Performance of heat recovery in passive stack ventilation systems

L Shao, S B Riffat and G Gan
Institute of Building Technology, University of Nottingham, Nottingham NG7 2RD

The large heat loss from Passive-stack ventilation (PSV) systems quite often makes natural ventilation systems unattractive and it is therefore desirable to implement heat recovery in PSV stacks. As the stack pressure is usually about a few Pascal, it is crucial that the heat recovery unit used in a PSV system produces even lower pressure loss, which is extremely difficult to achieve with the conventional plate heat exchangers. This work is concerned with an a low pressure-loss heat recovery device based on heat pipes. The heat pipe is a completely passive device without power consumption and its simple construction also means that it also has a low initial cost. Experimental investigation has been carried out using four types of heat pipe heat exchangers. Heat recovery efficiency of over 60% has been obtained using two banks of exchangers. It was also found that the efficiency decreases with increasing air velocity. Spine fin exchangers provided much lower efficiency than plain fin systems. Louver fined system produced the greatest efficiency but also the largest pressure loss. The wire-fin type produced a lower pressure loss than the plain fin type although its efficiency is also slightly lower. It was concluded that the wire fin type provided the optimum balance between the requirements for low pressure loss and high efficiency.

Introduction

Natural ventilation based on passive stacks (PSV) has been applied to many types of modern buildings [1, 2] but virtually all PSVs are designed without heat recovery leading to significant heat loss. It has been estimated that this heat loss amounts to 3 - 15 GJ per annum for a small family residence and much more for larger buildings, e.g. offices [2]. The absence of heat recovery is because the pressure loss caused by a conventional heat exchanger is large compared with the stack pressure and could cause the ventilation system to fail. Research work on heat recovery in natural stack ventilation has been carried out by Schultz and Saxhof [3] using a counterflow heat exchanger. This design had a high pressure drop and was not suitable for PSV systems.

Heat pipes offer an alternative approach for heat recovery in naturally-ventilated buildings and have several advantages. The heat pipe consists of a sealed pipe lined with a wick and partially filled with a working liquid. Its operating principle is described in a separate paper [4] by the authors. The heat pipe has very high thermal conductance. It does not require complicated channels for supply and exhaust air and individual heat pipes can be independently located in the stacks, making it easier to achieve lower pressure drops. These features make the heat pipe suitable for heat recovery in natural-ventilation systems. The heat pipe is a completely passive device without power consumption and its simple construction also means that it should have a low initial cost. This paper presents results of a experiment study and computer simulations of a heat-pipe heat recovery system for use in naturally-ventilated buildings.

Experimental set-up

Fig. 1 shows the schematic diagram of the test chamber. The external dimensions were 1.2 X 1.2 m floor area and 2.4 m high, as shown in the figure. The net internal volume of the chamber was 3.09 m^3. The cross-sectional area for both channels that contain the heat exchanger elements was 215 X 215mm. The chamber was divided into two zones by a horizontal partition with an opening in the middle of the partition. The partition serves to prevent possible short-circuiting of supply and return air. The chamber was made of plywood. There was a 25.4 mm layer of expanded polystyrene insulation on the interior of the chamber to reduce the influence of surroundings. A heat-pipe heat recovery unit was housed in the supply and exhaust ducts for heat exchange between return and supply air. A 500 W halogen lamp and ten 100 W general lighting services bulbs were used to simulate heat production in the chamber. The heat production rate could be adjusted in 100 W steps. To test the performance of the heat recovery unit at different air flow velocities, an axial fan with adjustable speed was used for inducing forced air flow in the chamber and no buoyancy driven flow was involved. As the heat pipe performance is primarily a function of air velocity and temperature difference rather than the way in which the flow is generated, such an arrangement was appropriate for this study and simplified the experimental set-up.

Investigation of the efficiency of the heat recovery unit requires measurement of air temperatures and flow rates. The heat recovery efficiency, η, is given by:

$$\eta = \frac{T_s - T_i}{T_r - T_i} x100\% \tag{1}$$

where T_i and T_s are the temperatures of air before and after the heat pipe condensers, respectively, and T_r is the temperature of return air. Thermocouples were used to measure temperatures upstream and downstream of the heat recovery unit in both supply and exhaust ducts. In addition, the temperature of air in the chamber was measured using a thermocouple in the middle of the partition opening. Before each experiment, the test chamber was heated under appropriate conditions for 2 hours to reach a steady state.

