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Investigation of a domestic heating system with ventilation heat recovery: Performance and integrity

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SUMMARY

Domestic heating systems with a heat exchanger are generally assessed for efficiency by the ratio of primary energy input / delivered energy output. In practice, performance depends on all the components in the heat delivery system and on their matching. In the air heating system addressed here, the components include: the gas burning air heater, supply ducts, return ducts, heat recovery system, controls, fans, filters and pumps. This paper describes experiments conducted on a test house in Bath during the years 1991 to 1993. The house has been retrofitted with an air heating system and a ventilation heat recovery unit which also recovers heat from flue gases. The aim of these experiments was to characterize the house and the retrofit system in terms of ventilation and energy performance. In particular, airflows through the heat exchanger, the heat losses occuring in the ducts and the air leakage from the ducts to the crawl space were investigated. To match the real values most of the experiments were conducted with the settings of the house as used by the occupants. This included the thermostat, the position of the delivery grilles and the window openings. Tracer gases have been used extensively to measure air change rates and an original method to assess air leakage from ducts has been developed.

1. INTRODUCTION

The house studied is part of a European Commission Demonstration project, with the aim of investigating the benefits of sunspaces and heat recovery systems as a mean of pre-heating ventilation air. The three bedroom test bungalow, located in Bath, has a floor area of 170 m^2 and a volume of 400 m^3. It has been retrofitted with a ducted warm heating system and a ventilation heat recovery unit (VHR) which recovers heat from both stale air and flue gases. The return ducts run in the attic whereas the supply run in the crawl space, outside the heated envelope. Temperatures across the heat exchanger, outside and in the lounge are recorded by six thermistors installed by British Gas and stored in a Grant squirrel data logger, together with the gas consumption. The performance of the VHR depends on the system as a whole.

In the present case, the underfloor duct system has been installed in an existing house. When first installed it suffered from major heat loss to the crawl space from the ductwork due to the difficulties of installation associated with retrofitting. These heat losses outweighed the benefits of the VHR unit. This has motivated a set of experiments, which investigated the efficiency of the installation and the source of the heat losses.

1.1 Coefficient of Performance (COP) of a Heat Exchanger

The total heat delivered to the incoming air is the sum of heat from the heat exchanger and the fans (Figure 1):

$$H_{hr} = H_{ex} + P_{fan} \quad (1)$$

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

$$H_{hr} = m_{in} C_p (T_{sup} - T_{int}) \quad (2)$$

The overall coefficient of the system is:

$$\eta_{hr} = \frac{H_{hr}}{P_{hr}} \quad (3)$$

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

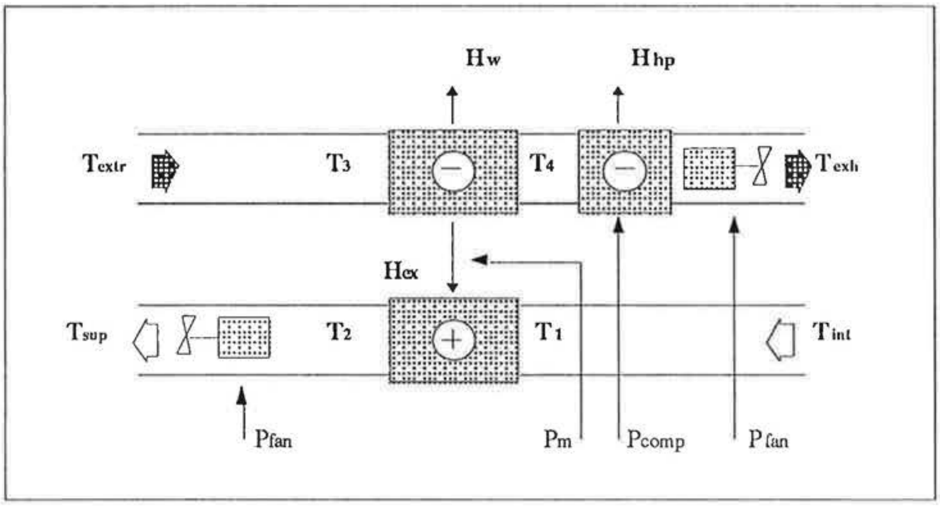

Figure 1 : Temperatures T, heat flows H and power inputs P for an air to air heat exchanger followed by a heat pump

