AIVC 11726

ELSEVIER

Energy and Buildings 28 (1998) 179–184

ENERGY AND BUILDINGS

Heat recovery with low pressure loss for natural ventilation

L. Shao *, S.B. Riffat, G. Gan

Institute of Building Technology, Nottingham University, Nottingham NG7 2RD, UK

Received 28 December 1997; accepted 22 February 1998

Abstract

Heat recovery is difficult to implement in passive stack ventilation because the pressure loss is usually too high in conventional heat exchangers compared with the stack pressure. Laboratory investigation and computer simulation have been carried out on a low pressure-loss heat recovery device based on heat pipes which is suitable for application in passive stack systems and other systems where a low pressure loss is essential. It was found that heat recovery efficiency decreased with increasing air velocity. Heat recovery efficiency of close to 50% have been achieved using a single-bank plain-fin unit and the efficiency of a double bank unit was 40% higher than that of a single bank unit. It was also found that the pressure loss coefficient reduces as velocity increases but the reduction is around 10% over the entire range of common velocities in natural ventilation systems. The pressure loss across the plain fin unit is in the order of 1 Pa at a flow velocity of 1 m/s. The wire fin and plain fin type heat-pipe units are superior to other types investigated in this study.

Keywords: Heat recovery; Low pressure loss; Natural ventilation

1. Introduction

Passive stacks ventilation (PSV) is a form of natural ventilation driven by buoyancy created by air temperature difference between the interior and the external environment of a building. It is increasingly used in modern buildings [1,2] but virtually all PSVs are designed without heat recovery leading to wasteful heat loss. It was estimated that this heat loss amounts to 3–15 GJ per annum for a small family residence and much more for larger buildings [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 counter-flow heat exchanger. This design had a high pressure drop and was not suitable for PSV systems.

Heat pipes is more attractive for heat recovery in naturally-ventilated buildings and one of the possible arrangements of the heat pipes for this purpose is shown in Fig. 1. Its operating principle of heat recovery based on heat pipes is described in a separate paper [4] by the authors. The heat pipe offers several advantages over conventional devices for heat recovery in ventilation systems. The heat pipe has very high thermal conductance. It does not require complicated, flow resistant 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. This paper presents a study of pressure loss and heat recovery efficiency of heat pipe units, using both experimental and computational approaches.

2. Experimental and computational set up

Experiments were carried out in a two-zone test chamber with a heat-pipe heat recovery unit. Fig. 2 shows the schematic diagram of the chamber. The external dimensions were 1.2×1.2 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 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 10 100-W general lighting

* Corresponding author.

0378-7788/98/$ - see front matter
PII S0378-7788(98)00016-4

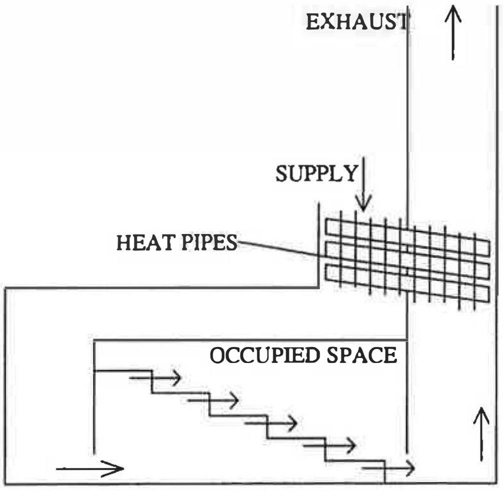

Fig. 1. PSV heat recovery using heat pipes.

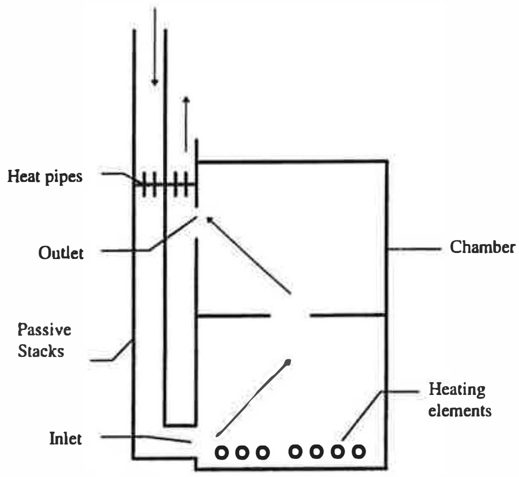

Fig. 2. Schematic of the experimental set-up.

