoing air stream temperature to the outside air temperature. The energy effectiveness is then defined as

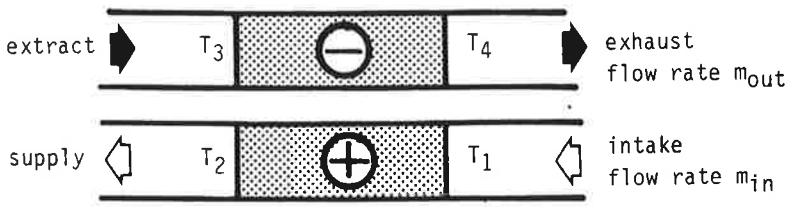
$$\varepsilon = m_{in}\left(T_2-T_1\right)\Big/\left(m_{out}\left(T_3-T_1\right)\right) \qquad (3.3)$$

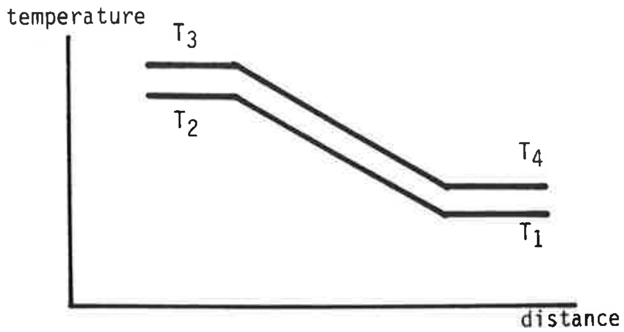
$$= \theta m_{in}/m_{out}$$

This equation assumes that the specific heat of air at constant pressure is constant over the temperature ranges involved. Figure 3.2 illustrates the concepts of efficiency and effectiveness.

For the fan position shown in the diagram the intake fan produces an additional temperature increment in the air, and the supply temperature is given by

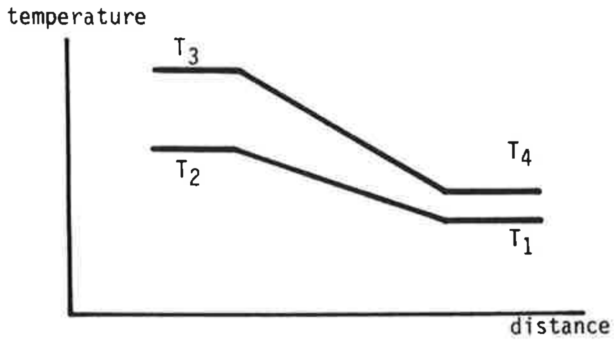
$$T_{sup} = T_2 + P_{fan}\Big/\left(m_{in}\ C\right) \qquad (3.4)$$





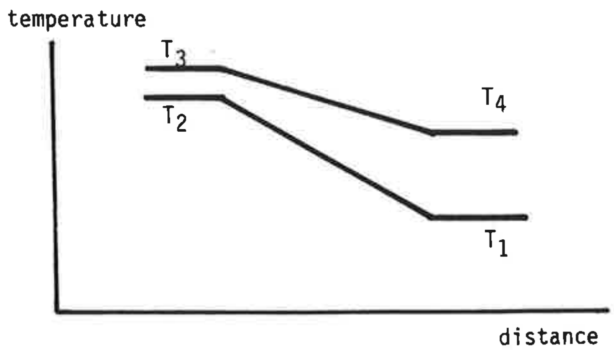
Case 1. $m_{in} = m_{out}$

Energy effectiveness equals temperature efficiency



Case 2. $m_{in} > m_{out}$

High energy effectiveness
Low temperature efficiency



Case 3. $m_{in} < m_{out}$

High temperature efficiency
Low energy effectiveness

Figure 3.2 Heat exchanger efficiency.

The temperature efficiency and energy effectiveness of a heat exchanger are affected by the ratio between intake and extract flow rates. The figures are defined as:

Temperature efficiency $\theta = (T_2 - T_1)/(T_3 - T_1)$
Energy efectiveness $\varepsilon = m_{in}(T_2 - T_1)/(m_{out}(T_3 - T_1))$

The diagram shows a heat loss H_w in the heat exchanger. This can occur in practice if a poorly lagged heat exchanger unit is situated in an unheated space, such as the loft. The definitions of efficiency and effectiveness above take this into account automatically.

Some forms of heat exchanger require input power, e.g. to operate the regenerator wheel or the pump of a run around system. It is assumed that this power P_m is wasted and does not appear as heat in the input air stream. The overall coefficient of performance of the system is the heat input to the incoming air stream, divided by the total power consumption.

$$\eta_{vhr} = \frac{m_{in}\, C \left(T_{sup} - T_o\right)}{P_m + 2P_{fan}}$$

The COP varies strongly with outside temperature.

For heating analysis, the most important quantity is the actual heat input to the ventilation air.

$$H_{hr} = m_{in}\, C \left(T_2 - T_1\right) + P_{fan} \qquad (3.5)$$

$$= \varepsilon\, C\, m_{out} \left(T_i - T_o\right) + P_{fan}$$

This shows that the heat input from the heat exchanger is proportional to the temperature difference between inside and outside temperature.

3.2.2 Heat pump

A heat pump is used to transfer heat from exhaust to supply air streams. The fan arrangement shown in Figure 3.3 is that of the Clarel Ete; putting the exhaust fan upstream of the evaporator allows its heat dissipation to be collected by the evaporator coil. The coefficient of performance of the heat pump alone is

$$\varepsilon = H_{hp} / P_{comp}$$

and that of the whole system is

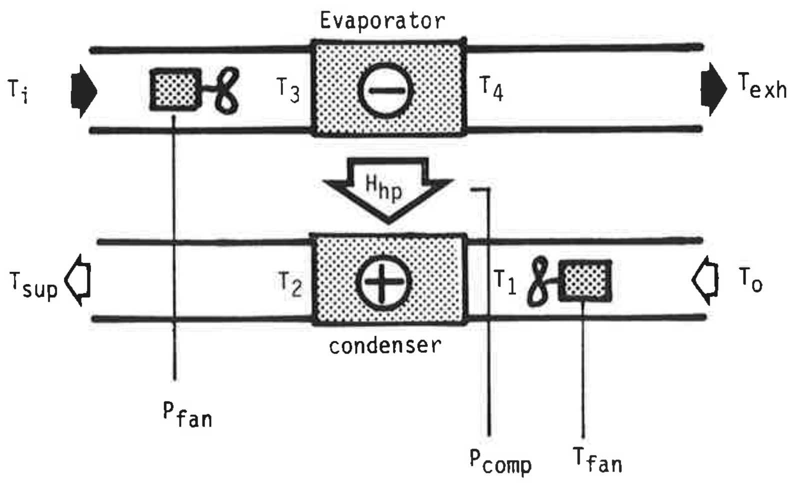


Figure 3.3 Energy flows for heat pump heat recovery. Fan placement is that of the Clarel Ete.

