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THERMAL PERFORMANCE OF PARALLEL MINIATURE HEAT PIPE SYSTEM

C. M. Feroz, **M. Mamunur Rahman**, **Md. Hasibul Alam**, **and Auvi Biswas**

Department of Mechanical Engineering, BUET, Dhaka, Bangladesh

ABSTRACT
The experimental analysis presented here is based on the heat transfer performance of parallel miniature heat pipe (mHP) system intended for desktop computer processor cooling. The system consists of six single copper tube mHPs slotted into two copper blocks at the evaporator section and fifteen parallel copper sheets at the condenser section. Stainless steel wicks are inserted inside the copper tubes, while methanol and iso-propanol serves solely as working fluids. Heat transfer characteristics of mHPs are determined by conducting the experiment at various levels of heat inputs and analyzed to evaluate the performance. The overall heat transfer coefficient of the system is determined as a performance parameter. The results show that, the behavior of the heat pipe varies significantly for different heat inputs and different working fluids.

Keywords: Parallel Miniature Heat Pipes, Heat Flux, Overall Heat Transfer Coefficient, Working Fluids.

1. INTRODUCTION
There was a time when people were satisfied with conventional fan cooling in electronic devices despite the huge noise generation and power consumption. But the realization changed quickly because of the increasing heat generation with increasing working speed and also the space constraint. Specially, the cooling of computer processors demanded a whole new outlook. The CPU of a desktop and server computer releases 80 to 130 W and notebook computer 25 to 50W of heat energy [1]. It became more challenging because the chip surface temperature should not be allowed to go beyond 100^{o}C [2]. Scientists started to apply liquid submersion cooling, active and passive heat sink cooling, thermoelectric cooling etc. in computer cooling. But soon these became more or less obsolete for integration and reliability issues. Even the most recent technique of integrated chip cooling is not acclaimed by all because of the huge cost. With massive development in the concept and technology of two-phase and porous media heat transfer systems heat pipes have come up as one of the most potential candidate to meet these challenging needs.

A typical heat pipe consists of a sealed pipe or tube made of a material with high thermal conductivity. After evacuating, the pipe is filled with a fraction of a percent by volume of working fluid, chosen to match the operating temperature. Due to the partial vacuum that is near or below the vapor pressure of the fluid, some of the fluid will be in the liquid phase and some will be in the gas phase. Having a vacuum eliminates the need for the working gas to diffuse through another gas and so the bulk transfer of the vapor to the cold end of the heat pipe is at the speed of the moving molecules. Inside the pipe's walls, an optional wick structure exerts a capillary pressure on the liquid phase of the working fluid. At the hot interface, the fluid turns to vapor absorbing the latent heat resulting in a phase change, and the gas flows and condenses back to liquid in the cold interface releasing the latent heat. The liquid is moved back by capillary gravity action to the hot interface to evaporate again and repeat the cycle.

The very first advent of heat pipe was in 1944 by R.S. Gauglar [3] of General Motors, which was rediscovered by George Grover and his co-workers [4] of the Los Alamos Scientific Laboratory in 1963. Since then continuous research and development have gone behind heat pipe. Starting in the 1980s Sony began incorporating heat pipes into the cooling schemes for tuners & amplifiers in electronic products in place of both forced convection and passive finned heat sinks. Cao, Y. and Gao, M. [5] designed, fabricated, and tested wickless, cross-grooved thermal spreaders, which were made of Copper or Aluminum. The maximum heat flux achieved was about 40 W/cm^2 for methanol and 110W/cm^2 for water with a total heat input of 393W.

In the development process, as the electronic devices became more mobile in use and tiny in size, scientists started to concentrate more on micro and miniature heat pipes. Zhang, J. and Wong, H.[6] studied heat transfer and fluid flow in an idealized micro heat pipe with the support of NASA and LaSPACE. They made an analysis for four different values of length to width ratio of an idealized micro heat pipe, viz. 20, 50, 100, and 200. In a study of micro and miniature heat pipes, developed by A.R. Anand [7], attempts have been made to develop a one dimensional numerical model of micro heat pipes, taking into account the effect of liquid-vapor interfacial shear stress. In 1991, Wu and Peterson [8] developed a transient numerical model capable of predicting the thermal behavior of micro heat pipes and compared their results with the steady state results obtained by Babin et al. [9] in 1990.