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. For natural ventilation, a chimney with a height of 4.5 m above the chamber was used as an extension of the exhaust duct. As the chimney would be subjected to the influence of the outdoor environment, it was made of 50.8-mm thick polystyrene to provide thermal insulation.

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:

$$\eta=\frac{T_s-T_i}{T_r-T_i}\times 100\% \tag{1}$$

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 h 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. Fig. 3 shows the schematic representation of flow measurement. 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 (ppm), is monitored in the exhaust duct. The air flow rate, Q (m^3/s), is given by

$$Q=\frac{q}{C}\times 10^6 \tag{2}$$

The duct mean velocity, V (m/s), is then calculated from the measured flow rate and duct cross-section area ($A=0.215$ m$\times$0.215 m):

$$V=\frac{Q}{A} \tag{3}$$

Four types of heat pipe heat exchangers have been tested and they are shown schematically in Fig. 4. The first type of heat recovery unit consisted of one or two banks of externally finned heat pipes. Each bank had seven heat pipes 12.7 mm in diameter and 450 mm 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 215×215 mm. The overall dimensions of each bank were 450×215×48 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 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

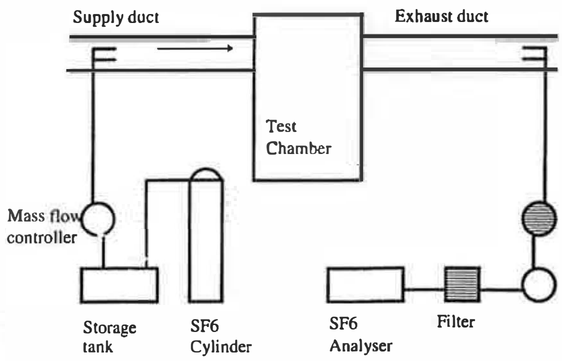

Fig. 3. Schematic of the set-up for air flow measurements.

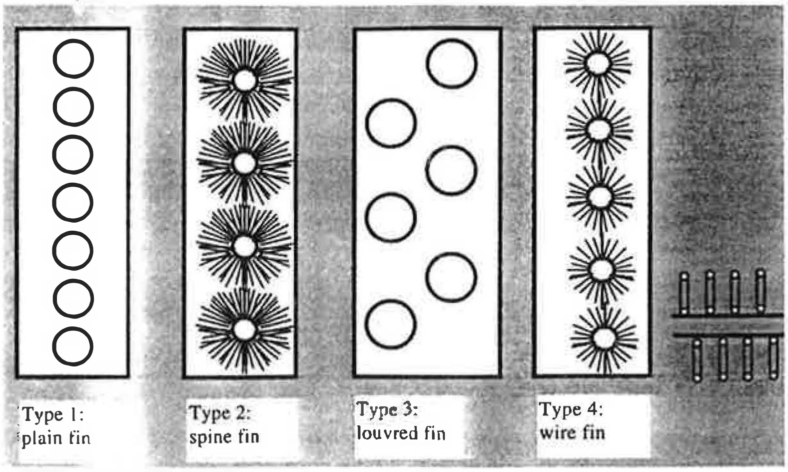

Fig. 4. Schematic of the four types of heat recovery units used in the study.

fins on each of these pipes. Each row had about 300 spine fins and each spine fin was 30 mm 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 × 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 450 × 215 × 43 mm. The total surface area for heat transfer for each of the evaporator or condenser sections was 0.6035 m^2.

