EXPERIMENTAL STUDY ON THE PERFORMANCE LIMITATION OF MICRO HEAT PIPES OF NON CIRCULAR CROSS-SECTIONS

by

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An experimental study of three different cross-sections (circular, semicircular and rectangular) of micro heat pipes having same hydraulic diameter (D = 3 mm) is carried out at three different inclination angles (0°, 45°, 90°) using water as the working fluid. Evaporator section of the pipe is heated by an electric heater and the condenser section is cooled by water circulation in an annular space between the condenser section and the water jacket. Temperatures at different locations of the pipe are measured using five calibrated K type thermocouples. Heat supply is varied using a voltage regulator which is measured by a precision ammeter and a voltmeter. It is found that thermal performance tends to deteriorate as the micro heat pipe is flattened. Thus among all cross-sections of the pipes circular cross-section exhibits the best thermal performance followed by semicircular and rectangular cross-sections. Moreover maximum heat transfer capability of the pipes also decreases with decreasing of its inclination angle. A correlation is developed using all the gathered data of the present study to predict the heat transfer coefficient of micro heat pipes of different cross-sections placed at different inclination angles.

Key words: *micro heat pipe, non circular, cross-section, inclination*

Introduction

The size of microprocessors has been reducing day by day with the development of electronics. Consequently, the number of active semi-conductor devices per unit chip area has been increasing. In the last decade, the number of active semi-conductor devices per unit chip area has almost quadrupled [1]. The minimum feature size in microprocessors has reduced from 0.35 m in 1990 to 0.25 μm in 1997 and which will go down further to 0.05 μm by the year 2012 [2]. This has increased the heat dissipation density for desktop microprocessors. The current heat dissipation rates for some of desktop computers are approximately 25 W/cm^2. It is expected that microprocessor chips for some of the next generation work stations will dissipate 50-100 W/cm^2. Thus reduction in size also brings severe limitations to the conventional cooling techniques [3]. Development of efficient thermal management scheme is essential to dissipate these high heat fluxes and maintain suitable operating temperature of the device. Micro heat pipes (MHP) are increasingly filling this role. To keep up with today's thermal solution challenges, MHP must improve efficiency and integrate remote heat transfer into thermal management solutions. Recent research is going on several different types of MHP other than simple tubular form. Flat MHP is one of them. The significance of flat MHP over tubular MHP is its ability to distribute the working fluid over a wide surface area. Therefore it can produce a surface with very small temperature gradients across it. This near isothermal surface can be used to even out and remove hot spots produced by the heaters

[4]. Also, by mounting a number of heat generating components on a flat MHP, they can be operated at similar temperatures because of the fact that the vapor space will be at a fixed uniform temperature. Thus flat MHP allow a number of heated surfaces to be mounted. Zhuang *et al*. [5] compared the performance of MHP placed at different inclination angles in terms of maximum heat transfer capacity using three different structural wicks. It is found that all structures of wicks have little influence on the heat transfer capacity of miniature heat pipes working with the aid of gravity. Under the condition of anti-gravity, the structure of wicks has obvious influence on the heat transfer capacity of miniature heat pipes. Zhang [6] studied the heat transfer and fluid flow in an idealized MHP. The idealized MHP is a rectangular heat pipe, the top portion of which is made of a non-wetting material and the bottom portion is a wetting material. The lower portion is filled by a liquid, and the upper portion is filled by its vapor. It is found that the interface can be separated into two regions, an inner region near the wall where evaporation occurs and an outer region away from the wall. Yamamoto *et al*. [7] studied the high-performance MHP. It was found that pipe diameters have substantial effects on its performance such that larger the pipe diameter, greater the maximum heat transfer rate. Moreover, high performance type heat pipe exhibits greater effects of improvements in the maximum heat transfer rate compared to conventional heat pipes. A wickless network MHP for high heat flux study spreading applications was developed by Cao *et al.* [8]. Yuichi *et al.* [9] experimentally confirmed steady-state heat transfer characteristics of flat MHP in detail and proposed a method for determining its maximum heat transfer rate. Moon *et al*. [10] studied the performance of triangular MHP mounted horizontally and found its heat transport limit 6-15 W/cm^2 for the operating temperature ranging from 45-80 °C. Sreenivasa *et al*. [11] experimentally determined the optimum fill ratio of miniature heat pipes and found that it is not required to fill the evaporator section 100% rather if it is filled around 50% then also its performance is similar as 100% fill ratio. Akhanda *et al*. [12] tested an air cooled MHP and investigated the effects of working fluids and inclination angles on its thermal performance. According to Babin *et al.* [13], maximum heat transport capacity of a MHP with 0.01-0.5 mm hydraulic radius was 0.03-0.5 W and corresponded to 1 W/cm^2 in the heat flux based on the surface area of the evaporator. Wu *et al.* [14] also reported that Q_{max} = 4-5 W for a flat micro heat pipe of 1 mm hydraulic diameter. The objective of this study is to carry out an experimental investigation to compare thermal performances of MHP of three different cross-sections (circular, semicircular, and rectangular) having same hydraulic diameter (D = 3 mm) placed at three different inclination angles (0°, 45°, 90°) using water as the working fluid. It may be mentioned here that a number of research works on MHP have been carried out but very limited research works have been done so far to determine effects of MHP cross-sections on its thermal performances. That's why this study has been taken.

