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Summary The performance of four types of heat-pipe heat recovery unit for naturally ventilated buildings was determined in terms of effectiveness and pressure drop. The effectiveness of the heat recovery units was tested in a two-zone chamber. The pressure loss characteristics of the heat recovery units were determined using computational fluid dynamics (CFD) and experimental measurement. CFD was also used to evaluate the performance of a solar chimney for heat recovery in naturally ventilated buildings.

A study of heat-pipe heat recovery for natural ventilation

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List of symbols

A	Cross-sectional area of a duct (m^2)
C	Concentration of tracer gas in exhaust duct (ppm)
k	Pressure loss coefficient (dimensionless)
n	Number of heat-pipe banks
Q	Air flow rate ($m^3\ s^{-1}$)
q	Injection rate of tracer gas ($m^3\ s^{-1}$)
T_i	Temperature of inlet air (°C)
T_r	Temperature of return air (°C)
T_s	Temperature of supply air (°C)
V	Mean velocity of air in duct ($m\ s^{-1}$)
ΔP_s	Static pressure loss across a heat-pipe unit (Pa)
ε	Effectiveness of a heat-pipe unit (%)
ρ	Air density ($kg\ m^{-3}$)

1 Introduction

It has been estimated that ventilation accounts for 30% or more of space conditioning energy demand but that as much as 70% of this energy can be reclaimed through efficient ventilation heat recovery(1). Commercially available ventilation heat recovery systems are exclusively for mechanically ventilated buildings. However, most domestic buildings in the UK are naturally ventilated. In recent years, a number of effective natural ventilation systems have also been developed for nondomestic buildings such as the Queens Building at De Montfort University(2), but no consideration has been given to heat recovery from naturally ventilated buildings. Pressure loss is a crucial parameter that limits the use of heat recovery with natural ventilation. Developing a heat recovery system with low flow resistance is a challenging task. A system that has the potential to provide substantial heat recovery without significant pressure loss could be one employing heat pipes.

A heat-pipe heat recovery unit is a heat exchanger consisting of externally-finned sealed pipes using a working fluid such as methanol or water. The unit is divided into two sections, i.e. the evaporator and the condenser, for heat exchange between exhaust and supply air(3). The performance of a heat recovery unit is normally assessed according to the effectiveness of heat recovery and pressure loss across the unit. The effectiveness and pressure loss depend on air velocity and both can be determined experimentally and numerically.

Solar chimneys and/or wind towers are employed to enhance movement of room air and hence to effect the heat exchange between supply and exhaust air in naturally ventilated buildings. To increase the solar heat absorption and ventilation rate, a south-facing wall of the solar chimney is glazed and the interiors of other walls are blackened and the exteriors are insulated. The insulated walls of a solar chimney serve to collect and store solar energy so as to boost the buoyancy effect and enhance stack ventilation. The solar heat gain is stored in the chimney wall to temper the effect of the changing outdoor environment so that the extra buoyancy effect by the solar chimney can still be induced when solar radiation is temporarily low, for example owing to the passage of clouds. Insulating the storage wall reduces the heat loss to the cold outdoor air and thus helps the buoyancy effect of the solar chimney. The air flow in such chimneys is complicated and their performance needs to be investigated.

The objective of this project was to determine the optimum construction of a heat-pipe heat recovery unit for naturally ventilated buildings. This was achieved by designing four heat-pipe units and then assessing their performance on the basis of measured effectiveness and pressure loss characteristics(3). This paper presents a methodology for determining the performance of a heat-pipe unit, either standalone or integrated into a passive stack system. For the latter, computational fluid dynamics (CFD) is used to predict the performance of a glazed solar chimney for heat-pipe heat recovery in naturally-ventilated buildings.

2 Effectiveness of heat recovery

The effectiveness of a heat-pipe heat recovery unit for heat exchange between supply and exhaust air, ε(%), is defined as:

$$\varepsilon = \frac{T_s - T_i}{T_r - T_i} \tag{1}$$

where T_i is the temperature of inlet air before the condenser section (°C), T_s is the temperature of supply air after the condenser section (°C) and T_r is the temperature of return air before the evaporator section (°C).

