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Summary A ventilation heat recovery unit was installed in an Electricity Council Research Centre test house. The unit consists of a cross flow heat exchanger augmented by a small heat pump. It supplies warm air to all inhabited rooms, while extracting air from kitchen and bathroom. Trials were run during the winter of 1986/87. The unit provided a ventilation air flow of 180 $m^3 h^{-1}$, and a gross heat input to the house of 1.7 kW; this figure is averaged over time, and includes the effect of defrost periods. An overall coefficent of performance of 3 was achieved. Such a system can provide the greater part of the heating requirement of a low-energy house, and has its own advantages over other heating systems.

Domestic ventilation unit with heat exchanger and heat pump

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List of symbols

C	Specific heat of air at constant pressure (J kg^{-1} K^{-1})
m_{in}	Supply air mass flow rate (kg s^{-1})
m_{out}	Exhaust air mass flow rate (kg s^{-1})
m_{mech}	Greater of m_{out} and m_{in} (kg s^{-1})
P_{fan}	Power consumption of one fan (W)
P_{comp}	Power consumption of heat pump (W)
P_m	Power consumption of auxiliary 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)
H_w	Heat loss from VHR unit (W)
V_{mech}	Mechanical ventilation rate at house temperature (m^3 s^{-1})
ε	Energy effectiveness of heat exchanger
η_s	Seasonal COP of system
η_{hp}	COP of heat pump
η_{hr}	COP of whole system
θ	Temperature efficiency of heat exchanger
T_{sup}	Supply air temperature external to VHR unit (°C)
T_{exh}	Exhaust air temperature external to VHR unit (°C)
T_{int}	Intake air temperature external to VHR unit (°C)
T_{extr}	Extract air temperature (°C)
T_1, T_2, T_3, T_4	Heat exchanger temperatures (°C)
t_{df}	Defrost duration (h day^{-1})

1 Introduction

Full-house mechanical ventilation with heat recovery (VHR) is increasingly being adopted as part of the specification of a low-energy house in the UK. 'Full house' implies that fresh air is supplied to all living rooms and bedrooms. Heat is recovered from exhaust air and transferred to the incoming air by means of a heat exchanger. By this means, positive ventilation is provided at a controlled rate, and ventilation heat loss is reduced. Some 16 out of the 35 house types displayed at the Milton Keynes Energy World exhibition in 1986 incorporated VHR, and the system has been adopted as standard for the Electricity Council's low energy house. Energy monitoring has demonstrated its effectiveness, and interviews with occupants have elicited that the ventilation system is regarded as a positive asset, contributing to a better environment in the house as well as saving energy(1).

It is possible to augment the system by incorporating a small heat pump. This extracts heat from the exhaust air downstream of the heat exchanger, and transfers the heat to the supply air. Air is rejected to outside at a temperature below ambient, and so the system makes a positive contribution to the heating of the house. This paper analyses the energy balance of such a unit, and reports trials of a Danish unit in an ECRC test house.

2 Principle of operation

A ventilation heat recovery system consists of intake and extract fans with their associated ductwork, and a heat exchanger to transfer heat from extract to inlet air stream. The fans and heat exchanger are usually mounted together in a single unit, which may be mounted in the loft space of the house, over the cooker or in some other convenient place. Several types of heat recovery system are available, most of which have found application in domestic use. Extensive surveys are given in References 2 and 3.

The plate heat exchanger is probably the most common form used in domestic VHR. The heat exchanger consists of a series of thin metal plates separated by small gaps. The two air flows pass through adjacent gaps, separated by a plate through which heat transfers by conduction. The exchanger is normally configured to give a cross flow operation, and temperature efficiencies of up to 70% are possible.

The performance may be augmented by the incorporation of a heat pump. The evaporator of the heat 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 COPs can be obtained. Since the heat pump only handles part of the total heat exchange, it can be relatively small. Such systems have been introduced by some manufacturers, but are not yet common in domestic use. Background information on the systems may be found in References 4 and 5.

3 Performance of a VHR system

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. These figures do not necessarily give the information required to calculate the energy consumption of the house. The approach adopted here is to consider the energy recovery of the system as a separate energy input to the house, treated independently of the ventilation loss. This approach avoids confusion, and is particularly helpful where the incoming and outgoing air flows in the system are not equal.

This paper considers the energy exchanges entirely in terms of sensible heat. There is no point in reclaiming moisture in domestic dwellings, since one of the primary functions of ventilation is the removal of moisture. Reclaim of latent heat by condensation improves the efficiency of the system, and happens with a heat pump VHR system, and in cold weather with a heat exchanger.

