Measurement and simulation of air flow in a two-zone chamber with heat-pipe heat recovery

G. Gan and S. B. Riffat***

SYNOPSIS
The performance of a heat-pipe heat recovery unit was tested in a two-zone chamber with a horizontal partition. Air velocity was found to have a significant effect on the effectiveness of heat recovery. The effectiveness decreased with increasing air velocity.

Simulation of air flow was carried out for the test chamber under natural ventilation conditions. It was shown that a heat-pipe heat exchanger can be used to reclaim exhaust heat in naturally-ventilated buildings to effect energy conservation.

INTRODUCTION

The objective of this study is to assess the performance of a heat-pipe heat recovery unit in a two-zone chamber with an opening in a horizontal partition. The chamber is designed to model air flow in a two-storey building. The control of air flow through horizontal openings such as stairwells and ventilation shafts in buildings is important for energy conservation and indoor air quality. Riffat [1] and Klobut and Siren [2] have reviewed investigations which have been carried out for air flows through openings in a vertical partition but relatively little is known about flows through horizontal openings. Besides, most investigations of horizontal openings are concerned with purely buoyancy-driven flows in enclosures without considering net flows created by external pressures. Recent work on buoyant flows through horizontal openings includes experimental measurement and numerical simulation of interzonal flow [3–5] and flow in a stairwell [6, 7]. Klobut and Siren [2] carried out laboratory measurements to explore the influence of parameters such as the direction and rate of the net flow, the temperature difference between the zones and the dimensions of the opening on the combined natural and forced air flows through large openings in a horizontal partition. The forced flow here refers to the net flow of air through an enclosure by external pressures due to stack or wind effects for example. In contrast, the natural flow is the purely buoyancy-induced flow within a single-zone enclosure or between multi-compartments of an enclosure. Such combined natural and forced flows occur in naturally-ventilated buildings.

In recent years natural ventilation has been recognised as an environmentally responsible means for controlling the indoor environment and as a result priority is now given to natural ventilation instead of air-conditioning when designing new buildings. However, unlike air-conditioned or mechanically-ventilated buildings, no consideration has been given to heat recovery in naturally-ventilated buildings. This may be partly attributed to the relatively large pressure loss caused by a conventional heat recovery system compared with driving pressures due to wind

* Dr G. Gan, Research Associate, Institute of Building Technology, Department of Architecture and Building Technology, University of Nottingham, Nottingham NG7 2RD, UK.
** Professor S. B. Riffat, Head of The School of the Built Environment, Department of Architecture and Building Technology, University of Nottingham, Nottingham NG7 2RD, UK.

and/or stack effects. Using a heat-pipe heat exchanger reduces the pressure loss so that sufficient ventilation rates can be attained. A recent numerical study of room air movement by the authors [8] has demonstrated that a heat-pipe heat recovery system can produce adequate thermal comfort with minimum energy consumption in naturally-ventilated buildings. Based on the numerical study, this paper describes measurement of the effectiveness of a heat-pipe heat recovery system. Numerical simulation is then carried out to predict air flow in a naturally-ventilated two-zone chamber with heat recovery.

MEASUREMENT OF EFFECTIVENESS OF HEAT RECOVERY

The effectiveness of a heat recovery unit for sensible heat exchange between supply and exhaust air of the same flow rate, ε (%), is defined as:

$$\varepsilon = \frac{T_s - T_i}{T_r - T_i} \times 100\% \qquad (1)$$

where T_i is the temperature of the inlet air (°C), T_s is the temperature of the supply air after the heat exchanger (°C) and T_r is the temperature of the return air (°C).

A heat-pipe heat exchanger consists of evaporator and condenser sections. Exhaust heat is recovered from the evaporator section and transferred to the condenser section. According to the above equation, the temperatures of air before and after the condenser section and the temperature of return air need to be measured. To investigate the effect of air velocity, the effectiveness was determined for a range of air flow rates.

Measurements were carried out in a vertical two-zone test chamber with a heat-pipe heat recovery unit. The two-zone chamber was designed so that good mixing of supply air with room air in the lower zone at a higher temperature could be achieved and return air in the upper zone maintained at a uniform temperature to ensure reliable temperature measurement.

Test chamber. Figure 1 shows the dimensions of the test chamber. The chamber was made of plywood. A layer of polystyrene insulation 0.0254 m thick was fitted on the interior of the chamber to reduce the influence of surroundings. The chamber had a net interior base area of 1.169 × 1.133 m and a total height of 2.335 m. It was divided into two zones with a horizontal partition. There was an opening (0.215 × 0.215 m) in the middle of the partition to allow air to flow from one zone to another. The net internal volume of the chamber was 3.09 m^3. Supply and exhaust ducts were connected to the chamber on one of the vertical walls. The air ducts were also made of plywood. When in operation, air was supplied to the lower zone of the chamber via the supply duct and return air exited from the upper zone to the exhaust duct. The connection of the plywood panels was sealed with glue and rolls-in-seal. The air tightness of the chamber was examined by smoke-testing. The whole chamber was installed in a laboratory.