Fabrication of MHP

Cleaning of container

Distilled water is passed through the MHP twice or thrice to ensure that there is no foreign matter inside the tube which may hinder capillary action or may create incompatibilities. Then the MHP is heated for some time to make it dry again.

Fitting of wick

Cleaning of the MHP is best carried out before insertion of the wick, as it is then easy to test the wick for wettability. As the mesh wick layer is not bonded to the pipe wall, a coiled spring is inserted to retain the wick against the wall. This is done by coiling the spring tightly around a mandrel giving a good internal clearance in the MHP. When the spring holds the wick against the wall nicely, then the spring along with the mandrel is taken out of the pipe.

MHP filling

After fitting the wick with the wall properly, one opening of the pipe is sealed permanently and the other end is temporarily sealed with plastic cap. Opening the cap, water is put into the pipe by a syringe. The amount of water inserted into the tube is equal to the total volume of the evaporator (100% fill ratio). After putting the desired amount of fluid, the pipe is heated so that the amount of air inside the pipe is replaced by water vapor and in this condition the open end of the MHP is also sealed permanently. Then it can be assumed that there is no air present inside the MHP but the fill ratio is not 100%. Sreenivasa [11] concluded that heat pipe with fill ratio 85 and 100% gives more or less same result. Therefore fill ratio a bit less than 100% will not significantly alter the thermal performance of the MHP.

Experimental setup and test procedure

The schematic diagram of the experimental setup is shown in fig. 1. It mainly consists of four parts – the MHP (evaporator section, condensing section, and adiabatic section), heat supply system, measurement system, and data acquisition system. Detailed dimensions of the micro heat pipe are shown in fig. 2.

The MHP of three different cross-sections (circular, semi circular, and rectangular) having same hydraulic diameter are used in this study which are shown in fig. 3. Water is used as the working fluid to transport heat from evaporator to condenser. Performance tests for each cross-sections of MHP are carried out at three different inclination angles 0°, 45°, and 90°.

Figure 4 shows a layout of the test rig. As shown in fig. 4, the MHP is provided with a Ni-Cr thermic wire heater for heat input. Insulated Ni-Cr thermic wires having diameter of 0.28 mm (10 /m) are wound around one side of the evaporator wall at a constant interval of 1.5 mm. This setup is done in a way as to reproduce the mode of MHP heating applications close to realistic. Temperatures at different locations of MHP are measured using five calibrated K type thermocouples. Locations of thermocouples are shown in fig. 5.

Electrical power supply and temperatures are measured during each experimental run. Using these data, thermal performances of MHP of different cross-sections and orientations are measured in terms of heat transfer coefficient.

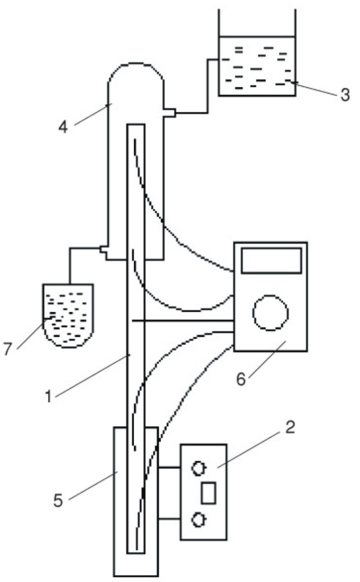

Figure 1. Schematic diagram of experimental setup
1 – heat pipte, 2 – power supply unit, 3 – elevated water tank, 4 – water jacket, 5 – evaporator, 6 – thermocouples, 7 – measuring cup

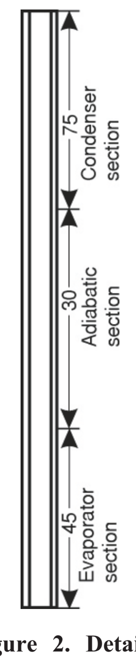

Figure 2. Detailed dimensions of MHP (dimensions are in mm)

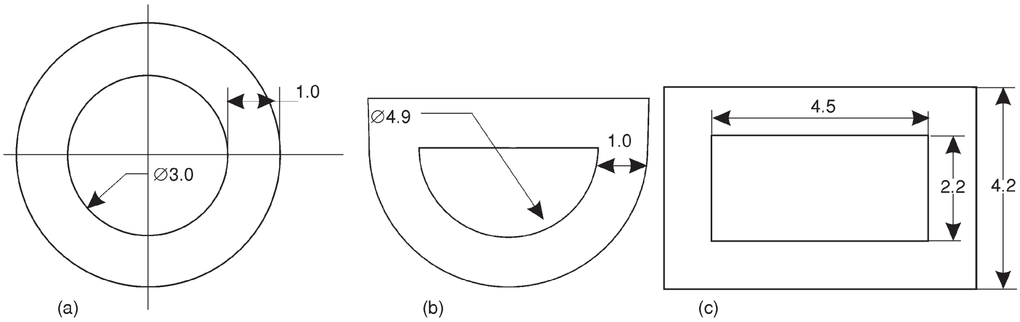

Figure 3. Different cross-sections of MHP (all dimensions are in mm)
(a) circular, (b) semicircular, (c) rectangular