Figure 1 shows heat flows and temperatures in a schematic heat pump VHR unit. The placing of fans affects the analysis, since this determines whether heat dissipation in 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

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

and so a 100% efficient heat exchanger would deliver incoming air at a temperature $T_2 = T_3$, i.e. at a temperature equal to the extract temperature. The efficiency θ is in general a function of flow rate, with the efficiency rising as the flow rate through the heat exchanger is reduced.

The mass flows through the two halves of the heat exchanger are not necessarily equal, and so the temperature efficiency 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 therefore defined as

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

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

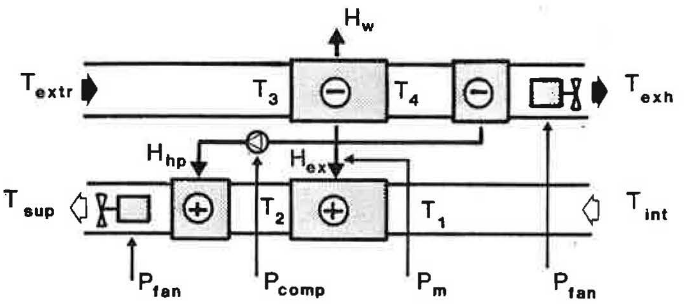

Figure 1 Temperatures T, heat flows H and power inputs P for a ventilation unit incorporating heat exchanger and heat pump

where m_{in} and m_{out} are the intake and exhaust mass flow rates.

This equation assumes that the specific heat of air at constant pressure is essentially constant over the temperature ranges involved. 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 take this into account.

The temperature change in the incoming air is related to the heat transfer in the exchanger by

$$H_{\text{ex}} = m_{\text{in}}\, C(T_2 - T_1)$$

The heat delivered by the heat pump H_{hp} 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, the heat exchanger and any useful fan power:

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

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

$$H_{\text{hr}} = m_{\text{in}}\, C(T_{\text{sup}} - T_{\text{int}})$$

The overall coefficient of performance of the system is

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

where P_{hr} is the total power consumption $2P_{\text{fan}} + P_{\text{comp}} + P_{\text{m}}$. For a plate heat exchanger $P_{\text{m}} = 0$, but some devices may consume additional power, e.g. a heat wheel regenerator. This COP is highly dependent on outside temperature, and quoting a single value measured at design conditions may give a misleadingly high impression.

Any practical heat pump system will require defrosting when ice builds up in the evaporator. The most common method is to divert hot gas from the compressor to the evaporator, while simultaneously turning off the intake fan. The defrost operation reduces the COP, and hence increases the cost per unit of heat delivered to the house, as well as reducing the net heat output. The most useful is the concept of seasonal coefficient of performance

$$\eta_{\text{s}} = \Sigma H_{\text{hr}}/\Sigma P_{\text{hr}}$$

where the summations are made over the whole heating season and include the effect of defrost.

The above treatment has shown that the use of 'efficiency' alone as a descriptor of performance is incomplete. It is better to describe performance in an absolute form. Three quantities must be given:

(*a*) The ventilation rate. Mechanical ventilation rate is denoted by the mass flow rate m_{mech}, which is the higher of the supply and extract rates. Some manufacturers recommend an extract rate 10% higher than the supply rate. The difference between the two is made up by leakage induced by the fan pressure. It is more common to give the ventilation rate as the volumetric flow rate V_{mech} ($\text{m}^3\,\text{s}^{-1}$), where the volume is measured at internal house temperature. As before, the flow rate is taken as the higher of inlet and extract flows.

(*b*) The total heat delivered to the supply air by the VHR system. This includes heat from all sources, i.e. reclaimed heat, heat pump plus any useful heat dissipated from fans or compressors. This will be a function of both inside and outside temperatures.

(*c*) The total electrical power consumption of the system, including all fans, motors and compressors:

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

The power consumption may be a function of temperature, particularly for the compressor.

4 Service trials

4.1 Equipment

A Danish heat pump ventilation unit was bought and installed in one of the Capenhurst test houses. The object of the trial was to gain experience of a commercial unit operated in a practical manner, and to investigate its role in an integrated heating system. The unit is self contained and floor standing, designed to be installed in a utility room or cupboard. Its overall dimensions are 560 × 540 × 1180 mm, with the four 125 mm diameter duct connections in the top of the unit. The mode of operation is made clear by Figure 2. Air is extracted from kitchen and bathroom, plus any other WC or utility room by the extract fan (1). The air passes through a filter (2) and through a cross flow heat exchanger (3), where it gives up some of its heat to the incoming fresh air. The outgoing air then passes over the heat pump evaporator (4), where it is cooled still further before being rejected to outside. Fresh air is brought in from the outside by the fan (5). After being filtered (6), it passes through the heat exchanger. The air then passes over the compressor (8) and the condenser (7).