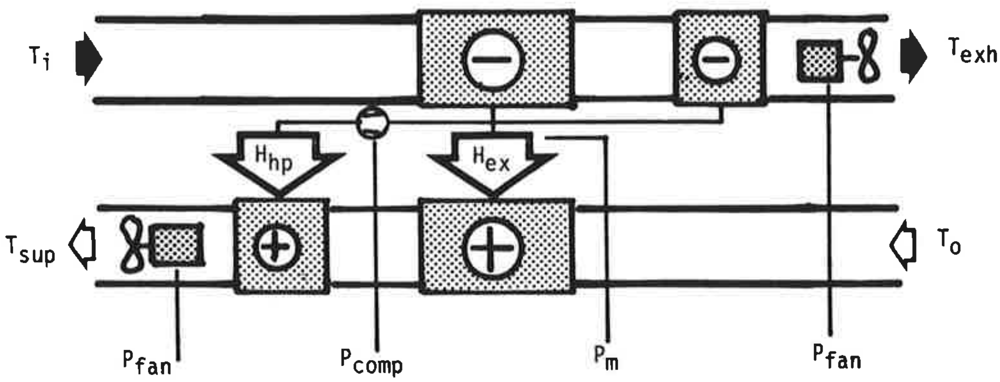


Figure 3.4 Energy flows for heat pump plus heat exchanger. Normally P_m = 0 for a heat exchanger.

$$\eta_{vhr} = \left(H_{hr} + P_{fan}\right) / \left(P_{comp} + 2P_{fan}\right)$$

H_{hp} is the heat delivered by the system to the incoming air stream. It includes the heat collected from the exhaust fan, and any useful heat dissipated by the compressor.

The coefficient of performance of the heat pump increases as the outside temperature falls, given a constant internal temperature. Thus in contradiction to the normal outside air source heat pump, more heat is delivered in cold weather.

3.2.3 Heat exchanger with heat pump

A typical layout is shown in Figure 2.4. The heat exchanger parameters are defined as in Section 3.2.1. The temperature efficiency of the heat exchanger is

$$\theta = \left(T_2 - T_1\right) / \left(T_3 - T_1\right)$$

and the energy effectiveness

$$\varepsilon = m_{in} \left(T_2 - T_1\right) / \left(m_{out} \left(T_3 - T_1\right)\right)$$

The heat delivered by the heat exchanger is

$$H_{ex} = \varepsilon m_{out} C \left(T_1 - T_o\right)$$

The heat pump coefficient of performance is

$$\eta_{hp} = H_{hp} / P_{comp}$$

The heat delivered by the heat pump, H_{hp} (W) includes any useful heat dissipated by the compressor. The total heat H_{hr} delivered to the incoming air is the sum of heat from the heat pump, heat exchanger plus any useful fan power.

$$H_{hr} = H_{hp} + H_{ex} + P_{fan}$$

This heat produces a rise in temperature of the incoming air stream of $\left(T_{sup} - T_o\right)$ which is given by the relation:

$$H_{hr} = m_{in} C \left(T_{sup} - T_o\right)$$

The overall coefficient of performance of the system is

$$\eta_{vhr} = H_{hr}/P_{hr}$$

where P_{hr} is the total power input

$$P_{hr} = 2P_{fan} + P_{comp} + P_m$$

As before this COP is highly dependent on outside temperature, and quoting a single value measured at design conditions will give a misleadingly high impression. More useful is the concept of seasonal coefficient of performance

$$\eta_s = \sum H_{hr}/\sum P_{hr}$$

where the summations are made over the whole heating season.

3.3 Absolute description of system performance

The above treatment of VHR efficiencies has shown that the use of "efficiency" alone as a descriptor of performance is incomplete, and moreover can be misleading. For the purposes of the seasonal energy analysis described below, it is necessary to describe the performance of all the ventilation systems in a common form. Three quantities must be given:

1. The ventilation rate. The mechanical ventilation rate is denoted by the mass flow rate m_{mech}, which is the higher of the supply rate m_{in} and the extract rate m_{out}; any difference between the two is made up by leakage induced by the fan pressure, so that the make up ventilation flow is $m_u = m_{in} \sim m_{out}$. However it is more common to give the ventilation rate as the volumetric flow rate V_{mech} (m^3/sec), where the volume is measured at internal house temperature. As before, the flow rate is taken as the higher of the inlet and extract flows.

2. The total heat delivered to the supply air by the VHR system. This includes heat from all sources: i.e. reclaimed heat, plus any useful heat dissipated from fans or compressors. This is a function of both inside and outside temperatures

$$H_h = H\left(T_i, T_o\right)$$

For any particular analysis which holds T_i constant, it will normally be sufficiently accurate to use a linear approximation

$$H_{hr} = a + b\ T_o$$

3. The total electrical power consumption of the system. This includes all fans, motors and compressors.

$$P_{hr} = 2P_{fan} + P_m + P_{comp}$$

The power consumption may be a function of temperature, particularly for heat pump compressors. For an analysis with constant internal temperature, we can use a linear approximation

$$P_{hr} = c + d\ T_o$$

No consideration has been given to defrosting, which in practice may reduce the performance of the heat pump system.

4. COST OF HEAT SUPPLIED

If the heat pump is considered purely as a provider of heat, then it must supply that heat at a cost which is less than that of any suitable alternatives. For an all-electric house, comparison is made between a heat pump and heat from a storage heating system. Denote the cost of electrical energy as k_d(p/kWh) during the daytime and k_1(p/kWh) during the low rate period. The heat pump always provides energy at a lower cost than off-peak heating if the coefficient of performance is greater than the day/night cost ratio i.e. $\eta < k_d/K_1$. A typical 1985 tariff is $k_d = 5.42$ and $k_1 = 2.03$. With these figures the heat pump provides lower cost energy if the COP is $>$ 2.7.

When the space heating energy is costed over the whole heating season, the average cost per kWh of space heating from a normal (i.e. non heat pump) storage based system will be somewhat greater than k_1, since it is normal to size a storage system so that about 5% of the energy over the heating season is provided at day rate. The effective cost per unit is then $k_{sp} = 0.05\ k_d + 0.95\ k_1$. If the heat pump is operated 24 hours per day, then the energy cost per unit is lowered by the proportion of time spent operating at the low rate period. On a seasonal basis, the heat pump shows lower costs than a storage system if

$$\left(0.05\ k_d + 0.95\ k_1\right) > \left(7\ k_1 + 17\ k_d\right)/24\eta$$

i.e. $\eta > 2.0$

Thus, if $\eta > 2.7$, the heat pump can always produce heat at a lower running cost than a storage system. If $2 < \eta < 2.7$, the heat pump system will have a lower average cost per unit of heat supplied compared with a storage system, assuming continuous 24 hour running of the heat pump. However, if such a system were used, it would be advantageous in mild weather to use any available storage capacity in preference to the heat pump, since the marginal cost of low rate heat is lower than the cost of the heat supplied by the heat pump during the daytime. An optimum operating strategy would therefore restrict the operating hours of the heat pump during mild weather, representing an inefficient use of the capital investment. This report does not attempt any economic analysis of the optimum use of capital employed. However, it seems clear that it would only be worth considering a heat pump on economic grounds if its COP is greater than 3.

The several air to air heat pumps for use in ventilation heat recovery systems that have been marketed, all show COPs greater than 3 and therefore offer the potential of reduced running costs. The actual amount that will be saved depends on the actual equipment, and the house in which it is installed, together with the weather pattern. Energy analyses have been made in this report for three commercial systems, installed in a low energy house built to the Capenhurst specification, and operated over a standard weather year.

5. CALCULATION OF SEASONAL ENERGY REQUIREMENTS

In low energy housing, solar gains and internal free heat gains provide a substantial contribution to the gross space heating energy requirement. This has the effect of reducing the heating season, and reducing the hours of operation of any space heating equipment. The effectiveness of additional energy saving measures may therefore be reduced, since there is simply less energy to be saved. It is important therefore in considering the usefulness of energy saving measures to use calculation methods which accurately deal with heat gains.

