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HEAT-PIPE HEAT RECOVERY FOR PASSIVE STACK VENTILATION

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SYNOPSIS

Four types of heat-pipe heat recovery systems were tested for application in passive stack ventilation. The effects of fin shape, pipe arrangement and air velocity on the heat recovery effectiveness were investigated. The air velocity was found to have a significant effect on the effectiveness of heat recovery; the effectiveness decreasing with increasing air velocity.

The pressure loss coefficient for heat pipe units was also determined. It was found that at low velocities for natural ventilation the pressure loss coefficient decreased with increasing air velocity but the total pressure loss increased with the velocity. It is recommended that in naturally-ventilated low-rise buildings, without the wind effect or solar energy, the design duct mean velocity should be less than 1 m/s in order for a heat recovery system to function properly. The use of a solar chimney and/or wind turbine could increase the range of air velocity and so the amount of heat recovery.

INTRODUCTION

Ventilation accounts for 30% or more of space conditioning energy demand but as much as 70% of this energy can be recovered by the use of ventilation heat recovery systems [1]. Until now, effort has mainly been devoted to the design and development of heat recovery systems for mechanically-ventilated buildings. However, most domestic buildings are naturally ventilated and little consideration has been given to heat recovery from these buildings. A crucial parameter that limits the use of heat recovery with natural ventilation is pressure loss. The total pressure loss through a natural ventilation system should be much lower than in a mechanical ventilation system so that sufficient air flow can be achieved in the building. A system employing heat pipes would have the potential to provide substantial heat recovery without significant pressure loss.

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 (see Fig. 1). In addition to low flow resistance, a heat-pipe heat exchanger has a number of other advantages over conventional heat exchangers such as high reliability, no cross-contamination, compactness and suitability for both heating and cooling.

The objective of this study is to assess the performance of heat-pipe heat recovery units for naturally-ventilated buildings. The effectiveness of four heat-pipe units was measured in a two-zone chamber. The pressure loss characteristics of the units were determined by CFD modelling as well as measurement.

EFFECTIVENESS OF HEAT RECOVERY

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

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

where T_i and T_s are the temperatures of inlet air before and supply air after the condenser section of heat exchanger in the supply duct (°C) respectively, and T_r is the temperature of return air before the evaporator section of heat exchanger in the exhaust duct (°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 was designed to allow good mixing of supply air with room air in the lower zone and maintain a uniform temperature and concentration of return air in the upper zone. This ensured the reliability of temperature and air flow measurements.

Temperatures up and downstream of the heat recovery unit in both supply and exhaust ducts were measured using thermocouples (type T). The temperatures were recorded by a data logger. The air flow rate was measured using the constant-injection tracer-gas method [2].

Test chamber

Fig. 2 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 1.169 X 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 X 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 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. An axial flow fan with adjustable speed was used to generate air flow through the chamber.

Heat pipes

Four types of heat-pipe heat recovery units were constructed and tested. Fig. 3 shows the cross-section of the heat pipes. The working fluid in the pipes was methanol with an operating temperature range from -40° to 100°C.

The first heat recovery unit (Type I) consisted of a bank of seven externally finned heat pipes. Each pipe was 0.0127 m in outside diameter and 0.45 m in length with 72 continuous plain fins on both the condenser and evaporator sections. 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 at the middle of the bank to prevent cross-contamination of return and supply air. The cross-sectional area of both the condenser and evaporator sections was 0.215 X 0.215 m. The total surface area of each finned pipe including fins and exposed pipe was 0.196 m^2. The whole unit was made of copper.

The second type heat pipe had cylindrical spine fins. The fins were made of copper wire. The unit with this type of fin consisted of three heat pipes of the same size and material as for Type I. There were eight continuous rows of fins on each of these pipes. Each row had about 300 spine fins and each spine fin was 0.7 mm in diameter and 30 mm long. The fins were soldered on the pipes and the tips of fins were fixed in such a way that they were uniformly distributed circumferentially. The estimated total surface area of the spine fins of each heat pipe was 0.158 m^2, which is about 19% less than that of the continuous plain fins.