The cooling of the exhaust air produces condensation in the heat exchanger and icing in the evaporator, and a condensate drain must be provided. On defrost the inlet fan is turned off, and the extract fan runs at full speed, blowing air at house temperature through the exchanger and over the evaporator. Simultaneously a magnetic valve directs hot gas from the compressor through the evaporator. Each defrost period lasts about 15 min and occurs several times a day depending on conditions.

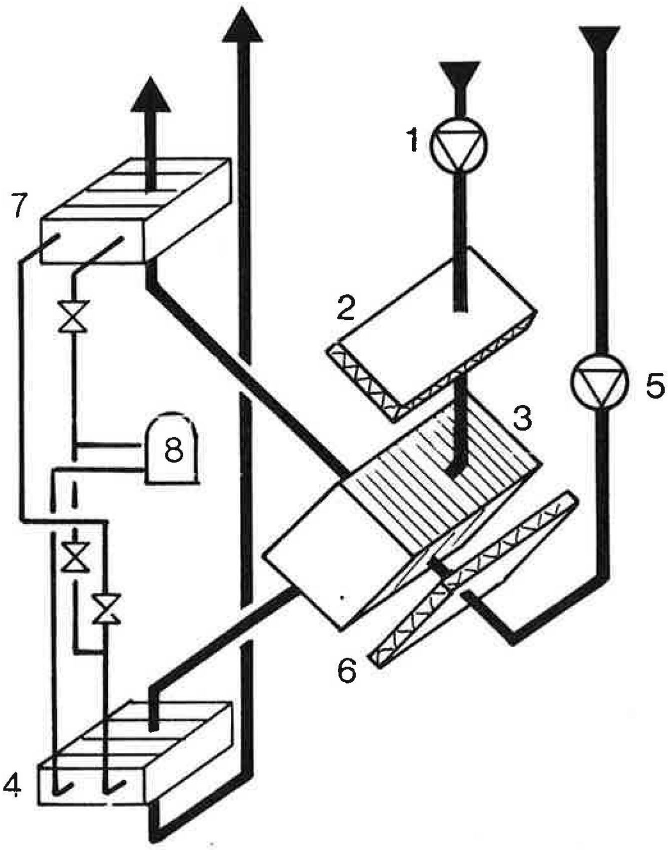

Figure 2 Working principle of the VHR unit on trial. Key: 1 Extract fan; 2 Filter; 3 Heat exchanger; 4 Evaporator; 5 Intake fan; 6 Filter; 7 Condenser; 8 Compressor.

The available control on the unit is compressor on/off and fan speed high/low. Since the unit provides necessary ventilation to the house, the fans are designed to run continuously. The manufacturers recommend that the compressor run under the control of a room thermostat. Since the heat pump provides heat at low unit cost, the thermostat should be installed so that the unit provides the base heating load, and also so that the compressor does not undergo frequent on/off cycles. In this trial, the thermostat was positioned in the hall. This position provides a stable temperature and so reduces the frequency of cycling of the compressor. The thermostat should be of modest differential (1 or 2 K) and have no accelerator fitted.

4.2 Installation

The unit was installed in a Capenhurst test house, semi-detached and of 90 m^2 floor area. The level of insulation is substantially better than the 1985 Building Regulations standard, and is summarised in Table 1. The house was pressure tested and was found to have 10 air changes per hour at a pressure difference of 50 Pa. This flow rate was achieved after all accessible and obvious leaks had been sealed with tape. This leakage is better than average for a conventional house, but worse than for a modern house constructed with airtightness in mind.

The installation in the test house followed the layout of a real system, except that all ductwork was left exposed for accessibility. The unit was installed in the kitchen, with the intake and exhaust ducts running to a balanced terminal in a nearby wall. Warm air was supplied to each room using a wall supply grille mounted over the doorway, except in the dining room, where it was mounted on a side wall. A cooker hood was fitted in the kitchen, and a wall-mounted extract in the bathroom. All ductwork was inside the envelope of the house. The extract ductwork is uninsulated. The supply ductwork was initially uninsulated, and the effect of subsequent insulation is treated below.

Air flow rates were measured near the main unit in the supply and extract ducts, using a pair of flow measuring nozzle sections. The system was set to give equal supply and extract flows. The supply air was adjusted to give roughly equal volume flows upstairs and downstairs by means of a butterfly valve in the main supply duct.