Measurements of the effectiveness were carried out in a vertical two-zone test chamber with a heat-pipe heat recovery unit. The two-zone chamber (see Figure 1) was designed to allow good mixing of supply air with room air in the lower zone and maintenance of a uniform temperature and concentration of return air in the upper zone, which was confirmed by detailed air flow simulations(4). This ensured the reliability of temperature and air flow measurements.

2.1 Test chamber

Figure 1 shows the schematic diagram of the test chamber. The chamber was made of plywood insulated with a layer of polyurethane. The chamber had a net interior base area of 1.324 m^2 and a total height of 2.335 m. It was divided into two zones with a horizontal partition. There was an opening (0.215 m × 0.215 m) in the middle of the partition to allow air to flow from one zone to another. Supply and exhaust ducts were connected to the chamber on one of the vertical walls. The air ducts were also made of plywood. When the unit was in operation, air entered the lower zone of the chamber via the supply duct and return air was extracted from the upper zone through the exhaust duct. A heat-pipe heat recovery unit was housed in the supply and exhaust ducts for heat exchange between return and supply air. Light bulbs were used to simulate heat production in the chamber. An axial-flow fan with adjustable speed was connected to the exhaust duct by means of a flexible duct to generate air flow through the chamber.

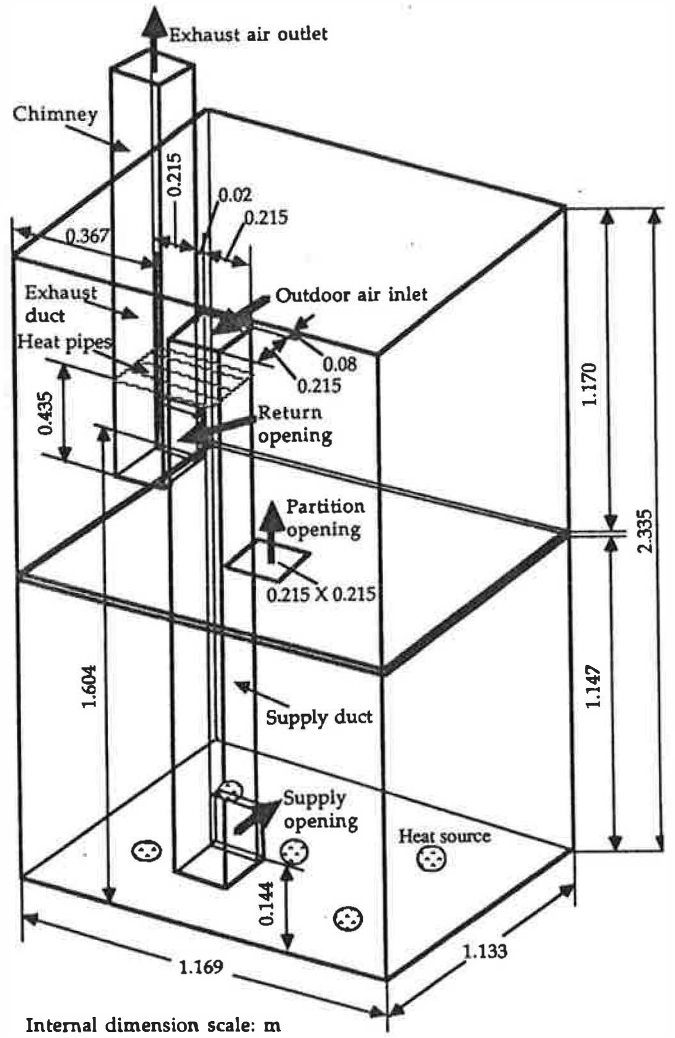

Figure 1 Schematic diagram of the two-zone test chamber

2.2 Heat pipes

Four types of heat-pipe heat recovery unit were constructed and tested. They included heat pipes with plain fins, spine fins, louvred fins and wire fins[3]. Only the heat-pipe unit with plain fins is discussed here.

