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Novel Groove-Shaped Screen-Wick Miniature Heat Pipe

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Miniature heat pipes (MHP) are passive heat transport devices mainly considered in electronics packaging for high heat flux acquisition and transport. Their applications could include thermal management of a variety of electronic devices such as computer processors and laser diodes. A novel screen-wick design suitable for MHP is described. The new design promises improved performance and ease of fabrication and is recommended in place of the familiar forms of rectangular groove design MHP. Design and fabrication details, along with steady-state horizontal orientation performance test results of a proof-of-concept rectangular copper–water heat pipe are presented. Heat flux, temperature difference, and heat transfer coefficient data are compared with the literature data of a comparable flat MHP with a machined groove wick. The performance of the new design matches the comparable groove design. The highest applied evaporator heat flux at the heater surface was 115 W/cm^2 at an operating temperature of 90°C and evaporator-to-adiabatic temperature difference of 37°C. Results on the heat transfer coefficients are also presented.

Nomenclature

A	=	area
h	=	heat transfer coefficient
k	=	thermal conductivity
L_F, L_w	=	dimensions of screen folds (Fig. 1)
Q_{in}	=	heat input
q	=	heat flux
r_c	=	capillary radius
r_h	=	hydraulic radius
T	=	temperature
t_w	=	thickness of wall
w	=	width of groove
ΔT_{AC}	=	temperature difference between adiabatic and condenser
ΔT_{EA}	=	temperature difference between evaporator and adiabatic
δ	=	depth of groove

Subscripts

A	=	adiabatic
C	=	condenser
E	=	evaporator
$_{,}B$	=	bottom surface
$_{,}i$	=	inner wall
$_{,}M$	=	average or mean value
$_{,}o$	=	outer wall
$_{,}S$	=	side surface
$_{,}T$	=	top surface
w	=	wall

Introduction

IT is well known that heat pipes have proliferated over the years from nuclear industry to space and many terrestrial and military applications. The size and performance parameters have improved considerably. One significant recent industrial application is the notebook computer, wherein the processor chip is cooled by a pencil sized (3-mm-diam × 250-mm-long) miniature heat pipe. Importance and keen interest in this area are further evidenced from the volume of recent research publications.[1,2] The heat pipe industry has projected an annual demand of millions of these and other heat pipes. These heat pipes are heat-flux limited (5–25 W/cm^2), whereas the next generation of miniature heat pipes are expected to handle up to 200 W/cm^2. Miniature heat pipe technology has been evolving, and it is time that it be readied for exploitation by the high volume (millions of pieces per year) military and industrial markets. To achieve this, the mass-production issue has to be addressed. The present heat pipe manufacturing process employs conventional technologies and processes, and the production capabilities are very small. It is estimated that by the year 2010, the total requirement of miniature heat pipes will exceed tens of millions per year.

Future advanced electronics packages for the U.S. Air Force and space programs will require high-power devices. These devices are packaged in high-density modules, necessitating innovative thermal management solutions at all levels of packaging, namely, chip, board, and box. Miniature heat pipes (MHP) are one of the viable, accepted, fail-safe, and passive solutions of waste heat removal methods in use in the industry. High-performance, high-heat-flux heat pipe technology provides a simple, reliable solution to power electronics cooling problems. However, the high-performance groove structure needed for the MHP is currently manufactured in laboratory quantities via expensive and time-intensive machining processes. Heat pipe manufacturers are keen on advancing the MHP technology to the level of readiness and affordability, as in the cases of the electronics and materials fields.

The main objective of this research is to investigate a new MHP design and fabrication process that will provide a simple and low-cost solution to the manufacturing of miniature wick structures. The wick design, fabrication, and performance aspects are demonstrated by experimental verification of a proof-of-concept MHP. This design approach is expected to eliminate the difficulties encountered in the expensive machining of fine rectangular grooves.

