

NATURAL VENTILATION WITH HEAT RECOVERY USING HEAT PIPES

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Natural ventilation based on Passive-stacks are currently designed without incorporating heat recovery leading to wasteful heat loss. Heat recovery is not used because the pressure loss caused by conventional heat exchangers is large and could cause the ventilation system to fail. This paper presents laboratory investigation and computer simulation of a low pressure-loss heat recovery device for passive stack systems. It was found that the heat recovery effectiveness decreases with increasing stack velocity and heat recovery effectiveness of over 50% has been obtained in the experiments. It was also demonstrated that acceptable pressure losses can be achieved if the stack mean velocity is kept under approximately 1 m/s.

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. 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 AND COMPUTATIONAL SET UP

Experiments were carried out in a two-zone test chamber with a heat-pipe heat recovery unit. Fig. 1 shows the schematic diagram of the 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 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. The 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. A total of 72 plain copper fins (fin spacing was approximately 3mm) were mounted on the heat pipes within both the supply and exhaust ducts, each with a cross-sectional area of 215 X 215 mm. The dimensions of each fin were 215 mm long, 48 mm high and 0.45 mm thick. The whole unit was made of copper and the working fluid in the pipes was methanol. 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, 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 effectiveness of the heat recovery unit requires measurement of air temperatures and flow rates. The effectiveness of heat recovery, e, is given by:

$$\varepsilon = \frac{T_s - T_i}{T_r - T_i} x100\% \qquad (1)$$

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 (type T) 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. Fig. 2 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

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

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

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

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.

RESULTS AND DISCUSSION

Fig. 3 shows the variation of effectiveness 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 using two banks of heat pipes than using one bank. The air velocity has a significant effect on the effectiveness of heat recovery. The effectiveness decreases with increasing air velocity. The relationship between the effectiveness and velocity can be represented by the following correlations for the velocity ranges investigated:

for one bank,

$$e = 1.37\,V^2 - 12.77\,V + 49.93 \quad (r = 0.99) \tag{4}$$

for two banks,

$$e = 1.30\,V^2 - 12.74\,V + 66.72 \quad (r = 0.99) \tag{5}$$

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

$$k = \frac{\Delta P_h}{\frac{1}{2}\rho V^2} \qquad (6)$$

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

It is generally known that the pressure loss coefficient across a duct fitting is a constant when the Reynolds number (Re) based on the duct mean velocity and hydraulic diameter is greater than 2×10^5. For the duct used in the test, this would require the mean velocity to be over 10 m/s. When Re is less than the critical value, the coefficient may be dependent on Re. Fig. 4 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, within the velocity range between 0.25 and 7.5 m/s. The rate of decrease is particularly significant at low velocities. The maximum variation is however less than 12% for the velocity range. When the velocity is above 7.5 m/s which corresponds to about half of the critical Reynolds number, the pressure loss coefficient becomes a constant. The data of pressure loss coefficient for the velocity range from 0.25 and 7.5 m/s can be represented by the following correlation:

$$k = 3.7674 - 0.2684 \log V \qquad (r = 0.99) \qquad (7)$$

The pressure loss at a given velocity can be obtained from the pressure loss coefficient (= $\frac{1}{2}k\rho V^2$). It was found that at a velocity of 0.5 m/s, the pressure loss through one section of the heat pipe unit is about 0.57 Pa and total pressure loss through the whole unit (both condenser and evaporator sections) is 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. At the velocity 1 m/s, the pressure loss through both sections of the unit is 4.5 Pa. Without the wind effect, this would require a stack height of about 10 m at a temperature difference between inlet and exhaust openings of 10 K or 4 m height at 25 K temperature difference. In naturally-ventilated low-rise buildings, the average driving pressures are unlikely to exceed this value. Therefore, in designing ventilation ducts for housing this type of heat recovery unit, the duct mean velocity should be less than 1 m/s. Therefore, if a design requires higher flow rates, they should preferably be achieved by using a larger stack cross-section rather than higher velocities.