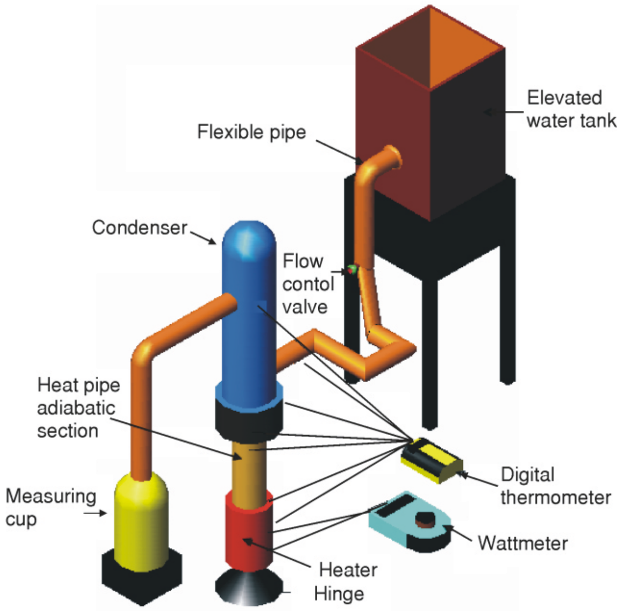

Figure 4. Layout of the experimental rig

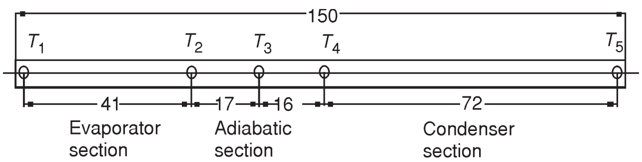

Figure 5. Locations of thermocouples on MHP surface (all dimensions are in mm)

Results and discussions

Using collected data, various curves are plotted as shown from fig. 6 to 13. Results of Moon *et al.* [10], Sreenivasa *et al.* [11] and Akhanda *et al.* [12] are included in some of these plots to compare their results with results of the present study.

Figures 6(a) to 6(f) show the axial wall temperature distribution for different cross-sections of MHP placed at different inclination angles. Comparing figs. 6(a) and 6(b) it is evident that at 90° inclination angle wall temperature of evaporator section is higher when there is a wick structure inside it. Moreover, the temperature of the evaporator section is lowest for circular cross-section under identical working condition, while comparing figs. 6(b) and 6(c). Figures 6(c) and 6(d) indicate that at a particular heat input, the temperature of evaporator section is higher when MHP is placed at a lower inclination angle. From these figures it is also possible to find out the maximum heat transfer rate of a MHP of certain cross-section, placed at a particular inclination angle. From these figures it is evident that the slope of evaporator wall temperature distribution with heat input is almost same up to a certain limit of heat input but beyond that limit the slope changes suddenly. This sudden change in the slope indicates the seizure of MHP operation (this situation is also known as dry out condition of MHP). Therefore, maximum heat transfer capability of a MHP is