1.2 Use of Tracer Gas to Detect Leaks

The poor accessibility of the duct system suggested the use of a technique based on tracer gas to determine the leakage. Tracer gas was injected into the return air duct located in the corridor. The boiler ran continuously and provided fresh air to the whole

house. Obvious cracks as well as 13 out of the 16 ventilation air bricks around the crawl space were obstructed to prevent unwanted infiltration and of the three remaining bricks two were used as air inputs and one as output (sampling). Input and output were located on two opposites sides of the house to provide the maximum air mixing in the crawl space. A fan provided an extract rate of 120 m^3/h and the air was then sampled downstream using a Brüel and Kjaer multi-gas analyser (photoacoustic detector). As a fraction of the return air is reinjected in the ducts it has been necessary to monitor the tracer concentration both at the delivery grilles and around the return grille. Due to leakage, a certain amount of tracer left the duct and entered the crawl space (CS) at a concentration C_0 and a flow rate F_0. The air still flowing in the ducts was delivered to the house at concentration C_1 and flow rate F_1. The air sampled downstream of the fan was at concentration C_2 and flowrate F_2. Therefore the different mas balances are given as follows :
in the ducts:

$$q_1(t) = \Sigma C_1(t)F_1 + \Sigma C_0(t)F_0 \tag{4}$$

in the crawl space (C.S):

$$C_{out}F_{out} + \Sigma C_0(t)\,F_0 = C_2(t)F_2 \tag{5}$$

Assuming that the concentration of tracer in the outdoor air is negligible and that there is a good mixing in the ducts and in the crawl space (C_1 identical for all grilles and C_0=C_1) then :

$$q_1(t) = C_1(t)\,\Sigma F_1 + C_0(t)\,\Sigma F_0 \tag{6}$$

$$C_1(t)\,\Sigma F_0 = C_2(t)F_2 \tag{7}$$

The air delivered is a mixing of fresh air (C_{out}, F_{out}) and return air (C_{ra}, F_{ra}). As the injection of tracer takes place in the return air at a rate q, the mass balance can then be applied as follows :

$$C_1(t) = \frac{q}{F} + \frac{F_{ra}}{F}C_{ra}(t) \tag{8}$$

(F, F_{ra}, q being constant with time).

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e VHR fans on and the air heater on. A separate
set of points, the separation being made on the
extract temperature. Efficiency ranges from 50%
10°C to 20°C (air heater off), and from 40% to
.0°C to 70°C (air heater on). Despite a relatively high efficiency (50 to 70%) the rise in temperature stays modest at around 3.5°C, which gives a power of 0.22 kW (with an airflow of 185 m^3/h). If one assumes a consumption of 85W for each fan, this gives a COP of 1.28. This study on the efficiency of the heat exchanger puts forward two types of behaviour related to the setting of the boiler (on or off). The reason is that the flue gases are mixed with extract air and thus provide more heat when on. When the boiler is off, a linear regression can be found between supply and inlet temperatures (Figure 3) and efficiency can rise up to 70%. But the low temperature difference between supply and inlet (3.5°C) finally gives a power of only 0.22 kW. When the boiler is on, efficiencies vary between 40 and 70%, but lead to a power of 1kW due to a temperature rise across the heat exchanger of 17°C.