The heat-pipe heat recovery unit consisted of a bank of seven externally finned heat pipes. Each pipe was of 0.0127 m outside diameter and 0.45 m long with 72 continuous plain fins on both the condenser and evaporator sections. Each fin was 0.215 m long, 0.048 m high and 0.45 mm thick. There was a 0.02 m divider on the outside of the heat pipes and at the middle of the bank to prevent cross-contamination of return and supply air. The cross-sectional area for both the condenser and evaporator sections was 0.215 m × 0.215 m. The total surface area of one bank of heat pipes, including fins and exposed pipes, was 1.372 m^2. The whole unit was made of copper. The working fluid in the pipes was methanol with an operating temperature range from −40° to 100 °C.

2.3 Measurement

Temperatures upstream and downstream of the heat recovery unit in both supply and exhaust ducts were measured using thermocouples (type T: copper–constantan). The temperatures were recorded by a data logger.

The air flow rate was measured using the constant-injection tracer-gas method. Figure 2 shows the schematic represen-

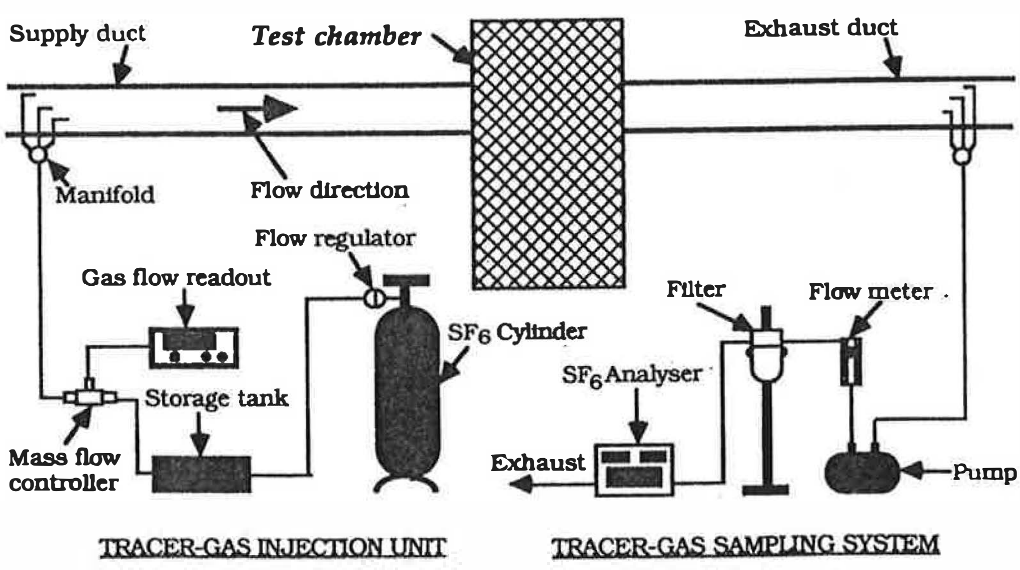

Figure 2 Schematic diagram of air flow measurement

tation of air flow measurement. The method involves release of a tracer gas (SF_6) at a constant rate at the entrance of the supply duct. The concentration of tracer gas is monitored in the exhaust duct. The air flow rate, Q ($m^3\ s^{-1}$), is given by

$$Q = \frac{q}{C_v} \tag{2}$$

The mean air velocity, V ($m\ s^{-1}$), is then calculated from

$$V = \frac{Q}{A} = \frac{q}{CA} \times 10^6 \tag{3}$$

In equations (2) and (3), q is the injection rate of tracer gas ($m^3\ s^{-1}$), A is the cross-sectional area of the duct (m^2), C_v is the concentration of tracer gas in volume ratio, and C is the concentration of tracer gas in unit of parts per million (ppm) and so requires the multiplier 10^6 on the right-hand side of equation (3).

2.4 Results and discussion

Tests were performed at mean air velocities ranging from 0.3 to 5.3 $m\ s^{-1}$. This velocity range encompasses both natural and forced ventilation for heat-pipe heat exchangers in practical use. The design mean air velocity for forced ventilation normally ranges from 2 to 4 $m\ s^{-1}$; for natural ventilation the velocity is below 1 $m\ s^{-1}$, and for the application to this work it is between 0.5 and 1 $m\ s^{-1}$.