For electronic equipments, heat pipes of diameter 3 to 6 mm and length less than 400 mm are preferred [10]. Most preferable length is 150 mm [11]. An experimental study is performed by Tanim *et al.* [12] to investigate the performance of cooling desktop processors using miniature heat pipes of 5.78 mm ID and a length of 150 mm with respect to the normal fanned CPU unit. They reported that four mHPs system shows better performance than that of two mHPs. Another experiment has been performed similar to this one by Imtiaz and Feroz [13] on Parallel mHPs for cooling desktop computer processor. They concluded that the addition of a cooling fan in the condenser section provides the lowest temperature of the surface of the processor. This particular experiment is further development of that one. Instead of drawing heat directly from the CPU, external variable heat source is used to observe the performance of the heat pipe under different heating conditions. Finally the performance of the mHPs is also checked for two different working fluids.

2. EXPERIMENTAL SETUP

The experimental setup for this study mainly consists of – 6 parallel mHPs along with heating and cooling sections, a variac, 5 selector switches, 30 thermocouples and a temperature controller, as shown in Fig. 1.

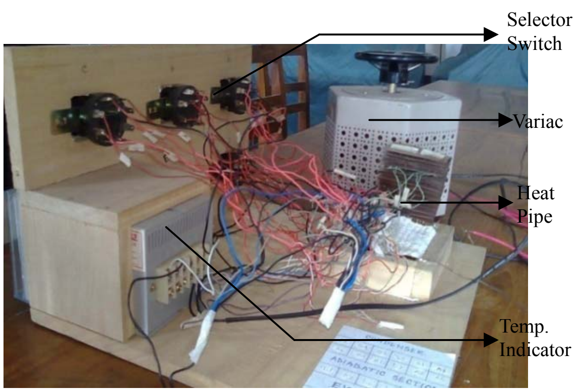

Fig 1. Experimental setup

Six mHPs are placed parallel to each other for cooling purpose. Every mHP has an inner diameter of 1.8 mm and outer diameter of 2.8 mm having a length of 150 mm. There are three sections in every mHP: evaporator, adiabatic section and condenser, as shown in Fig. 2. The condenser sections of MHPs are made of fifteen copper sheets of 67mm×50mm (thickness 0.5mm) placed parallel as extended fins at a constant interval of 5 mm. Plates are joined with the mHPs with Araldite for better heat transfer. Having the space constrain in field of its application in mind, the MHPs are bend at 90o in adiabatic section. The evaporator sections of MHPs are inserted in to the grooves of copper blocks. Two copper blocks of 67mm×50mm×8mm are made very precisely to mate with the MHPs. Grooves are cut inside the blocks. The blocks are precise in dimension and surfaces are finished highly to reduce the contact resistance as well as to increase the heat transfer rate. Nichrome wires are wound around the evaporator section and then it is electrically and thermally insulated using insulating tape and asbestos respectively.

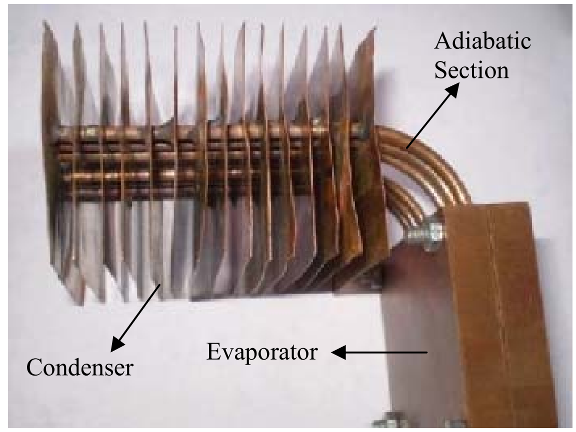

Fig 2. mHPs with Evaporator, Adiabatic and Condenser sections

Before bending, wick of stainless steel of 200 meshes are inserted into the MHPs. After inserting the wick, mHPs are bent to the desired angle. One end of the mHPs is sealed and working fluid with charge ratio 0.9 is poured into that. Five calibrated K-type (Φ = 0.18 mm) thermocouples are attached at the wall of each mHP using adhesive to measure the wall temperature: two units at the evaporator section, one unit at the adiabatic section and two units at the condenser section. Locations of thermocouples connected on different points along the length of the mHP are shown in the Fig. 3. All thermocouples are connected to a digital temperature indicator through selector switches.