Table 1 Heat loss at 16 Manorfield Close

House volume	200 m^3
Infiltration rate (say); hence loss	0.2 air changes h^{-1} 0.3 kWh K^{-1} day^{-1}
Fabric loss	2.6 kWh K^{-1} day^{-1} (ECRC/N1537)
Mechanical ventilation; hence loss	180 m^3 h^{-1} 1.4 kWh K^{-1} day^{-1}
Total heat loss	4.3 kWh K^{-1} day^{-1}
i.e. heat loss coefficient	180 W K^{-1}

Free heat simulation was provided in the house using tungsten filament lamps controlled by a time clock, plus a clock controlled radiator in the kitchen. The free heat schedule is intended to represent a typical day's activities. The total input per day is 15 kWh, distributed as follows: lounge 2.5 kWh, dining room 3 kWh, kitchen 6.9 kWh and hall 2.6 kWh. Auxiliary heating was provided in lounge and dining room between the hours of 0700 and 2400 by means of thermostatically controlled fan heaters set to 20°C. Some humidification was provided by a spinning disc evaporative humidifier mounted on the landing. This was connected to the mains water supply, and operated continuously, controlled by a hygrostat set to 60% RH.

Comprehensive data logging was carried out. Temperatures were measured in all rooms together with supply temperatures at each inlet. Temperatures of air entering and leaving the VHR unit were measured with a thermocouple centred in the duct 200 mm above the top of the unit. The dewpoint of the extract air was measured by a lithium chloride dewpoint probe. Air flow was derived from pressure drop across a measurement nozzle. Both supply and extract flow rates could be measured, but only the extract rate was routinely logged. Energy consumption for the VHR unit and whole house were measured separately.

5 Results

The auxiliary heating in the test maintained the two living rooms at constant temperature during the day, while other rooms had no form of heating other than warm air from the VHR unit. The mean house temperature therefore fell in cold weather. The supply temperature of the air leaving the unit is a strong function of outside temperature; see Figure 3. However, even in the coldest weather the supply does not fall below 20°C, i.e. the supply is always warm compared with the house temperature.

The temperature efficiency of the heat exchanger was measured by running the unit for a week with the compressor off, and heaters switched on continuously in kitchen and bathroom to elevate the extract temperature. Analysis of the results showed that the heat exchanger has a temperature efficiency of just over 50%, after allowing for the temperature rise in the supply air produced by the intake fan. At first sight this seems somewhat low compared with other claims.

Table 2 Analysis of results

Temperature rise through unit† (K)	$15.8-0.225\ T_{int}+0.736\ T_{extr}$
Power consumption of unit† (W)	Total $P_{hr}=575+5.41\ T_{int}$ One fan $P_{fan}=85$
Exhaust temperature† (°C)	$T_{exh}=-7.4+0.439\ T_{int}+0.380\ T_{extr}$
Defrost duration‡ (h day^{-1})	$t_{df}=1.86-0.165\ T_o\quad(-3<T_o<11.5)$

† Regression analysis of 6944 quarter-hour periods, defrost excluded, normal air flow rate.

‡ Analysis of 110 days with $(-3<T_o<14)$

However, the figure represents the temperature efficiency measured across the entire unit, rather than the heat exchanger itself. The temperatures were measured external to the unit, and so the air had to traverse the entire unit, including the switched-off condenser and evaporator coils. There is presumably some degradation of performance because of heat loss to the internal equipment and to the casing of the unit. The measured performance is in line with the manufacturer's claims.

Analysis of heat pump operation is complicated by the defrost cycle. Logging took place every 15 min, and the analysis of heat pump performance is based on an analysis of some 7000 quarter hour periods in which no defrost occurred. The temperature rise of the air as it goes through the unit gives a measure of the power input to the supply air. This rise is a function of both extract (i.e. house) and intake (outside) air temperatures. Over the range of conditions experienced in this test, it is quite adequate to express the results as a linear regression. The results of the analyses are summarised in Table 2. The power drawn by the compressor increases with intake temperature, and is shown in Table 2, together with the exhaust temperature as a function of intake and extract temperatures. Note that the exhaust temperature is below the intake temperature.