Figure 3 shows the measured effectiveness for the heat recovery unit with one and two banks. It can be seen that at the same air velocity the heat recovery was between 16% and 17% more efficient using two banks of heat pipes than when using one bank.

The air velocity was found to have a significant influence on the effectiveness of heat recovery. The relationship between the effectiveness and velocity can be represented by the following correlations.

For one bank:

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

For two banks:

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

where r is the correlation coefficient.

3 Pressure loss across heat pipes

The pressure loss across a heat-pipe unit is represented by the pressure loss coefficient (k) as follows:

$$k = \frac{\Delta P_s}{\frac{1}{2}\rho V^2} \tag{6}$$

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

The pressure loss coefficient for the heat-pipe unit was determined using CFD modelling and measurement.

3.1 CFD prediction

It has been shown that CFD can be used to predict pressure losses through duct fittings[5]. We therefore applied this method to predicting the pressure loss coefficient for the heat-pipe unit. A commercial CFD package was used[6]. The heat pipe unit was modelled as one to four banks of rectangular cylinders such that it had the same free-area ratio and thickness as did the real heat pipes. The fins were modelled as uniformly distributed rectangular studs on both sides of heat pipes such that the total cross-sectional area of the studs was the same as the sum of those of fins. This ensured that the mean velocity and Reynolds number of air gaps were the same as for the heat pipes.

Figure 4 shows the predicted pressure loss coefficient of the heat pipes; it is seen that the loss coefficient, like effectiveness, varies with air velocity. The predicted pressure loss coefficient is related to the air velocity by:

$$k = (2.6 + 1.177n)\,V^{-0.03n^{3/4}} \tag{7}$$

where n is the number of heat-pipe banks.

The pressure loss at a given velocity can be obtained from the pressure loss coefficient (which equals $\frac{1}{2}k\rho V^2$). For example, at a velocity of 0.5 $m\ s^{-1}$, the pressure loss through one section of one bank of heat pipes 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 $m\ s^{-1}$. At the velocity 1 $m\ s^{-1}$, 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 of 4 m height at a temperature difference of 25 K. In naturally ventilated low-rise buildings, the average driving pressures are unlikely to exceed this value. In any case, it is not practical to provide a 10 m stack on top of a building. Therefore, in designing ventilation ducts for housing this type of heat recovery unit, the duct mean velocity should be less than 1 $m\ s^{-1}$.

3.2 Measurement

For assessment of the accuracy of CFD predictions, the flow resistance of the heat pipes was measured; the same instrumentation was used as shown in Figure 2, but the chamber

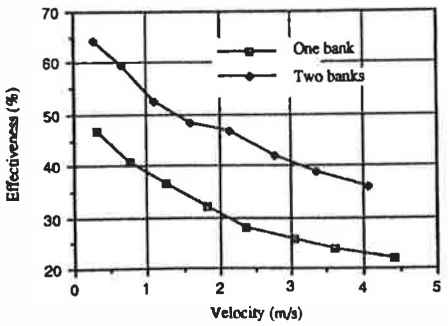

Figure 3 Measured heat recovery effectiveness for the heat pipes

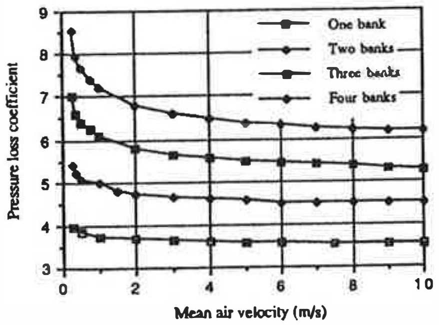

Figure 4 Predicted pressure loss coefficient of the heat pipes

was replaced by the heat pipes and pressure taps were fitted on the upstream and downstream ducts. The ducts had the same cross-sectional area as each section of heat pipe units (0.215 m × 0.215 m). Details of the experimental measurement of pressure loss coefficient of duct fittings are given in reference[5].

