INTERNATIONAL ENERGY AGENCY
energy conservation in buildings and
community systems programme

Technical Note AIVC 45

Air-to-Air Heat Recovery in Ventilation

December 1994

Air Infiltration and Ventilation Centre
University of Warwick Science Park
Sovereign Court
Sir William Lyons Road
Coventry CV4 7EZ
Great Britain

This report is part of the work of the IEA Energy Conservation in Buildings & Community Systems Programme.

Publication prepared by
Annex V Air Infiltration and Ventilation Centre

Document AIC-TN-45-1994
ISBN 0 946075 83 2

Participating countries:

Annex V

Belgium, Canada, Denmark, Germany, Finland, France, Netherlands, New Zealand, Norway, Sweden, Switzerland, United Kingdom, and United States of America

Distribution: Annex V only

Additional copies of this report may be obtained from:

The Air Infiltration and Ventilation Centre
University of Warwick Science Park
Sovereign Court
Sir William Lyons Road
Coventry CV4 7EZ
Great Britain

Air-to-Air Heat Recovery in Ventilation

Steve Irving

©Copyright Oscar Faber plc 1994

All property rights, including copyright are vested in the Operating Agent (Oscar Faber Consulting Engineers) on behalf of the International Energy Agency.

In particular, no part of this publication may be reproduced, stored in a retrieval system or transmitted in any form or by any means, electronic, mechanical, photocopying , recording or otherwise, without the prior written permission of the Operating Agent.

PREFACE

International Energy Agency

The International Energy Agency (IEA) was established in 1974 within the framework of the Organisation for Economic Co-operation and Development (OECD) to implement an International Energy Programme. A basic aim of the IEA is to foster co-operation among the twenty-one IEA Participating Countries to increase energy security through energy conservation, development of alternative energy sources and energy research development and demonstration (RD&D).

Energy Conservation in Buildings and Community Systems

The IEA sponsors research and development in a number of areas related to energy. In one of these areas, energy conservation in buildings, the IEA is sponsoring various exercises to predict more accurately the energy use of buildings, including comparison of existing computer programs, building monitoring, comparison of calculation methods, as well as air quality and studies of occupancy.

The Executive Committee

Overall control of the programme is maintained by an Executive Committee, which not only monitors existing projects but identifies new areas where collaborative effort may be beneficial.

To date the following have been initiated by the Executive Committee (completed projects are identified by *):

I	Load Energy Determination of Buildings*
II	Ekistics and Advanced Community Energy Systems*
III	Energy Conservation in Residential Buildings*
IV	Glasgow Commercial Building Monitoring*
V	Air Infiltration and Ventilation Centre
VI	Energy Systems and Design of Communities*
VII	Local Government Energy Planning*
VIII	Inhabitant Behaviour with Regard to Ventilation*
IX	Minimum Ventilation Rates*
X	Building HVAC Systems Simulation*
XI	Energy Auditing*
XII	Windows and Fenestration*
XIII	Energy Management in Hospitals*
XIV	Condensation*
XV	Energy Efficiency in Schools*
XVI	BEMS - 1: Energy Management Procedures*
XVII	BEMS - 2: Evaluation and Emulation Techniques
XVIII	Demand Controlled Ventilating Systems*
XIX	Low Slope Roof Systems
XX	Air Flow Patterns within Buildings*
XXI	Thermal Modelling*
XXII	Energy Efficient Communities
XXIII	Multizone Air Flow Modelling (COMIS)
XXIV	Heat Air and Moisture Transfer in Envelopes
XXV	Real Time HEVAC Simulation
XXVI	Energy Efficient Ventilation of Large Enclosures
XXVII	Evaluation and Demonstration of Domestic Ventilation Systems
XXVIII	Low Energy Cooling Systems
XXIX	Energy Efficiency in Educational Buildings
XXX	Bringing Simulation to Application

Annex V Air Infiltration and Ventilation Centre

The Air Infiltration and Ventlation Centre was established by the Executive Committee following unanimous agreement that more needed to be understood about the impact of air change on energy use and indoor air quality. The purpose of the Centre is to promote an understanding of the complex behaviour of air flow in buildings and to advance the effective application of associated energy saving measures in both the design of new buildings and the improvement of the existing building stock.

The Participants in this task are Belgium, Canada, Denmark, Germany, Finland, France, Netherlands, New Zealand, Norway, Sweden, Switzerland, United Kingdom and the United States of America.

Contents

1. Introduction and Summary

This technical note discusses the issues which influence the performance of heat recovery devices within typical building applications. The report is intended to cover the three main types of devices installed in ventilation systems in residential and commercial buildings.

- -Run-Around Coils
- -Plate Heat Exchangers
- -Thermal Wheels (rotary regenerators)

Other systems such as heat pipes are also described briefly.

As with any energy efficiency measure, the successful implementation of air-to-air heat recovery depends on several factors

a) careful equipment selection
b) proper integration of that equipment into the ventilation system
c) good commissioning
d) proper maintenance and operation
e) building characteristics (e.g. airtightness)

Section 4 gives information to assist the designer and building operator understand some of the factors that influence the performance of heat recovery systems. This section highlights some of the key points that have emerged from the analysis of existing information, detailed simulation work and site measurements.

a) Manufacturers' catalogue data can be used to give a reasonable indication of the best possible field performance of heat recovery effectiveness. Site measurements have shown that the level of recovery given in the catalogues can be realised in practice, **provided the system is properly installed and maintained**. If inadequate attention is given to the installation, then the level of heat recovery can be very low, and indeed there may be nett energy consumption because of the additional fan power requirements.

b) In energy terms alone, recirculation of return air is the most efficient form of heat recovery, since it involves little or no energy penalty. However re-circulation is only possible if the ventilation rate is fixed by cooling rather than ventilation needs, and is therefore only applicable in all-air air conditioning systems. If the recirculation rate is more than about 60%, there is little benefit in considering heat recovery. However it must be stressed that the air quality implications of recirculation must always be considered (see point d) below).

c) The different forms of heat recovery device considered in this report each have their specific advantages and disadvantages (see section 3). It is not possible therefore to give definitive recommendations on the best system, but the following general guidance is supported by the analysis undertaken on the data available

- optimum performance is not always obtained by seeking to maximise recovery efficiency, since this is often at the expense of a significantly increased fan penalty and/or capital cost.

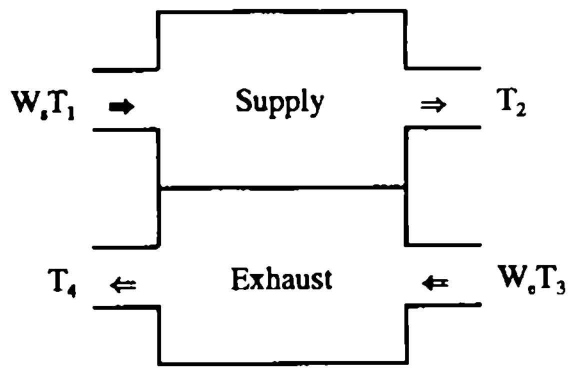
- inappropriate specification of controls and/or a lack of proper maintenance of sensors and set points may ruin performance. In that respect, plate heat exchangers have certain advantages, in that as passive devices, there is less opportunity for mistakes to be made. This benefit is also their weakness, in that because they cannot be controlled (except by using a by-pass duct), it is possible to over-recover heat and impose an over-heating or increased cooling penalty.

d) The different forms of heat recovery provide different levels of protection against the contaminants in the exhaust stream being recirculated back into the supply. Run around coils completely isolate the two air streams and use an independent heat transfer fluid, thus making cross contamination impossible. Plate heat exchangers also isolate the two fluid streams, but there can be some cross leakage of air if there are poor seals or mechanical damage. With a thermal wheel, the heat transfer medium rotates between the two air streams, and so significant cross contamination is inevitable. This can be minimised with a purge section, but this will also reduce efficiency.