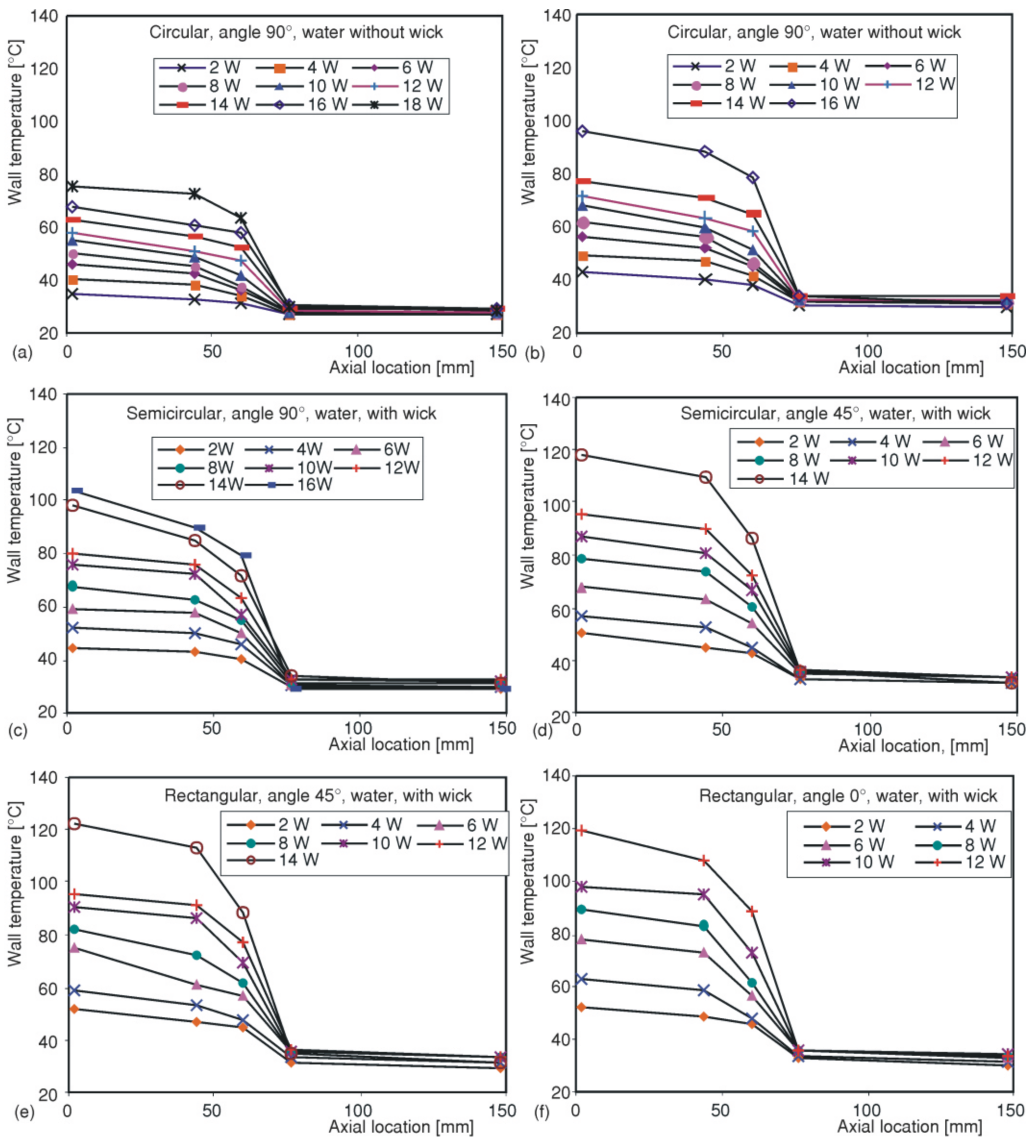

Figure 6. Axial temperature distributions of MHP

defined as the maximum heat input to it up to which no dry out occurs and this limit is known as stable operational zone of the MHP. Within stable operational zone, heat transfer coefficient increases with increasing of heat input. In dry out condition rate of vapor transport through the vapor space is so higher than the rate of condensate transport through wick that evaporator becomes empty and liquid slug starts to deposit in the condenser section.

Figures 7(a) to 7(c) show the effects of capillary structure in a MHP when it is placed at 90° inclination angle. It is evident that within stable operational zone, overall heat transfer coefficient of MHP with wick is lower than that of MHP without wick. This implies that the MHP working with the aid of gravity is mainly affected by the radius of vapor space and the influence of wick is even worse.