MHP

Description

MHP are the class of heat pipes that function in the same way as conventional heat pipes, with the exception of the physical sizes. The heat pipe researchers have the following general understanding on the definition of a micro heat pipe. A micro heat pipe can be conceived as a functional heat pipe with at least one capillary channel such that $r_c/r_h \geq 1$ where r_c is the capillary radius and r_h is the hydraulic radius of the flow channel.[3,4] MHP, on the other hand, can be

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imagined as conveniently manufactured design arrangements of several micro capillary channels in combination with a common vapor core. Basically, heat pipes have some form of a machined or inserted capillary wick lining on the inside walls of suitable metal containers, which are hermetically sealed after placing inside just enough pure working fluid to saturate the wick structure. The materials of the wick and the container vary depending on the application. The wick structure could be machined grooves, wrapped metal screen mesh, or forms of sintered porous powder metal.

A number of literature references containing the previous art and technology of the MHP exist. MHP of sizes ranging from $0.6 \times 0.1 \times 25$ mm^3 to $7 \times 2 \times 120$ mm^3 were fabricated and tested by various researchers. A good review of these is available in Ref. 4. The heat transfer performance numbers vary from 1 to 60 W/cm^2 for evaporator heat flux in near-room temperature operation. More recent publications describe three miniature heat pipes with machined trapezoidal or rectangular grooved wicks.[5,6] The ratio, r_c/r_h of these grooves varied from 1.24 to 2.12. A metal rolling method was used for making the trapezoidal grooves and a dicing saw cutting method was used for the rectangular grooves. These fabrication methods pose a number of limitations and disadvantages:

1) Physically manufactured rectangular or trapezoidal grooves are still too big ($w \geq 0.2$ mm), and their capillary pumping radii are not the smallest compared to those possible with the sintered wick or wire meshes ($w = 0.04$–0.15 mm).

2) The groove forming methods such as rolling, dicing saw cutting, electrical discharge machining, etc., are difficult and expensive, and it is difficult to form deep grooves ($w/\delta > 4$) without damaging the walls of the grooves.

3) In the solid-walled grooves, circumferential fluid distribution (an essential feature for evaporator priming) is not possible.

4) Longitudinal seam welding of the half boxes of the envelope tube adds to the fabrication difficulties.

To alleviate these difficulties, a simple and new wick design has been created and investigated.

Novel Wick Design

The present wick design concerns a relatively simple fabrication methodology for manufacturing a porous-wall groove-shaped screen-wick structure. This design simplifies the fabrication process and improves the heat transport performance of the MHP. The new wick is formed out of a commercially available metal wire cloth stock. A square- or rectangle-shaped wavy or corrugated pattern with constant pitch is formed using a special industrial fin-making process developed by a commercial firm (Robinson Fin Machines, Inc., Kenton, Ohio).[7] A sample wick design for a rectangular cross section MHP using 59×59 cm^{-1} mesh copper screen cloth is shown in Fig. 1. The width of the screen stock that is needed to create a certain length of the wick L_f, screen cloth thickness x, groove width w, and the associated pitch p of the fold/groove required are subject to the limitations of the fin machine and tooling. A straight tube with rectangular cross section 6.35×12.7 mm and a 1.22-mm wall is selected from commercially available standard refrigeration copper tubing. In the present study, a folded screen rectangular groove design with $w = 0.2$ mm, $p = 0.305$ mm, groove aspect ratio of $\delta/w = 4.5$, and $L_f = 100$ mm is selected. A total of 72 screen folds (36 U shaped and 36 inverted U shaped), measured along L_w and cut along the L_f direction (Fig. 1) from the screen preform, are accommodated within the internal cross section of the selected rectangular tube as shown in Fig. 2. This design is comparable to the machined groove design with $w = 0.2$ mm, $\delta/w = 2.0$, and axial length $L_f = 120$ mm found in the literature.[5,6] The length of the present MHP is limited to 108 mm because of the tooling constraint. To minimize cost of fabrication, it was decided to use the available tools. The MHP is fitted with two welded end caps and a fill tube to facilitate filling of the working fluid.

The following advantages are claimed over the current methods used in manufacturing MHP:

1) It is easier to form the screen cloth to the desired rectangular groove geometry and insert into the heat pipe tube than to machine solid-wall rectangular grooves.