The third type of heat recovery unit was made of two rows of staggered heat pipes. Each row consisted of three heat pipes. Each pipe was 18 mm in diameter and 365 mm in length with 70 continuous louvred aluminium fins on both the condenser and evaporator sections. The dimensions of each fin were 180 mm long and 60 mm high. Each fin had 96 louvres with 2 mm spacing and 0.65 mm gap. The length of the louvres varied from 5.5 mm at the centreline of each pipe row to 8.5 mm near the edge of pipes. The cross-sectional areas of the condenser and evaporator sections were 180 X 180 mm and 175 X 180 mm, respectively. The total surface area for heat transfer of the evaporator section was 1.5418 m^2; this is about 8% more than that for Type I heat pipes.

The fourth type of heat recovery unit was made of five in-line heat pipes with wire fins. Each pipe was 19.05 mm in diameter and 450 mm long with 34.5 turns of copper wire fins on both the condenser and evaporator sections. Each turn of fins had 65 loops of wire 0.65 mm in diameter. The height of fins was about 12 mm. The cross-sectional area of the unit is the same as that of the first type of heat pipe. The total surface area for heat transfer of the evaporator or condenser section was 0.6035 m^2; this is less than half of the first type.

Results and discussion

Fig. 4 shows a comparison of the effectiveness for the four types of heat pipes. It can be seen that the rate of heat recovery increases with decreasing air velocity. However, 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 velocity is the lower pressure loss through the ventilation system since the flow resistance is proportional to the square of velocity.

The effectiveness for the spine-fin heat pipes presented in Fig. 4 was for seven equivalent heat pipes. The effectiveness for this type heat recovery unit was much lower than that of plain-fin heat pipes. The main reason for the ineffectiveness of spine-fin heat pipes is the poor thermal contact between fins and pipes, resulting in a high contact resistance.

For the same cross-sectional area, the staggered heat pipes with louvred fins were more effective than plain fins, particularly at lower velocities. This may be attributed mainly to the increased external surface area available for heat transfer per unit cross section (55% more).

The effectiveness of the unit with wire fins was lower than that with plain fins. This is due to the lower external surface area for the wire-fin heat pipes. If the surface area was increased by say 50%, which would still be less than that of the plain-fin heat pipes, the effectiveness would be higher than that of the plain-fin heat pipe unit.

The effectiveness of heat pipes could be increased by employing more than one bank. For example, for Type I heat pipes, the measured heat recovery was between 16% and 17% more efficient using two banks than using one bank. It may be postulated that the effectiveness could be further increased by employing more banks of heat pipes but this would increase flow resistance and cost of installation. For natural ventilation the resulting pressure loss must be smaller than the driving force so that adequate air flow rates can still be achieved.

PRESSURE LOSS ACROSS HEAT PIPES

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

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

where ΔP_s is the static pressure loss across the unit (Pa), V is the mean velocity of air flowing over the unit (m/s) and ρ is the air density (kg/m^3).

The pressure loss coefficient for Type I and III heat-pipe units was predicted by means of CFD modelling. The predictions were carried out using the CFD package FLUENT [3]. The CFD technique was validated for predicting the pressure loss coefficients for a number of duct fittings [2]. In the predictions, each row of heat pipes was modelled as one bank of rectangular cylinders such that it had the same free-area ratio and thickness as 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 that of fins.

To determine the effect of fin shape on the pressure loss characteristics, flow resistance was also measured for two types of heat pipes - Type I and Type IV with pressure taps fitted on the up and downstream ducts of 0.15 m in diameter. Since the pressure loss at low values was difficult to determine, measurements were made at velocities higher than 2 m/s such that the resulting pressure loss was higher than the precision of instrumentation (1 Pa).

The pressure loss coefficient was found to decrease with increasing mean air velocity. The pressure loss coefficient for Type I heat pipes can be correlated to velocity between 0.25 and 10 m/s as follows:

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

where n is the number of heat-pipe banks.

The pressure loss at a given velocity can be obtained from the pressure loss coefficient ($=\frac{1}{2}k\rho V^2$). For example, at a velocity of 0.5 m/s, the pressure loss through a 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. At a velocity of 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 heat recovery unit, the mean air velocity should be less than 1 m/s.