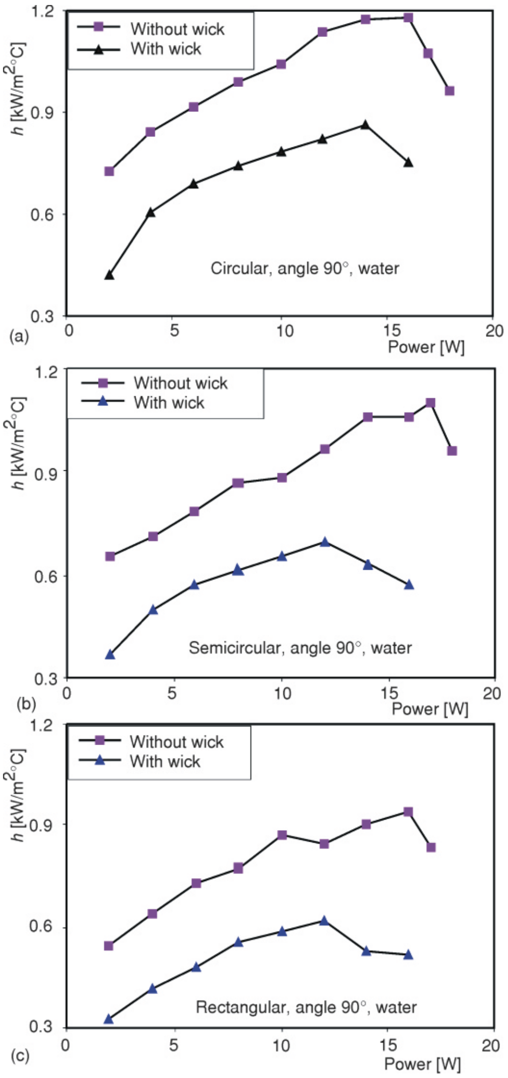

Figure 7. Variation of overall heat transfer coefficient in MHP

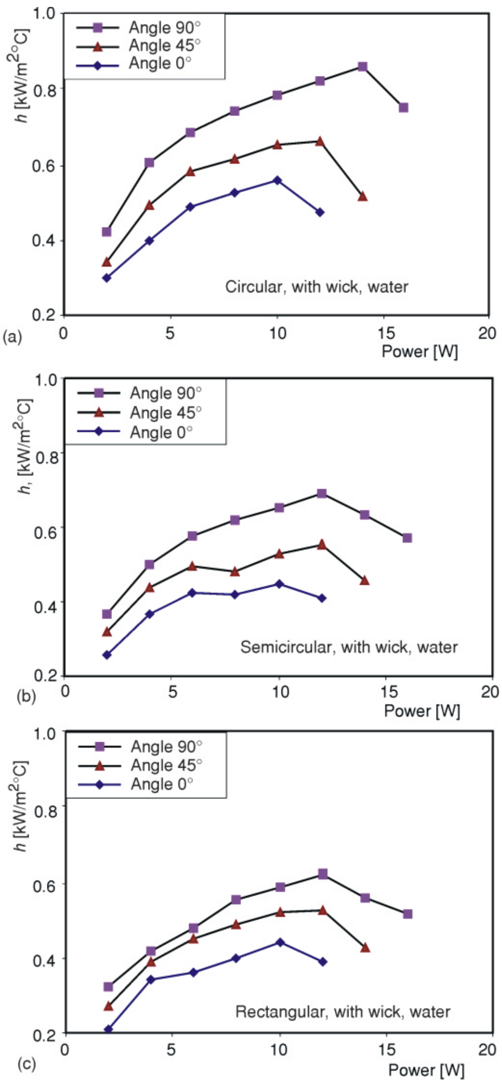

Figure 8. Variation of overall heat transfer coefficient with heat input at different inclinations

Figures 8(a) to 8(c) show effects of MHP inclination angles on overall heat transfer coefficient. From all these figures it is evident that overall heat transfer coefficient of circular MHP is highest at 90° inclination angle within the stable operating zone and for any MHP cross-sections overall heat transfer coefficient decreases with decreasing of inclination angle.

Figures 9(a) to 9(d) show the effect of MHP cross-sections on evaporator heat transfer coefficient. From all these figures it is evident that evaporator heat transfer coefficient of circular MHP is higher than any other cross-sections at any inclination angle.

Figure 9(a) shows that within stable operational zone at maximum heat input situation and at 0° angle, evaporator heat transfer coefficient of circular MHP is 24% and 30% higher than that of semi circular and rectangular cross-sections, respectively.

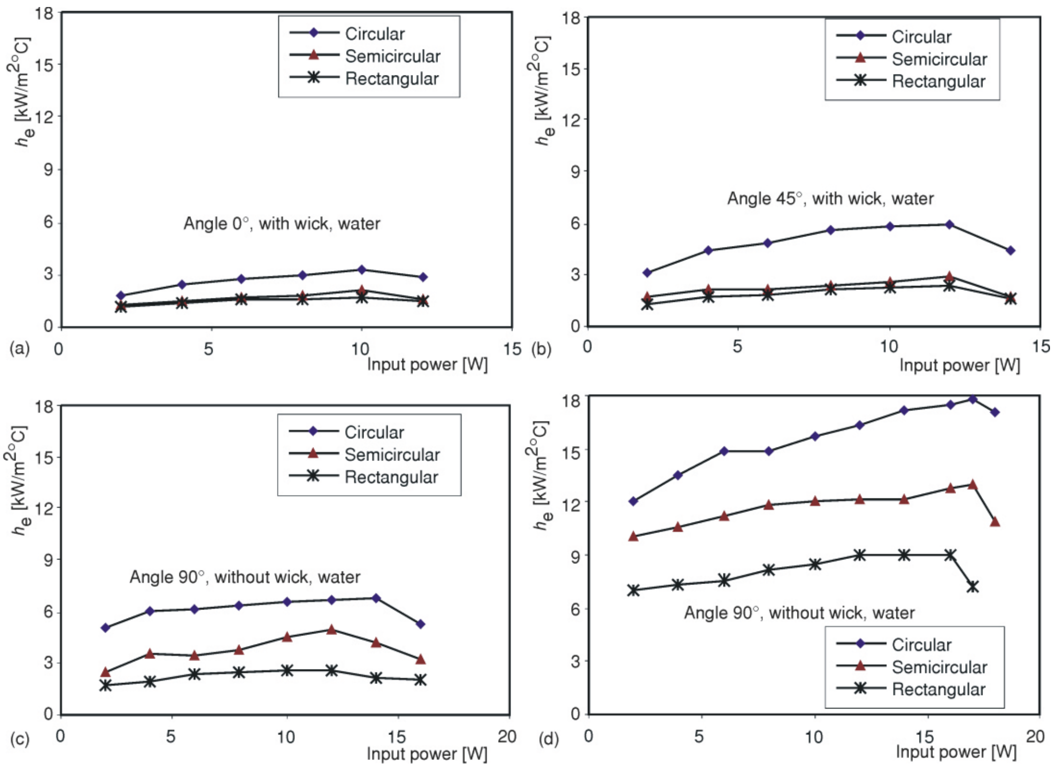

Figure 9. Variation of evaporator heat transfer coefficient in MHP with heat input

From fig. 9(b) it is evident that within stable operational zone at maximum heat input situation and at 45° angle, evaporator heat transfer coefficient of circular MHP is 56% and 60% higher than that of semi circular and rectangular cross-section, respectively.

At 90° inclination angle evaporator heat transfer coefficient of circular MHP is 38% and 68% higher than that of semi circular and rectangular cross-sections, respectively, as shown in fig. 9(c).

In case of MHP without wick, evaporator heat transfer coefficient for circular cross-section is 27% and 55% higher than that of semicircular and rectangular cross-section, respectively, within the stable operational zone and at maximum heat input situation, as shown in fig. 9(d).