Defrost operations were recorded directly on an event channel, by monitoring operation of the internal defrost thermostat. Frequency of defrosting increased with lowered outside temperature; see Figure 4. The mean duration of each defrost episode was 14 min; this increased with increas-

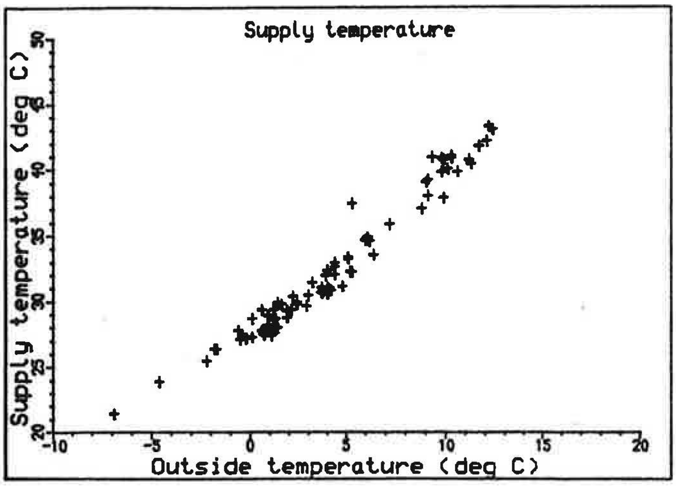

Figure 3 Supply temperature as a function of outside temperature. Each point is a 24 h average, and is uncorrected for the effects of defrosting or internal temperature.

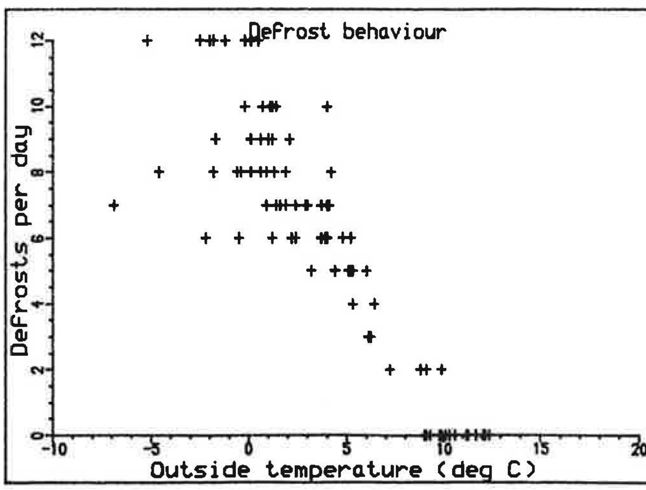

Figure 4 Frequency of defrost operation as a function of outside air temperature

Table 3 Heating performance

Continuous operation i.e. defrost excluded; $T_{extr} = 20°C$, $V = 180\ m^3\ h^{-1}$	Whole unit: Heat exchanger:	$H_{hr} = 1830-13.5\ T_{int}$ (W) $H_{ex} = 670-30.6\ T_{int}$ (W)
Coefficient of performance	Whole unit: Heat pump:	$\eta_{hr} = 3.18-0.049\ T_{int}$ $\eta_{hp} = (1160+17.1\ T_{int})/(400+5.41\ T_{int})$ $= 2.90+0.003\ T_{int}$
Average performance including effects of defrost ($T_o < 115°C$)	Average heat output over day (W)	$H_{av} = 1685$
	Average power consumption (W)	$P_{av} = 565+6.8\ T_o$
	Coefficient of performance	$\eta = 2.99-0.034\ T_o$

ing ambient temperature. The total defrost duration over 24 h as a function of outside temperature is shown in Table 2. Most of the test was conducted with the fan on high speed. Part of the test was carried out with the fan at low speed, corresponding to an air flow of $130\ m^3\ h^{-1}$. The defrost system was unsatisfactory at low outside temperatures when the fan was at low speed. At a mean ouside temperature of $-5°C$ with the fan at low speed, the unit defrosted 58 times in a 24 h period, with a mean defrost duration of only 4.6 min.

The effect of defrosting is to reduce the performance. During defrost, the intake fan is switched off; however, since the extract fan continues to run, cold oustide air is drawn into the house by other routes, and the ventilation heat loss of the house is unchanged. There is no heat input to the house during defrost; so the mean rate of heat input is reduced in proportion to the total defrost duration. The mean power consumption is reduced by the power of one fan, since the compressor continues to run during defrost. The coefficient of performance of the unit is therefore reduced. The heat output of the unit was estimated from the measured temperature rise across the unit, the intake flow rate and the specific heat of air. This was done for defrost-free periods and then was recalculated to give an average value over the day allowing for defrost duration. The results are shown in Table 3.