CFD modelling was also carried out to simulate pressure loss through the heat pipe unit. The predictions were carried out using the CFD package FLUENT. In the predictions, the heat pipe unit was modelled as a bank of rectangular tubes such that it had the same free-area ratio and thickness as the real heat pipes. Because it would require an enormous number of cells to represent each fin, it was not possible to model individual fins of heat pipes. Therefore, the fins were modelled as uniformly distributed rectangular studs on both sides of heat pipes such that the total cross-sectional area of studs was the same as the sum of that of fins. This approximation would affect accuracy of the prediction but nevertheless would still allow indicative values and comparative performances to be obtained.

3. Results and discussion

Fig. 5 shows the variation of the efficiency of the heat recovery unit with air velocity in the duct (air velocity in PSV systems are typically around 0.5–1 m/s). It can be seen that at the same velocity the heat recovery is between 16% and 17% more efficient (about 40% relative increase) using two banks of heat pipes than using one bank. The air velocity has a significant effect on the efficiency of heat recovery. The efficiency decreases with increasing air velocity. The relationship between the efficiency and velocity can be represented by the following correlations for the velocity range investigated: for one bank,

$$\eta = 1.37V^2 - 12.77V + 49.93 \quad (4)$$

for two banks,

$$\eta = 1.30V^2 - 12.74V + 66.72 \quad (5)$$

Fig. 6 shows the performance for heat recovery of the spine fin type unit, together with that of the plain fin unit. The heat recovery efficiency for the three-pipe unit with wire fins was

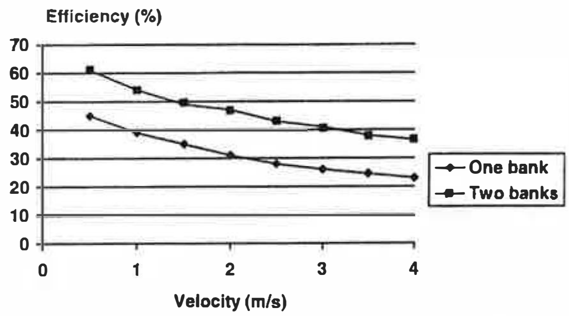

Fig. 5. Heat recovery efficiency of plain fin units.

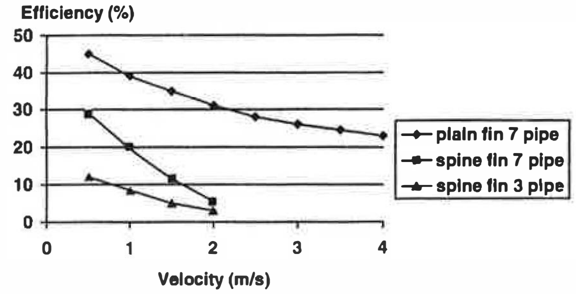

Fig. 6. Heat recovery efficiency of spine fin and plain fin units.

less than 25% of that of the plain fin type and even when the performance is extrapolated for a unit with seven equivalent heat pipes the efficiency is still far lower, particularly at higher velocities. The poor performance of the wire fin type is probably due to the limited contact areas between the fins and the heat pipe, which restrict the flux of heat flow to the fins from the heat pipe.

Heat recovery efficiency of the louver fin type unit is compared with that of the plain fin type in Fig. 7. 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.

Fig. 8 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

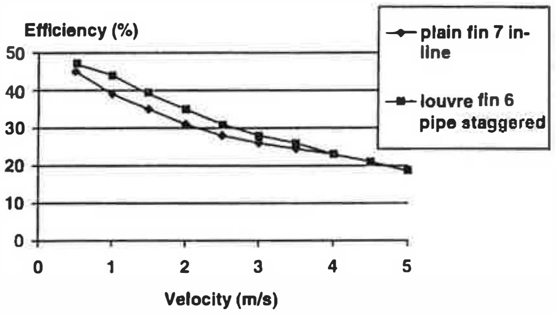

Fig. 7. Heat recovery efficiency of louver fin and plain fin units.