2 Fundamentals

2.1 Terminology

The various terms used throughout this document are defined below, with reference to the following figure.



Definition of symbols

Where:

T = dry-bulb temperature (K)	1 = Supply on to device
W_s = Supply flow rate (kg/s)	2 = Supply off device
W_e = Exhaust flow rate (kg/s)	3 = Exhaust on to device
W_{min} = Minimum value of W_s or W_e	4 = Exhaust off device

<u>Device Efficiency</u>: Where devices are being compared outside of an actual installation, the performances are defined in terms of their efficiencies. These may be temperature efficiencies (sensible heat transfer only) or total efficiencies (sensible and latent heat transfer).

Temperature (or sensible) efficiency η_t is defined as:

$$\eta_t = \frac{(T_2 - T_1)}{(T_3 - T_1)} \qquad (1)$$

Total efficiency $_T$ is defined as:

$$\eta_T = \frac{(H_2 - H_1)}{(H_3 - H_1)} \qquad (2)$$

Where H is the specific enthalpy (kJ/kg) at the points indicated in fig. 1. Some devices (notably thermal wheels) are able to effect total energy recovery, i.e. transferring latent as well as sensible heat from the exhaust air to the supply air. However most devices can recover latent heat in the exhaust air and transfer it to the supply air, if the heat recovery reduces the temperature of the exhaust air below its dewpoint. In the analyses that follow in sections 4-6, this latter factor has been ignored.

Sensible Effectiveness: Where a heat recovery device is within a system, the efficiency term is not strictly valid due to variations in supply and extract flows. The device is then rated in terms of its effectiveness as defined by the ratio of the actual heat transfer to the thermodynamically limited maximum heat transfer possible. For example, the temperature effectiveness is defined as -

$$\varepsilon_T = \frac{W_s(T_1-T_2)}{W_{min}(T_1-T_3)} = \frac{W_e(T_4-T_3)}{W_{min}(T_1-T_3)} \qquad (3)$$

Heating Savings: This is the annual reduction in heating energy as a result of heat recovery, expressed as a percentage of the energy loss for the air passing through the ventilation system.

$$\frac{mC_p(T_2-T_1)}{MC_p(T_{room}-T_1)} \qquad (4)$$

m	the mass flow through the supply side of the heat recovery device (kg/s)
C_p	the specific heat capacity of air (kJ/kgK)
T_{room}	the temperature of the room (K)

Net Energy Savings: This is the annual total energy savings (heat recovered less any energy consumed as a result of the heat recovery process) expressed in primary energy units (peu). This is again referenced to the energy loss of the air passing through the ventilation system.

$$\frac{(Q_{rec} * PE_h/\eta_b)-(E_{fan}+E_{aux}) * PE_e}{mC_p(T_{room}-T_{out}) * PE_h/\eta_b} \qquad (5)$$

Q_{rec} is the heat recovered by the heat recovery device (kJ)
PE_h is the factor for converting the heating fuel into primary energy units (-)
E_{fan} is the additional energy consumed by the fan (kJ)
E_{aux} is the auxiliary energy consumption (kJ)
PE_e is the factor for converting electricity into primary energy units (-)
η_b is the seasonal efficiency of the boiler (-)

Primary Energy Units peu: The quantity of energy accounted for in terms of 'primary' energy, depends not only on the net consumption of energy but also in the form in which it is received. This includes the energy consumed in the production and transportation of the energy. Average factors for the U.K. are given below

Supplied Energy	Primary Energy Unit per unit of Supplied Energy (peu)
Electricity	*** 3.37
Oil	1.08
Natural Gas	1.06
Coal	1.02

*** NOTE:This factor may vary from country to country depending on the mix of fossil fuel/nuclear/hydro power generation. The CO_2 impact will vary even more significantly.

2.2 Effects of Outside Conditions

For most purposes, the efficiency of a heat recovery device can be considered to be largely independent of outside weather conditions. However it is very important to realise that although the efficiency may be constant, the absolute amount of heat recovered will reduce as the outside temperature increases. The energy penalty of the heat recovery device (increased fan power, ancillary energy use etc) will remain constant no matter what the outside condition. This means that as the outside temperature rises, the nett energy recovery actually decreases, and can indeed go negative. Figure 1 illustrates these variations for a plate heat exchanger of efficiency of 0.58 with a pressure drop of 485Pa at an airflow of $2m^3/s$.

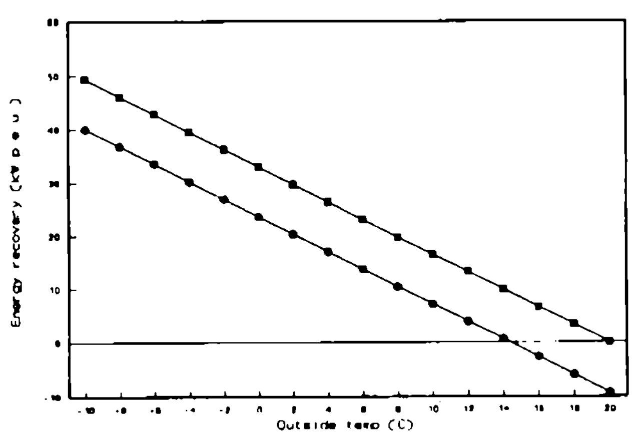


Figure 1 Heat and energy recovery as a function of outside air temperature

3 Description of Devices

3.1 Run-around Coils (Water Circuit System)

Run-around coils comprise two fin-type heat exchangers one installed in the supply duct and the other in the exhaust. A liquid (normally a water/glycol solution) is used as the heat transfer medium. Heat in the exhaust air stream is thus transferred via the liquid to a coil in the supply duct, which is then picked up by the incoming air passing through it.

Advantages:

- As the supply and exhaust air streams are kept totally separate, run-around coils are ideal for situations where the contaminants in the exhaust air prohibit recirculation, or where the fresh air and recirculation ducts are not adjacent to each other. Run-around coils are therefore often the only practical heat recovery solution when retrofitting an existing ventilation system.

- Multiple supply and exhaust systems can be combined in a single loop.

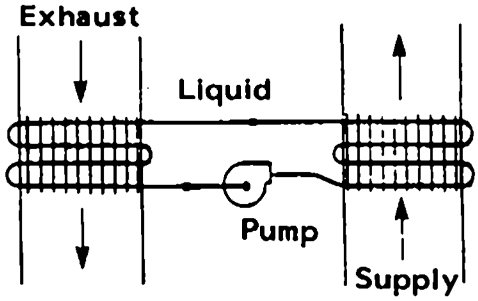


Figure 2 Run-around coil configuration

Disadvantages:

- This type of system can generally only transfer sensible heat. It also has a relatively low efficiency.