The pressure predictions also underline the difficulty in accurate determination of the pressure loss through a heat recovery unit at low velocities by experimental measurement. Since the maximum allowable pressure loss for natural ventilation is usually about 1 Pa, it would require sophisticated and expensive equipment to determine such small pressure differences accurately. A high degree of measurement accuracy is difficult to achieve even with such a instrument, because turbulent air flow and conditions of ducts and pressure tappings could result in errors of a similar magnitude to that of the total pressure loss.

CONCLUSIONS

The measurements show that air velocity has a significant effect on the effectiveness of heat-pipe based heat recovery. The effectiveness decreases with increasing air velocity. At a given velocity, the heat recovery is typically between 16% and 17% more efficient using two banks of heat pipes than using one bank. The pressure loss coefficient for the heat recovery unit decreases with the increasing duct mean velocity. When applying the heat pipe system for heat recovery in natural ventilation systems in low-rise/medium height buildings, the stack mean velocity should be less than 1 m/s and preferably below 0.5 m/s in order to achieve acceptable pressure loss. For higher buildings with taller stacks, higher design velocities would also be feasible.

ACKNOWLEDGMENT

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

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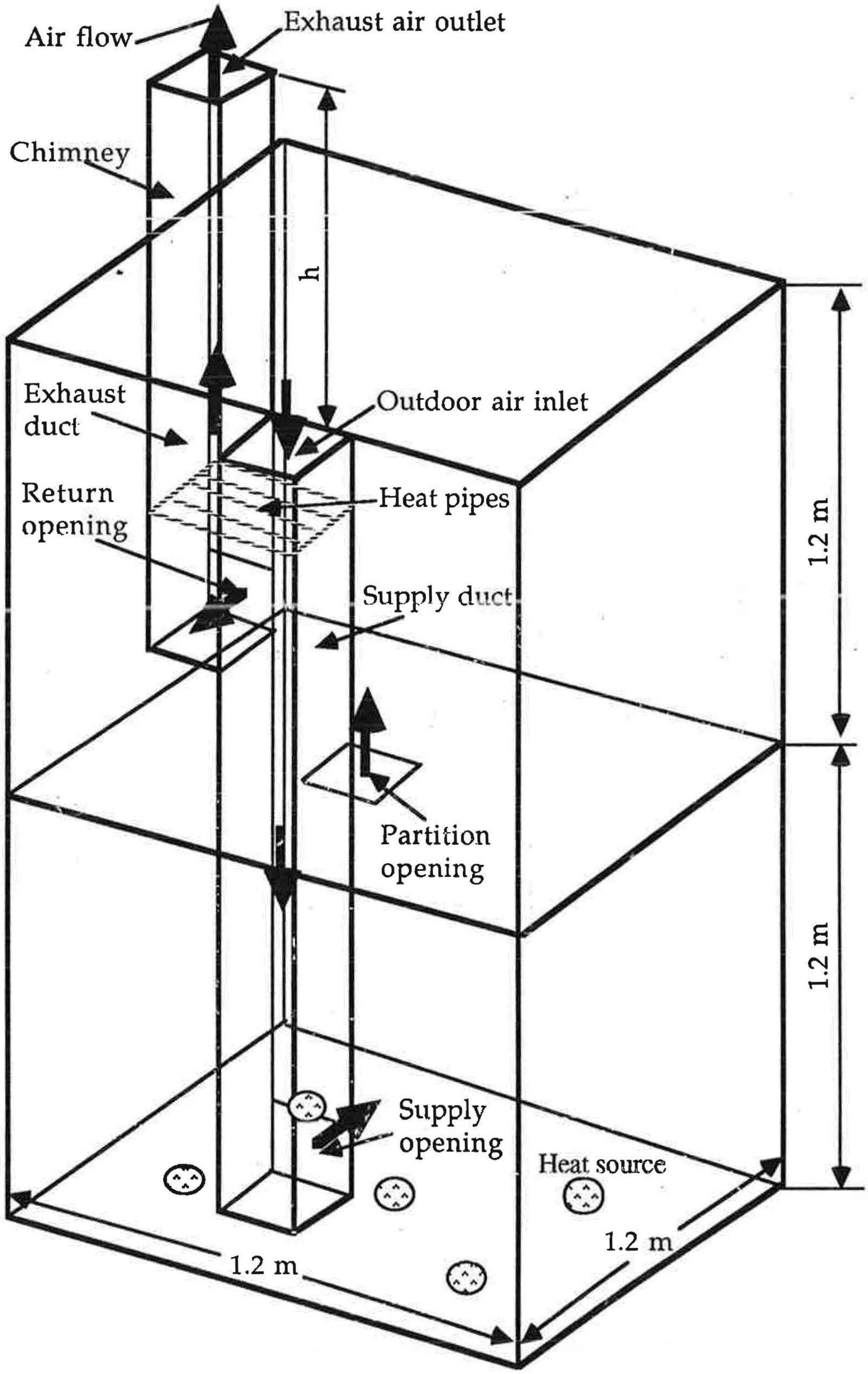