Figure 5 shows the heating performance of the unit as a function of outside air temperature. The results have been standardised to an internal temperature of 20°C and a volume flow rate of $180\ m^3\ h^{-1}$; the manufacturer's data are quoted under these conditions. The measured lines shown are obtained from the linear regressions of measured data. The trial figures for total heat output are some 6% below the manufacturer's figures. This is within experimental error, and can be considered good agreement. The coefficient of performance is shown in Figure 6. The overall COP of the system increases with falling ambient temperature, as the heat exchanger delivers more heat with no associated increase in power consumption. The COP of the heat pump itself increases with rising ambient temperature. The COP is rather below the manufacturer's figure. The measured performance of the unit is summarised in Table 3; where the quantity is a function of temperature, linear approximations have been used. The measured heat output is below the manufacturer's value, while the compressor power consumption is at the top of the range. Each ventilator fan had a power consumption of 85 W, compared with a maximum of 63 W quoted by the

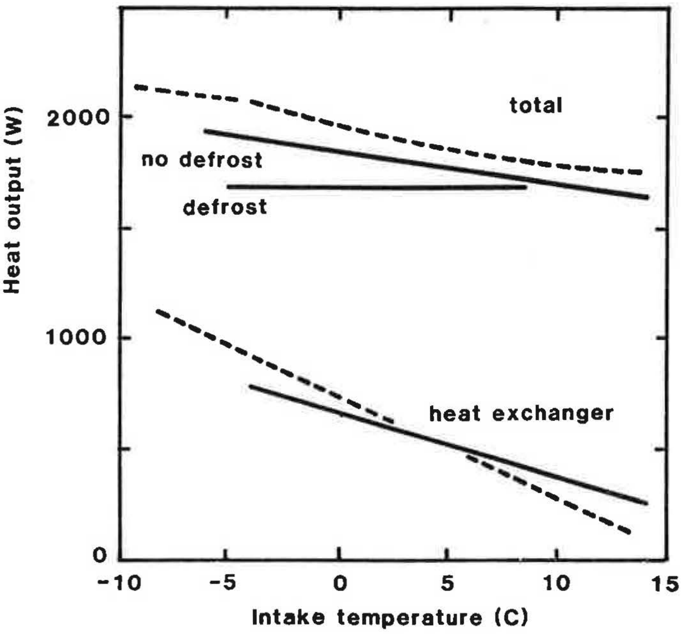

Figure 5 Heat output of unit as a function of intake temperature, normalised to a house extract temperature of 20°C and a flow rate of 180 $m^3\ h^{-1}$. Broken curve manufacturer's data, full curve measurements.

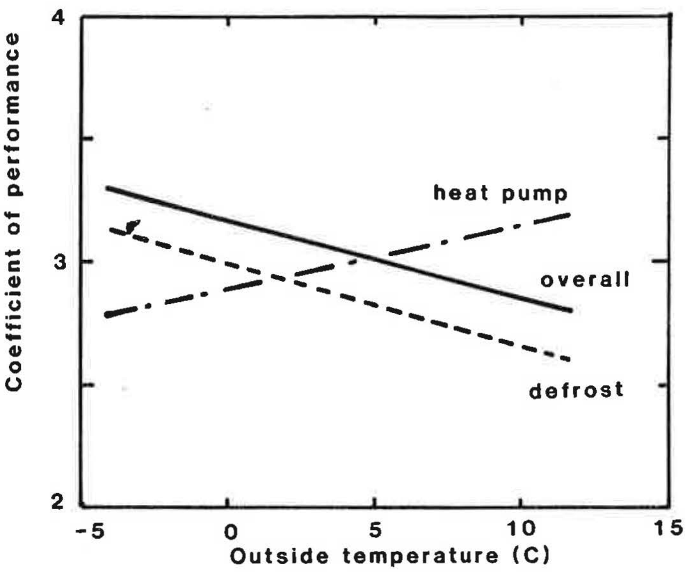

Figure 6 Coefficient of performance of the unit as a function of intake temperature, normalised to a house temperature of 20°C and a flow rate of 180 $m^3\ h^{-1}$. COP is shown for the heat pump alone, and for the overall system, both with and without the effects of defrost operation.

manufacturer. The differences are explained by the fact that the unit is designed to run at 220 V, but was run directly from the UK mains at a nominal 240 V.

6 Heat distribution

The warm air supply was conducted to the inhabited rooms through uninsulated ductwork suspended below the ceiling. Heat loss from the ducting produces a temperature drop between the unit and room terminal. Towards the end of the experiment, ductwork to the bedrooms was insulated using 25 mm of mineral wool. The temperature drop was analysed for each terminal, both before and after insulation. The temperature drop is a function of inside and outside temperature, and was normalised to an extract temperature of 20°C and an outside temperature of 10°C. Before insulation, the dining room register, which is the one closest to the unit, showed a temperature drop of about 2 K, while the remote bedroom terminals showed a drop of over 4 K. This was reduced to below 2 K by insulating the ducts. It is by no means obvious that this reduction in temperature drop is worth the cost and space penalty of insulation. Heat loss from the ducting does not represent a heat loss from the building, as long as the ductwork is installed inside the heated envelope. A room register typically handles an air supply of 35 $m^3 h^{-1}$, and a temperature drop of 2 K in the air supply represents a reduction of 25 W in the heat supply to that room.