- The additional circulating pump energy costs have to be offset against predicted energy savings. This tends to be small relative to the increased fan power.

- The circulating pump presents additional maintenance requirements.

3.2 Plate Heat Exchangers

Plate Heat Exchangers are static devices, where the interchange of the supply and exhaust air streams enable heat to be transferred from one to the other. They are more typically associated with continuous process use.

Advantages:

- Air to air plate heat exchangers are fairly simple devices with no moving parts.

- They tend to be modular and thus the appropriate number of modules can be selected for the appropriate flow rates.

- If properly constructed there is little possibility of cross contamination between air streams. Cross leakage can occur if there are faulty seals in modular units, or if mechanical damage has occurred.

- Once fitted there are little maintenance requirements.

- Apart from additional fan power requirements, they have no additional pump electrical penalties.

Disadvantages:

- Most types of plate heat exchangers can only transfer sensible heat. However, paper core heat exchangers are available which permit the transfer of moisture from the exhaust to the supply side.

- Unless a bypass is provided, there may be an overheating problem in the summer.

- They can only be used where supply and exhaust ducts can be brought together.

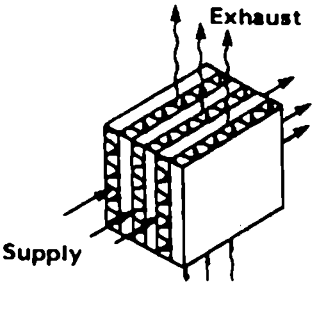


Figure 3

Figure 3: Typical plate heat exchanger

3.3 Thermal Wheels

A typical thermal wheel (rotary regenerator) is essentially a revolving cylinder divided into a number of segments packed with a coarsely knitted metal mesh, or some other material. It operates by rotating at between 10 and 20 revolutions per minute picking up heat from the warmer exhaust air stream and discharging it to the cooler supply side. Where it is essential to restrict cross contamination, a purge section is included.

Advantages:

- Depending on the medium, thermal wheels can transfer latent as well as sensible heat.
- A variable speed drive enables the efficiency of the device to be varied.
- The wide range of matrix materials and densities suit many applications.
- Since the whole metal mesh is constantly being used, the thermal wheel provides a large heating surface per unit volume.
- The static pressure drop will generally be low (although larger composite wheels may have higher pressure drops).

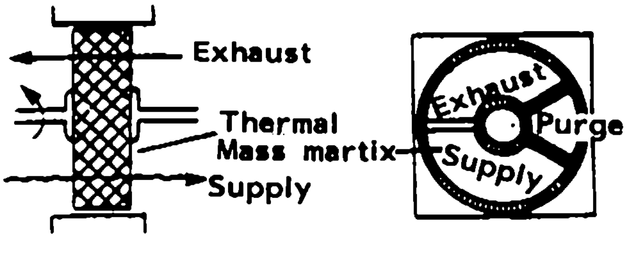


Figure 4 : Typical thermal wheel (or rotary regenerator)

Disadvantages:

- Exhaust and Supply ducts must be adjacent.
- Additional drive motor penalties (although this is usually low).
- Even with a purge section fitted, some cross contamination is unavoidable, and so thermal wheels cannot be used where particularly noxious fumes are exhausted, or

where the presence of any exhaust air in the supply stream is undesirable. The purge section also reduces the heat recovery efficiency of the device.

3.4 Other Heat Recovery Strategies

The previous paragraphs have listed the most commonly used ventilation heat recovery strategies. A number of other approaches have been used, and these are summarised below.

3.4.1 Heat Pipes

This process uses a refrigerant, and the heat transfer process follows the normal refrigerant evaporation / condensation cycle, with the working fluid being returned from the condensing section to the evaporator section via a capillary wick. The efficiency of the heat pipe is in the range 0.45-0.65, and the performance can be modulated by adjusting the slope of the pipe. The air streams are kept completely separate, and so there can be no cross contamination. Heat pipe units also tend to be very compact.

3.4.2 Heat Pumps

Heat pumps use the conventional vapour compression cycle to transfer heat from one fluid to another. It has been used in some air to air applications where there is a high energy recovery potential, and it is not possible to have any recirculation of exhaust air back to the supply (e.g. swimming pools). Because the source to sink temperature differential is low, a good CoP can be achieved. Heat pumps can be particularly useful if there is a high latent heat content in the exhaust air. As well as the fan energy, the energy consumed by the compressor has to be accounted for in the overall evaluation.

Heat pumps can be used in combination with heat exchangers to provide very high efficiency systems. In this arrangement, the evaporator is placed downstream of the exhaust side of the heat exchanger, and the condenser downstream of the supply side. Heat pumps are also used in extract ventilation systems to transfer heat from the exhaust air to the domestic hot water.

3.4.3 Heat Recuperators

This system uses a chamber with a significant thermal capacity and a damper to cycle the supply and exhaust flows between the two halves of the chamber. In the first part of the cycle, the exhaust air flowing through one half of the chamber heats up the thermal mass. The damper is then moved so that the supply air now flows through that part of the chamber, absorbing the heat from the structure, and reducing its temperature ready for the beginning of the next cycle. Efficiencies can be quite high for these systems.

3.4.4 Dynamic Insulation

This concept brings ventilation air into the building through the fabric of the wall, thus recovering a significant proportion of the conduction heat loss. This relies on good distribution of the incoming air through the fabric.

4 Factors Influencing The Selection Of Heat Recovery Devices.

Section 4 is intended to give a comprehensive picture of the various choices open to a designer when considering heat recovery systems, and the impact those choices will have on overall performance.

4.1 Illustrative Examples of Net Energy Savings

In order to present the information, two main performance criteria are used (see section 2 for definitions)

- heating savings
- nett energy savings

The difference between these two parameters is the energy penalty (mainly the additional fan power, but also auxiliary energy use such as pumps in a run around coil). When calculating the nett energy savings, the heating savings and energy penalty must be calculated on the same basis of primary energy units to be consistent in terms of likely cost and environmental impact (primarily CO_2 emissions).

Nett energy savings is used as the primary criteria, since this will most closely relate to the energy cost savings accruing from heat recovery. These performance parameters are then related to the typical range of temperature efficiencies and device pressure drops of each device type as indicated in the manufacturers literature. These ranges are summarised in the table below

Performance parameter	Run around coil	Plate exchanger	Thermal wheel
Temperature efficiency (%)	50-65	40-70	60-85
Pressure drop (supply + exhaust) (Pa)	200-900	50-250	75-500

It should be noted that the quoted ranges of efficiency and pressure drops do not necessarily correspond (i.e. a run around coil giving 50% efficiency will not necessarily have an air side pressure drop of 200 Pa). It should also be stressed that the pressure drops given above are the <u>sum</u> of the supply and exhaust side pressure drops.

When investigating a case for heat recovery, data for assessment purposes should always be taken from manufacturers catalogues, but the above data provides a useful guideline to the typical ranges of values. It should also be noted that even within a given category of device type, there is a significant variation in the efficiency / pressure drop characteristics between different manufacturers.