Fig. 1 Schematic of the naturally-ventilated two-zone chamber with heat pipes

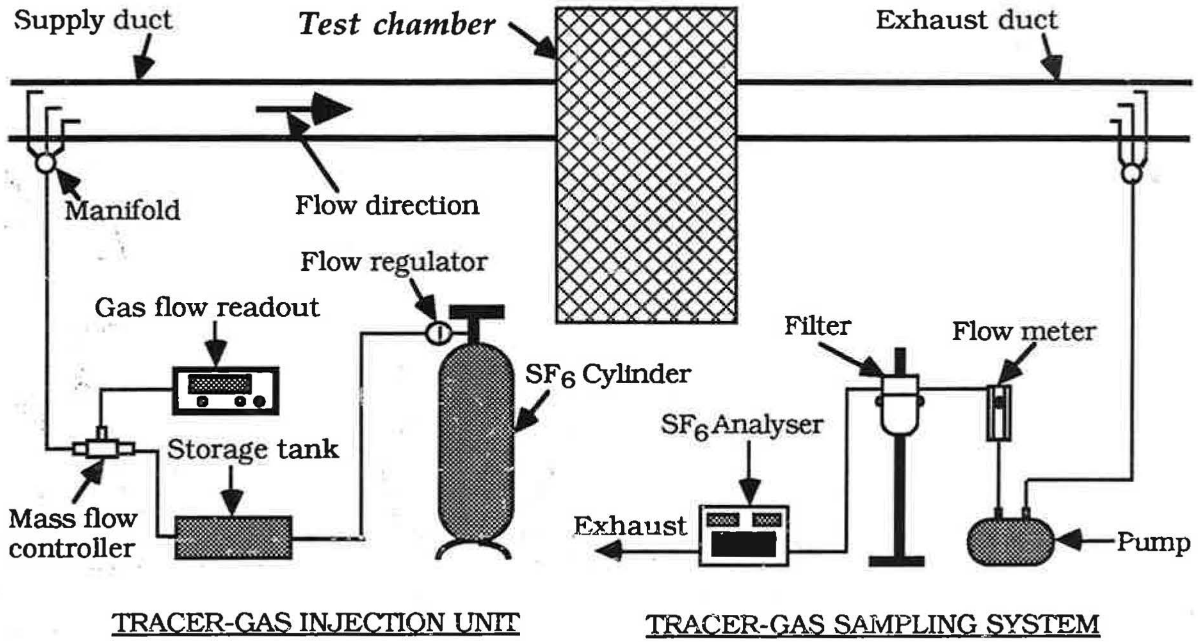

Fig. 2 Schematic diagram of the experimental setup for measurement of air flow rate