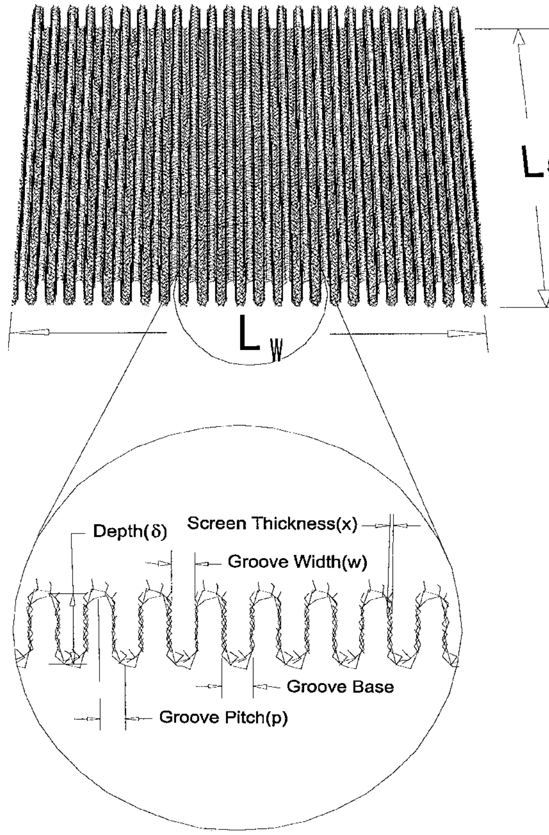

Fig. 1 New folded screen-wick design of MHP.

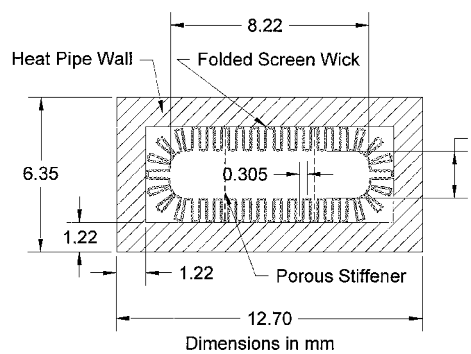

Fig. 2 Cross-sectional view of MHP.

2) An inverted-meniscus evaporator, known to support high evaporator heat flux, is automatically created in this new design as illustrated in Fig. 3. In other words, the new wick design is dry-out tolerant.

3) The working fluid wicking in the circumferential direction (necessary to delay evaporator dry out at high heat flux operation) is also automatically introduced in the new design.

4) The new design derives the benefit of a composite wick performance where the pores of the screen work as pumping wick and the rectangular grooves work as transport (artery) wick.

Design of an MHP

It is a fairly simple task to complete the heat pipe design after choosing the physical dimensions of the wick and envelope from a practical standpoint. Basic steps found in heat pipe texts are followed to obtain the design parameters such as the operating pressure and temperature, end cap and fill-tube sizes, fill volume, liquid pressure drop, capillary limit, sonic limit, entrainment limit, and boiling limit. Capillary limit calculation is tricky for this design because of the porous-wall groove nature of the wick. Neither the known capillary pore radius of the screen nor that of the groove alone can be used to calculate the pumping head available for the wick. A true value for

Table 1 Design details of heat pipe

Parameter	Value
Operating conditions	
Temperature range	20–150°C
Power	10–150 W
Material	
Envelope tube	Copper
Wick, end caps, and fill tube	Copper
Working fluid	Water
Physical dimensions	
Length	107.9 mm
Rectangular tube cross section, inner/outer size	10.26×3.91 mm^2/ 12.7×6.35 mm^2
Evaporator/adiabatic/ condenser length	18.5 mm/50.3 mm/ 32.5 mm
Vapor core (4.73 mm equivalent diameter)	8.43×2.08 mm
Wick[a]	
Screen mesh	59×59 cm^{-1}
Screen thickness	0.11 mm
Groove size	0.2×0.9 mm
Groove pitch	0.31 mm
No. of grooves	72
Fluid inventory	1.0 ml at 25°C

[a]Single-layer folded screen with corrugations in the form of grooves.