In order to illustrate the influence of various parameters on overall performance, catalogue data has been taken and used in a simplified spreadsheet model to predict heating savings and net energy recovery. These are discussed in the following sections for a standard case where the supply and exhaust air flowrate are both 2 m^3/s for 24 hours/day, 365 days/year. The climate data is based on a central European climate (Zurich), assuming that the heat recovery is displacing heat from a gas fired heating system. The model makes a number of simplifying assumptions as listed below.

a) the heat recovery is useful, provided the temperature of the heat recovery device is no greater than the required room temperature. This simplifying assumption results in no credit being taken for heat recovery during those parts of the year when heating is not required.

b) the device efficiency is constant across the range of outside air conditions.

c) the infiltration rate is constant through the year (assumed to be zero for the first series of analyses in section 4.1).

d) boiler efficiency is constant with heating demand.

e) the effect of duct leakage is ignored

f) the effect of defrost cycles is ignored. Although it is recognised that these effects can be significant in some situations, this simplified analysis is useful in highlighting the important trends in heat recovery performance. *Specification and selection of heat recovery equipment should always be made on the basis of a specific evaluation of performance.*

4.1.1 Run-around Coils

The efficiency of the device is primarily related to the number of coil rows; 6, 8, 10 or 12 coil rows are typically offered. The number of coil rows is also the primary factor in determining the fan penalty.

As well as the fan penalty, run around coils consume energy through the pump motor. Typically this represents around 5% of the energy available for recovery, and because of the higher energy cost of electricity, is a significant impact on overall performance. Because the pump energy is usually considerably less than the additional energy consumed by the fan, it is generally preferable to have fewer coil rows but with higher liquid velocity, than to have more rows with a lower liquid velocity.

Because heat is transferred from one coil to another by liquid flowing through interconnecting pipework, it is essential that this be properly protected from freezing (see 5.4). Effectiveness is maximised if water is used as the heat exchange medium, but this can cause freezing problems, and so a glycol solution is generally used. For example, a 20% glycol mix provides protection down to -10°C, but does increase the water loop pressure drop by 15%. It will also reduce temperature efficiency by 10-20%, and so it may be beneficial to explore other forms of frost protection, e.g. trace heating (putting a blanket containing electric resistance heaters around the pipework), thermostatically controlled immersion heater, continuous running of the pump). Such systems are inherently less reliable, and this may be important in very cold climates.

Where a device is operating towards its maximum duty, it is possible for condensation to occur on the exhaust-air coil. This can be transferred as sensible heat to the supply coil, but condensate build up will also increase air side pressure drop by up to 30%. This can also result in freezing problems in extreme climates. If condensation is likely to occur, drop eliminators (which prevent the accumulation of moisture droplets) are recommended to avoid damage to the coils, but these also impose an additional fan penalty.

The performance of a run-around coil can be modulated using either a variable speed pump or a three way valve. Although this offers a potential benefit, site observations indicated that this was also a cause of potential problems. In one case the temperature sensor monitoring the off-coil condition was significantly out of calibration, resulting in less than the desired heat recovery. In another case, the three way valve movement had been manually restricted, again limiting the potential for recovery. This highlights the importance of good commissioning and maintenance.

To illustrate the effect of the various parameters, the test case described above has been simulated. In this case, the glycol content of the circulating water is taken as 30%, with a water circulation rate of 0.85 l/s. The following data is for two different model sizes appropriate to the selected duty.

Model	No of coil rows	Efficiency	Air side pressure drop per coil (Pa)	Water side pressure drop (kPa)
A	6	0.44	170	31
	8	0.52	235	41
	10	0.58	290	51
	12	0.58	350	38
B	6	0.46	110	17
	8	0.55	145	24
	10	0.57	175	29
	12	0.60	210	25

Table 4.1.1:Characteristics of different run around coil selections for a ventilation rate of 2000 l/s, and circulating water/glycol flow of 0.85 l/s.

The heat and nett energy recovery for these two devices has been calculated for the example case described in section 4.1, and the results illustrated below.

It illustrates the point that increasing efficiency does not always result in more cost effective performance. Increasing the number of coil rows increases efficiency and the amount of heat recovered, but the maximum nett energy recovery occurs for a 10 row coil. Bearing in mind the extra capital costs of the unit with more rows, these results indicate the need to carefully analyse each potential application for heat recovery.

This is further illustrated by the fact that calculations for the model B coil above (same manufacturer but the next size up in the range), indicates that there is only a small increase in heat recovery (approx 5%), but the net energy recovery increases by nearer 20% because of the substantially reduced fan penalty).

<u>It must be stressed that these trends may vary depending on climate and on displaced heating fuel.</u>

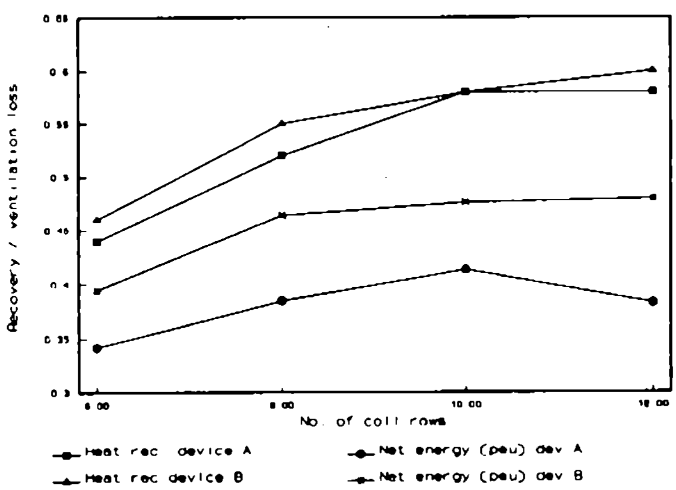


Figure 5 Example performance of different run-around coil configurations

4.1.2 Plate Heat Exchangers

The efficiency of the device is primarily associated with spacing between plates, and the level of embossment on the plate surfaces (promoting turbulence and enhanced heat transfer coefficients). Where a device is operating towards its maximum duty, it is possible for condensation to occur on the exhaust side of the exchanger. This can be transferred as sensible heat to the supply. This can also result in freezing problems in extreme climates (see section 5.4). If condensation is likely to occur, drop eliminators are recommended to avoid damage, but these impose an additional fan penalty of typically 20%.

Because it is a passive device, the performance of a plate exchanger cannot be modulated except by providing a bypass damper. This can avoid over recovery of heat, but will only reduce the fan penalty outside the heating season if there is appropriate fan control to respond to a lower bypass resistance.

Plate spacing (mm)	Efficiency	Air side pressure drop per side (Pa)
3.0	0.58	485
3.8	0.51	285
4.5	0.47	219
6.2	0.39	154

Table 4.1.2:Different plate heat exchanger selections for a ventilation rate of 2000 l/s.

Table 4.1.2 contains a summary of the performance of a number of plate heat exchangers selected to meet the duty of the example system as described in section 4.1.The performance of these configurations is illustrated in figure 6. A very similar trend is seen, in that the maximum net energy recovery occurs at intermediate plate spacings, and indeed the closest spacing although giving the maximum heat recovery, gives only slightly better nett energy recovery than the widest spacing. It must again be stressed that this is a function of the particular case analysed, and will vary from situation to situation.

