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The length and bending angle effects on the cooling performance of flat plate heat pipes

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ABSTRACT

The effects of length and bending angle on the cooling performance of flat plate heat pipes (FPHPs) were examined experimentally in this study. The length effect was explored with four lengths of 80, 150, 200, and 300 mm; each FPHP was filled by acetone in cold condition to its optimum filling ratio. The bending angle effect was examined with four angles of 0°, 30°, 60°, and 90°, using the length of 200 mm and the volumetric filling ratio of 31.4%. All FPHPs had the same cross sectional area of 50 mm (width) by 2.5 mm (thickness). Experimental results showed that by increasing the length from 80 to 150 mm, to 200 mm, and to 300 mm, the minimum thermal resistance, $R_{th(min)}$, increased by the factors of 2.4, 6.0, and 17.9, respectively from that of 0.103 K/W of the 80 mm FPHP. A rapid increase in $R_{th(min)}$ occurred around the length of 150 mm. For the FPHPs with lengths smaller than 150 mm, $R_{th(min)}$ could be smaller than 0.252 K/W. The maximum heat transport capability Q_{max} decreased quickly from 109.5 to 49.6 W (a factor of about 0.452) when the length was increased from 80 to 150 mm, and then slowly decreased to the minimum value of 35 W (a factor of about 0.318) for the length of 300 mm. In contrast, the results of bending angles showed that by increasing the bending angle, the thermal resistance decreased; $R_{th(min)}$ reduced by a factor of about 3.3 from 0.6207 K/W of 0° bending to 0.1885 K/W of 90° bending. The corresponding maximum effective thermal conductivity, $K_{eff(max)}$, increased from 1933.4 to 6365.6 W/m K and Q_{max} increased from 45 to 85 W. That is, a short FPHP preformed better than those of longer ones, and the thermal performance of FPHPs could be enhanced by proper bending.

1. Introduction

Heat pipes have several unique features, such as requiring no external powers for operation, operating temperatures can be controlled, light, and small form factors. They are widely used in consumer electronics, especially the mobile devices. Conventionally, the shape of a heat pipe is round for its cost-effective manufacturing process. On the other hand, high performance mobile devices always have a large heat flux enclosed in a thin package. Therefore, round heat pipes are typically flattened so that both requirements of small form factor and high heat flux can be met. However, material strength and wick structures impose limitations on the flattening technique. Thus, flat plate heat pipes (FPHPs) were developed to overcome these limitations.

The main capillary structures of FPHPs include mesh, groove, and porous types. The FPHPs of both mesh- and groove-type capillary structures are popular because of their reasonable performance and relative simplicity in manufacturing. For mesh types, Wang and Peterson [1] showed that the maximum heat transport capability of their micro heat pipes could be increased by enlarging the diameter of the Al mesh capillary. Launay et al. [2] numerically showed that the effective thermal conductivity could be increased by a factor of 1.3 over that of the empty heat pipe for their micro heat pipe with a Cu mesh capillary structure. Chien and Shih [3] examined the effect of mesh size, filling ratio, and tilt angle on the thermal resistance and showed that the condition of mesh size of #80, filling ratio of 25%, and 0° tilt angle would result in the smallest thermal resistance for the Cu–water FPHP. Lefevre et al. [4] experimentally showed that the thermal performance of the CuSn 325 square screen mesh and coarse screen mesh associated with rectangular grooves in methanol was not significantly different.

For groove-type capillary structures, the one dimensional model of Longtin et al. [5] indicated that the pressure drop in liquid was larger than that in vapor by about one order of magnitude. Gao et al. [6] showed that the effective thermal conductance of their flat miniature heat pipe (FMHP) could be 60–110 times that of a copper bar of the same size. With the groove structure on one side, Fan et al. [7] concluded that the gravity effect on the performance

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Nomenclature

A	the cross-sectional area for FPHP of thermal conductivity direction, m^2
D	distance between T_{h1} point and T_{c1} point, m
F_r	liquid filling ratio for a single channel, %
K_{eff}	effective thermal conductivity, W/m K
L	the length of FPHP, mm
P_i	input heating power, W
Q_{max}	maximum heat transport capability, W
R_{th}	thermal resistance, K/W
T_{a1}	temperature of the right adiabatic section, K
T_{a2}	temperature of the left adiabatic section, K
T_{c1}	the center temperature of the condenser section, K
T_{h1}	the center temperature of the evaporator section, K
V_i	liquid volume, cm^3
V_o	Channel space, cm^3
α	bending angle, °
ΔT	temperature different, K

of the FMHP was obvious in the axial direction. Rulliere et al. [8] showed that the experimental and numerical results of their methanol-filled two-phase heat spreader (TPHS) were consistent. And the optimal thermal resistance at the saturation temperature of 70 °C was 0.035 K/W [9]. Lefèvre et al. [10] indicated that the minimum thermal resistance was 3.7 times lower than that of the empty system for their grooved silicon-methanol flat heat pipe. Lips et al. [11] showed that a small vapor space thickness could reduce the thermal resistance of the FPHP; hence, the filling ratio and vapor space thickness should be optimized. Moreover, the liquid–vapor interface should be accurately assessed [12]. Lips et al. [13] and Lefevre et al. [14] established a database for the FPHPs developed by their group, including the vapor space thickness, filling ratio, and tilt angle, and suggested that new laws are required to better describe both the permeability and the equivalent thermal conductivity of the capillary structures involved. Chen and Chou [15] showed that the optimum liquid filling ratio was 25% for their Al extruded FPHP and the minimum resistance was 0.254 K/W.