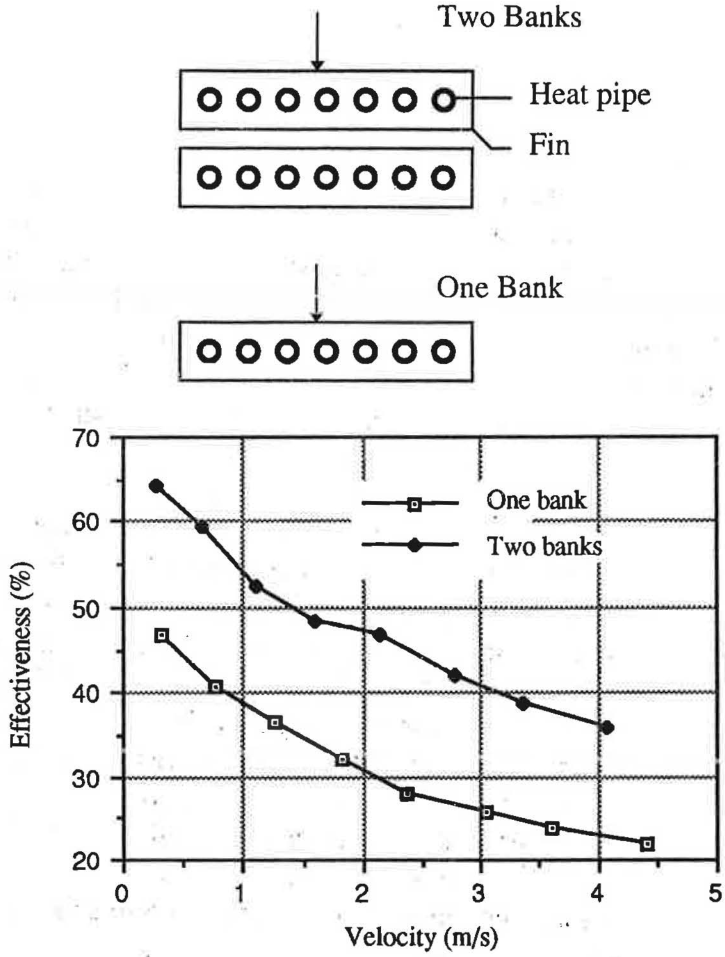

Fig. 3 Measured effectiveness of heat-pipe heat recovery

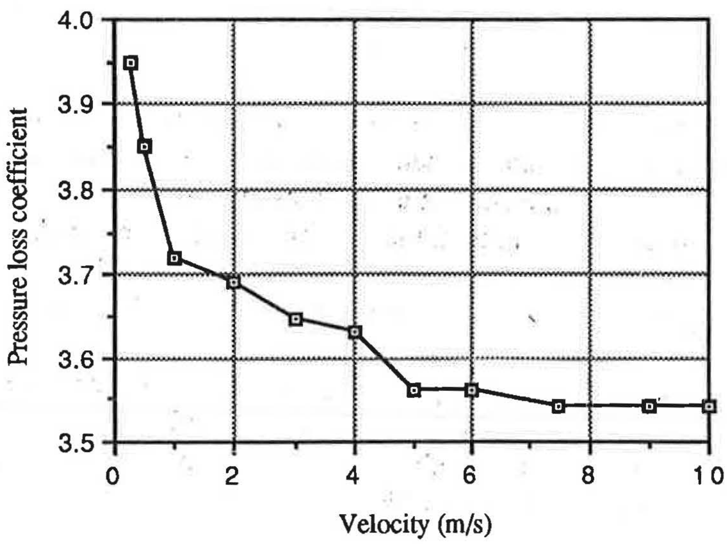

Fig. 4 Predicted pressure loss coefficient for one bank of heat pipes