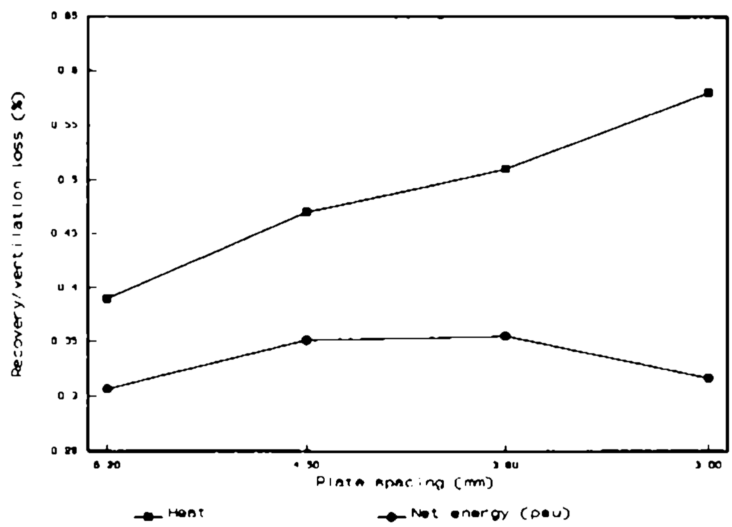


Figure 6 Example performance of various plate heat exchanger configurations

4.1.3 Thermal Wheel

The efficiency of the device is mainly a function of the packing type and organisation. Different packing materials can be used where there is a benefit in recovering moisture as well as sensible heat. This is a unique benefit of thermal wheels. Thermal wheels also offer the highest efficiency of all devices, which combined with a relatively low airside pressure drop will tend to maximise net energy savings. The design pressure drops for these devices tends to be fairly constant for system flow rates up to $10m^3s^{-1}$, but increases rapidly above that. This combined with the rapid price increase above that same duty means that the cost effectiveness of thermal wheels reduces at the bigger sizes.

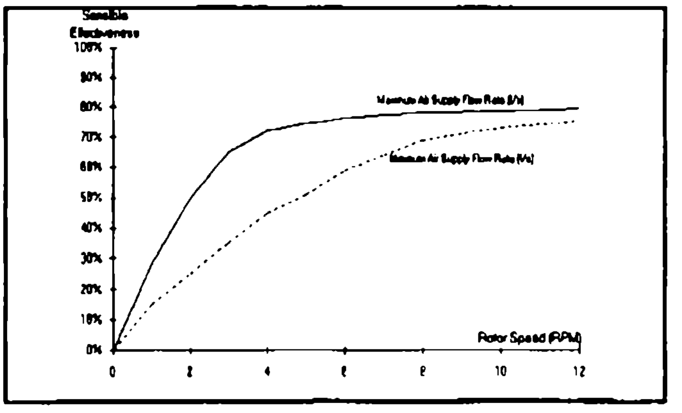


Figure 7 Plot of temperature effectiveness according to rotation speed

The thermal wheel also consumes energy due to the drive motor, but this is generally small relative to the fan penalty. The performance of a thermal wheel can be modulated by varying the speed of rotation of the wheel. Figure 7 shows how the effectiveness varies with rotational speed for the range of allowable flows through the device. The reduction in efficiency with rotor speed occurs much sooner at reduced volumes, and these effects must be carefully considered when setting up a control strategy.

As far as thermal wheel performance is concerned, the only real variant within one manufacturer is the wheel size. Consequently this effect has been analysed below, where the increased size results in a lower pressure drop, but an effectively constant efficiency.

Face Dimensions	Efficiency	Air side Pressure Drop (Pa)
1516x1532	0.790	260
1666x2132	0.795	230
1816x2132	0.805	155
2079x2732	0.820	110

Table 4.1.3: Characteristics of different thermal wheel selections, ventilation rate of 2000 l/s.

Figure 8 shows the performance of these various thermal wheels under the same operating conditions as discussed above. The increased size improves the heat recovered slightly, but the reducing airside pressure drop increases the net energy recovery. This results in the designer needing to trade off the capital cost of the device against the improved energy cost savings.

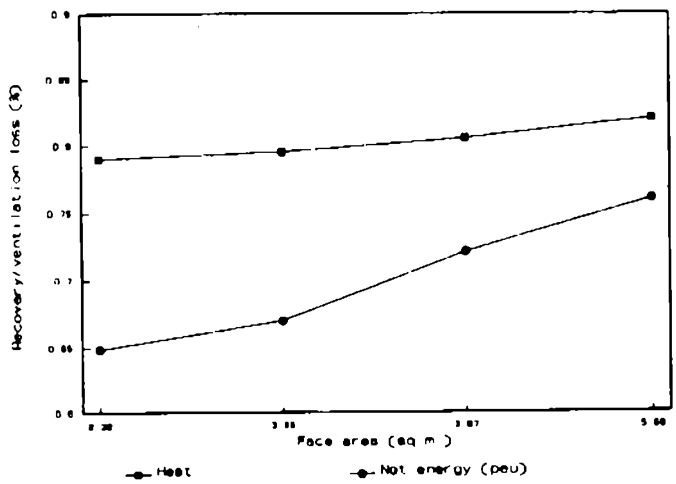


Figure 8 Example of effect of differing size thermal wheels

4.2 Effect of Different Climate Zones

To obtain a picture of the performance of the different devices in different climate classes of the AIVC participating countries, the performance of the 10 row run around coil has been analysed for 8 different locations. These locations have been selected from data available in reference zz as typical of the range in the AIVC countries. The minimum and maximum average monthly temperatures are shown in figure 9, along with the annual average temperature. The results are shown in fig 10 which indicates that the nett energy performance increases in colder climates because the relative significance of the fan and auxiliary consumption reduces.