The constant-injection tracer-gas method was used for the measurement of air flow rate. The facility for the test was developed and proved accurate as part of an EPSRC funded research at Nottingham. The air tightness of the system was checked using smoke testing to prevent tracer gas leakage which may affect measurement accuracy. The method basically involves release of a tracer gas (SF_6) at a constant rate, q (m^3/s), at the entrance of the supply duct. The concentration of tracer gas, C, is monitored in the exhaust duct. The air flow rate, Q (m^3/s) is given by q/C and the duct mean velocity is Q/A where A is the duct cross-section area.

Four types of heat pipe heat exchangers have been tested and they are shown schematically in Figure 2. The first type of heat recovery unit consisted of one or two banks of externally finned heat pipes. Each bank had seven heat pipes 12.7mm in diameter and 450mm in length with 72 continuous plain fins on both the condenser and evaporator sections. The fins were also made of copper. The dimensions of each fin were 215 mm long, 48 mm high and 0,45 mm thick. The cross-sectional area for both the condenser and evaporator sections was 215 X 215mm. The overall dimensions of each bank were 450X215X48 mm. The total surface area or heat transfer including fins and exposed pipes for each section was 1.4229 m^2.

The second type of heat pipes had needle-like cylindrical spine fins. The cross-sectional areas of the evaporator and condenser were identical to those for type 1. The fins were made of copper wire 0.7 mm in diameter. A unit with this type of fin consisted of three heat pipes. The pipes had the same dimensions as those used in the Type 1 heat pipes. There were eight continuous rows of fins on each of these pipes. Each row had about 300 spine fins and each spine fin was 30mm long. The estimated total surface area of the spine fins for each heat pipe was 0.158 m^2.

The third type consisted of two rows of staggered heat pipes, each row having three heat pipes. Each pipe was 18 mm in diameter and 365 mm in length with 70 continuous louvered aluminium fins. The louvered fins are basically plain fins pressed to form regular array of louver openings. The dimensions of each fin were 180 mm long, 60 mm high and 0.45 mm thick. Each fin had 96 louvers with 2 mm spacing between neighbouring louvers. The gap of the louver opening is about 8.5 mm long and 0.65 mm wide. The cross-sectional areas of the heat recovery unit were 180 mm X 180 mm in both the condenser and evaporator sections. The total surface area for heat transfer within each section is 1.5414 m^2.

The fourth type of heat recovery unit was made of five heat pipes with wire fins. Wire fins were made by winding a coil around the heat pipe to cover its full length. The coil was soldered to the pipe external wall to give a metallic bond between the wire fins and the pipe. Each pipe was 19.05 mm in diameter and 450 mm in length with 34.5 turns of coil along the length of the heat pipe contained within each of the condenser and evaporator sections. Each turn had 65 loops, which have an outer diameter of 12 mm. The wire had a diameter of 0.65 mm. The overall dimensions of the unit was 450X215X43 mm. The total surface area for heat transfer for each of the evaporator or condenser sections was 0.6035 m^2.

Results and discussion

Figure 3 shows the heat recovery efficiency of the wire fin type unit in comparison with that of the plain fin type unit. The performance of the former was about 9% lower at 1 m/s and the reduction was smaller at other test velocities. There may be two main reasons for the reduction: The total surface area available for heat transfer in the wire fin type was about 55% lower than that of the plain fin type and there were 5 heat pipes in the former compared to 7 heat pipes in the latter. If the heat transfer area and number of heat pipes were increased to the same levels of the plain fin type, it is expected that the heat recovery efficiency would be comparable or higher than that of the plain fin type.

It was also found that at the same velocity the heat recovery is between 16% and 17% more efficient (about 40% relative increase) using two banks of plain heat pipes than using one bank. Heat recovery efficiency of over 60% has been obtained using two banks of exchangers. The efficiency decreases with increasing air velocity. The efficiency of a one-bank exchanger reduced from 45% at 0.5m/s to 23% at 4m/s. The heat recovery efficiency for the 3 pipe unit with wire fins was less than 25% of that of the plain fin type and even when the performance is extrapolated for a unit with 7 equivalent heat pipes the efficiency is still far lower. The louver fin type was consistently more efficient over the entire range of velocities that are likely to be encountered in real stacks. This is likely to be due to the larger surface area available for heat transfer in louver fins and the flow disturbance/turbulence generated by the louvers which generally help to improve heat transfer.