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 two banks of externally finned heat pipes. Each bank had seven heat pipes 0.0127 m in diameter and 0.45 m in length with 72 continuous plain fins on both the condenser and evaporator sections of the unit. The dimensions of each fin were 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. Thus, the cross-sectional area for both the condenser and evaporator sections was 0.215 × 0.215 m and the overall dimensions of each bank were 0.45 × 0.215 × 0.048 m. The whole unit was made of copper. The working fluid in the pipes was methanol with a temperature range from – 40° to + 100°C. The two banks of heat pipes were horizontally mounted in series and one of them could be removed from the system in order to compare the performance of arrangement.

A 500 W halogen lamp and ten 100 W light 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 flow fan with adjustable speed was used to generate controlled air flow in the chamber. 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 0.0508 m thick polystyrene to provide good insulation.

Measurement. Temperatures up and downstream of the heat recovery unit in both supply and exhaust ducts were measured using thermocouples (type T). In addition, the temperature of air in the chamber was measured using the same type of thermocouples in the middle of the partition opening. 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 representation of air flow measurement. The method 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:

$$Q = \frac{q}{C} \times 10^6 \qquad (2)$$

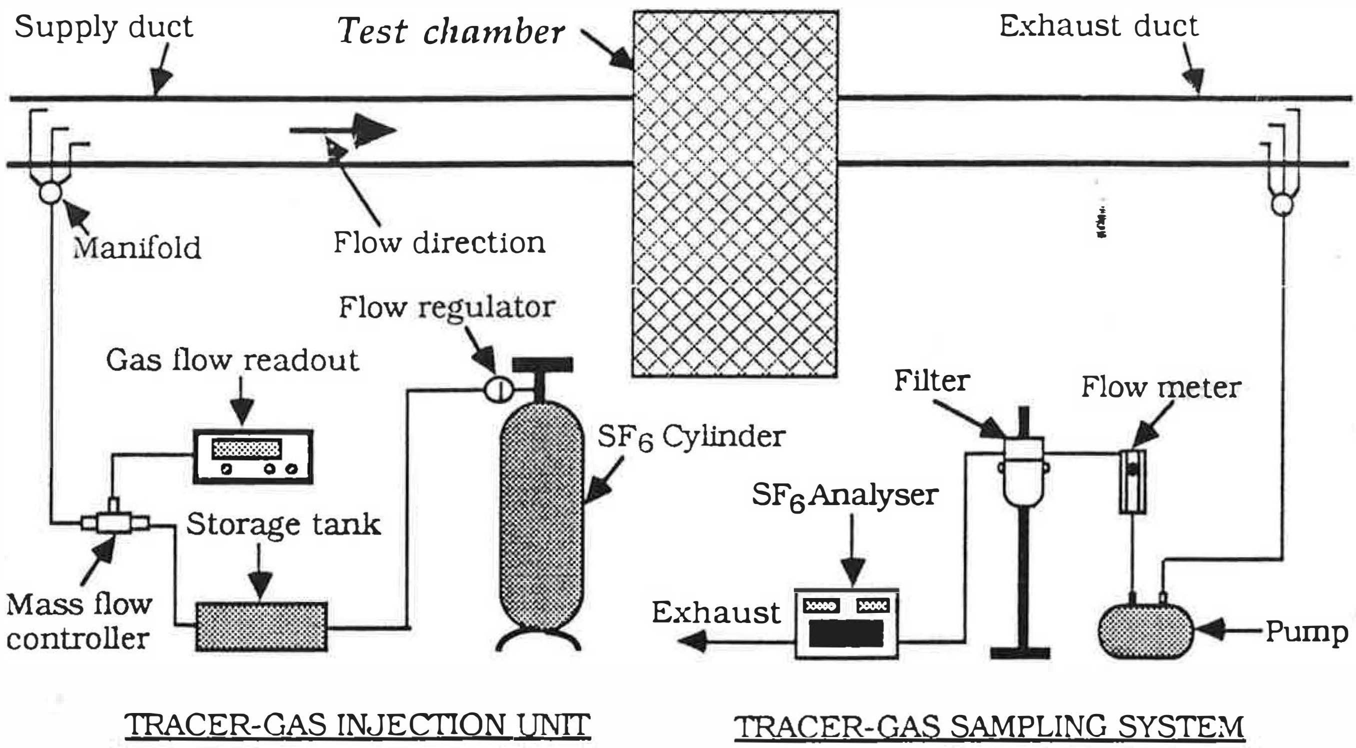


Figure 2 Schematic diagram of air flow measurement.

2 to 4 m/s whereas for natural ventilation the velocity is typically between 0.5 and 1 m/s. Table 1 shows a comparison of the measured effectiveness for the heat recovery unit with one and two banks of heat pipes. It is seen that at the same fan speed, the duct mean velocity with two banks of heat pipes is lower than that with one bank. This is due to the increased flow resistance from the second bank of heat pipes. It is also seen by interpolation that at the same air velocity the heat recovery is between 16% and 17% more efficient using two banks of heat pipes than using one bank. It may be postulated that the effectiveness could be further increased by employing more banks of heat pipes but this will inevitably increase flow resistance. For natural ventilation the resulting pressure loss must be smaller than the driving forces such that sufficient air flow rates could still be induced.

Table 1 Measured effectiveness of heat-pipe heat recovery.