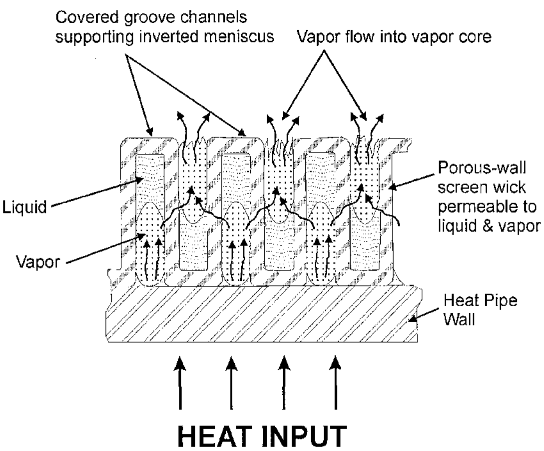

Fig. 3 Inverted meniscus formation in new wick design.

this composite wick can be obtained only by practical measurement of the wicking height, which is planned to be done in the future. However, a conservative estimate based on the groove radius is used for determining the capillary limit in the present study. The envelope of all of the heat transfer limits of this MHP is found to lie much higher than the intended operating regime of the heat pipe for a 20–150°C and 0–150 W operation. In addition, the goal here is to test the heat transfer characteristics and not the maximum transport capability. Table 1 provides the design details of the heat pipe.

Experimental Work

Fabrication

Two identical MHP test articles were fabricated to the design specifications described earlier. Standard soft rectangular copper tube was cut in straight lengths to form the heat pipe envelope and machined at the ends to fit the end caps and the fill tubes. A piece of plain woven copper screen cloth, 100×3000 mm, was cut and sent to preform the precise folding by the fin-making process. This yielded an approximately 750-mm-long, 100-mm-wide wick preform that was further sliced to 100×24 mm strips to form the special wick. Machined parts were cleaned as per standard procedure, and the wick was inserted into the tube using a special assembly mandrel. Two stiffeners made of copper strips were inserted into the wick grooves (as shown in Fig. 2) for maintaining proper wick contact with the tube wall. The end caps and fill tubes were welded in a glove box. The heat pipe was pumped, baked, and filled with analytical-grade distilled water in a vacuum-transfer fill process. The fill tube was pinched and sealed by welding. Weight measurements on the heat pipe before and after the filling, along with the displacement volume, determined the exact fill amount. Fill inventory could be changed as necessary by cutting the seal off and reprocessing.

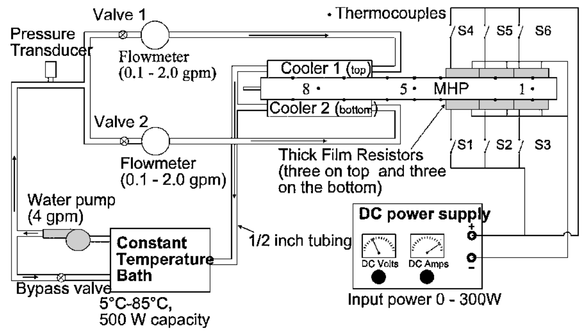

Fig. 4 Schematic diagram of test setup.

Test Setup and Test Procedure

The schematic sketch of the test setup showing the heating and cooling arrangements and the instrumentation is given in Fig. 4. There are a total of 19 copper–constantan (type-T) thermocouples, 8 at the top surface, 8 at the bottom surface, and 3 on the side wall of the MHP. Six thick film resistors on ceramic substrates 6.1×12.7 mm^2 are used to simulate the heat source of electronic components. The resistors are soldered onto the top and bottom walls of the evaporator of the MHP. The thermocouple locations and the attachment details are illustrated in Fig. 5. Four thin intermediate copper plates with slots are soldered to the heat pipe to facilitate thermocouple attachment. A dc power supply unit is used to power the electric heater. The heaters could be turned on or off (Fig. 4) and regulated independently or collectively to create a combination of input heat flux scenarios. Two modes of heating are used. Mode A uses one heater chip in the middle of the bottom surface to simulate high flux at peak load conditions of an electronic device that may use this heat pipe. Mode B uses all six heater chips to simulate a distributed flux at average load conditions. The condenser is cooled by two identical flat water-circulated coolers mechanically clamped onto the top and bottom surfaces of the condenser of the MHP. The operating temperature of the heat pipe can be controlled by regulating the flow rate and bath temperature of the constant temperature bath integrated into the setup along with the circulating pump, bypass valve, pressure transducer, control valves, and magnetic flow meters.