The pressure loss through a heat pipe unit is represented by the pressure loss coefficient or k-factor as follows:

$$k = \frac{\Delta P_h}{\frac{1}{2}\rho V^2} \tag{2}$$

where ΔP_h is the static pressure loss across the unit (Pa) and ρ is the air density (kg/m^3). Experimental measurements of pressure losses across heat recovery units were carried out and the results are shown in Figure 4. It is seen that, at air velocities greater than 2 m/s, the pressure loss of the wire-fin unit was higher than that of the plain-fan unit. At 2 m/s, the pressure loss across the two units was nearly equal. Because it is difficult to measure small pressure differences accurately, pressure loss measurement below 2 m/s was not carried out. However, pressure loss in that velocity range can be estimated by extrapolation based data presented in Figure 11. For example, at velocities lower than 2 m/s, the pressure loss of the wire-fin type would be lower than that for the plain type by about 10-20%. The greater rate of pressure loss increase of the wire fin type may be explained by the opposite effects of the two types of fins on turbulence. The plain fins could act as a flow straightener, helping to keep flow laminar whereas wire fins generally enhances turbulence generation.

CFD simulation of the heat pipes reported by the authors [5] showed that at a velocity of 0.5 m/s, the pressure loss through one section of the plain heat pipe unit was about 0.57 Pa and total pressure loss through the whole unit (both condenser and evaporator sections) was just over 1 Pa. The computer simulation also indicated that pressure loss of the louver fin type unit was much higher. This is because the louvers tend to increase flow disturbance or turbulence which generally cause greater pressure losses

Table 1 shows the relative performance of the four types of heat recovery units investigated in this study. The pressure loss performances are given as a ratio of that of the plain fin type unit. The heat recovery performances are listed as ratios of that of the plain fin type unit as well as in absolute values. All values are for an average stack velocity of 0.5 m/s and based on experimental measurement (except the pressure loss for the louver fin unit). It can be seen from the table that the heat recovery performance of the spine fin unit was far lower than those of the other types. The louver fin unit produced a higher heat recovery efficiency but also a much higher pressure loss. The value for the pressure loss would actually be even greater because the fins were considered smooth in the computer simulations. The high pressure loss made this type unattractive to natural ventilation where the driving force is normally weak. The wire fin type offers lower pressure loss compared to the plain fin unit but also a slightly lower heat recovery efficiency. Its low pressure loss is particularly desirable for application in natural ventilation. To reduce the costs of manufacturing units of one-off designs, commercially available units or units requiring limited tooling changes had to be used, and heat exchangers with different dimensions and number of heat pipes etc. have been used in the experiments. As a result, comparing the performance of the units was sometimes not a straightforward matter. This would need to be improved on in future studies.

Conclusions

Experimental investigation has been carried out using four types of heat pipe heat exchangers. The heat pipe is a completely passive device without power consumption and its simple construction also means that it also has a low initial cost. Heat recovery efficiency of over 60% has been obtained using two banks of exchangers. It was also found that the efficiency decreases with increasing air velocity. Spine fin exchangers provided much lower efficiency than plain fin systems. Louver fined system produced the greatest efficiency but also the largest pressure loss. The wire-fin type produced a lower pressure loss than the plain fin type although its efficiency is also slightly lower. It was concluded that the wire fin type provided the optimum balance between the requirements for low pressure loss and high efficiency.

Acknowledgement

The authors wish to acknowledge the financial support of the European Union for this work under the JOULE Programme.