No. of Banks	Fan speed* (%)	Air flow rate (m^3/s)	Air velocity (m/s)	Effectiveness (%)
1	2.5	0.0145	0.31	46.81
1	5.0	0.0352	0.76	40.81
1	7.5	0.0580	1.26	36.47
1	10.0	0.0840	1.82	32.23
1	12.5	0.1096	2.37	28.12
1	15.0	0.1410	3.05	25.73
1	17.5	0.1663	3.60	23.78
1	20.0	0.2044	4.42	22.07
2	2.5	0.0121	0.26	64.15
2	5.0	0.0306	0.66	59.40
2	7.5	0.0510	1.10	52.63
2	10.0	0.0742	1.61	48.42
2	12.5	0.0994	2.15	46.77
2	15.0	0.1287	2.78	41.85
2	17.5	0.1548	3.35	38.93
2	20.0	0.1887	4.08	35.92

* in percentage of maximum speed

The air velocity was found to have a significant effect on the effectiveness of heat recovery. Figure 3 shows the variation of effectiveness with air velocity in the duct. The relationship between the effectiveness and velocity can be represented by the following correlations for the velocity ranges investigated:

for one bank:

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

for two banks:

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

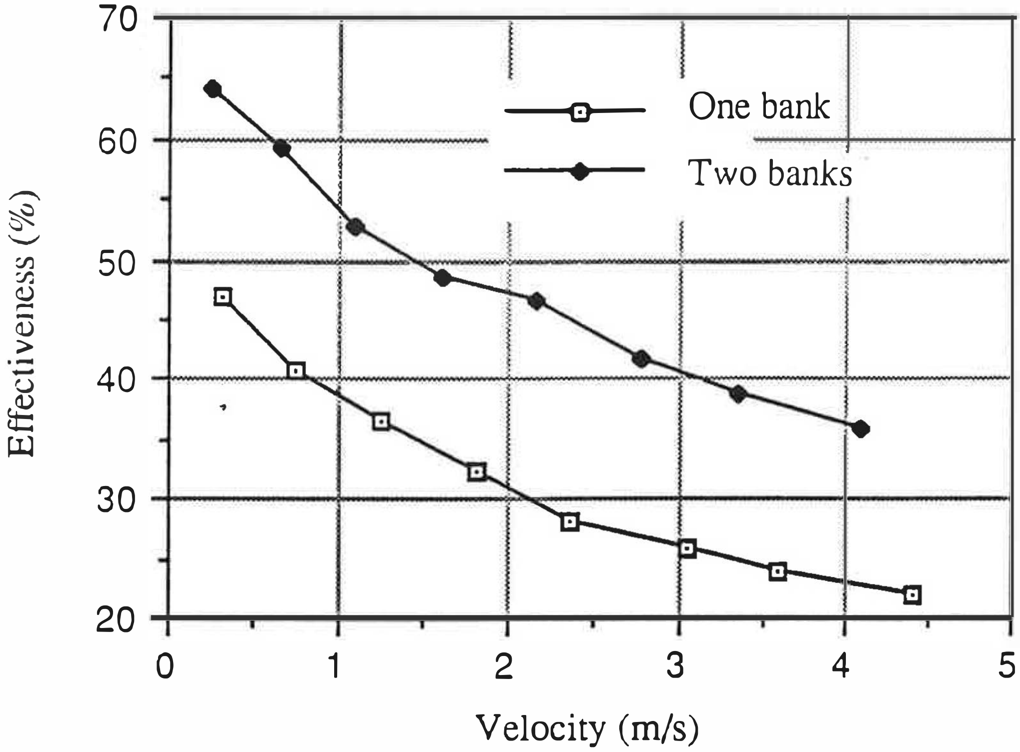


Figure 3
Effect of air velocity on the heat recovery effectiveness.

It follows that decreasing the duct velocity can increase the rate of heat recovery but for a given heat recovery unit this does not necessarily increase the total amount of heat recovery (proportional to velocity). To achieve a required quantity of heat recovery at a lower velocity, the size of a heat exchanger needs to be increased but this will result in a higher initial cost. The main benefit of a lower duct velocity is the lower pressure loss through the ventilation system since the flow resistance is proportional to the square of velocity ($\frac{1}{2}\rho V^2$ where ρ is the air density). In naturally-ventilated buildings, as the pressure loss is one of the major considerations, use may be made of low duct velocity to reclaim exhaust heat while providing sufficient air flow through a relatively large heat reclaim unit. On the other hand, at low velocities, the effect of transient nature of the outdoor environment increases and, occasionally, there may be no heat recovery at all.

Therefore, when designing a heat-pipe heat recovery system for naturally-ventilated buildings on the basis of required quantity of heat recovery and pressure differences between inlet and outlet openings, consideration should be given to optimising air velocity, number of heat-pipe banks and the size of each bank as well as construction of heat pipes. A numerical method described in the following section can assist the optimisation.

NUMERICAL SIMULATION

Numerical simulation was carried out for the test chamber under natural ventilation conditions.