Figures 10(a) to 10(d) show the effect of MHP cross-sections on its condenser heat transfer coefficient. From all these figures it is evident that condenser heat transfer coefficient of circular MHP is higher than any other cross-sections at any inclination angle. But comparing with evaporator heat transfer coefficient as shown in fig. 9 it is clear that the effect of MHP cross-section is more significant in the condenser section compared to evaporator section. It implies that the channel height reduction tends to impede the thermal performance of the condenser section more significantly.

Figure 10(a) shows that within stable operational zone, at maximum heat input situation and at 0° angle, condenser heat transfer coefficient of circular MHP is 16% and 28% higher than that of semicircular and rectangular cross-section, respectively.

From fig. 10(b) it is evident that at 45° angle, the condenser heat transfer coefficient of circular MHP is 8% and 15% higher than that of semicircular and rectangular cross-section.

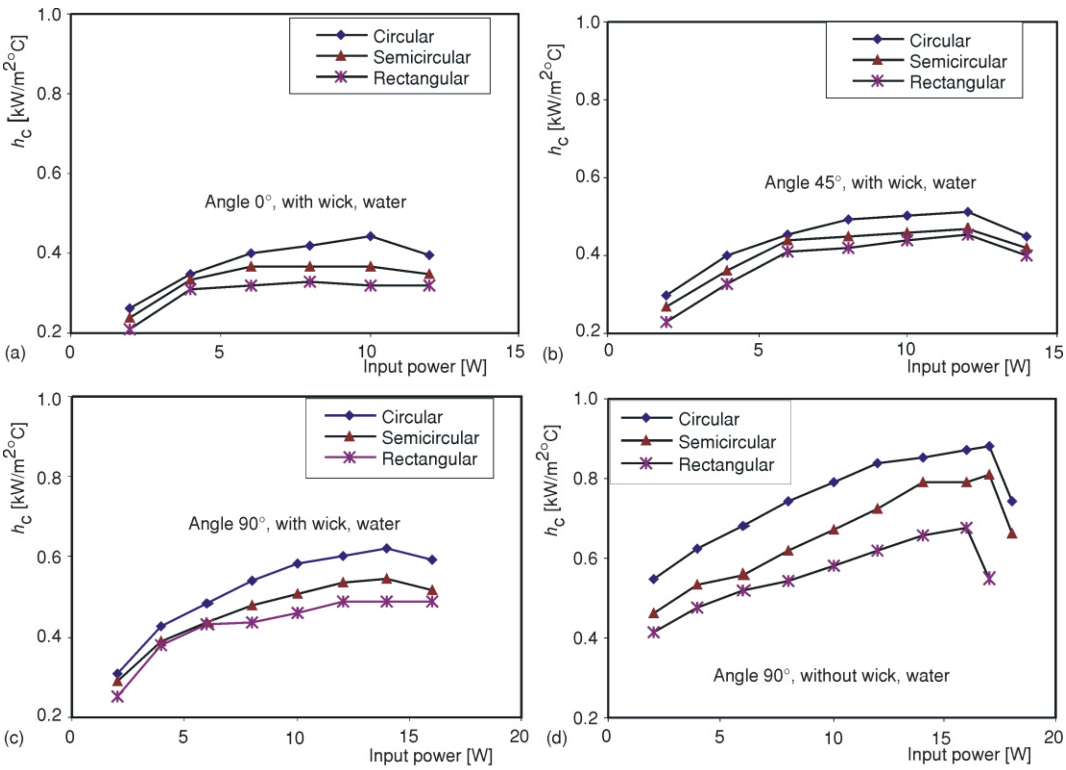

Figure 10. Variation of condenser heat transfer coefficient in MHP with heat input

At 90° inclination angle condenser heat transfer coefficient of circular MHP is 19% and 26% higher than that of semi circular and rectangular cross-section as shown in fig. 10(c).

Similarly, in case of MHP without wick, condenser heat transfer coefficient of circular cross-section is 10% and 38% higher than that of semi circular and rectangular cross-section respectively, as shown in fig. 10(d).

Figures 11(a) to 11(d) show the effect of MHP cross-section on its effective heat transfer coefficient. Figure 11(a) shows that within stable operational zone, at maximum heat input situation and at 0° angle, effective heat transfer coefficient of circular MHP is 20% and 27% higher than that of semi circular and rectangular cross-section, respectively.

From fig. 11(b) it is evident that within stable operational zone, at maximum heat input limit and at 45° angle, effective heat transfer coefficient of circular MHP is 19% and 26% higher than that of semicircular and rectangular cross-sections, respectively.

At 90° inclination angle effective heat transfer coefficient of circular MHP is 28% and 37% higher than that of semicircular and rectangular cross-sections, respectively, as shown in fig. 11(c).

Therefore comparing figs. 11(a), (b), and (c) it can be concluded that the effect of MHP cross-section on effective heat transfer coefficient is much higher at 90° inclination angle.

In case of MHP without wick effective heat transfer coefficient for circular cross-section is 9% and 30% higher than that of semicircular and rectangular cross-section, respectively, as shown in fig. 11(d).