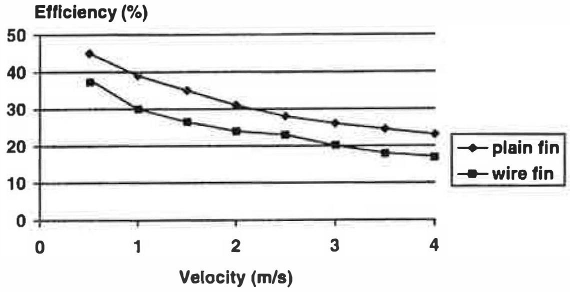

Fig. 8. Heat recovery efficiency of wire fin and plain fin units.

55% lower than that of the plain fin type and there were five heat pipes in the former compared to seven 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.

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

$$k=\frac{\Delta P_{\mathrm{h}}}{1/2\rho V^2} \tag{6}$$

where ΔP_{h} is the static pressure loss across the unit (Pa) and ρ is the air density (kg/m^3).

Fig. 9 shows the predicted pressure loss coefficient for one bank of heat pipes with plain fins. It is seen that the pressure loss coefficient decreases with the increasing duct mean velocity and the rate of decrease is particularly significant at low velocities. The overall reduction is however less than 12% for the velocity range. When the velocity is above 8 m/s which corresponds to a Reynolds number of about 10^5, the pressure loss coefficient levels off to becomes a constant. This is in agreement with the well known observation that above a critical Reynolds number of 2×10^5, the pressure loss coefficient would be independent of the Reynolds number.

It was found that at a velocity of 0.5 m/s, the pressure loss through one section (condenser section or evaporator section) of the 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. Thus, if the driving pressure available for ventilation is, say, 1 Pa, the mean velocity through the heat pipe unit should not be more than 0.5 m/s.

Fig. 10 shows the predicted pressure loss of the louver fin type unit. As simulation of the louver openings would mean excessive requirements for computing resources, they are ignored in the computer modelling. The resultant pressure loss data therefore would be approximate and only reflect the effect of fin/pipe area, spacing and dimensions. The predicted loss coefficient for the six staggered heat pipes was higher than that for the plain fin unit despite the porosity (free-area ratio) of the former being higher than that of the latter. Obviously, if the louver effect are taken into account the pressure loss would be even higher as the louvers tend to increase flow disturbance or turbulence which generally cause greater pres-

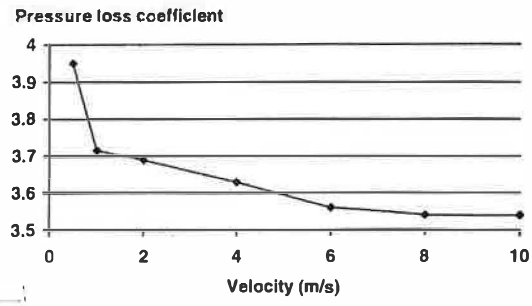

Fig. 9. Predicted reduction of pressure loss coefficient with velocity in the plain fin unit.

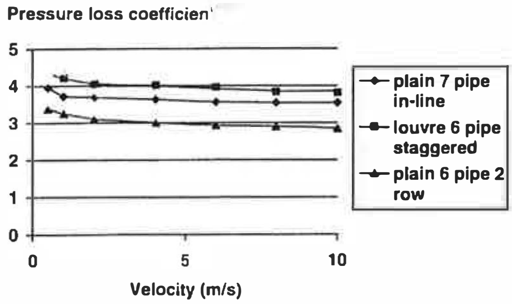

Fig. 10. Predicted pressure loss coefficients in louver fin and plain fin units.

sure losses. Fig. 10 also shows that if the heat pipes in a heat recovery unit are arranged into two identical rows, which are parallel to each other but perpendicular to the flow, the flow loss will be reduced.

Experimental measurements of pressure losses across heat recovery units were also carried out and the results are shown in Fig. 11. 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 Fig. 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. As a result, the pressure loss increase is more in the wire fin type unit.