The method adopted in this memorandum is to use weather data for the CIBSE example weather year. Energy consumption is calculated day by day. The outside air temperature is assumed to vary sinusoidally about the mean 24 hour temperature, using monthly values of peak to peak temperature swings. This instantaneous temperature is used to calculate the performance of the ventilation heat recovery unit, since this is strongly temperature dependent. Conduction losses are estimated from the 24 hour mean temperature, and solar gains from the daily insolation. A constant internal temperature is assumed, and constant internal heat gains.

The ventilation fans run continuously over the whole heating season. The space heating system is assumed to be well controlled. As the space heating energy requirement reduces in mild weather, the space heating system is turned off before the VHR heat pump. On days when only part running of the heat pump is required to maintain the internal design temperature, the heat pump is switched off preferentially during the warm part of the day.

A computer program was written to compute seasonal energy requirements, and is described in more detail in Appendix I.

Comparison of heating systems must be made at stated ventilation rates. Ideally, one estimates the 'correct' ventilation rate for a dwelling, based on human and physical requirements (odour, moisture, etc.), and then compares the costs of heating and ventilating by the different methods. In practice, this is not so straightforward, since the ventilation system itself controls the ventilation rate. Experience in the Capenhurst low energy house shows that full mechanical systems, with ducted inlet and extract, produce satisfactory internal conditions at lower total air flow rates than those conventional assumed for buildings with natural ventilation through opening windows. This is because the correct positioning of supply and extract terminals removes contaminants at source, and ensures a supply of fresh air in the occupied living spaces. There is, however, some field evidence of the opposite effect in cold climates. In a comparison between extract only and full ventilation VHR, Svensson (1982) found that house owners ran an extract only system at a lower rate than for a full system; this was to avoid the discomfort of cold draughts induced by the extract system. The VHR system, since it introduced tempered air through a properly designed inlet, did not cause this problem, and was operated at a higher ventilation rate.

In order to achieve optimum efficiency, the compressor and heat exchange coils of a heat pump VHR system must be matched to the ventilation rate provided by the fans. The ventilation rate in a house is therefore decided by the choice of VHR system, and is necessarily made at the design stage of the house. It is not readily altered if the ventilation needs of the house change at a later date.

6. EQUIPMENT

Three VHR units have been chosen as a basis for the energy comparison; they are representative of heat exchanger alone, heat pump alone, and a combination of heat exchanger with heat pump. The examples chosen are all designed for domestic use, and were chosen as being up to date and with published performance data. Performance data has been taken from manufacturer's literature, and the equipment itself has not been inspected.

6.1 Bahco minimaster

This is a whole house mechanical ventilation system, based on a compact heat exchanger. The heat exchanger and both ventilation fans are incorporated in a single unit, which incorporates a cooker hood and is designed to be installed in the kitchen.

The manufacturer's literature shows the temperature efficiency curve given in Appendix II. The diagram implies that the efficiency is based on the supply temperature, and so includes the effect of the heat dissipated by the inlet fan (see equations 3.1 and 3.2). On this basis it is possible to calculate the energy efficiency, for a given air flow rate. The rate chosen is an extract air flow of 110 cubic metres per hour (30.6 l/s), which was the typical mechanical ventilation rate found to be satisfactory in the Capenhurst low energy houses. The power consumption of the fans is taken to be 65 watts each.

The intake fan produces a temperature increment

$$T_{sup}-T_2 = 65 \times 1000/(1200 \times 30.6 \times 0.9) = 2.0 \text{ K}$$

Assume the performance curve is established at design conditions $T_i = 20°C$ and $T_o = 0°C$. The temperature efficiency $\theta_{sup} = .76$, from the figure in Appendix II, and so from equations (3.1 and 3.2), the exchanger temperature efficiency $\theta = 0.66$. The energy effectiveness

$$\varepsilon = 0.66 \times 0.9 \simeq 0.6$$

For the standard operating conditions, $T_i = 20°C$, $P_{fan} = 65$ W and $V_{mech} = 110\ m^3/h$, the heat supplied to the input air by the Bahco unit is

$$H_{hr} = 505-22T_o \text{ (W)}$$

6.2 Clarel Ete heat pump

This is an air to air heat pump, which is designed to be the heart unit of a full mechanical ventilation system. The unit incorporates an air to air heat pump and two ventilation fans, and is designed to be installed in the loft space. The manufacturer's performance data and a general description of the equipment is given in Appendix III. The system is not marketed in the U.K.

6.3 Thermal-Werke heat pump (West Germany)

This unit incorporates a cross flow heat exchanger and an air to air heat pump. The unit is designed to be installed as the heart unit of a full mechanical ventilation system. The manufacturer's performance data and a general description are given in Appendix IV. The manufacturer's literature differentiates between the heat supplied by the heat exchanger and that provided by the heat pump. The graph shown corresponds to a temperature efficiency and energy effectiveness of $\theta = \varepsilon = 0.68$. The temperature efficiency of the heat exchanger is similar to that of the Bahco unit; the energy effectiveness of the Bahco unit is lowered because of the reduced supply rate.

6.4 Genvex heat pump (Denmark)

After completion of the analysis in this report, information became available on a new system produced by Genvex. A description of the unit is given in Appendix V, but it has not been included in the main body of the report.

Performance data for all three systems is summarised in Table 2. It must be remembered that the data has been obtained from different sources, and has not necessarily been measured under identical conditions. The table includes the performance of the Thermal-Werke unit with the heat pump switched off; this allows direct assessment of the benefits of the heat pump.

Table 2. Performance data on VHR units

	Bahco Minimaster	Clarel Ete	Thermal-werke LWP1	
			Standard	(No heat pump)
Extract rate (m^3/h)	110	200	180	180
Fan Power (single) (W)	65	80	40	40
Total heat to inlet air (W)	$505-22T_o$	$2480-29T_o$	$1820-25T_o$	$896-41T_o$
Compressor input (W)	0	$640+6.4T_o$	$245+T_o$	0

Performance data is taken from manufacturer's figures, and is calculated for an inside temperature of 20°C.

6.5 Thermal properties of house

The performance of different heat reclaim systems is compared in this note assuming they are installed in a well insulated, detached house of about 80 m^2 floor area. The levels of insulation are taken to be that of the Capenhurst low energy house i.e. substantially better than that required by current Building Regulations. It is thought that it is only likely that an advanced ventilation system such as that described here would only be installed in a house of better thermal performance than average. The Capenhurst house project is described by Siviour and Millar (1985). Four examples of each house were built, and were fitted with a Bahco VHR unit to provide mechanical ventilation with heat recovery. All four occupants were very pleased with the mechanical ventilation system because of the quality of the internal environment it produced. They saw it as a benefit in its own right, quite apart from any financial savings to be made.

The thermal constants of the house are summarised in Table 3. Values are based on our experience of monitoring the performance of the houses over a period of a year. An adventitious ventilation rate of 0.2 ac/h has been assumed; this value implies good sealing of the house against air leakage. A well sealed house is a prerequisite of all mechanical ventilation systems.

Table 3. House data

House volume	200 m^3
Fabric heat loss coefficient	140 W/K
Adventitious ventilation at 0.2 ac/h	13 W/K
Solar gain constant	130 Wh per W/m^2
Internal free heat	15 kWh per day

7. SEASONAL ENERGY REQUIREMENTS

7.1 Energy consumption

Energy consumption and costs were calculated for the various VHR systems. The comparison was based on the Capenhurst low energy house, heated to a constant internal temperature of 20°C, for a 34 week heating season during the CIBSE example weather year. Table 4 shows a sample output from the computer program.