Fig. 5 shows a comparison of pressure loss coefficient for Type I and Type III heat pipes. The predicted loss coefficient for the six staggered 18 mm heat pipes was higher than that for one bank of seven heat pipes of 12.7 mm in diameter despite the porosity (free-area ratio) of the former being higher than that of the latter. When the six pipes were arranged as a two-row in-line bank, the predicted loss coefficient became lower than that of one bank of seven smaller heat pipes. For example, at a velocity 1 m/s, the predicted loss coefficient for the two-row in-line six pipes was 3.3, compared with 4.2 for the staggered pipes and 3.7 for the in-line seven smaller heat pipes.

In Fig. 6 the measured pressure loss coefficient for Type I and Type IV heat pipes is compared. At air velocities higher than 2 m/s, the pressure loss through the wire-fin heat pipe unit was higher than that for the plain-fin heat pipes particularly at high velocities. This may be explained by the opposite effect of the two types of fins on flow turbulence. The plain fins could act as a flow straightener whereas wire fins as a turbulence generator. The former decreased flow resistance whereas the latter increased flow resistance as velocity increased.

CONCLUSIONS

The experimental measurements show that air velocity has a significant effect on the effectiveness of heat-pipe heat recovery. The effectiveness decreases with increasing air velocity. For heat pipes with plain fins, at the same velocity the heat recovery is between 16% and 17% more efficient using two banks than using one bank. Poor thermal contact between fins and pipes can drastically reduce the effectiveness of heat pipes.

The numerical modelling indicates that at low velocities the pressure loss coefficient decreases with increasing air velocity but the total pressure loss still increases with the velocity. It is recommended that in naturally-ventilated low-rise buildings, without the wind effect or solar energy, the design mean air velocity should be less than 1 m/s in order for a heat recovery system to function properly. The use of a solar chimney and/or wind turbine could increase the range of air velocity and so the amount of heat recovery. For use in a natural ventilation system where low pressure losses are required, a staggered heat-pipe unit does not provide better overall performance than the in-line counterpart.

ACKNOWLEDGEMENT

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3. FLUENT User's Guide, Fluent Inc. USA, 1993.

* duct Δp loss → passive cooling!

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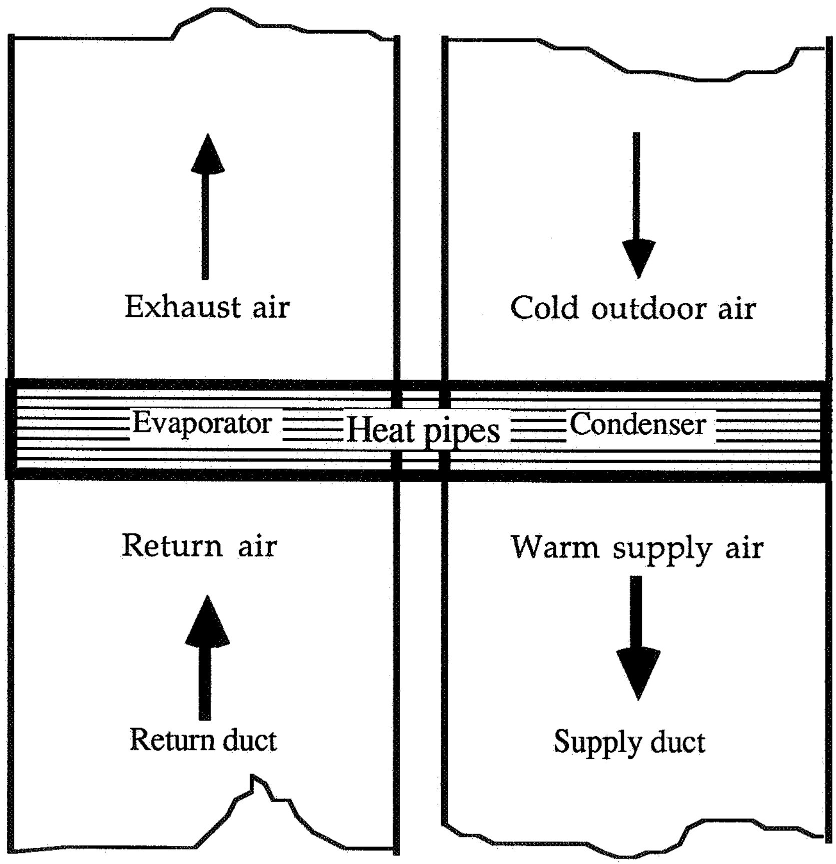
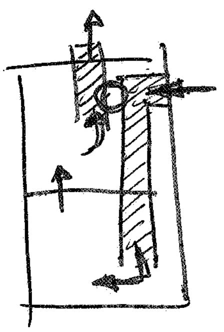