The measured pressure loss coefficient of the heat pipes for velocities higher than 0.4 m s^{-1} can be correlated as follows.

For one bank:

$$k = 2.10\, V^{-0.44} \qquad (r = 0.99) \qquad (8)$$

For two banks:

$$k = 4.56\, V^{-0.517} \qquad (r = 0.99) \qquad (9)$$

Figure 5 shows that the effect of air velocity on the measured pressure loss coefficient was larger than that on the predicted loss coefficient. The measured pressure loss coefficient was lower than the prediction for velocities above 1 m s^{-1} and higher than the prediction for velocities below 0.5 m s^{-1}. However, the uncertainty in the loss coefficient at low velocities was also large because the precision of instrumentation for pressure measurements was 1 Pa, which corresponded to velocity 0.5 m s^{-1} approximately. The pressure loss and loss coefficient for velocities below 0.5 m s^{-1} should therefore be used with caution. On the other hand, in practice, the cross-sectional area of a solar chimney is much larger than that used in the test, so that the air velocity in a natural ventilation stack is often below 0.5 m s^{-1}. Although exact values of pressure loss are difficult to determine for very low velocities, extrapolation suggets that the pressure loss through this type of heat-pipe unit will be less than 1 Pa when the air velocity is below 0.5 m s^{-1}.

For velocities between 0.5 and 1 m s^{-1}, the predicted values lay almost in the middle of variation of the measurements. Since the design air velocity for this type of heat recovery unit was within the range 0.5 to 1 m s^{-1}, the predicted pressure loss coefficient could also be used for calculating the pressure drop across heat pipes.

4 Performance of a solar chimney for heat-pipe heat recovery

The performance of a glazed solar chimney that would house the evaporator section of heat pipes in Switzerland[7] was simulated using an in-house CFD program. Details of the program model and solution of the model equations are described by Gan and Riffat[8]. The program was validated by comparing the numerical prediction with the experimental results of Bouchair[9] for natural convection in a heated solar chimney.

Simulation was carried out for steady-state stack ventilation in winter with heat recovery using the plain-fin heat pipes. Figure 6 shows the section of the glazed solar chimney. The chimney was 3 m tall and its horizontal cross-section was 0.2 m × 1 m with the major dimension along the eastwest orientation. The south face was double glazed and the other faces were made of brickwork blackened on the interior surface and insulated on the exterior surface. The glazing had an absorptivity for direct solar radiation of 0.2 and this was used to calculate the glazing solar heat gain for given solar irradiance. The inlet opening had the same dimensions as the chimney cross-section, i.e. 0.2 m high and 1 m wide. As a base simulation, heat pipes were first assumed to be absent in the chimney. The effect of installing heat pipes on the chimney performance was then compared.

The chimney solar heat gain was calculated from the mean total solar irradiance and mean solar gain factor. The calculated mean solar heat gain on the vertical south surface between 0800 h and 1600 h sun time on December 21 at 45° north latitude (close to Switzerland)[10] was 280 W m^{-2}. The outdoor air temperature was taken to be 0°C. The exhaust air from a building that entered into the chimney was assumed to be at 20°C and 50% relative humidity. This humidity level is higher than that of outdoor air at, say, 0°C. However, the room exhaust air would have higher humidity than the outdoor air owing to production of moisture by occupants.

Figure 7 shows the predicted air flow pattern and temperature distribution on the centre plane near the exit of the solar chimney. The predicted ventilation rate through the chimney was 0.106 m^3 s^{-1}. The air temperature near the insulated heated wall was higher than that near the glazing with conduction heat loss. Consequently, the velocity profile showed a peak near the heated wall and low magnitude near the glazing.