Table 4 Sample output table from program

```
House type  Capenhurst LOWEN
  Conduction loss                 HCON        (W/K)     153.0
  Ventilation rate                VENT       (m³/h)     180.0
  Gross ventilation loss          HVEN        (W/K)      59.4
  Solar gain coefficient          CSOL    (Wh/W/m²)     130.0
  Free heat input                QFREEM    (Wh/day)   15000.0


VHR type  Thermal-Werke LWP 1 (with heat pump)
  Total fan power  PFAN (W) =   80.0
  Compressor input  (W)     =  245.0+    1.0*TDB
  Output to house   (W)     = 1820.0+  -25.0*TDB


Internal average temperature (deg C)  = 20.0
Heating season runs from day  1 to 224
                    and from  0 to   0


Seasonal energy breakdown
  Gross heat requirement          YGR        (kWh)   14795.6
  Useful solar gain               YSOL       (kWh)    1919.5
  Useful free heat                YFREE      (kWh)    3328.5
  Space heating requirement       YSPACE     (kWh)    9547.6
  Supplementary heat              YHEAT      (kWh)    2320.7
  VHR  output                     YVHR       (kWh)    7226.9
  Compressor input                YCOMP      (kWh)    1081.8
  Compressor on-time              YTIM         (h)    4316.3
  Fan consumption                 YFAN       (kWh)     432.0


Running costs
  Low rate   2.04 (p/kWh)      Day rate  5.43 (p/kWh)
  0.95 of seasonal heating taken at low rate,  i.e. 2.21 (p/kWh)
  VHR compressor and fans costed at 29% low rate i.e. 4.44 (p/kWh)
Seasonal costs
  Supplementary heating             51.28
  Fans (continuous running)         19.19
  Heat pump compressor              48.05
```

The first set of computer runs was carried out with the VHR systems each running at their own normal ventilation rate. The Bahco Minimaster is set to a ventilation rate of 110 cubic metres per hour; this was found to provide satisfactory ventilation in the trials of the Capenhurst low energy houses. A second set of comparisons was made with all systems constrained to run at the same ventilation rate. This involves an increase in the ventilation rate and fan power of the Bahco unit, and a 90% derating of the Clarel Ete heat pump.

The results are given in Tables 5 and 6. In both cases the combination of heat exchanger and heat pump gives both the lowest running cost and the lowest energy consumption.

Table 5. Energy and cost comparisons between ventilation systems

Ventilation unit	Ventilation rate (m^3/h)	Seasonal Energy Consumption (kWh) Fans	Compressor	Heating	To[illegible]
Natural ventilation					
1 ac/h	160	0	0	9100	9[illegible]
½ ac/h	60	0	0	6870	6[illegible]
Bahco Minimaster	110	702	0	6180	6[illegible]
Clarel Ete	200	860	2590	1060	4[illegible]
Thermal Werke LWP1					
normal operation	180	430	1080	2320	3[illegible]
heat pump disconnected	180	430	0	6390	6[illegible]

Ventilation unit	Ventilation rate (m^3/h)	Seasonal Running Cost (£1985) Fans	Compressor	Heating	To[illegible]
Natural ventilation		0	0	201	[illegible]
(1 ac/h)	160				
(½ ac/h)	60	0	0	152	[illegible]
Bahco Minimaster	110	31	0	137	[illegible]
Clarel Ete	200	38	115	23	[illegible]
Thermal Werke LWP1					
normal operation	180	19	48	51	[illegible]
heat pump disconnected	180	19	0	141	[illegible]

Costs are for heating the Capenhurst Low Energy House to a steady 20°C for a 34 week heating season in the CIBS Example Weather Year.

Electricity tariff: Low rate 2.04p Day rate 5.43p/kWh
Space heating charged at 95% low rate.

An additional 40 m^3/h leakage ventilation is assumed in all cases.

Table 6. Energy and costs comparisons at constant ventilation rate

Ventilation unit	Ventilation rate (m^3/h)	Seasonal Energy Consumption (kWh) Fans	Compressor	Heating	Total
Natural	180	0	0	9550	9550
Bahco Minimaster	180	860	0	6710	7570
Clarel Ete (derated)	180	860	2400	1310	4570
Thermal Werke LWP1	180	430	1080	2320	3830

Ventilation unit	Ventilation rate (m^3/h)	Seasonal Running Cost (£1985) Fans	Compressor	Heating	Total
Natural	180	0	0	211	211
Bahco Minimaster	180	38	0	148	187
Clarel Ete (derated)	180	38	107	29	174
Thermal Werke LWP1	180	19	48	51	119

Costs are for heating the Capenhurst Low Energy House to a steady 20°C for a 34 week heating season. An additional 40 m^3/h leakage ventilation is assumed in all cases.

7.2 Cost of ventilation fans

The two ventilation fans are required to run night and day during the heating season, and the fan power consumption may represent a significant part of the total running cost. This is partly offset by the fact that the heat dissipated may contribute to the space heating. If the system has two fans, each rated at P_{fan} (W), then the total consumption over a 34 week season is

$$224 \times 2 \times P_{fan} \times 0.024 \text{ (kWh)}$$

With the normal fan configuration of the exhaust fan downstream of the heat exchanger and the inlet fan in the inlet air stream, half this amount will contribute towards the space heating load. However, energy to run the fans is costed at 4.44p/kWh, assuming 24 hour operation, while the space heating is costed at 2.21/kWh, assuming 96% of space heating is taken at low rate. The net seasonal running cost of a pair of fans, each rated at P_{fan} (W) is then shown to be

£0.358 P_{fan}

i.e. if the fan efficiency can be improved to maintain the same flow rate at lower power consumption, the net running cost saving over a season is £3.60 for every 10 watts reduction in fan power; this figure is independent of fan size or ventilation rate.

8. DISCUSSION

The comparison in the previous section shows that the combination of heat pump and heat exchanger shows clear benefits over either used alone, whether analysed in terms of energy consumption or running cost. The analysis is based on performance data obtained from different sources, which may not be strictly comparable. However, the general result is clear, which is that the combination of heat exchanger and heat pump offers substantial reductions in running costs.

The provision of the heat pump does of course require capital expenditure. It is difficult to give an exact figure of the extra cost of providing a heat pump in a VHR system. The different pieces of equipment considered have been produced in different countries at different times and some have been withdrawn from the market; this makes cost comparisions difficult. It would appear that provision of a heat pump and associated controls will add about £400 to the cost of a VHR system. This figure must be regarded as extremely crude, being the difference in price between the Thermal-Werke system in Germany and the Swedish Minimaster in England. One would expect the price to fall with mass production. Table 5 shows that one could expect an annual saving of about £50 in space heating costs from the provision of a heat pump system. In strictly financial terms this is barely worthwhile, and it is important to consider what other advantages and disadvantages may stem from the installation of a heat pump VHR system.

One advantage of the heat pump system is that the temperature of the inlet air is increased when compared with a heat exchanger alone, and this provides a means of heat distribution within the house. A well insulated house needs little heat in bedrooms or other minor rooms (e.g. study). However, if completely unheated the temperature of such rooms can fall below comfort levels in cold weather. The Capenhurst low energy houses included a source of heat in each room. However, the bedroom heaters were little used, and represent an inefficient use of resources. The use of the heat pump should make it possible to dispense with separate heaters in bedrooms completely, representing a worthwhile saving in space and money. The size of the main storage heaters may also be reduced. The system would however, reduce the possibility of room by room control.

There are disadvantages associated with the introduction of a relatively unfamiliar piece of equipment into the house. Points to be considered would be noise, maintenance and defrosting performance. It must be emphasised here that we have no practical experience of these units at ECRC, and there is no reason to suppose that these problems have not already been successfully overcome.