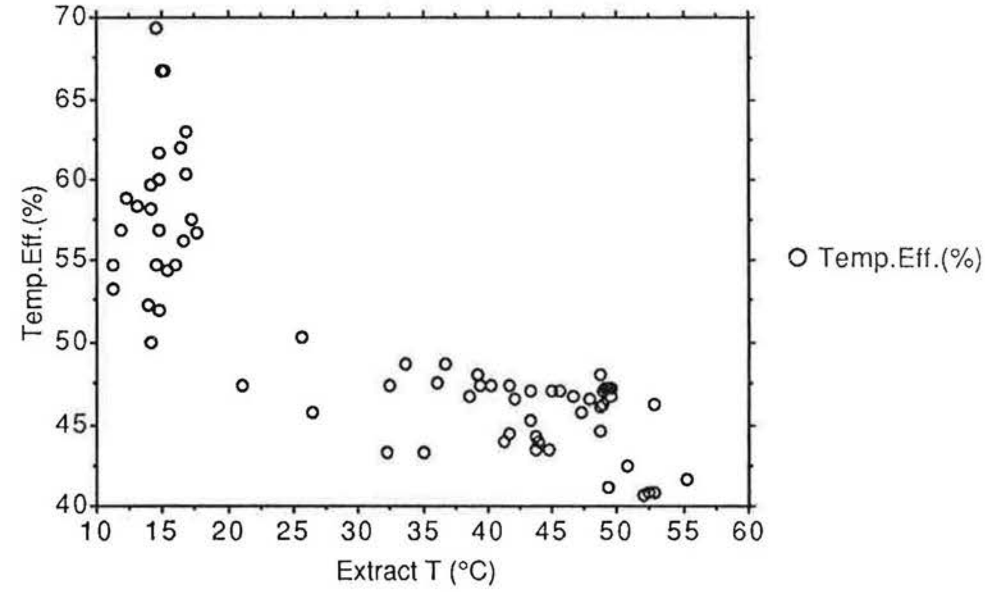

Figure 2 : Relation between temperature efficiency and extract temperature for the heat exchanger

Boiler off :	Boiler on :
$T_{rise} = 5.24 - 0.2\ T_{int}$	$T_{rise} = 1.93 + 0.41\ T_{ext} - 0.46\ T_{int}$
$T_{exh} = 0.88 + 0.16\ T_{ext} + 0.85\ T_{sup}$	$T_{exh} = 1.82 + 0.17\ T_{ext} + 0.75\ T_{sup}$
$\text{Temp Eff} = 76.53 - 5.12\ T_{rise}$	

Figure 3 : Heating performance of the heat exchanger

2.2 Heat Losses of the House

The measurement of heat loss in the ducts was conducted with the house in its operating mode as set by the occupants (thermostat set up, delivery grille position) and the air heater ran continuously until a relatively steady state was achieved. Two sets of measurements were performed, each on a different day, with the same conditions except the weather. The first set (1 record every 10 s) took place on a cloudy and slightly windy day whereas the second one (1 record every 30 s) enjoyed a calm, sunny day. Despite these differences, no significant changes in the results have been noticed. A constant injection tracer gas method was used to measure air flows off the boiler (which is the total airflow delivered to the house) and at the grilles where temperatures were also recorded. Although not all delivery temperatures were recorded, it is assumed that the temperatures measured at some specific grilles are likely to be the same for all the grilles, as the good repartition of heat throughout the house seem to indicate. Therefore, formula (10) gives the heat losses from the overall difference of temperature occurring between the boiler and the delivery grilles (about 20°C) :

$$P = \rho F C_p\, \Delta T \qquad (9)$$

This results to a duct heat loss of 3.8 kW assuming there is no air leakage. When the boiler is on, the supply air coming out of the heat exchanger is around 20°C in most cases. Assuming that the boiler heats this air to the mean temperature of 80°C as previously seen, formula 10 gives (for the total flow delivered to the house) a power of about 15.4 kW. Thus, the heat loss in the ducts accounts for 25% of the total power for space heating. This value does not take into account the losses due to air leakage. This duct heat loss can be compared to the heat recovered by the heat exchanger. When the air heater is off and assuming an airflow of 185 m^3/h and a temperature difference of 17°C between inlet and supply temperatures, one finally obtains a power of 1 kW for the VHR, to be compared to 3.8 kW for the heat loss in the ducts. The VHR power rises to 2.5 kW when we consider the case with the boiler on and a 40 °C inlet / supply temperature difference. These values show that the energy savings due to the heat exchanger are outweighed by the losses in the ducts. In this particular case, the 19 mm thick insulation was inadequate, especially when the ducts run in the crawl space.

2.3 Air Leakage from the Ducts

Two sets of experiments have been performed with one week interval. The airflow measurements in the main duct and across the heat exchanger suggest a fraction of recirculated air of 70% (390 m^3/h for recirculated air and 570 m^3/h for total flow). This value was confirmed experimentally by the ratio of concentration delivery grille / return air, proportional to the fraction of recirculated air. It was found to be 72%, compared to the design value of 76% (recirculated air : 670 m^3/h, total flow : 875 m^3/h). As a way of checking the results a comparison between the concentration out of the grilles which has been measured and the one calculated using C_{ra} and formula (8) has been carried out. The curve shows a very good correlation and a slope of 1.0005 which confirms the formula. As for the leakage, a linear relation between C_2 and C_1 has been found in both cases. The slopes are respectively 1.35 and 1.55 with a regression coefficient of about 0.98 (figure 4).