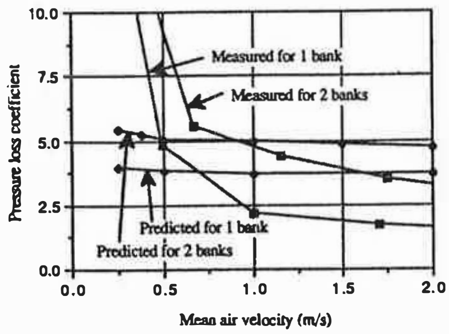

Figure 5 Comparison of predicted and measured pressure loss coefficient of the heat pipes

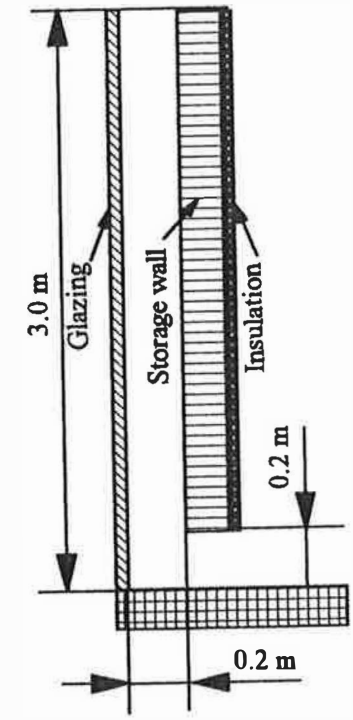

Figure 6 Section of the glazed solar chimney

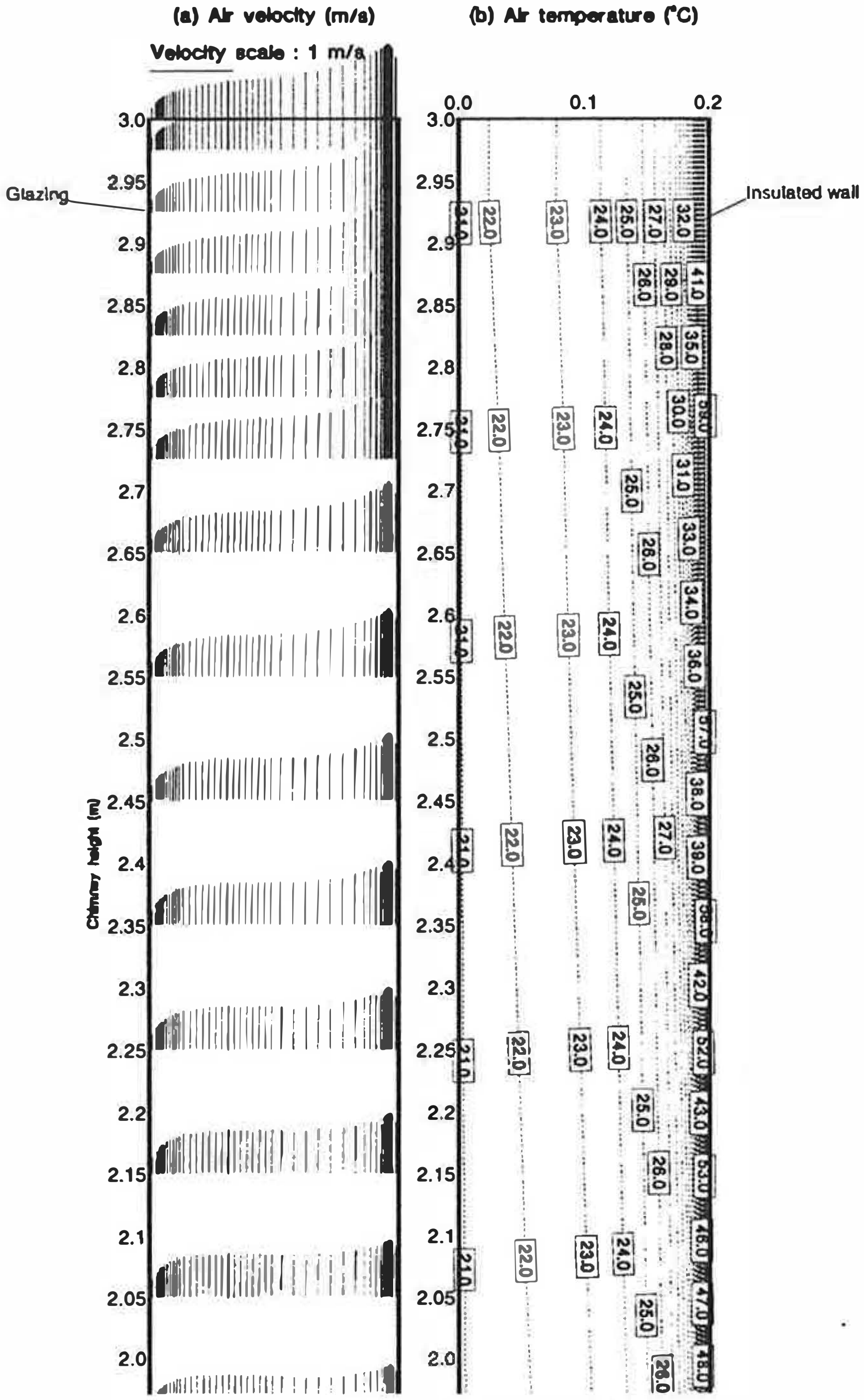

Figure 7 Predicted centreline air flow pattern and temperature distribution near the exit of the solar chimney

The ventilation rate was affected by the type of glazing. Using single glazing instead of double glazing would result in reduced ventilation rates owing to the effect of downdraught and moisture condensation on its interior surface for the simulated exhaust air conditions and ventilation rates(8).

The predicted ventilation rate would be lower if the effect of an internal flow obstruction such as a heat-pipe unit were taken into consideration. The flow resistance of a heat-pipe unit at a given velocity is represented by the pressure loss coefficient. For a unit of two banks of heat pipes with plain fins, it is given by equation (9). Thus, the predicted ventilation rate through the chimney with heat pipes, but without considering the cooling effect due to heat recovery, decreased from 0.106 $m^3 s^{-1}$ to 0.065 $m^3 s^{-1}$, a reduction of nearly 40%.

When in operation, heat pipes in the chimney not only increase the pressure drop but also decrease the air temperature after the evaporator section and so reduce draught. When the cooling effect of heat pipes in the chimney is taken into account, the predicted ventilation rate would decrease further. The predicted ventilation rate was 0.045 $m^3 s^{-1}$ when heat-pipe heat recovery was effected, a net decrease in ventilation rate of 0.020 $m^3 s^{-1}$ compared with the case without the cooling effect (0.065 $m^3 s^{-1}$). The overall effect of heat pipes due to increased flow resistance and decreased air temperature in the chimney was thus a reduction in the ventilation rate by nearly 60% (from 0.106 $m^3 s^{-1}$ for the chimney without heat pipes). This effect should be taken into account when designing a heat recovery system for naturally ventilated buildings.