Air flow model. The re-normalisation group turbulence model developed by Yakhot and Orszag [9] was used for air flow simulation. For an incompressible steady-state flow the time-averaged equations are represented by:

$$\underbrace{\frac{\partial}{\partial x_i}(\rho U_i \phi)}_{\text{convection}} - \underbrace{\frac{\partial}{\partial x_i}\left(\Gamma_\phi \frac{\partial \phi}{\partial x_i}\right)}_{\text{diffusion}} = \underbrace{S_\phi}_{\text{source}} \tag{6}$$

where ϕ represents the mean velocity component U_i in x_i direction, turbulent kinetic energy and its dissipation rate, mean specific enthalpy or mean concentration, Γ_ϕ is the diffusion coefficient for variable ϕ and S_ϕ represents the source term for variable ϕ.

A porous media model was incorporated in the general air flow model to predict air flow and pressure loss through the heat recovery unit and heat transfer between the unit and air. This involves modelling the heat-pipe heat recovery unit as a porous medium with heat generation (for the condenser section) and dissipation (for the evaporator section).

Details of the model equations and the boundary conditions are described by Gan [10] and Gan and Riffat [8]. The computer program for air flow simulation is based on the extensively applied two-dimensional finite-volume TEAM code [11]. In the program, the governing equations are solved for the three-dimensional Cartesian system using the SIMPLE algorithm and the Power-Law differencing scheme [12].

Validation of the program was performed by comparing the prediction with the experimental results for turbulent natural convection in a tall cavity by Betts and Bokhari [13]. The internal dimensions of the cavity were 2.18 m high, 0.076 m wide and 0.52 m deep. Temperatures of cold left and hot right walls were controlled at T_c = 14° and T_h = 35°C, respectively. Other walls were insulated. Air velocities in the cavity were measured using a laser-Doppler anemometry system. Thermocouples were used for temperature measurement.

Figure 4 shows a comparison between the predicted and measured velocity and temperature profiles at the cavity mid-height. As seen from the Figure, the velocity profile from the prediction agrees well with the measured results. The predicted temperature is also in reasonable agreement with the measurement. The predicted temperature distribution is anti-symmetrical whereas the measured temperature at the cavity centre is slightly higher than the mean value for the cold and hot walls.

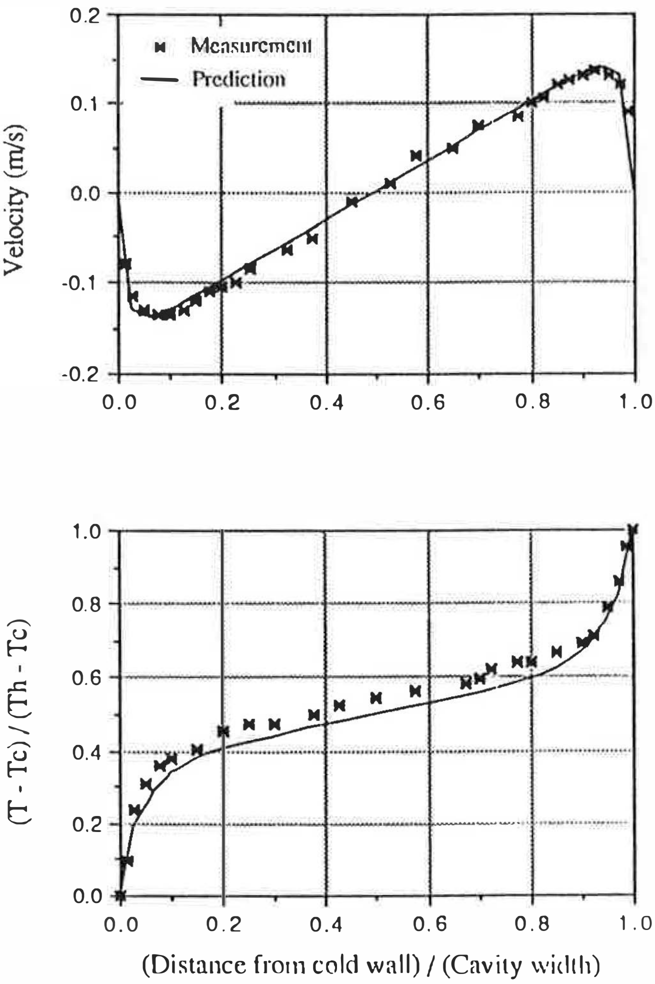


Figure 4 Comparison of the predicted and measured air velocity and temperature at the cavity mid-height.

Simulation. The validated program was used to predict air flow and temperature distribution in the test chamber under natural ventilation conditions.

As a boundary condition, the air flow rate through the chamber is calculated from [14]:

$$Q = C_d A_e \sqrt{\frac{2\Delta P}{\rho}} \qquad (7)$$

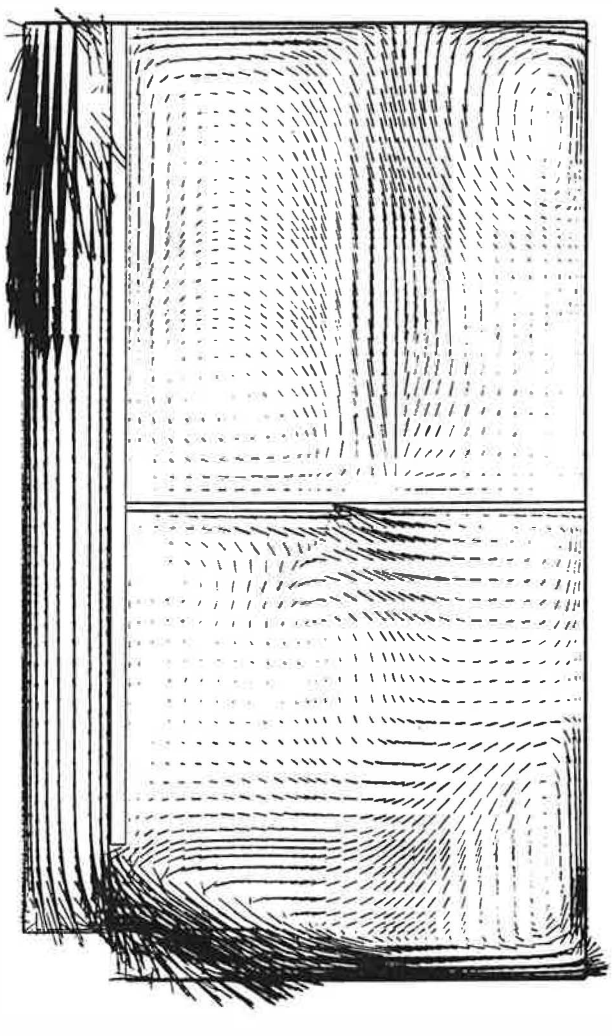
where C_d is the discharge coefficient and is dependent on the pressure loss coefficients for the flow system including entry, exit, bend and obstructions such as heat pipes as well as the friction loss coefficient of straight ducts [15], A_e is the effective opening area and ΔP is the driving pressure difference between inlet and exhaust openings due to wind and stack effects.

In the simulation, the chamber was assumed to have a uniformly-distributed heat source on the floor of 200 W in total. Two banks of heat pipes were in use for heat recovery and so Equation (5) was used to calculate the effectiveness of heat reclaim by the heat-pipe heat recovery unit. The outdoor air was assumed at 5°C and 90% rh. The chimney height was 4.5 m above the inlet opening. Initially the inlet and outlet openings were both assumed to have the same cross-sectional area as the ducts but it was found that the predicted air exchange rate was far too high (> 30 l/hr), resulting in too low a room air temperature (< 15°C). Therefore, the area of the exhaust opening was taken to be the same as the duct cross-sectional area (0.046 m^2) but the inlet area was reduced by half.

The solution was considered to have converged when the sum of the normalised residuals was less than 10^{-3} for each flow equation. Convergence was achieved after 13000 iterations. The slow convergence was caused by the inter-dependence between the air flow rate, heat recovery effectiveness which affects the heat production/dissipation rate of heat pipes, flow pattern and air temperature distribution. For a grid size of 50 × 70 × 44 for chamber length, height and width, respectively, the CPU time for the solution was about three minutes per iteration on a 6–processor Sun SPARCserver 1000E.

Results and discussion. Figure 5 shows the predicted air flow patterns and temperature distribution on three vertical planes through the inlet and exhaust openings and the partition opening. As seen from the Figure, the incoming air flows along the supply duct into the lower zone and spreads over the floor. It is noticed that air flow in the area between the inlet opening and heat recovery unit in Figure 5a is not uni-directional as a result of the reduced inlet opening and the presence of heat pipes. The cold air then flows upwards along the vertical walls (including side walls not shown in the Figure) but cannot immediately reach the top of the lower zone due to insufficient momentum to overcome the negative buoyancy effect. Instead, it moves away from the walls and mixes with air in the zone. The mixed air then flows upwards like an air jet through the partition opening into the upper zone (Figure 5b). Here the air jet hits the ceiling and then flows downwards along the side walls. The air in the chamber exits from the exhaust duct due to the stack effect (Figure 5c).

The outdoor air is heated from 5°C to about 10°C when passing the heat pipes in the supply duct. The temperature of the supply air is further increased after flowing over the heated floor and mixing with air in the lower zone. Figure 6 shows the variation of air temperature with chamber height. In the Figure, the temperature at any height is the weighted average for the whole horizontal cross-section. It is seen that near the floor air temperature decreases from the floor temperature (about 40°C due to the high heat production rate of over 100 W/m^2) to the temperature of cold supply air. It then increases with the distance from the floor in the lower zone but remains almost constant at about 21°C in the