The experimental investigation focuses on the heat transfer characteristics of the MHP with different fill charges (0.69, 1.0, and 1.1 ml) of the working fluid (water) at various heat rates Q and operating temperatures. The operating temperature $T_{A,o}$ is indicated by the thermocouple in the middle of the adiabatic section. Input power is varied in increments of 10 W from 30 W up to the power at which a temperature difference of 30°C exists between the evaporator and the adiabatic sections. The MHP was tested in a horizontal orientation of the wider sides.

Experimental Uncertainty

The dc power measurement is very accurate and is only subject to the instrument accuracy of ±1%. The temperature measurement accuracy was verified by calibration with a secondary standard platinum resistance temperature device and found to be within the thermocouple manufacturer's data of ±0.03°C for type-T in 0–150°C.

Results and Discussion

Effect of Fluid Inventory

Figure 6 shows the mean temperatures in the evaporator $T_{E,M}$, condenser $T_{C,M}$, and adiabatic section $T_{A,M}$ as functions of the heat

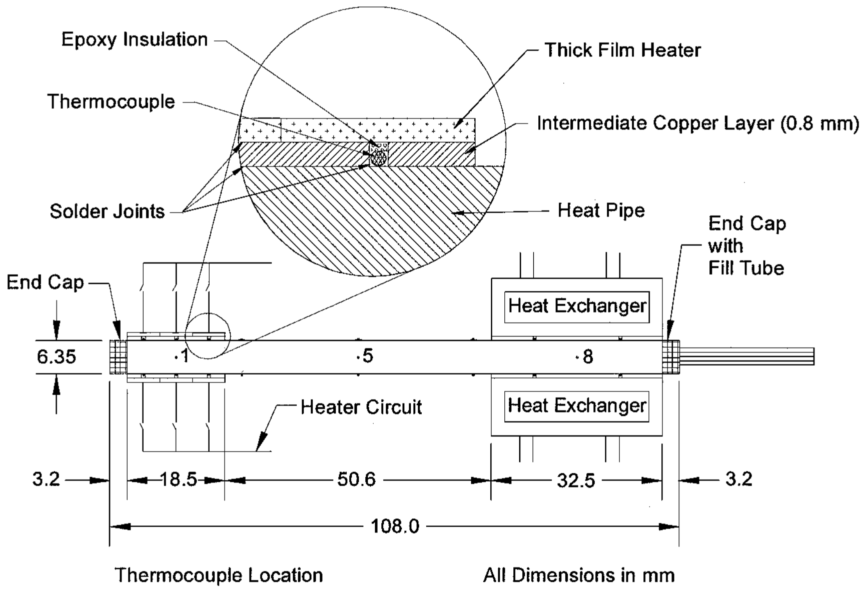

Fig. 5 Heater, cooler, and thermocouple mounting arrangements.

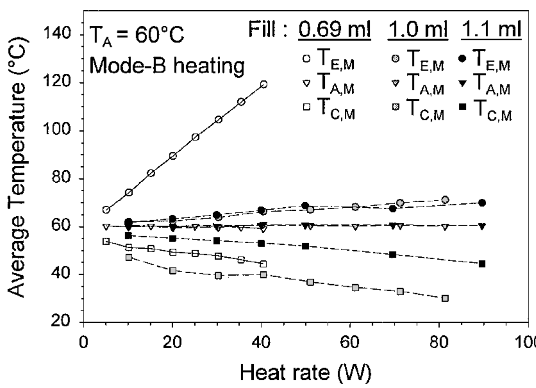

Fig. 6 Effect of fluid inventory on temperature profile for various heat input, $T_A = 60°$C.

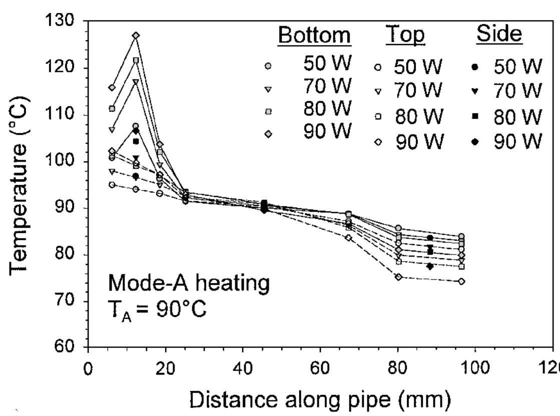

Fig. 7 Axial temperature profile for mode A heating, $T_A = 90°$C.

rate Q at a constant operating temperature of $T_{A,M} = 60°$C for the heat pipe with the mode B heating configuration for three fill conditions. The mean temperature in the evaporator for the case of the 0.69-ml fill is much higher than that of the other fill cases, indicating an underfill condition. The ΔT information derived from this graph clearly shows that a 10% overfill produced a desirable performance.