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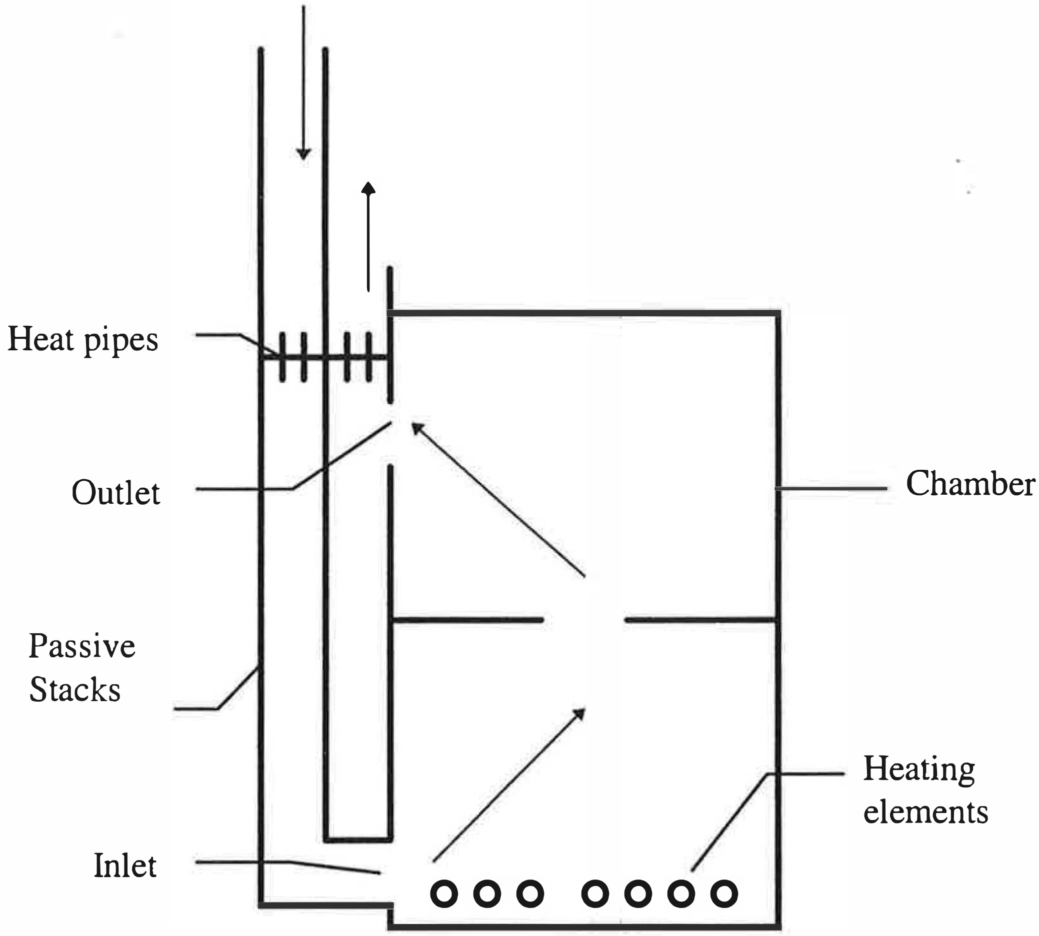

Fig. 1. Schematic of the experimental set-up.

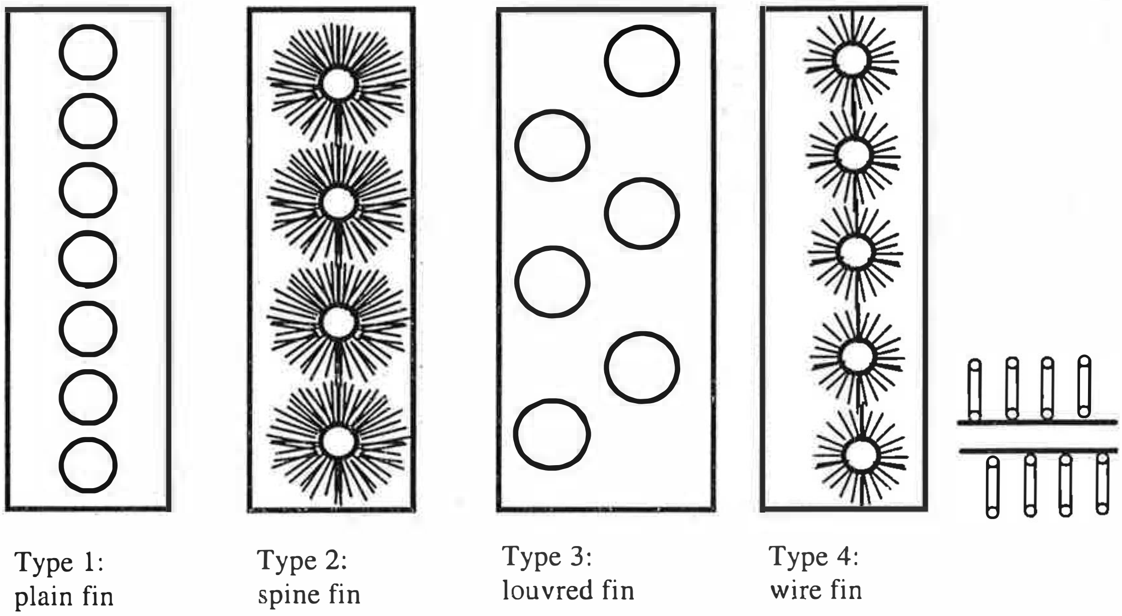

Fig. 2. Schematic of four types of heat recovery units.