5 Conclusions

The performance of heat pipes in terms of effectiveness and pressure drop has been determined using CFD modelling and measurement. The performance of a solar chimney for heat-pipe heat recovery in naturally ventilated buildings has been evaluated using CFD.

Air velocity is found to have a significant effect on the effectiveness of heat-pipe heat recovery. The effectiveness decreases with increasing air velocity. At the same velocity, heat recovery is between 16% and 17% more efficient using two banks of heat pipes with plain fins than when using one bank.

The pressure loss coefficient of the heat pipe unit decreases with increasing duct mean velocity. CFD overpredicts the effect of velocity on the pressure loss coefficient; that is, the predicted pressure loss coefficient is higher than that measured for velocities above 1 m s^{-1} and lower than that measured for velocities below 0.5 m s^{-1}. However, in the velocity range between 0.5 and 1 m s^{-1}, the predicted pressure loss coefficient is close to the measured value. When a heat recovery unit is used for natural ventilation without the full use of solar energy or wind force, the duct mean velocity should be less than 1 m s^{-1} according to pressure losses.

The performance of a glazed solar chimney is affected by the type of glazing for given indoor and outdoor air conditions. Double glazing is preferable to single glazing for solar chimneys due to possible condensation and downdraught in cold winter conditions. Installing heat pipes for heat recovery in the solar chimney increases pressure drop while reducing the stack effect, and consequently reduces ventilation rates. To achieve a design flow rate based on passive stack ventilation, the design should take account of increased pressure drop by the heat recovery system and reduced buoyancy of exhaust air after heat is removed from it.

Acknowledgement

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