Air is supplied to each room through a diffusing grille mounted over the door. This arrangement has considerable practical advantages, in that it keeps duct runs short and avoids the problem of running the ducts in the ceiling space. Air distribution was observed using a smoke puffer. With the compressor on, air was supplied at some 12 K above room temperature. A warm air stream moved steadily across the ceiling, fanning out from the register and losing speed on the way. Movement was still detectable at the far wall. There was no sign of any short circuiting, i.e. there was no sign of any warm supply air being extracted immediately through the doorway without having entered the room. Vertical temperature gradients in the room were small, and certainly no worse than would be expected with other heating systems. Horizontal gradients were small.

With the compressor off, the air entered some 3 K below room temperature. The air now falls as it enters the room. Air movement was just detectable on the face if one stood in front of the register and faced it; this was with an entry temperature of 17°C in a room at 20°C. In winter, with the compressor on, the entry air is warm, and no problem with draughts would be experienced. The intake fan is switched off during defrost. The compressor cycles on and off during mild weather when the inlet temperature may fall to about 16 or 17°C. No problems with draughts are anticipated.

7 Heating analysis

Figure 7 shows the weekly average temperature for a living room and bedroom; the two living rooms remained substantially constant at 20°C; there is a slight fall-off in cold weather, because the rooms were unheated during the night. The bedrooms were heated by the ventilation system only, and fell in temperature as the weather got colder. Mean bedroom temperatures of above 16°C were recorded for a week of mean temperature of 0°C. During this time the bedroom doors remained shut, and there was no source of free heat in the bedrooms. By comparison, the main bedrooms of the four ECRC low energy houses[6] had mean weekly bedroom temperatures between 12.5 and 16°C in a similarly cold week, even though the bedrooms were occupied, and the owners were free to add extra heating if they chose. The temperature distributions during these trials can be considered satisfactory, particularly the heat distribution to the bedrooms.

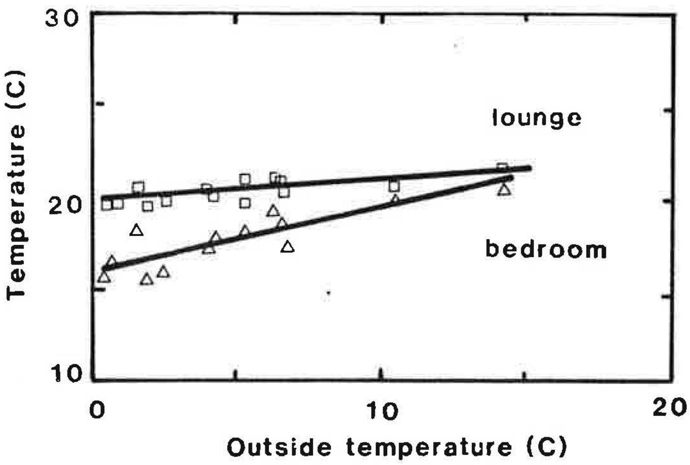

Figure 7 Weekly average room temperatures as a function of outside temperature

The heat supply to the house was partitioned into free heat, auxiliary heating from the two fan heaters, and heat supplied by the ventilation system. The analysis covered 84 days during the period 21 September 1986 to 26 April 1987 when the heating system was operating under the following conditions: ventilation rate 180 $m^3 h^{-1}$, lounge and dining room controlled at 20°C from 0900 to 2400; compressor controlled by hall thermostat at 18°C; free heat of 15 kWh day^{-1}.

Figure 8 shows a summary of the analysis. Results are presented directly, and are not corrected for the fall off in bedroom temperatures at low outside temperature.