Heat pump VHR units have been manufactured for some time. Most are large units, designed for the ventilation of commercial and industrial buildings. Small units for domestic installation have also been introduced; however, the impression gained by talking to manufacturers is that these units have not sold well. The Clarel Ete, used in this report as an example of heat pump VHR without heat exchanger, has been withdrawn from the market. Bahco, the Swedish manufacturer of the Minimaster and a leading firm in Europe in ventilation equipment, introduced domestic heat pump ventilation heat recovery systems, all of which have been subsequently withdrawn.

The control of a heat pump VHR unit presents some problems. The ventilation fans of the system are required to run continuously during the heating season to provide house ventilation. Temperature control can then be simply achieved by switching the heat pump compressor on and off via a room thermostat. This method has two potential disadvantages. The temperature of the inlet air in the occupied rooms will change when the heat pump is switched. When the heat pump is turned off, the inlet air is still tempered by the heat exchanger, and will fall to about 12°C when the outside temperature is freezing. The design and location of the air inlets is therefore important, to avoid this change of air temperature being directly perceptible by the occupants. The rate of change of room temperature should only be low, since the gross heat output of the heat pump system is small compared with the design heat loss of the house. The other problem is that the reliability of a heat pump may be reduced by frequent switchings, and it is best to allow the pump to run continuously for long periods. Heat pumps with variable speed compressors are now being developed. This would offer improvements in control. Where a heat pump has a water cooled condenser, it is common practice to provide a buffer heat store, which can accept heat from the heat pump when it is not immediately required within the house. An equivalent system would be difficult to achieve with the type of air/air system described in this report, since the condenser is mounted directly in the air stream. A simple step would be to arrange the control system to ensure long on/off periods. Since the heat pump's output is relatively low, the rate of change of temperature within the house is unlikely to be great. This aspect needs to be investigated and quantified.

It is possible to improve the efficiency, but at the cost of complications. A system produced by the Danish firm of Hosby uses a heat pump with evaporator in the extract air stream, but with two condensers: one in the inlet air and the other in a buffer water store. When the house is not calling for heat, the refrigerant is diverted to the condenser in the water store, which is used to provide heat to a supplementary underfloor system. This seems an expensive addition to a heating system which only has a maximum heat output of about 1½ kW.

A VHR heat pump is not powerful enough to provide full heating on a design day, though in a well insulated house, such as the ECRC low energy design, it could provide heating down to an outside air temperature of about 6°C. Storage heaters would provide the additional heat necessary at lower outside temperatures. There is scope here for designing a control system which would optimise the interaction of the two heating systems. In the mid-range, for instance, the heat pump would remain off in the daytime, and be switched on in the evening as the storage heater output decreased.

A VHR heat pump system suitable for the ECRC design could provide about 40 kWh heat per day at an outside temperature of 6°C, rising to 44 kWh at -1°C. This compares directly with the 42 kWh storage capacity installed in the Capenhurst low energy house. This raises the possibility that it would be quite realistic to consider a system of heat pump VHR and direct acting heaters. This offers the attractions of a low installed cost and accurate and flexible control of temperature.

9. CONCLUSIONS

(i) The use of an air to air heat pump for ventilation heat recovery is effective in reducing energy consumption. As long as the coefficient of performance is greater than the ratio of day/night electricity costs, it produces running cost savings compared with storage heating.

(ii) The combination of heat exchanger and heat pump is better than heat pump alone. It offers greater running cost savings coupled with capital cost reductions, because of the reduced size of the heat pump. Running cost savings of up to £50 per year may be expected using a heat pump plus heat exchanger system, compared with a heat exchanger alone. This comparison is based on manufacturer's data, for equipment installed in an 80 m^2 low energy house operated over a standard heating season. The incremental cost of the heat pump is about £400; it is difficult to get an exact figure of the cost.

(iii) The use of a heat pump VHR system imposes additional constraints on the installation design. In particular, the system is designed for optimum performance at a single predetermined ventilation rate.

(iv) Use of a heat pump VHR system would improve heat distribution throughout the house, and should eliminate the need to provide heating equipment in bedrooms. It would have the concomitant disadvantage that individual room temperature control is made difficult.

(v) For a well insulated house, the heat pump VHR system could provide nearly half the design heat loss. It would then be realistic to provide most or all of the remaining heating with direct acting heaters, giving flexible, accurate control.

10. REFERENCES

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CALCULATION OF SEASONAL ENERGY REQUIREMENT

A computer program was written to calculate the seasonal energy required to maintain a constant inside temperature in a house. The energy is calculated day by day. Conduction lost through the fabric is proportional to the difference between the constant internal temperature and the 24 hour mean outside temperature. Any uncontrolled ventilation due to natural leakage is lumped in as part of the conduction loss; the ventilation part of the program deals entirely with air handled by the VHR system. The program treats the gross ventilation loss and the energy recovered separately, i.e. the VHR unit is considered as a heat source, not as a loss reducer. This avoids any confusion over percentage reductions in the loss.

The heat output of the VHR unit is a function of inside and outside temperature, together with the rate of air flow. Each seasonal computation is carried out for constant internal temperature and ventilation rate, and the heat output of the VHR unit is specified as a simple linear function of outside temperature

$$RVHR = RVHR\phi + XVHR * TDB \qquad (A.1)$$

This Appendix follows the notation of the program, to allow direct comparison.

This expression gives the total power delivered to the incoming air stream and includes any useful heat from the fans. Similarly the power input to the compressor of any heat pump fitted is a function of temperature

$$= PHP\phi + XPHP * TDB \qquad (A.2)$$

In general, the constants in equation (A.1 and A.2) and above are functions of internal temperature.

The program approximates the diurnal variation of outside temperature by assuming a sinusoidal variation where the amplitude of the swing is constant for each calendar month, and is taken from the published data summary for the CIBSE example year.

The internal free heat is assumed to be at a constant rate over the season of QFREE (W) which is the internal (i.e. non solar) input from such sources as people, hot water, lights, etc. This heat input is assumed to be continuous and constant from day to day. All the energy in QFREE is available for heating, i.e. QFREE is the available sensible heat; this is less than the gross heat production in the house, since allowances have been made for any latent heat lost in cooking, water down drains, etc.

Weather data is available for daily insolation on a horizontal surface, expressed as an equivalent average continuous 24 hour irradiance SOL (W/m^2). It is assumed that the gross internal solar heat gain Q_{SOLM} (Wh) to the house is proportional to the daily insolation

$$Q_{SOLM} = CSOL \times SOL$$

The solar constant of the house CSOL is a function of house design, orientation and shading. It increases with window size, and decreases with better insulation or double glazing. Measurements on the ECRC test houses (Siviour 1977) taken at different times of the year show the solar constants for the houses to be about 165 (see Figure A1.1). This is equivalent to Siviour's observation that the solar gain in a house is equivalent to the total insolation over a horizontal area of about 7 m^2. The Capenhurst low energy houses have very good insulation, have moderate sized windows which are double glazed. We therefore assume a solar gain factor of 130 for these houses; this figure has not been checked, but gives results consistent with experience.