Temperature Profile and Heat Flux

The effect of heating modes on the temperature profile is illustrated in Figs. 7 and 8. Figure 7 shows the axial temperature profiles

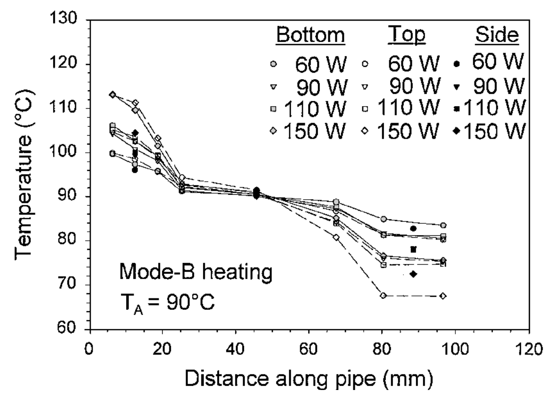

Fig. 8 Axial temperature profile for mode B heating, $T_A = 90°$C.

of the heat pipe for operations from 50 to 90 W with the mode A heating configuration. Note that the top and bottom surface temperatures vary quite dramatically in the evaporator region for mode A. With an increase of the heat rate, the average temperature difference ΔT_{EA} between the evaporator and adiabatic increases. The condenser ΔT_{AC} for the top surface is higher than that of the bottom surface because of the difficult-to-balance cooling rate regulation in the cooling blocks. Figure 8 shows a similar plot for mode B heating, wherein the evaporator ΔT is low even at 110 W, and, again, the condenser temperatures of the top and bottom faces reflect the uneven cooling rates.

Evaporator heat fluxes were calculated for both modes of heating based on the heat application area of $A_E = 0.777$ cm^2 for the mode A and $A_E = 4.66$ cm^2 for the mode B case, respectively. The highest value of heat flux (115.8 W/cm^2) at the heater surface occurs at $Q_{in} = 90$ W and ΔT_{EA} of 37°C in the case of mode A. Note that conduction heat spreading, if considered in the intermediate copper plate (0.8 mm thick), would lower the heat flux values. The highest value of heat flux (32 W/cm^2) occurs at $Q_{in} = 150$ W and ΔT_{EA} of 23°C in the case of mode B. The reader must exercise caution in interpreting the heat flux data. The ΔT_{EA} value is large in both cases, and the reason is apparently due to the nearness of the evaporator thermocouples to the heater.

Heat Transfer Coefficients

Figure 9 shows the evaporator and condenser internal (wall-to-wick) heat transfer coefficients computed from the measured temperature and heat input data using the following definitions:

Table 2 Comparison between present pipe and data from literature[5,6]

Details	Present MHP	Literature FMHP 3[5,6]
Wick	Folded screen	Machined groove
Groove aspect ratio	4.5	2.0
Overall length, mm	100	120
Fluid inventory, ml	1.1	0.84
Wick on narrow sides	Yes	No
T/C mounting location	Top, side, and bottom faces	Corner edge
Heating area of evaporator, mm	18.5 × 12.7 (on two faces)	10 × 0.7 (on two faces)

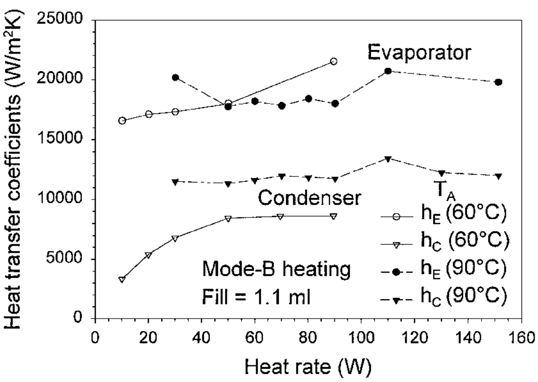

Fig. 9 Evaporator and condenser heat transfer coefficients for Q = 10–150 W at T_A = 60 and 90°C.