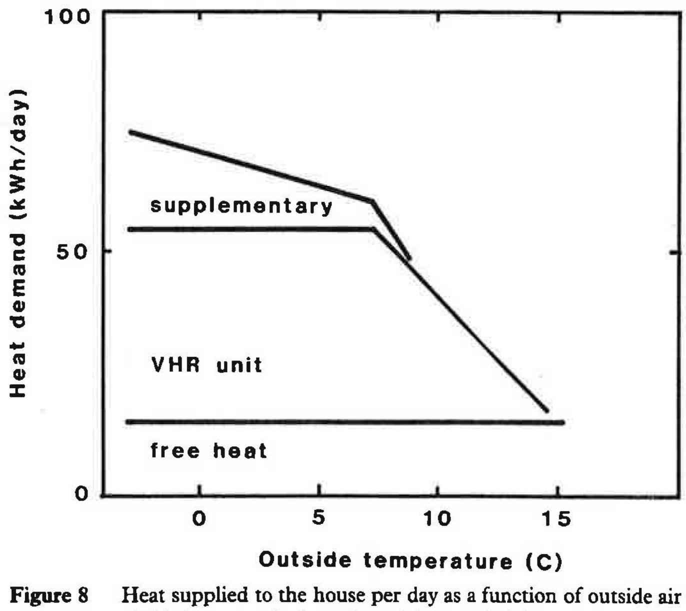

Figure 8 Heat supplied to the house per day as a function of outside air temperature. This is a smoothed version of the actual daily consumptions, and is not corrected for variations in internal temperature.

8 Discussion

The unit performed satisfactorily. The measured performance was only slightly below the manufacturer's figures, and the differences are of the same order as experimental error. Performance achieved in practice is, however, reduced by the defrost cycling of the unit. Nevertheless, the unit achieves a COP of about 3, including fan power and the effects of defrost. The unit thus achieves its objective of the provision of heat at a unit cost below that of off-peak electricity, and good distribution of warm air throughout the house. Energy analysis shows that when installed in a well insulated house, the VHR unit can provide a substantial part of the heat requirement over the year, giving a reduction in running costs and benefits of good heat distribution.

Fan power consumption was relatively high at 85 W per fan; this is partly explained by running at 240 V instead of 220 V. A 25 W reduction per fan consumption would reduce the seasonal cost by about £13.

The main unit is free-standing, and designed to be mounted in a cupboard. The unit has good sound insulation, but nevertheless would be best mounted and installed in a separate utility room. No silencers were installed in the test house trial. They would normally be fitted in both supply and extract ducts; this has been standard practice when installing VHR units in the Capenhurst low energy house.

Routing of ductwork through the house is always a problem, as has been found with the VHR system in the Capenhurst low energy house. This trial showed that inlets mounted over a door are satisfactory, and offer simple installation. Unless the duct runs are excessively long, it is acceptable to use uninsulated ducts. There is a slight drop in supply air temperature, but this does not represent a heat loss to the house. Any ductwork which passes through an unheated space such as the loft must of course be well insulated. Both the provision of a condensate drain and routing of duct work must be fully planned at the design stage.

The energy analysis in this paper has treated the VHR unit as a heat supplier, i.e. both the heat delivered to the incoming air stream by the heat exchanger and the heat supplied by the heat pump are regarded as energy inputs to the house. Consequently the air extracted by the unit is treated as pure ventilation loss which is not reduced by the heat recovery. This treatment gives a clear analysis. It must of course be remembered that the ventilation loss is caused by the VHR unit itself. Thus a large unit, with higher ventilation rates, will deliver more heat to the house, but at the same time increase the ventilation heat loss. It is therefore essential to decide on the required ventilation rate of the house first, and then size the unit accordingly. This sets an upper limit on the air handling capacity of the unit. This in turn decides the size of the heat pump, since any attempt to oversize the heat pump will produce unattainably high condensing temperatures. It is therefore not possible to increase the power of a VHR unit at will, and in practice the system cannot provide whole-house heating over the whole season.

The same manufacturer produces a similar unit, but designed to be mounted in the roof space. The installation and performance of such a unit has been reported[7]. The unit was installed in a well insulated house near Mannheim, West Germany, where the design temperature is -12°C. The unit was judged satisfactory, though the overall COP was below 2, compared with 3 achieved in this trial. This is partly explained by a fall-off in performance at low temperatures, where the defrost behaviour was found to be unsatisfactory. In addition, the paper comments on the problems of insulating the unit in the cold loft space.

9 Conclusions

The unit performed satisfactorily, and achieved a performance a little below the manufacturer's stated figures. At an air flow of 180 $m^3\,h^{-1}$, it supplies about 1.7 kW of heat at a COP of 3. These figures include the effects of defrost cycling and fan power consumption.

In a well insulated house the system can provide a substantial part of the seasonal heating requirement at a unit cost less than that of off-peak electricity. This is combined with the advantages of the circulation of warm fresh air throughout the house.

Installation of the ductwork presents no new problems over those with a conventional VHR system. It is advantageous to contain all ductwork within the envelope of a house, in which case it need not be insulated. The unit requires a low-level condensate drain, which has to be planned at an early stage.

The heating capacity of the unit is limited by the ventilation requirement of the house, and the system cannot be expanded to provide full house heating. An additional resistance duct heater used at night or in very cold weather would be useful.

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