Fig. 1 Schematic representation of heat-pipe heat recovery



Hoe krijg je zo'n flow met stackeffect??

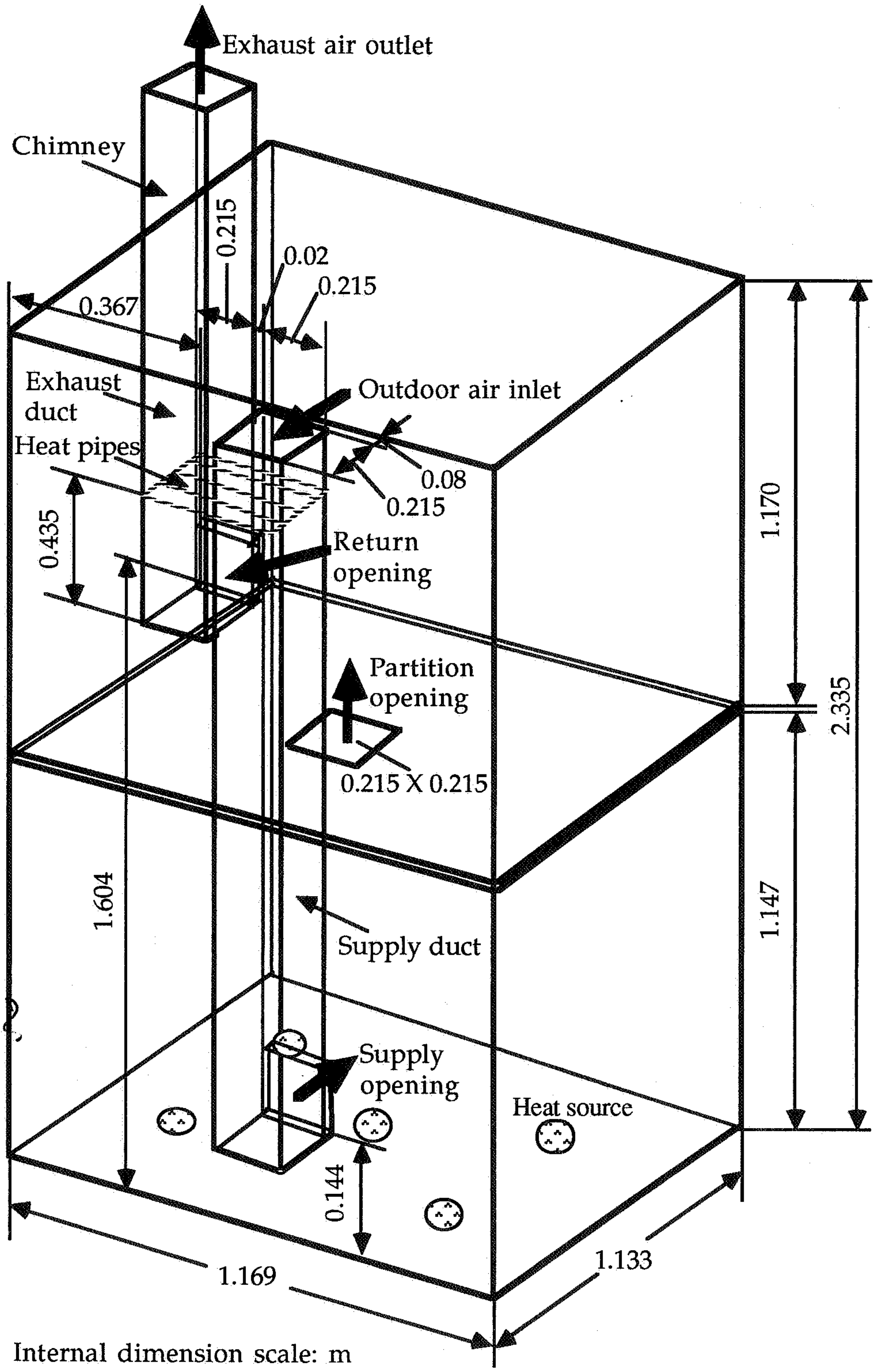


Fig. 2 Schematic of the two-zone test chamber with heat pipes

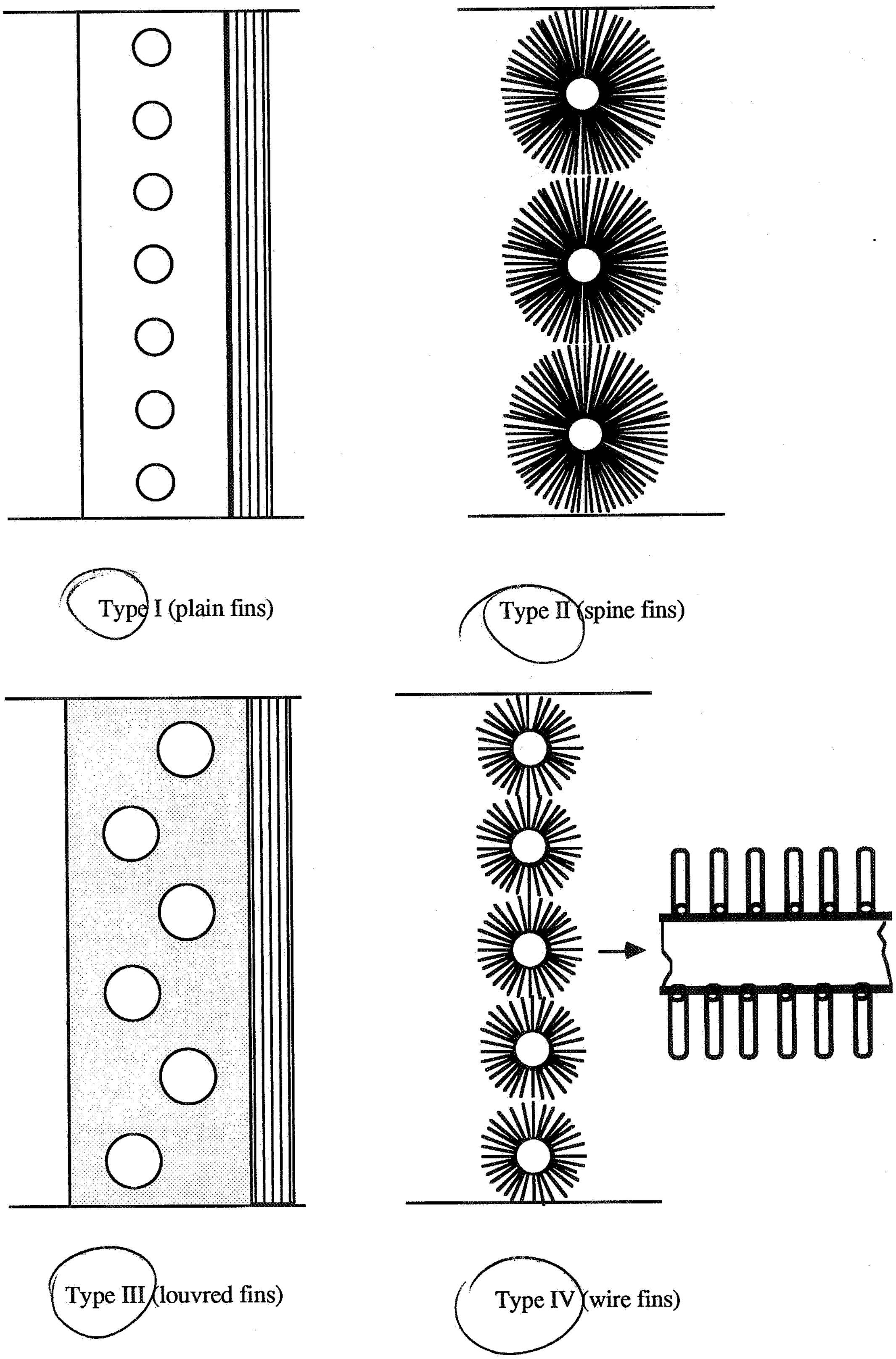