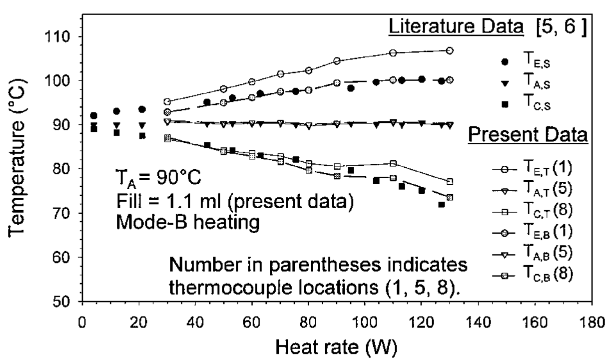

Fig. 10 Comparison of present results with literature data.[5,6]

$$q_E = Q_{\text{in}}/A_E, \qquad q_C = Q_{\text{in}}/A_C$$

$$h_E = q_E/(T_{E,i} - T_{A,o}), \qquad h_C = q_C/(T_{A,o} - T_{C,i})$$

where $T_{E,i} = T_{E,o} - q_E(t_w/k_w)$ and $T_{C,i} = q_C(t_w/k_w) + T_{C,o}$; $A_E = 6 \times 0.777 \times 10^{-4}$ m^2 and $A_C = 6.716 \times 10^{-4}$ m^2; and $T_{A,o}$, $T_{E,o}$, and $T_{C,o}$ are average measured wall temperatures. The plots correspond to mode B test conditions for 60 and 90°C operation for the case of a 1.1-ml fill. The evaporator heat transfer coefficients vary from 16,000 to 22,000 W/m^{2}K, whereas the condenser heat transfer coefficients vary from 4,000 to 14,000 W/m^{2}K over the 10–150 W input power range. This is in the expected range and matches very well with the typical range known for copper–water heat pipes.[8]

Comparison with Literature Data

The temperature profiles of the present MHP is compared with the literature data as shown in Fig. 10 (Ref. 5, 6). It is difficult to establish an exact basis for comparison between the present pipe and the literature data because of various minor differences in the heating arrangements, thermocouple locations, and physical sizes. Both heat pipes have many similarities, except for a few differences, as listed in Table 2.

The average evaporator and condenser temperatures of the heat pipes are compared for the case when the adiabatic section is maintained at an average temperature of 90°C. It is clear that the present MHP exhibits the same trend as that of the reference pipe. The top surface temperature plot of the present MHP is slightly higher than the case of the reference pipe.

Conclusions

A new folded screen-wick, copper-water MHP has been designed, fabricated, and tested successfully. It is demonstrated that an expensive and difficult process of fine groove machining currently employed by the industry in making the MHP wick could be easily replaced with the new affordable and simple wick design without compromising performance. The fabrication process of the new wick design can be easily adapted for mass producing the MHP that will suit the requirement for the electronics cooling application. Heat transfer characteristics of the present 108-mm-long, 12.7 × 6.35 mm rectangular MHP were evaluated at power inputs up to 150 W and compared with a similar design reported in the literature. The horizontal mode performances of both present MHP and the literature reference heat pipe at a representative operating temperature of 90°C are found to be the same in heat transport and evaporator/condenser ΔT aspects. In a high flux mode of heating, the present heat pipe has performed at an applied heat flux of 115 W/cm^2 (calculated at the heater surface with a ΔT_{EA} of 37°C). In a Comparison of results under similar operating conditions, the literature data show a higher value of heat flux (150 W/cm^2 at an unknown ΔT_{EA}). The heat transfer coefficients were calculated for the case of distributed heat flux mode of testing and found to vary from 16,000 to 22,000 W/m^{2}K for the evaporator and from 4,000 to 14,000 W/m^{2}K for the condenser. These data match with the results of typical copper–water heat pipes available in the literature. One important observation made during this research is that the permeability and wicking height data for this wick are unavailable. Hence, theoretical prediction of the capillary limit for this folded screen wick is not clearly established and future experimental measurement is recommended.

Acknowledgments

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