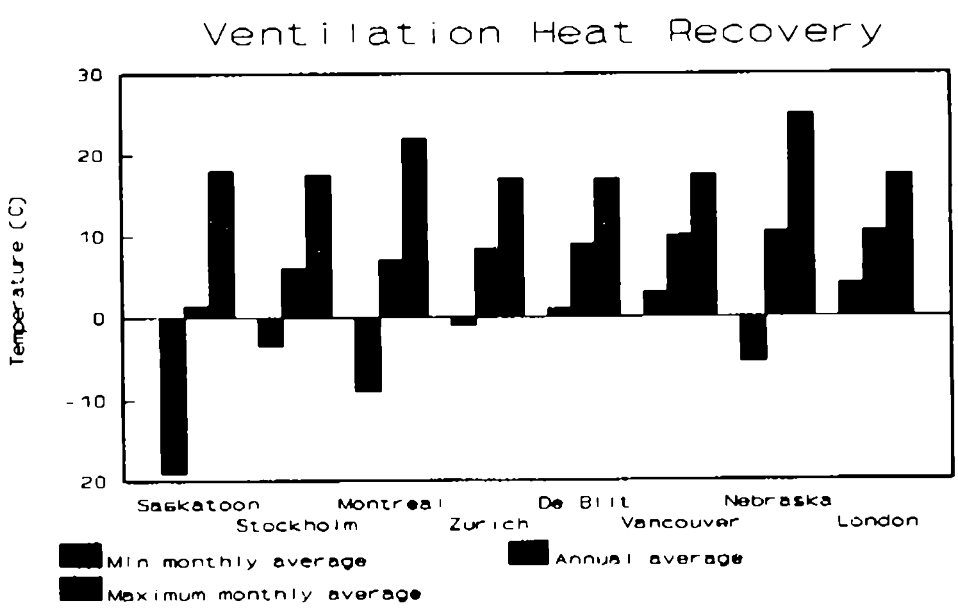


Figure 9 Average temperatures for 8 climate zones

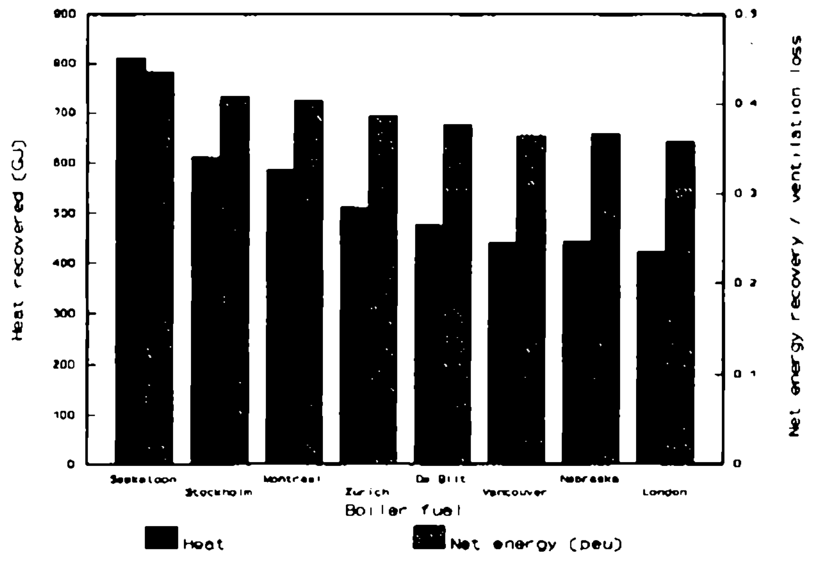


Figure 10 Performance of 8-row coil in various climate zones

4.3 Cross Contamination

As detailed in section 4, in respect of cross contamination, the various system types rank in exact reverse order to their net energy saving potential. Run around coils completely separate the flows, and so there is no possibility of cross contamination. Plate exchangers can allow contaminants to pass if there is mechanical damage or in modular units, the seals are faulty. The pressures can be arranged such that any leakage occurs from supply to exhaust, but this is in opposition to achieving maximum heat recovery (supply pull through, exhaust blow through). Cross contamination is inevitable with thermal wheels, but can be minimised if a purge section is fitted.

4.4 Space Requirements

Another important factor in selection of heat recovery equipment is the space requirement. It is almost impossible to make generalisations about this, but comparisons of face areas of equipment at different flow rates suggest that at the smaller size range ($2.5m^3s^{-1}$), run around coils and plate exchangers take up the least space. This takes no account of the space requirements needed to bring the supply and exhaust ductwork adjacent to one another. This is not required for a run around coil, which is perhaps its biggest advantage.

4.5 Capital and Maintenance Costs

The costs of the various devices will vary from country to country, and so it is only possible to give general guidance. In terms of capital cost, at the smaller sizes, run around coils tend to be more expensive because of the additional cost of installation associated with the pump circuit. Above a design flowrate of $10m^3/s$ the costs of all devices begins to increase much more rapidly. Investment in heat recovery equipment is sometimes justified by savings in the required capacity of boiler and chiller plant (where installed). Care should be taken before reducing the size of boiler plant, especially in intermittently heated buildings, since if the building has cooled down during the 'plant-off' period, the heat is not available in the exhaust air to provide the recovery. This may result in unacceptable warm up periods. It also highlights the need for good part load performance from the boiler plant, since the boiler demand will reduce significantly once the system is recovering heat from the exhaust air.

All systems require routine cleaning of the heat exchange media, and regular replacement of the filters which protect that media. In very general terms, the heat exchanger requires the least maintenance because it is a passive device, but in all cases, careful attention should be given to the manufacturers instructions. Based on U.K. information, the annual cost of maintenance for new equipment is much the same for all three devices, although earlier models of thermal wheel required the complete replacement of the media after a few years of operation.

5 General Factors Influencing The Operation Of Heat Recovery Devices.

Whilst manufacturers and guides can provide information on likely efficiencies of heat recovery devices and also highlight potential areas of savings, once installed within a system the device will often succeed or fail as a result of the quality of the setting up or control of the installation as a whole. Some areas where care should be taken are listed below:

5.1 Control Set-points

A device can be allowed to operate 'wild' with just a simple on-off control, or can be permitted to modulate relative to a sensed temperature. Unless strict temperature control is required, modulating control is usually unnecessary.

On-off control:

Where a device is operating 'wild' it is normal to attempt to achieve the highest heat recovery possible. With this definition the device is usually turned off completely (or bypassed for a plate heat exchanger) once a pre-set off-device temperature has been reached.

From site observations it has became obvious that care must be taken to ensure that this 'off temperature' is the optimum for the system in which the device is installed. If the temperature is set too low, then the heat recovery device will fail to provide the heating savings initially hoped for. Where the threshold is too high, an additional cooling penalty, or overheating may be placed on the building.

Modulated Control:

In this situation the heat recovered across the device is regulated either in response to the external temperature or in order to achieve the main air handling unit control set-point. Modulation is usually performed by means of a three-way control valve with a run-around coil, or by a variable speed drive for thermal wheels. (Note that the effectiveness of thermal wheels drop sharply once the speed of rotation has dropped below 30% of its maximum). Because plate heat exchangers are static devices, their performance can only be modulated if a bypass damper is fitted.

For similar reasons to those stated for 'wild' operation, care should be taken too when defining the control bands of the sensed and controlled temperatures.

5.2 Plant Location Relative To The Heat Recovery Device

The full benefits of a heat recovery device will only be realised if the unit is properly integrated into the overall HVAC system. For example, it is preferable to locate the fan to draw air through the device on the supply and blow air through it on the exhaust in order to maximise the benefit of the fan pick-up. Bearing in mind the pick-up can be of the order of 1K per fan (depending on the pressure drop), this effect can be significant. However this approach will result in a higher pressure on the exhaust side of the unit, and any leakage between the air streams will carry contaminants from the exhaust side to the supply.

If a <u>system</u> frost coil is provided to protect filters etc, it is best to place this downstream of the heat recovery device (provided the device is itself properly protected), otherwise the recovery potential will be significantly reduced. In a number of field installations investigated in the UK, frost coils had been installed upstream of the heat recovery, and were heating the air onto the heat recovery device to temperatures of between 3 and 8^{o}C.

5.3 Device sizes

For a given design flow rate there may well be several sizes of a particular heat recovery device that would be suitable. Where space provision is flexible, care should be taken to ensure that the optimum device is selected, balancing the effectiveness for probable air flows against the resulting electrical penalties.

5.4 Provision for Defrosting

When a device is operating between extreme temperatures, it is possible for the exhaust air to be cooled down below its dew point and for moisture to collect on the heat transfer surfaces. This can enhance heat transfer, but will also increase the pressure drop. If the outside air is sufficiently cold, this can freeze; in such situations de-frosting is necessary.