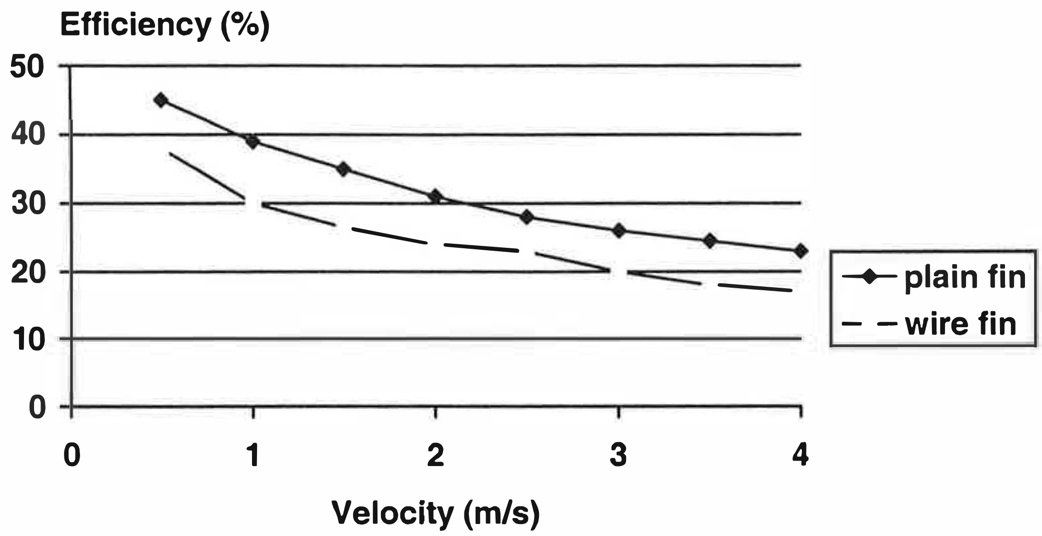

Fig. 3. Heat recovery efficiency of wire fin and plain fin units

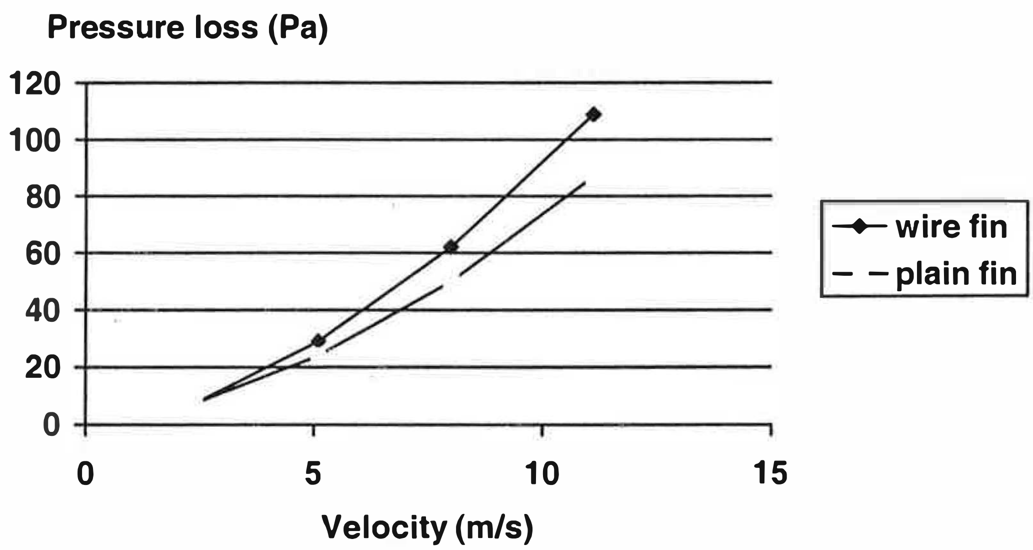

Fig. 4. Measured pressure loss in wire fin and plain fin units.

V=0.5 m/s	ΔP	η	
Type I (plain fin)	ΔP_I	43%	η_1
Type II (spine fin)	—	28%	$0.65\ \eta_1$
Type III (louver fin)	$1.27\ \Delta P_I$	47%	$1.09\ \eta_1$
Type IV (wire fin)	$0.9\ \Delta P_I$	37%	$0.86\eta_1$

Table 1. Performances of four types of heat recovery units at a stack velocity of 0.5m/s.