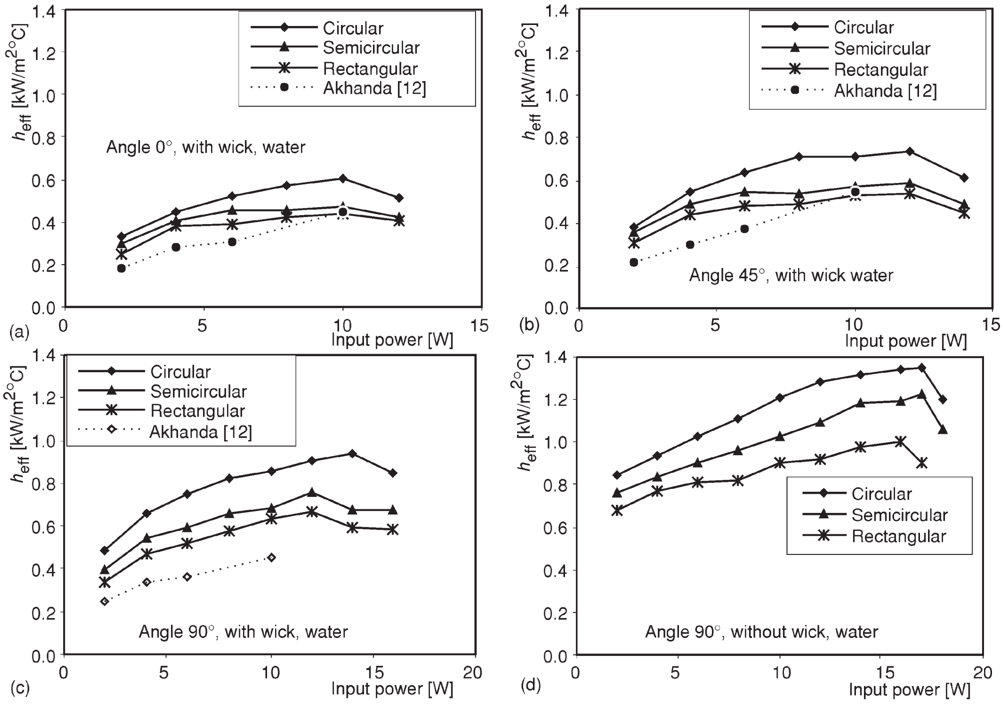

Figure 11. Variation of effective heat transfer coefficient in MHP with heat input

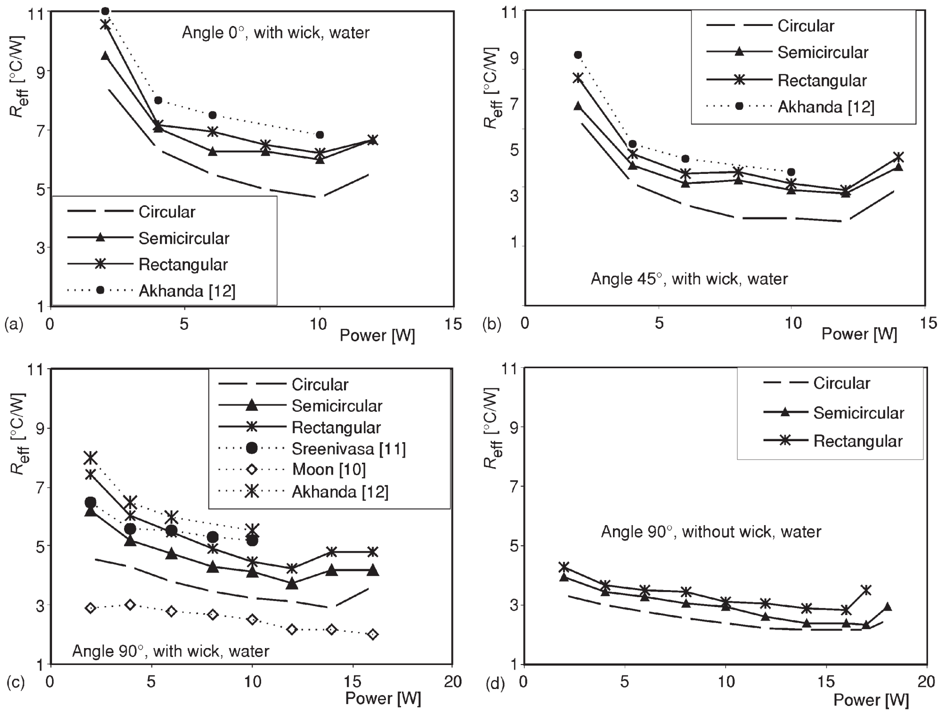

Figure 12. Variation of effective thermal resistance in MHP with heat input

Figures 12(a) to 12(d) show the effects of MHP cross-sections on effective thermal resistance. Results of Moon *et al.* [10], Sreenivasa *et al.* [11], and Akhanda *et al.* [12] are included in these plots to compare their results with results of the present study.

Figure 12(a) shows that, within stable operational zone, at maximum heat input situation and at 0° angle, effective thermal resistance of circular MHP is 21% and 31% lower than that of semicircular and rectangular cross-sections, respectively.

At 45° inclination angle, effective thermal resistance of circular MHP is 25% and 28% lower than that of semicircular and rectangular cross-sections, respectively, as shown in fig. 12(b).