The computer program evaluates energy requirement for each day of the year. On days when the free heat gains exceed the gross heating requirement no space heating is required. Any excess solar and internal gains are reduced in the same proportion, to give the total of useful gains over the season. On cold days, the VHR unit is run 24 hours per day and the supplementary heating system makes up the required space heating load. On milder days when no supplementary heating is used, the VHR heat

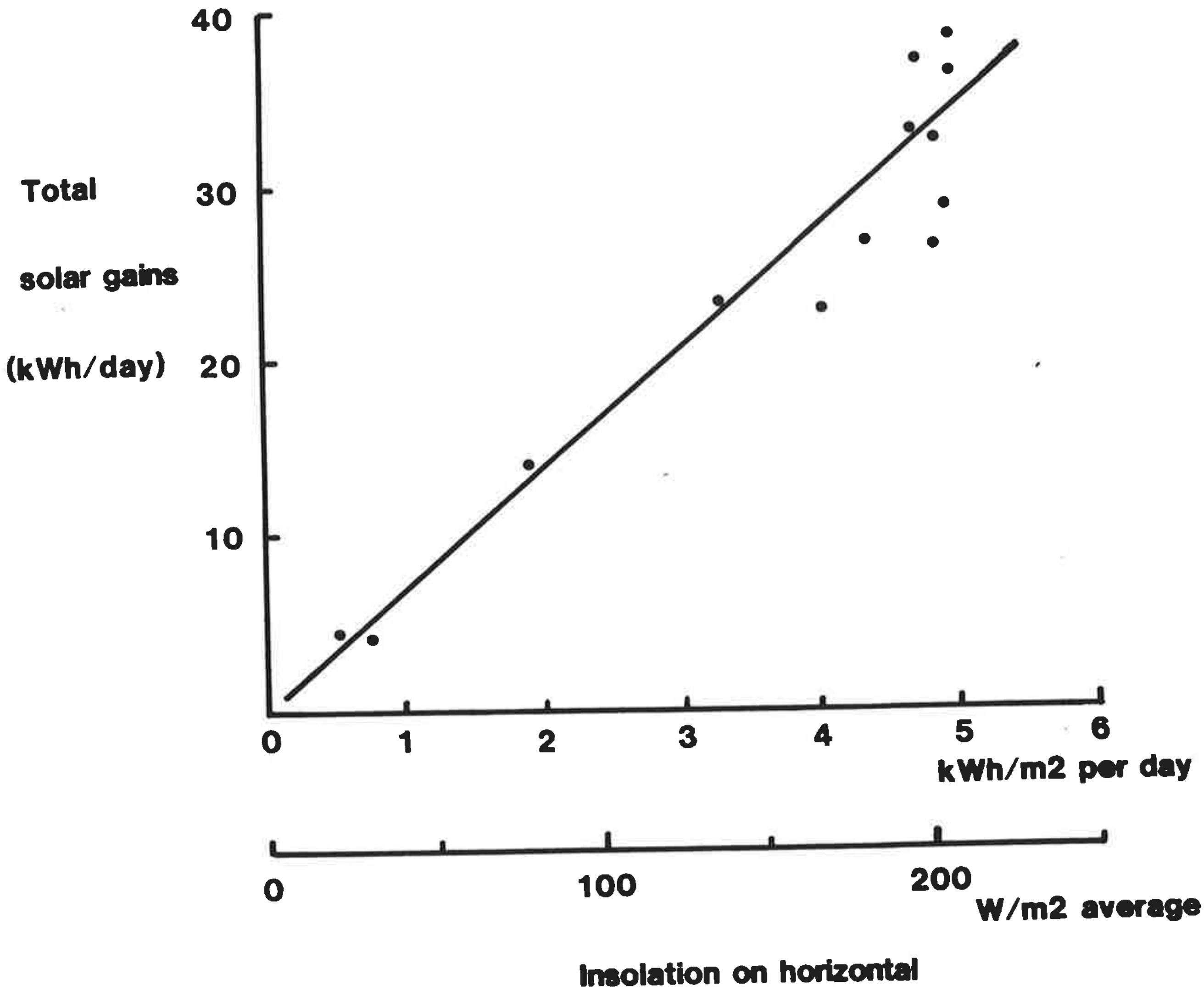


Figure A1.1 Relation between solar gains and insolation. Each point is an average of 10-15 days measurement in a Capenhurst test house. Slope of line CSOL = 165 $Wh/W/m^2$. Data from ECRC/M1070.

pump may not be required for the full 24 hours. It is assumed that the pump is progressively turned off during the warmest part of the day. All runs have been made using the CIBSE example weather year (Holmes & Hitchin, 1978). This is an actual year (October 1964 to September 1965, measured at Kew) chosen as a typical year. The use of a standard year allows comparisons to be made based on common data, and the use of an actual year means that the weather may be treated to any desired level of complexity. In this treatment we use daily mean air temperatures and daily mean insolations on a horizontal surface, taken from results published by Letherman & Wai (1980 and 1981).

The present method, using actual weather data, has an advantage in that it links sunshine and air temperature day by day, allowing a sensible estimate of the usefulness of solar gains. The utilisation of solar gains varies over the year, being highest in the winter, when all solar gains are useful, down to zero in summer when no heating is required. The beginning and end of the heating season in spring and autumn experience variable weather, and solar gain estimates based on average figures for sunshine and temperature can give rise to large errors. Basing the heating requirement on actual daily measurements of temperature and sunshine effectively eliminates the error. Seasonal space heating requirements were calculated for two houses: one a very well insulated house fitted with VHR, and the other a house built to current building Regulations. Figure A1.2 shows the space heating consumption as a function of the length of the heating seaon. It can be seen that the calculated energy consumption is insensitive to the length of the calculation period, showing that the program is effective in rejecting excessive solar gains.

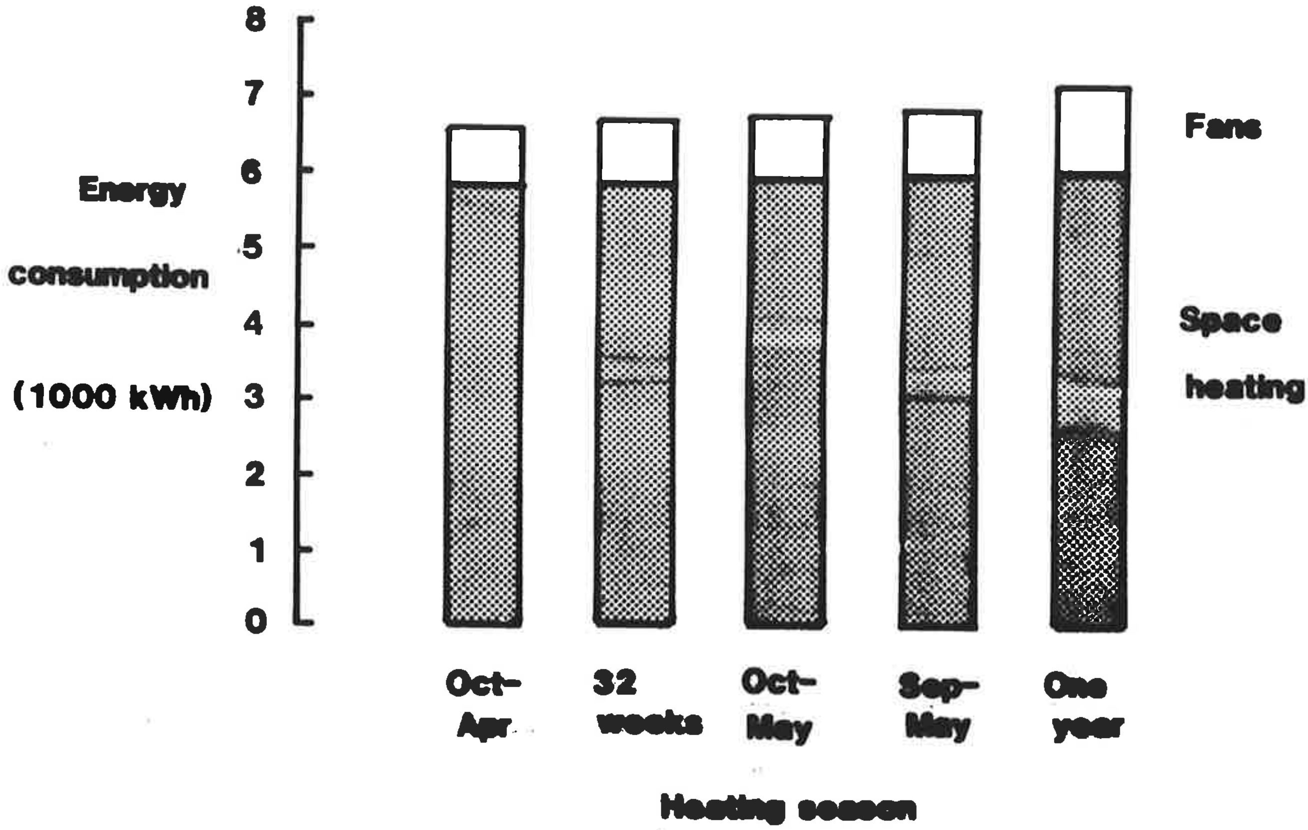
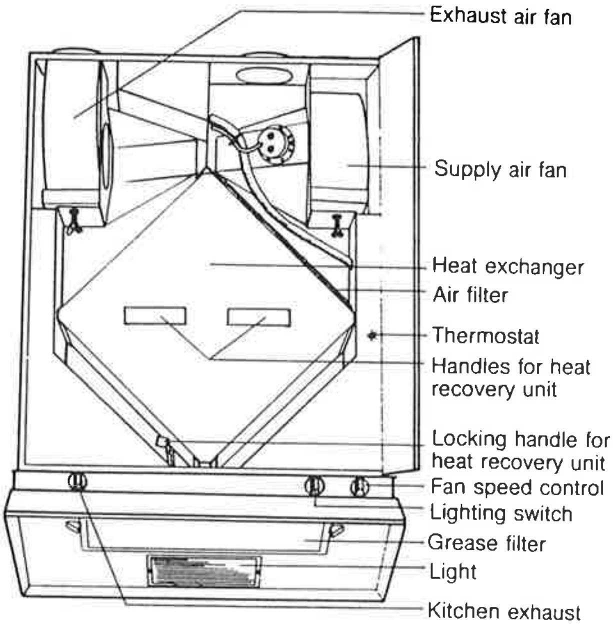