Comparison between computer prediction and experimental measurement of pressure loss in the plain fin unit showed that the former produced higher values than the latter. For example, at the velocity of 0.5 m/s the predicted pressure loss across a single bank unit was 0.57 Pa while that based on the measurement would be about 0.25 Pa. At the velocity of 1.0 m/s the predicted pressure loss was 2.25 Pa compared to 1.0 Pa based on experimental measurements. The error is probably attributable to the approximation used for the modelling of fins as described in the previous section. Nevertheless, the ratio of the CFD predicted value to measure value

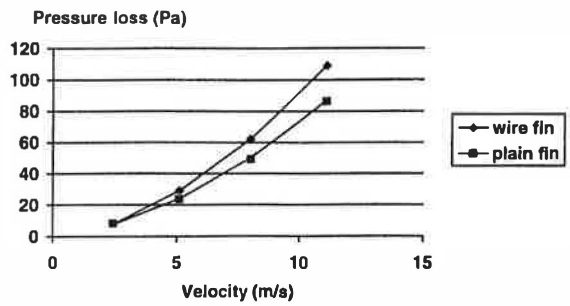

Fig. 11. Measured pressure loss in wire fin and plain fin units.

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

$V = 0.5$ m/s	ΔP		η
Type I (plain fin)	ΔP_1	43%	(η_1)
Type II (spine fin)	–	28%	(0.65 η_1)
Type III (louver fin)	1.27 ΔP_1	47%	(1.09 η_1)
Type IV (wire fin)	0.9 ΔP_1	37%	(0.86 η_1)

remains relatively constant, indicating that CFD may be used for examination of the relative performances of two units.

Comparison of the performances of plain fin units with one or two banks of heat pipes have shown that the two-bank configuration produced a heat recovery efficiency 40% higher than that of the single-bank unit. However, the two-bank unit is not completely superior to the one-bank unit because computer simulation showed that it also produced a pressure loss 30% higher than the single bank unit. Selection of the configuration to use has to be made according to the specific requirements and circumstances in individual cases.

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. The plain fin unit listed have a single bank of heat pipes and the data for spine fin type are for seven heat-pipes. 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 and was considered inferior to others for heat recovery in natural ventilation stacks. (The pressure loss of the unit, although not tested due to practical limitations, is expected to be larger than that of the wire fin unit because it has nearly twice as much surface area but a similar fin structure). 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 calculations. 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 and its overall performance is broadly comparable to that of the plain fin unit.

4. Conclusions

Experimental and computational investigations have been carried out into low pressure-loss heat-recovery in passive-stack natural-ventilation systems. The measurements showed that air velocity has a significant effect on the efficiency of heat recovery based on heat pipes. The heat recovery efficiency decreased with increasing air velocity. Heat recovery efficiency of close to 50% has been achieved using a single-bank plain-fin unit and adopting a double-bank configuration would increase the efficiency by about 40%. (e.g., to just under 70% at a flow velocity of 0.5 m/s). It was also found that the pressure loss coefficient reduces as velocity increases but the reduction is small, at around 10% over the entire range of common velocities found in PSV systems. Both computer simulations and experimental measurements have indicated that the pressure loss across the plain fin unit is in the order of 1 Pa at a flow velocity of 1 m/s. Pressure loss predicted by CFD is about twice the value measured in experiments but the consistency of CFD prediction found in this study meant that it is useful for examining the relative performance of two units. Based on the pressure and heat recovery results it was concluded that the wire fin and plain fin type heat-pipe units were superior to other types investigated in this study.

Acknowledgements

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

References

[1] R. Bunn, Learning curve, will natural ventilation work, Bldg. Service, October, 1993, pp. 20–42.
[2] M. Woolliscroft, The relative energy use of passive stack ventilators and extract fans, Proceedings of the 15th AIVC Conference, UK, 1994, pp. 245–256.
[3] J.M. Schultz, B. Saxhof, Natural ventilation with heat recovery, Air Infiltration Rev. 15 (1994) 9–12.
[4] L. Shao, S.B. Riffat, Flow loss caused by heat-pipes in natural-ventilation stacks, Appl. Thermal Eng. 17 (4) (1997) 393–399.