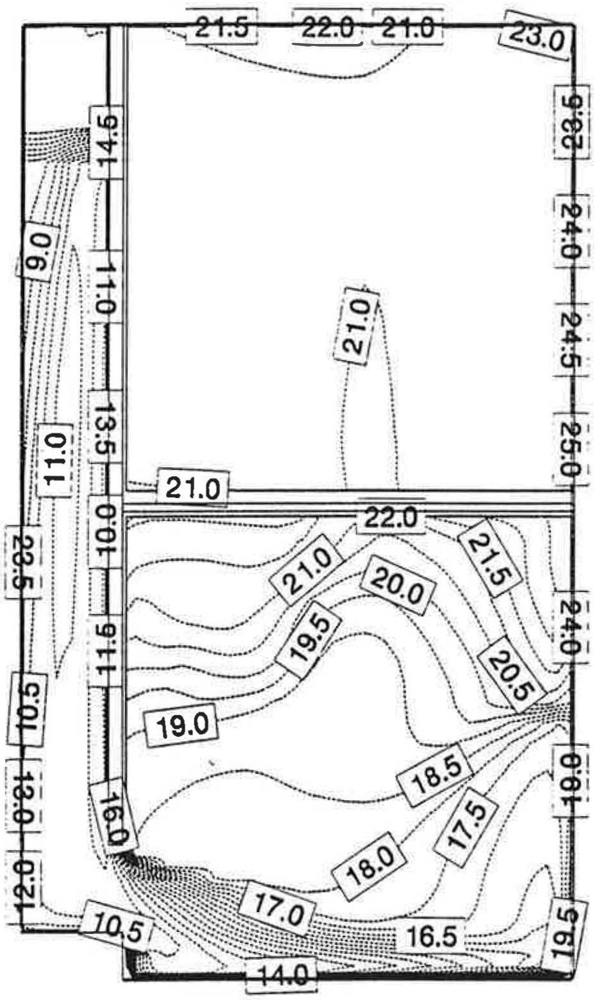
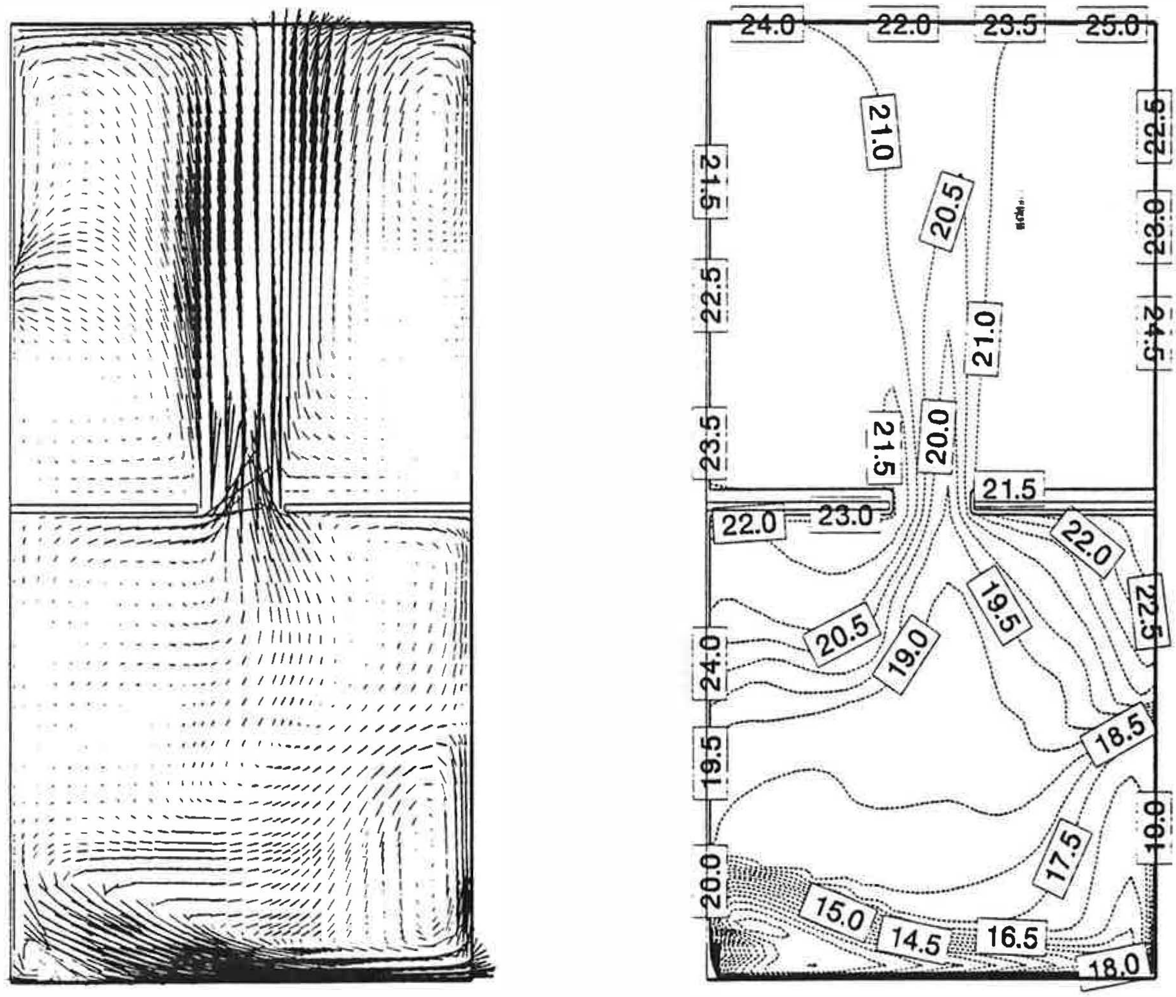


Figure 5a Predicted air flow patterns and temperature distribution through the inlet opening vertical plane.



Velocity vectors (m/s)

Temperature (°C)

Velocity scale : 1 m/s

Figure 5b Predicted air flow patterns and temperature distribution through the partition opening vertical plane.

upper zone (height > 1.2 m). Hence, the two-zone chamber provides an essentially uniform return air temperature.