Fig. 3 Cross sections of heat pipe units

passive cooling: natur. ventilation

1 & 3 are performing best

sheets

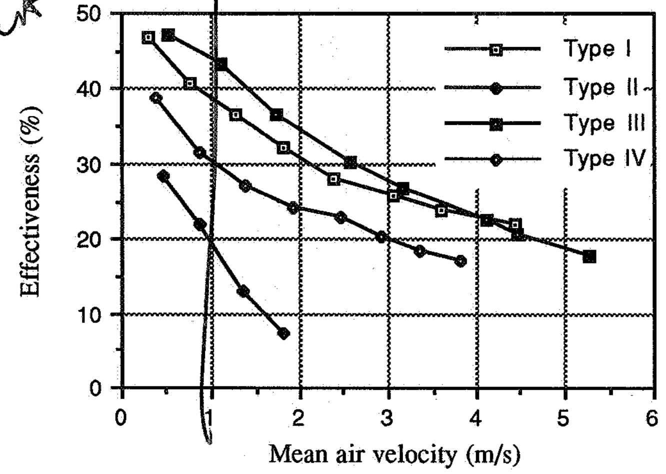


Fig. 4 Comparison of effectiveness for four types of heat pipes

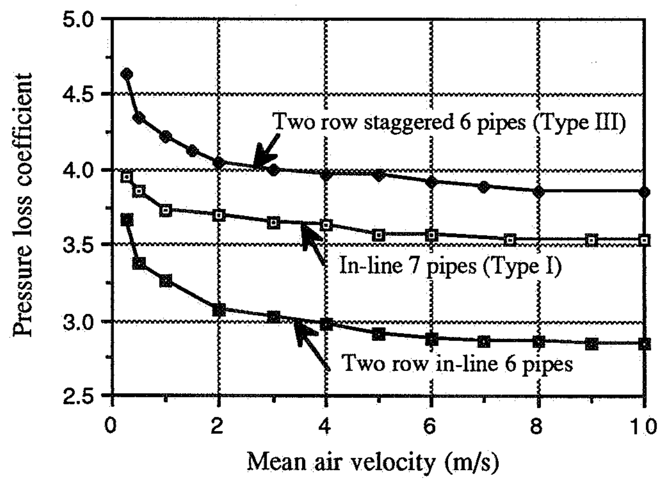


Fig. 5 Effect of pipe arrangement on the pressure loss coefficient

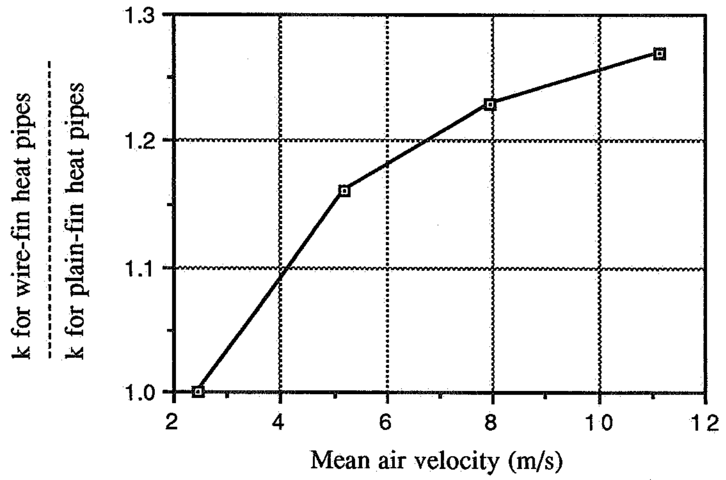


Fig. 6 Effect of fin shape on the pressure loss through heat pipes