Figure 12(c) is for 90° orientation of MHP which shows that, effective thermal resistance of circular MHP is 21% and 37% lower than that of semicircular and rectangular cross-sections, respectively. Similarly, incase of MHP without wick, effective thermal resistance of circular MHP is 9% and 32% lower than that of semicircular and rectangular cross-sections, respectively, as shown in fig. 12(d).

Correlation of thermal performance

In this study an empirical correlation has been developed which correlates all the experimental data within 7% by the following equation:

$$\frac{h}{h_{\mathrm{eff}}} = 0.894\left(\frac{P}{P_{\max}}\right)^{0.0412}(Fi)^{0.0085}(Sk)^{0.0083}$$

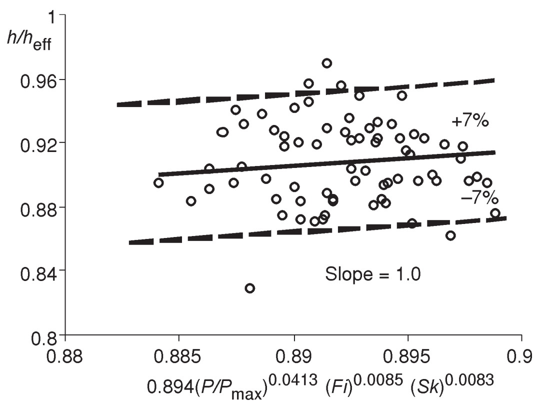

Figure 13. Graphical representation of the correlation equation

Graphical representation of correlation is shown in fig. 13.

Conclusions

The following conclusions can be drawn from this investigation.

At a particular inclination angle and at a particular heat input, thermal performance of circular MHP is the best followed by semicircular and rectangular cross-sections.

For a particular cross-section of MHP and at a particular heat input, thermal performance of MHP is better at 90° inclination angle compared to other inclination angles.

At 90° inclination angle, thermal performance of MHP without wick is better than that of MHP with wick and it is true for any cross-sections of MHP.

Among all cross-sections of MHP, circular MHP placed at 90° inclination angle possesses maximum heat transport capability.

Steady-state temperature of evaporator section increases with increasing of heat loads.

For all cross-sections and at a particular heat input, wall temperature of evaporator section goes higher when MHP is placed at lower inclination angles and wall temperature of the evaporator section decreases as inclination angle increases.

Dry out point varies with the variation of inclination angles and cross-sections. Among all MHP cross-sections and orientations circular MHP placed at 90° inclination angle experiences dry out at maximum heat input and rectangular MHP placed at 0° inclination angle experiences dry out at minimum heat input.

Experimental data are satisfactorily correlated within 7% deviations.

Nomenclature

A_c – condenser surface area, [m^2]
A_e – evaporator surface area, [m^2]
D – hydraulic diameter, [mm]
Fi – proposed dimensionless number, (= H/D), [–]
H – profile height, [mm]
h – overall heat transfer coefficient, [$kWm^{-2}°C^{-1}$]
h_c – heat transfer coefficient of condenser, [$kWm^{-2}°C^{-1}$]
h_e – heat transfer coefficient of evaporator, [$kWm^{-2}°C^{-1}$]
h_{eff} – effective heat transfer coefficient, [$kWm^{-2}°C^{-1}$]
P – power input, [W]
P_{max} – maximum power input, [W]
R_{eff} – effective thermal resistance, [$°CW^{-1}$]
Sk – proposed dimensionless number, (= 1 + sinϕ), [–]
T_1, T_2, T_3, T_4, T_5 – thermocouple readings at locations shown in fig. 5, [°C]
T_c – average temperature of condenser section, [°C]
T_e – average temperature of evaporator section, [°C]
T_{sat} – saturated vapor temperature, [°C]
ϕ – inclination angle of MHP, [deg]

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Appendix

$$h_{\mathrm{eff}} \quad \frac{Q}{A_{\mathrm{e}}(T_{\mathrm{e}} \quad T_{\mathrm{c}})} \ [\mathrm{kW/m^2 °C}] \tag{1}$$

$$h \quad \frac{Q}{A_{\mathrm{e}}(T_1 \quad T_5)} \ [\mathrm{kW/m^2 °C}] \tag{2}$$

$$h_{\mathrm{e}} \quad \frac{Q}{s_{\mathrm{e}}(T_{\mathrm{e}} \quad T_{\mathrm{sat}})} \ [\mathrm{kW/m^2 °C}] \tag{3}$$

$$h_{\mathrm{c}} \quad \frac{Q}{A_{\mathrm{c}}(T_{\mathrm{sat}} \quad T_{\mathrm{c}})} \ [\mathrm{kW/m^2 °C}] \tag{4}$$

$$R_{\mathrm{eff}} \quad \frac{T_{\mathrm{e}} \quad T_{\mathrm{c}}}{Q} \ [\mathrm{°C/W}] \tag{5}$$

$$T_{\mathrm{e}} \quad \frac{T_1 \quad T_2}{2} \tag{6}$$

$$T_{\mathrm{c}} \quad \frac{T_4 \quad T_5}{2} \tag{7}$$

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