The predicted mean velocity in the chamber is 0.1 m/s and air flow rate is 19 l/s, i.e. 22.3 air changes per hour which is still rather high. The predicted mean air temperature in the chamber is 20.1°C and relative humidity is 33.6%. The temperature in the lower zone is on average 2°C lower than in the upper zone. The temperature along the air stream from the supply duct is about 16°C (Figure 6) and this is too low for thermal comfort. The low temperature is due to the high flow rate in terms of air exchange rate. To decrease the flow rate and so to increase air temperature, the inlet opening needs to be further reduced, to say 30% of the duct cross-sectional area, which in practice can be achieved using a flow control damper. However, if the flow rate is converted into the air exchange rate for an enclosure with the same size as a practical two-storey house, the ventilation rate is perhaps only around one air change per hour and the fresh air supply rate is only sufficient for one or two residents. When both the room temperature and flow rate are taken into account, it is clear that reducing the inlet opening further is not an appropriate solution for this case. Therefore, to achieve thermal comfort for a realistic house under the same indoor and outdoor conditions, it is necessary to increase the quantity of heat reclaim rather than decrease the flow rate. This can be

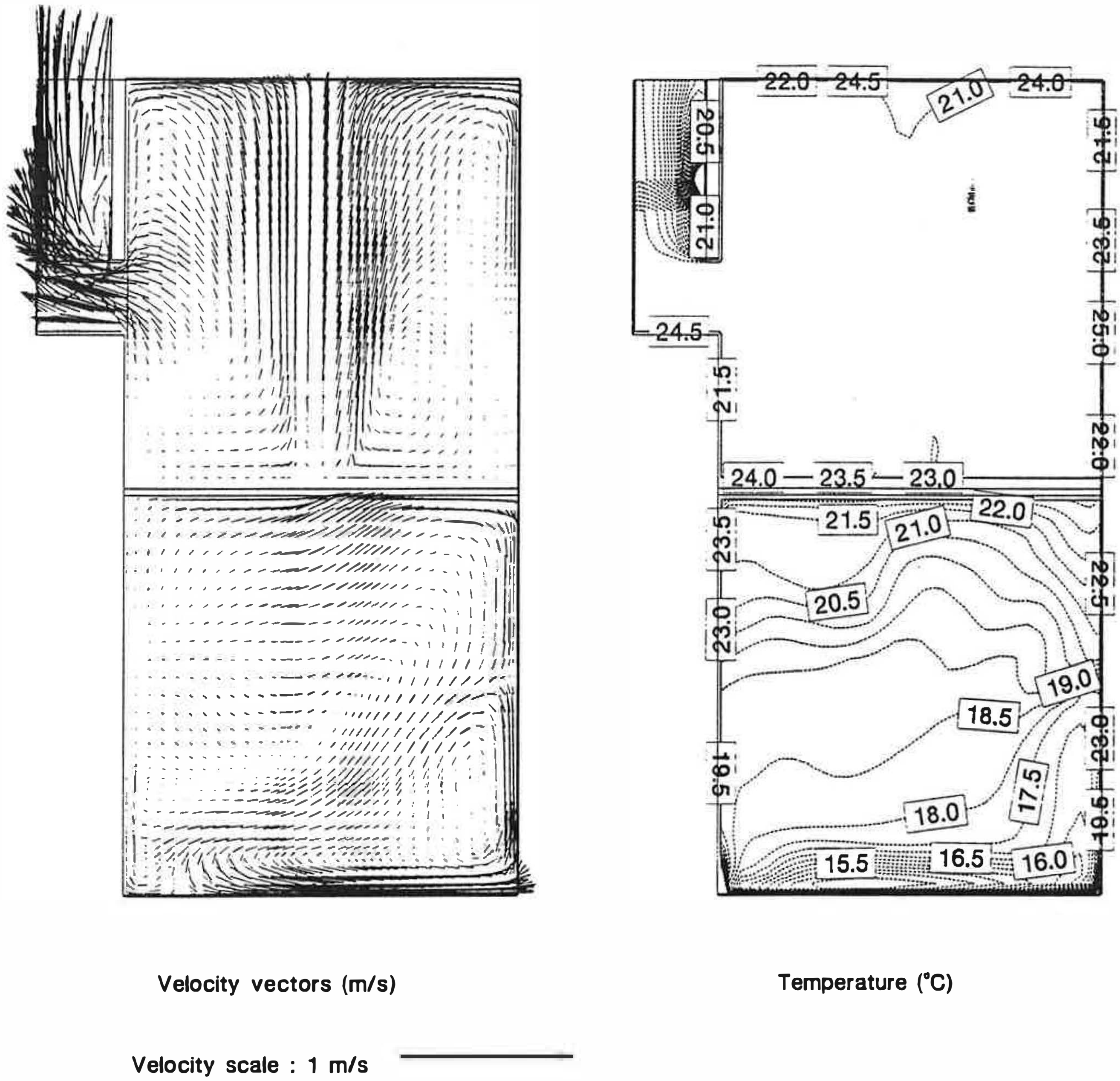


Figure 5c Predicted air flow patterns and temperature distribution through the exhaust opening vertical plane.

achieved by increasing the size of the heat recovery unit or employing more banks of heat pipes if the resulting flow resistance would not be excessive.

For the purpose of simulation, in order to control the flow rate at given indoor and outdoor air conditions, an alternative to adjusting the size of the inlet opening is to alter the height of the chimney. Although this may not be practical for a real building, the idea can be used to optimise the design of the chimney and ventilation system. The chimney used in this study was however based on the required height above the roof of the laboratory in which the test chamber was installed.