Figure A1.2 Computed energy consumption. The space heating consumption is insensitive to the choice of length of season; this shows that the program is effective in rejecting excess solar gains.

Calculation is for the Capenhurst low energy house, fitted with a Bahco Minimaster VHR unit.

Bahco Minimaster (Sweden)



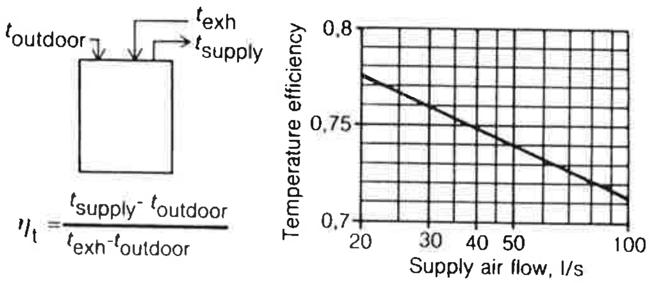
General view of unit, which is mounted over the kitchen cooker.

Technical specification

Voltage	220 V, single-phase
Power demand	1340 W
Rated current (amperage)	7 A
Motor power for boosted ventilation	105 W per motor
Motor power for normal speed	45 - 85 W per motor
Electric reheated rating	1000 W
Cooker lighting (normal bulb)	100 W max.

Heat exchanger

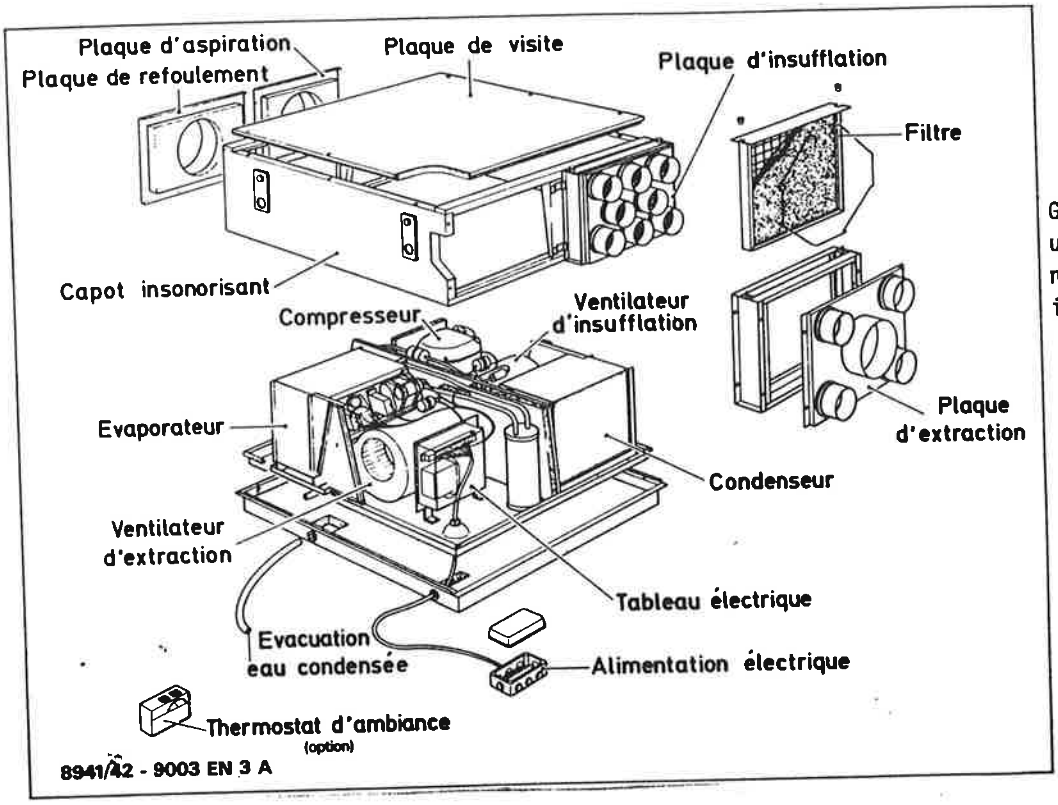
The capacity graph is given for normal design conditions, i.e. supply air flow = 0.9 exhaust air flow.



Performance specification. The temperature efficiency η_t in the diagram is denoted θ_{sup} in this memo.

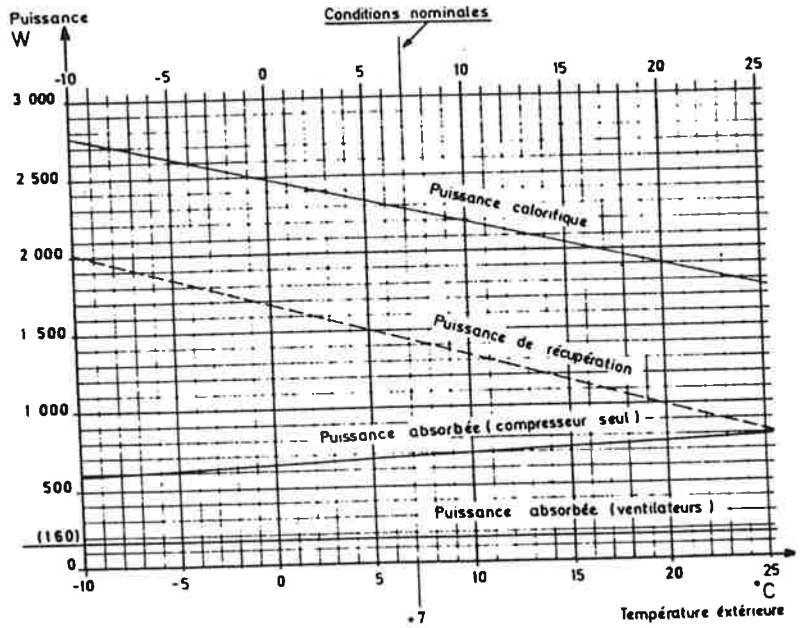
Bahco Minimaster. Above data is reproduced from the manufacturer's literature. Approximate price in UK is £500 (1985).