For FPHPs with porous capillary structures, Wang et al. [16] revealed that dry out would occur at a lower heating power for a longer condensation section length and performance would be better by equalizing the lengths of the condensation and evaporation sections. Boukhanouf et al. [17] showed that the thermal spreading resistance of their Cu–water porous FPHP was about 40 times smaller than that of a solid copper block.

For transient features of FPHPs during starting-up and shutting-down operations, Wang and Vafai [18,19] showed that the wick in the evaporator section created the largest thermal resistance. Hu and Jia [20] indicated that both heating power and filling ratio were important factors in the start-up pulsating situation. By coupling the wall heat transfer to the fluid flow in both vapor and liquid phases in the transient process, Sonan et al. [21] demonstrated that the FPHP clearly worked as a thermal spreader, providing a more uniform temperature distribution than a solid cooling plate, similar to the case of steady state operations [17]. Harmand et al. [22] reported that, for transient cooling of electronic components in a starter–alternator, local hot spots could be avoided even in complex and confined geometries.

The above studies indicate that FPHPs have two key features: good heat spreading capability and thermal performance strongly depending on the geometry, capillary structure, and liquid filling ratios. The capillary structures reported for FPHPs in the literature mainly focused on mesh sheets or grooves on one side. In contrast, with practical applications in mind, this study examined the performance of FPHPs with groove-type capillary structures on both sides. It is known that the thermal resistance of groove type heat pipes is larger than that of porous types. For example, for round heat pipes of 150 mm in length, the thermal resistances normalized by that of porous type are 1.36 and 1.68 for groove and mesh types [23], respectively. And gravity has a larger effect on the groove type round heat pipes [23]. However, the groove-type FPHPs were selected in this study because they can be made readily by Al extrusion and are very suitable for mass production for commercial usages.

The main focus of this study was on the length and bending effects of FPHPs. The bending effect was explored because mobile devices/systems always have space limitations and bending may be necessary to meet the spatial requirement. In this regard, Wang [24] demonstrated that bending adversely affected the heat transfer performance for slightly bent round heat pipes. On the other hand, similar investigations for FPHPs were not reported in the literature. Therefore, this study intended to fill in this gap so that field applications of FPHPs can be better assessed.

The performance of FPHPs was characterized by the thermal resistance, effective thermal conductivity, and maximum heat transport capacity as follows.

2. FPHP fabrication

The FPHPs were made of aluminum by extrusion which included several steps. First, Al 6061 was extruded into a long channel with grooves on both top and bottom sides. Second, the long channel was cut into a desired length and sealed together at one end by a clamping pressure of 1241 kN/m^2. Third, chemically pure (99.87% pure) acetone was filled into the cut channel by a syringe in cold condition. Fourth, the other end of the channel was sealed by the same clamping pressure after vacuum to 80 torrs. Last, the fabricated FPHP was inserted into hot water of about 100 °C for a reliability test to complete the fabrication process. The volumetric liquid filling ratio was varied according to the length and bending angle of the FPHP. Details will be presented in Sections 4.1 and 4.2.

A typical fabricated FPHP is displayed in Fig. 1. Fig. 1(a) is the top view of the real FPHP. The related width and thickness were 50 and 2.5 mm, respectively. They were the same for all the FPHPs. A cross-cut at the A-A section illustrates the structure of 14 small channels across the A-A cut as shown in Fig. 1(b). The size of each small channel was 3 mm by 1.7 mm. A close-up view of the seventh small channel from the left further shows that each small channel had 16 small grooves. Each small groove was 0.4 mm (depth) by 0.2 mm (width) by 0.2 mm (gap) in size as tabulated in Table 1. For the exploration of length effect, four lengths of 80, 150, 200, and 300 mm were tested.

For the bending angle effect, the length of 200 mm was adopted. The bending angle denoted by α had four values of 0°, 30°, 60°, and 90° as shown schematically in Fig. 2(a) and pictorially in Fig. 2(b). For all the bent FPHPs, the bending location was at the mid-span of each FPHP. The radius of curvature of bending was 13 mm. In Fig. 2(b), the evaporator section is on the right side of the FPHP. The evaporation side of the heat pipe was set horizontally for all the bending angle tests. Thus, the location of the condensation