Finally, simulation results for concentration distribution can be used to assess the reliability of air flow measurement. The accuracy of air flow measurement based on the downstream concentration, in this case the exhaust duct, depends on the degree of mixing of the tracer gas with air in the chamber. In the concentration simulation, it was assumed that the tracer gas was injected at the centre of the inlet opening (point source). The predicted concentration along the direction of chamber height is shown in

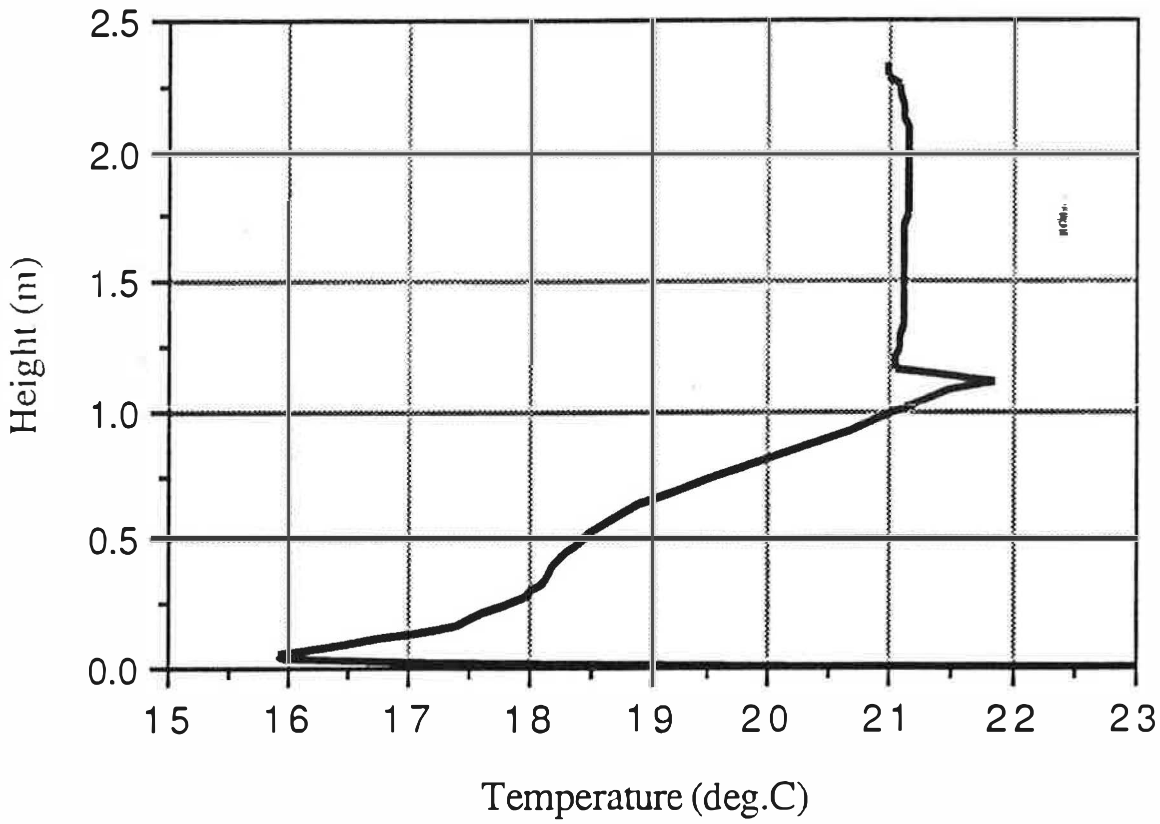


Figure 6 Variation of air temperature with height in the chamber.

Figure 7. The concentration is averaged for a horizontal cross-section and normalised based on the value at the exhaust opening. It is seen from the Figure that the concentration varies in the lower zone. The maximum plane-averaged variation occurs near the floor and is about 5%; the maximum spatial variation in the lower zone is 30% (from 0.85 to 1.15). In the upper zone (height > 1.2 m) the predicted concentration is uniform. Consequently, the concentration in the exhaust duct will also be uniform if there is no absorption of tracer gas by the duct surfaces and heat pipes. The uniformity of the concentration is further enhanced in practice by means of injecting and sampling the tracer gas at multiple points rather than at a single point. The simulation therefore confirms that the two-zone chamber enables reliable measurements of air flow rate as well as return air temperature.

CONCLUSIONS

This study shows that air velocity has a significant effect on the effectiveness of heat-pipe heat recovery. The effectiveness decreases with increasing air velocity. At the same velocity the heat recovery is between 16% and 17% more efficient using two banks of heat pipes rather than using one bank. Higher effectiveness can be achieved using more banks of heat pipes but for natural ventilation the resulting pressure loss should be taken into consideration.

A heat-pipe heat exchanger can be used to reclaim exhaust heat in naturally-ventilated buildings so as to minimise energy consumption. To achieve a comfortable indoor thermal envirnoment requires installation of a flow control system as well as proper sizing of the heat recovery unit. The numerical method can be used to optimise the ventilation system. The simulation shows that a two-zone chamber can provide adequate air conditions for determination of heat recovery effectiveness.

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