Depending on the device type defrosting can be achieved by various means.

a) stopping (or reducing) the outside air flow and using the exhaust air to defrost.

b) temporarily shutting of the fresh air and re-circulating exhaust air through the supply side of the device.

c) reducing the effectiveness of the heat recovery process (e.g. by reducing the speed of the thermal wheel, or the liquid flow rate through the exhaust coil in a run around coil).

d) use of a low duty heater.

Work by Phillips et al indicates that the choice of freeze control strategy has little effect on seasonal performance for climates less than 4200 centigrade degree-days. In very cold climates, the best strategy is either re-circulation or reducing fresh air flows. The imposition of a drop eliminator may also reduce the problem, but the additional 20-30% fan penalty associated with this will be borne all year.

5.5 Supply and Exhaust Airflow Volumes

Based on equation (3), figure 11 presents the likely influence on temperature effectiveness of the ratio of supply to exhaust air flows. The larger the exhaust volume, the higher will be the effectiveness of the device.

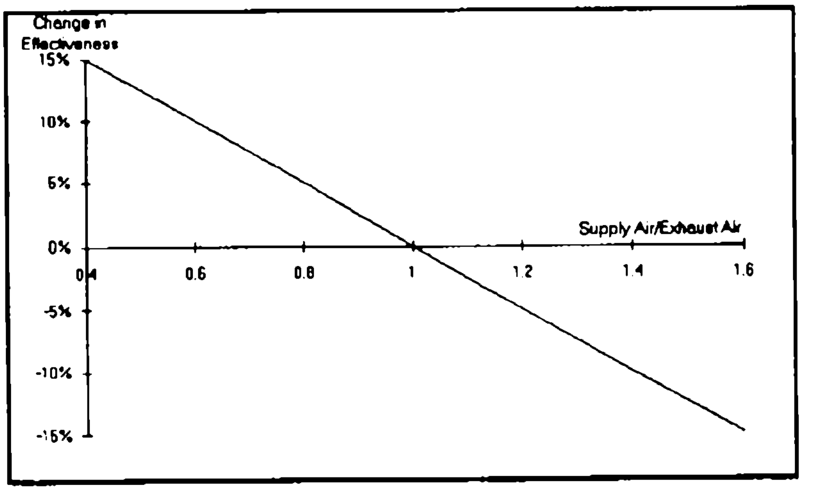


Figure 11 Influence of supply to exhaust air flow rates on temperature effectiveness

6 Influence of Building Related Issues

6.1 Building Airtightness

Airtightness has a significant impact on the viability of heat recovery. It is self evident that air to air heat recovery is only possible in a balanced ventilation system. It is generally accepted that the maximum building airtightness should be in the range of 2 - 7 air changes per hour at 50Pa, or the infiltration energy loss will outweigh any potential benefit of heat recovery. To counter this problem, the supply and exhaust fans can be rated differently; if the supply fan is larger, then the building will be slightly pressurized, thus reducing the infiltration but this may also result in moisture being driven into the fabric. If the exhaust fan is larger, the effectiveness of the system will be increased (see section 5.5), but the building under-pressure may result in unwanted draughts, and the inability to transfer the recovered energy to some of the incoming air.

The same model as used in section 4 has been applied to assess the importance of building airtightness. The airtightness (n_{50} value) has been varied across a range of 0-14 air changes/hr at 50Pa. The average infiltration rate has then been calculated from the simple ($n_{50}/20$) relationship, and the subsequent infiltration loss has been subtracted from the heat recovered.

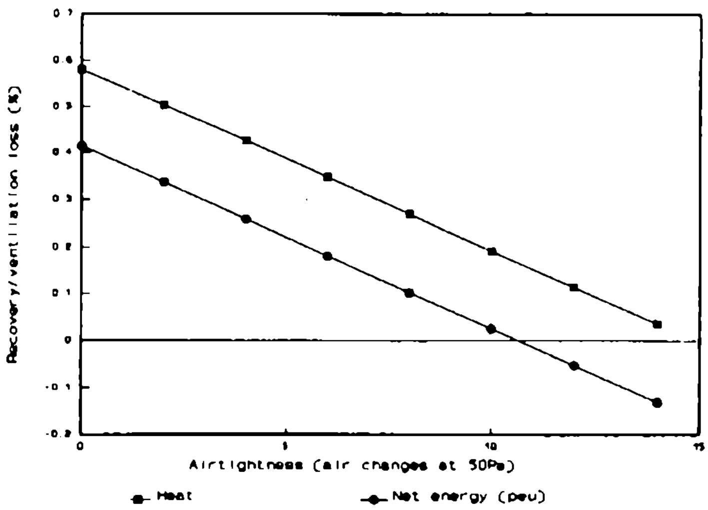


Figure 12 Effect of airtightness on overall efficiency

The analysis has been run with the 3mm plate spacing heat exchanger (see 4.1.2) in the climate of Zurich. The graph shows that reducing airtightness has a very significant effect on the overall performance of building and system. At an airtightness of about 15 a.c./hr, the nett heat recovered is zero, and the nett energy recovery goes negative at about 10 a.c./hr.

Figure 13 shows the effect of climate zone; the same analysis has been repeated for two airtightness values (2 and 10 a.c./hr) for four different locations representing the spread of climate zones. The locations are Saskatoon, Stockholm, Zurich and London.

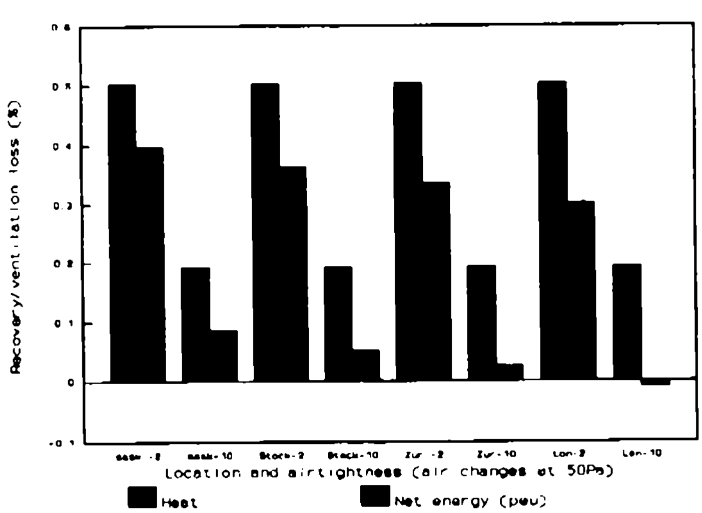


Figure 13 Effect of climate and airtightness

This shows the heat and energy recovery as fractions of the ventilation system heat loss. The figure re-inforces the point that the fan penalty increases as the climate is less extreme. The figure also shows that the point at which reduced airtightness results in energy wastage rather than energy recovery is reached much earlier in milder climates. However in all climates, reduced airtightness will have a negative impact on the economic performance of the heat recovery system.

6.2 Insulation

The level of insulation in a building will not directly influence the performance of heat recovery. However there will be an indirect effect in that it will influence the relative significance of the ventilation heat loss as a proportion of the total. The absolute heat recovery benefit is constant, but the proportionate reduction in total heating requirement will increase in a better insulated building. This balance may shift in a building where cooling is required, since heat recovery may be in conflict with the needs for free cooling if the temperature off the heat recovery unit is greater than the required plant temperature.