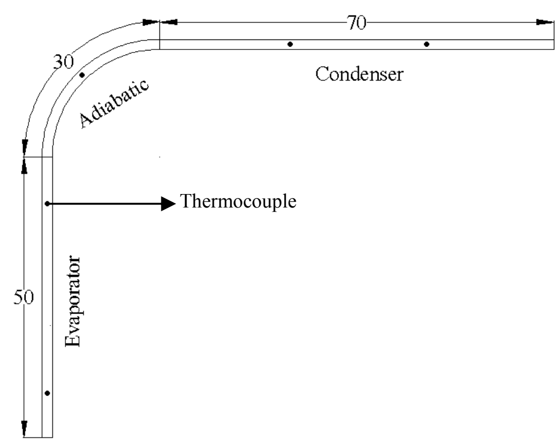

Fig 3. Location of thermocouples

3. TEST PROCEDURE

Experiment is conducted for two different working fluids: propanol-2 and methanol. After insertion of the fluid, the mHPs are sealed and alternating current (AC) line is connected to the nichrome wire via a variac. The heat input is increased gradually by changing the voltage with the variac. Wall temperatures of mHPs are recorded for 40 to 45 minutes (until steady values are attained) at an interval of 5 minutes for each heat input value. The whole procedure is repeated for both the fluids. The experimental parameters are stated in Table 1.

Table 1: Experimental Parameters

Parameters	Condition
Number of the heat Pipes	6
Diameter of the heat pipe(mm)	ID- 1.8; OD- 2.8
Length of the heat pipe(mm)	150
Length of the evaporator section (mm)	50
Length of the adiabatic section (mm)	30
Length of the condenser section (mm)	70
Working fluid	Propanol-2, Methanol
Dimension of the copper block(mm)	67×50×8
Dimension of the copper sheet(mm)	67×50×0.5
Charge ratio	0.9
Wick (SS)	200 meshes

4. TEST RESULTS AND DISCUSSION

As heat load is applied to the evaporator section, the temperature of the evaporator section rises and results in the vaporization of the working fluid. This vaporization of liquid absorbs heat from the evaporator section. The heating value is determined by multiplying the voltage indicated by the variac and the current drawn into the circuit.

Heat flux (φ) is calculated using Eq. (1) where Q is the input heat power and A is heating area of the mHP system, i.e. the outer surface area of all six mHPs at the evaporator section.

$$\varphi = Q/A \qquad (1)$$

4.1 Temperature Profile

Fig. 4(a) and 4(b) shows the temperature profiles along the length of the mHP for propanol-2 and methanol respectively. The temperatures are average of all the readings of the thermocouples in the same horizontal line, i.e. the average of the temperature readings of the six mHPs. Uniformity of temperature in the evaporator and condenser sections indicates the reliability of using mHPs for the cooling of desktop processors.

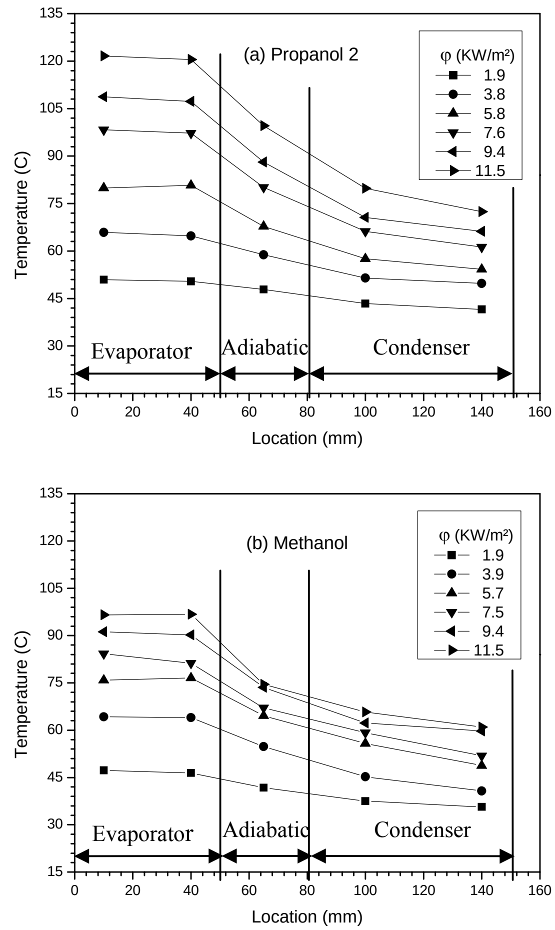

Fig 4. Temperature Profile along the length of mHPs

4.2 Overall Heat Transfer Coefficient

A graph of overall heat transfer co-efficient vs. heat flux for both the fluids is shown in Fig. 5. The overall heat transfer co-efficient (h_{ov}) is determined from Eq. (2). For a particular system, it depends solely on the temperature difference (ΔT) of the evaporator end and the condenser end.