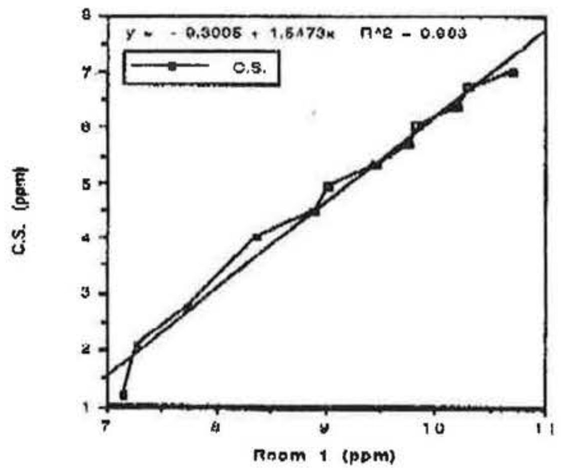

Figure 4 : SF6 concentration in the crawl space

From these values Formula (7) gives the total air leakage which is here between 160 and 180 m^3/h. This represents 30% of the total flow F for the whole house. Even by considering the 10% often tolerated by ventilation engineers between design and real values, this value clearly shows a big problem in the underfloor ducts. When a system is fitted in a new house, it is unlikely to produce such amount of leakage. Here the main problems must have arisen during the installation when ducts were successively joined and pushed under the floor.

The amount of heat loss due to air leakage (180 m^3/h, air at 62°C) can be evaluated from formula (9) at 3.7 kW. Therefore one needs to take this leakage into account when calculating the heat loss due to lack of insulation. One can assume that this loss corresponds to the temperature difference for the remaining flow rate (i.e. total flow less air leakage). These losses in the underfloor ducts are reduced from 570 to 390 m^3/h at 20°C, which is 2.6 kW. In summary, the heat losses are as follows :

Table 1 : Heat losses in the house

Heat losses	
Due to air leakage:	3.7 kW
Due to poor insulation:	2.6 kW
Total	6.3 kW

3. CONCLUSIONS

The house studied has been found very airtight with 0.16 and 0.50 ach with ventilation on and off respectively. Thus any leakage from the ductwork has a larger importance as it accounts for an important part of the air pattern. The method developed here, based on SF_6 detection to measure underfloor air leakage is valuable since it is easy to install and does not require large pieces of equipment, but it would require further investigation to confirm its reliability. The measurements carried out indicate a leakage rate of 180 m^3/h which accounts for 30% of the total flow delivered to the house (570 m^3/h). The heat loss associated is 3.7 kW, whereas the total heat losses rise up to 6.3 kW (including the loss due to poor insulation of the ducts : 2.6 kW). The comparison with the power both of the boiler (15.4 kW) and of the heat recovered by the exchanger (1 kW) clearly highlights that the performances of the heat exchanger are outweighed by the losses. The method for evaluating duct air loss into the crawl space is being refined and applied to the attic. This problem of duct air system needs to be taken into account when houses are to be retrofitted. Nowadays the trend in construction is for superinsulated houses and airtight buildings. In such buildings the need for a system with minimum leaks is more and more important as the energy requirements are low.

4. NOMENCLATURE

Cp = Specific heat of air at constant pressure, J kg^{-1} K^{-1}
η_{hr} = cop of whole system
ρ = Volumetric mass of air, kg m^{-3}
Hex = Heat transferred by heat exchanger, W
Hhr = Total heat supplied to incoming air, including useful fan power, W
P =Heat loss from ducts, W
P_{fan} = Power consumption of one fan, W
P_{hr} = Total power consumption, W
P_m = Power consumption of auxiliary motors, W
q = Injection rate of tracer, m^3 h^{-1}
t = time, h
Tint = Intake air temperature, °C
Text = Extract air temperature, °C
Texh = Exhaust air temperature, °C
Tsup = Supply air temperature, °C

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