Clarel Ete (France)



General view of unit, which is normally installed in the loft.

Power



Outside temperature

Performance curves, at a ventilation rate of 200m³/h and T_i = 20°C.
P_{fan} = 80W.

The Clarel Ete RAE 1500 heat pump. Data reproduced from manufacturer's literature. The unit is no longer manufactured.

Thermal-Werke LWP-1 (Germany)

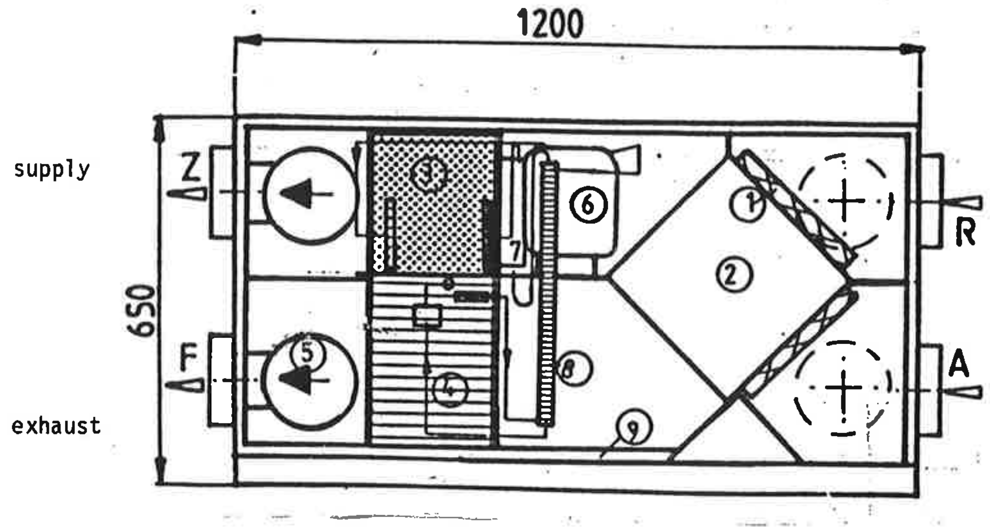
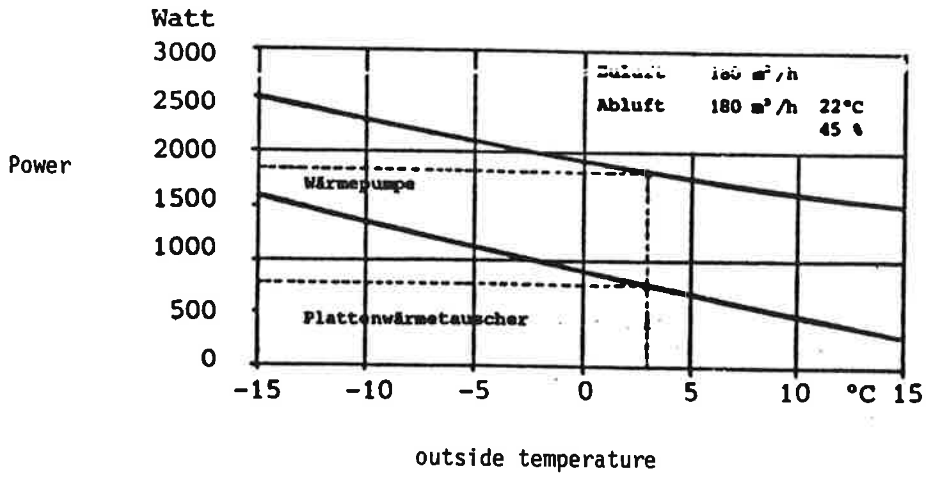


Diagram of unit, which is normally installed in loft.

1	Filter	2	Plate heat exchanger
2	Condenser	4	Evaporator
5	Fan	6	Compressor

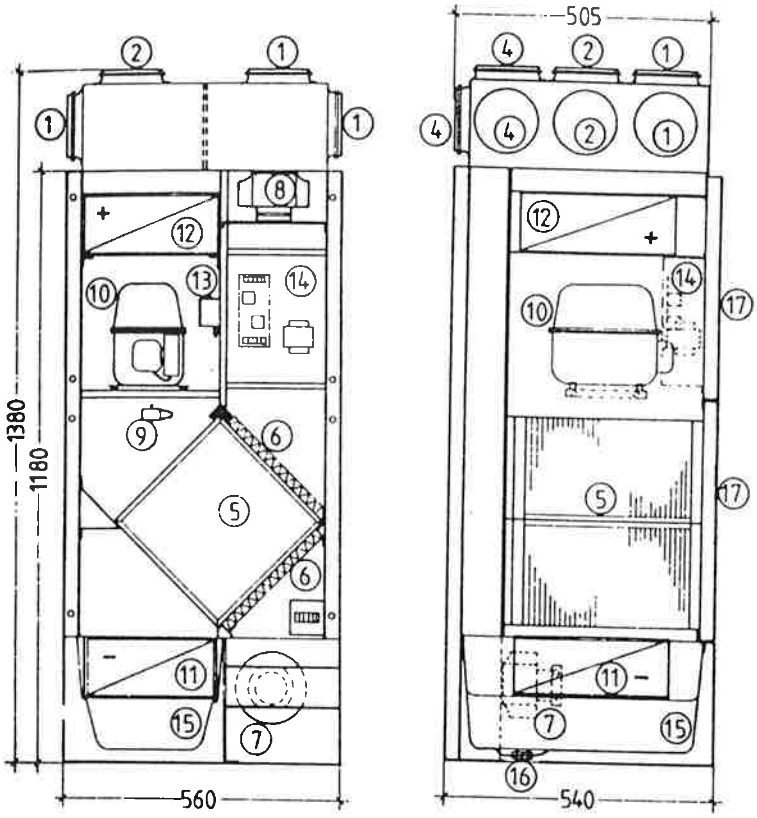


Performance curves, at

V_{mech} = 180m³/h
T_i = 22°C
P_{fan} = 40W

The Thermal-Werke LWP-1 heat pump with heat exchanger. Data is reproduced from manufacturer's literature. Approximate price is DM3,500 i.e. £920 (1985).

The Genvex VHR system. The central unit is designed to be mounted in a cupboard



Key:
1-4 Duct connections
5 Cross flow heat exchanger
6 Filter
7 Intake fan
8 Extract fan
9 De-icing thermostat
10 Compressor
11 Evaporator
12 Condenser
13,14 Controls
15,16 Condensation tray and drain
17 Double skinned front panel

Manufacturer's data		207i	215i
Ventilation rate (m^3/h)	min	110	150
	max	240	240
Fan consumption (W)	min	35	35
	max	63	63
Compressor consumption (W)	min	200	380
	max	280	600
Performance (c.f. Table 2)			
at $T_i = 20°C$ and ventilation rate:		150	180
Total heat to inlet air (W)		$1450-25T_o$	$1950-20T_o$