$$h_{ov} = Q / (A^* \Delta T) \qquad (2)$$

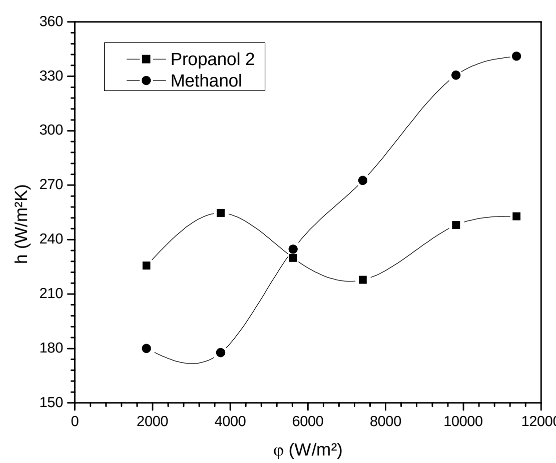

Fig 5. Overall heat transfer coefficient vs. heat flux

As observed from the graph, at lower heat flux value, the overall heat transfer coefficient for the propanol-2 based system is higher than that of the methanol based system. But for heat fluxes higher than 5600 W/m^2, the overall heat transfer coefficient is significantly higher for the later one.

4.3 Transient Temperature

Variation of evaporator surface temperature with time is shown in Fig. 6(a) and (b) for the lowest and the highest heat fluxes, respectively. Here, the temperatures of the evaporator section are only considered, because only this section of the mHP system is to be placed on the computer processor and its temperature resembles the temperature of the processor in field of application. The figure indicates that, at a lower heat flux, after attaining the steady state, both the systems show almost the same temperature, but at a higher heat flux, propanol-2 based system shows a little bit higher temperature. It can also be observed that the propanol-2 based system attains steady state temperature a little faster than the acetone based one.

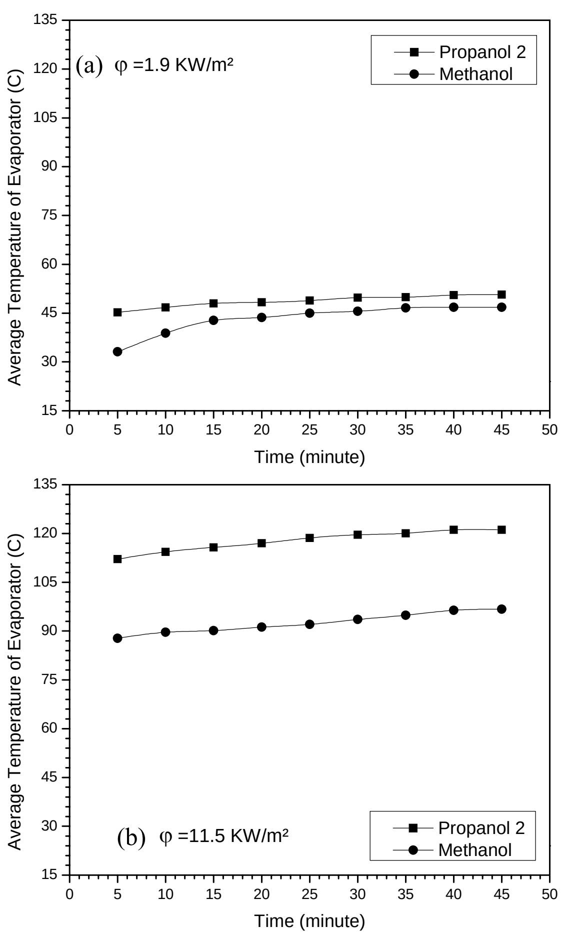

Fig 6. Variation of Evaporator surface temperature with time

5. CONCLUSION

The following conclusions can be drawn from the experimental studies:

(l) Insignificant fluctuation of temperature along the length of the MHPs indicates the stability and consistency of the system.

(2) Methanol is the better working fluid to be used in a mHP than propanol-2, as the overall heat transfer co-efficient is higher at higher heat fluxes. Besides, lower evaporator section temperature is achieved with the methanol based system.

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7. NOMENCLATURE

Symbol	Meaning	Unit
Q	Heat power input	(W)
φ	Heat Flux	(W/m^2)
h_{ov}	Overall heat transfer co-efficient	(W/m^2K)

8. MAILING ADDRESS

Prof. Chowdhury Md. Feroz
Department of Mechanical Engineering, BUET
Dhaka-1000, Bangladesh
Email: cmferoz@me.buet.ac.bd