6.3 Internal Gains

Internal gains will influence heat recovery in a similar way to improved insulation, in that there is a reduction in total heating demand, although the ventilation heat loss will be unchanged. Again, this situation will be somewhat different if the internal gains are so high that cooling is required for much of the year.

If a proportion of the internal heat gain is extracted with the exhaust air rather than being seen in the room (e.g. air handling luminaires), the return air will be warmer than the room air, and this can provide enhanced heat recovery.

6.4 Running Hours

Increased running hours are likely to maximise the effect of heat recovery. This is especially so where the extended hours include a greater proportion of the night time when external temperatures are lower. This means that the benefit of energy recovery will be enhanced by a proportion greater than the increase in run hours.

6.5 Boiler Efficiency

Because the heat recovery device displaces heat that would otherwise be provided from the main heating boiler, then the overall performance of the system is influenced by the boiler efficiency. This is shown in figure 14, where the performance is shown as a function of the seasonal boiler efficiency.

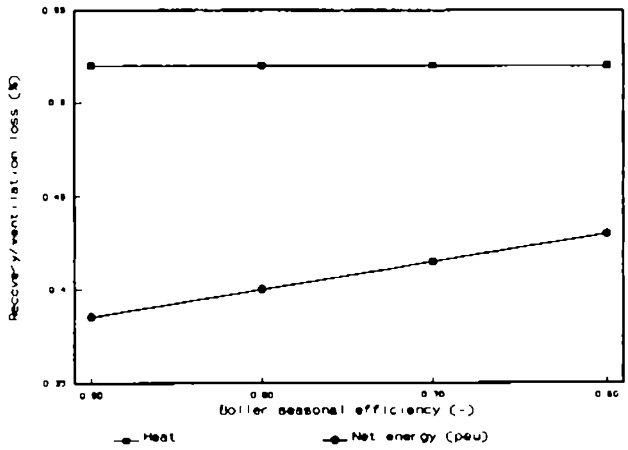


Figure 14 Performance of system shown as a function of seasonal boiler efficiency

This should not be taken as a means of justifying heat recovery by an inefficient boiler, but highlights the fact that the performance of any efficiency measure must be looked at in the context of the whole HVAC system.

6.6 Boiler Fuel

In a similar way, the boiler fuel will also influence the overall performance of the system. Figure 15 shows how the same heat recovery device will perform relative to a gas heating or electric heating system.

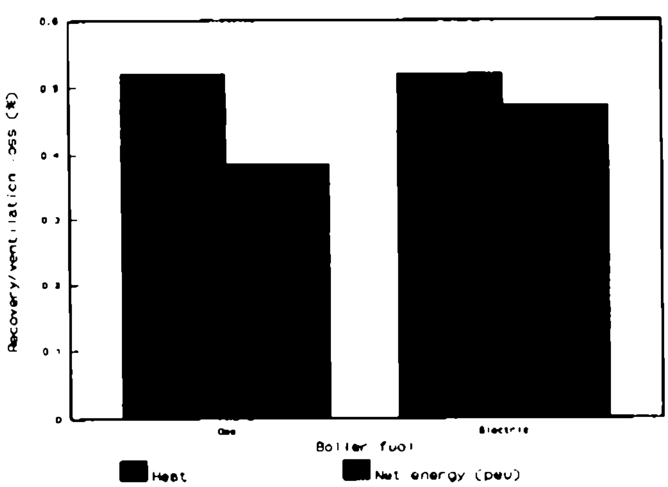


Figure 15 Performance of heat recovery device relative to a gas or electric heating system

This illustrates that the performance of heat recovery in primary energy terms is very much more effective in an electrically heated building. The heat recovery unit is recovering the same amount of energy in both cases, but the relative value of the recovered energy is much greater in the electrically heated building.

7. Acknowledgements

The work reported here developed out of a project on heat recovery for the Research Fund of CIBSE in the U.K. Their support of the earlier work is gratefully acknowledged.

8. Bibliography

1. Michael Holmes and Gavin Hamilton, "Heat Recovery with run-around coils", BSRIA TN2/78, 1978

2. G. Hamilton, "Seasonal heat recovery performance data for the air to air plate heat exchanger", BSRIA, June 1985

3. R. H. Rooley, "Heat recovery systems and possibilities"

4. G J Gudac et al. , "Effectiveness and Pressure drop characteristics of various types of air-to-air energy recovery systems", ASHRAE Trans 1981(1)

5. A.T. Howarth and K M Letherman, "Performance tests on an air-to-air thermal wheel in a swimming pool hall", BSER&T Vol.2 No. 3, 1981

6. John P Zarling, "Air-to-air heat recovery devices for small buildings" School of Engineering, University of Alaska, Fairbanks, Alaska, May 1982

7. T Mahlbacher, "Measurements on a Domestic Heat Recovery System" Electrowarme International (FRG), Vol 44, No Al pp 20 - 27, January 1986

8. B. S. Szabo, "The Economics of Heat Recovery Systems", ACHV June 1967, pp 59 - 64

9. W P Jones, "Energy Recovery in Ventilation in Systems", VI Course on Building Services Engineering, 5th - 6th January, 1977

10. Howard N Gawley & Donald R Fisher, "The Effectiveness and Rating of Air-to-air Heat Recovery", ASHRAE Trans 1975 V. 81 Pt 2

11. Canadian Standards Association, "Standard Methods of Test for rating the performance of Heat Recovery Ventilators", Preliminary Standard C439 - M1985

12. Kennard L Bowlen, "Energy Recovery From Exhaust Air",

13. Ove Strindenhag, "How Does Recovery and Air Recirculation affect the Air Quality", Flakt Evaporator AB, VVS N0. 9, 1979

14. A Persily, "Evaluation of An Air-to-Air Heat Exchanger", Environmental International, vol 8, pp. 453 - 459, 1982

15. Donald Pescod, "A Heat Exchanger for Energy Saving in an Air-Conditioning Plant", ASHRAE Trans 1979 (2)

16. L. Carnes, "Air-toAir Heat Recovery Systems for Research Laboratories", ASHRAE Trans. 1984 (24)

17. E G Phillips, R E Chant, D R Fisher & B C Bradley, "Comparison of freezing control strategies for residential air-to-air heat recovery ventilators", ASHRAE Trans, 1989, Vol 95 (2).

THE AIR INFILTRATION AND VENTILATION CENTRE was inaugurated through the International Energy Agency and is funded by the following thirteen countries:

Belgium, Canada, Denmark, Germany, Finland, France, Netherlands, New Zealand, Norway, Sweden, Switzerland, United Kingdom and United States of America.

The Air Infiltration And Ventilation Centre provides technical support to those engaged in the study and prediction of air leakage and the consequential losses of energy in buildings. The aim is to promote the understanding of the complex air infiltration processes and to advance the effective application of energy saving measures in both the design of new buildings and the improvement of existing building stock.

Air Infiltration and Ventilation Centre
University of Warwick Science Park
Sovereign Court
Sir William Lyons Road
Coventry CV4 7EZ
Great Britain
Tel:+44 (0)1203 692050
Fax:+44 (0)1203 416306
Email Address: AIRVENT@AIVC.ORG

Operating Agent for International Energy Agency, The Oscar Faber Partnership, Upper Marlborough Road, St. Albans, UK