International Journal of Heat and Mass Transfer 77 (2014) 874–882

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ijhmt

Cooling performance of flat plate heat pipes with different liquid filling ratios

Jung-Shun Chen *, Jung-Hua Chou

Department of Engineering Science, National Cheng Kung University, Tainan 70101, Taiwan

ARTICLE INFO

Article history:
Received 11 September 2012
Received in revised form 21 May 2014
Accepted 10 June 2014

Keywords:
Flat plate heat pipes
Liquid filling ratio
Maximum heat transport capability
Thermal resistance
Effective thermal conductivity
Leakage

ABSTRACT

The effects of liquid filling ratios and leakage on the cooling performance of flat plate heat pipes (FPHPs) were examined experimentally in this study. With the size of 150 mm × 50 mm × 2.5 mm for all Al 6061 FPHPs filled with acetone (99.87% pure), the results showed that the one with the liquid filling ratio of 25% performed thermally the best. The corresponding maximum heat transport capability, minimum thermal resistance, and maximum effective thermal conductivity were about 47 W, 0.254 K/W, and 3150 W/m K, respectively. In contrast, improper vacuum and leakage would decrease the maximum thermal conductivities greatly to about 200–306 W/m K and 164 W/m K, respectively. The latter was similar to that of an aluminum block and performed the worst among all FPHPs. This situation should be avoided or carefully assessed as the maximum effective thermal conductivity decreased from the best by a factor of about 19.2.

1. Introduction

Due to thinness in geometry, slim liquid crystal display TVs (LCD-TVs) are becoming the main stream in TV markets because they occupy a small space and can be installed on a wall as wall decorations. For slim LCD-TVs to be achievable, a thin LCD module is essential. However, thickness reduction of present LCD modules always poses thermal issues, including thermal strain problems [1,2]. Therefore, heat pipe cooling modules are typically used in slim LCD-TVs to handle these issues for their smaller form factors. These heat pipe cooling modules have to be flat so that the thinness of LCD TVs could be accommodated. Conventionally, a flat heat pipe is made by flattening a round one. However, flattening has limitations because of material strength and wick structures. Hence, flat plate heat pipes (FPHPs) have been developed to resolve these limitations.

The capillary structures of mesh, groove, and porous types are commonly used in conventional heat pipes. Their thermal performance is different. For horizontal installations, the thermal resistance in decreasing order is mesh, groove, and porous type. Taking a heat pipe of 150 mm in length (the length used in this study) as a specific example, the thermal resistances normalized by that of porous type are 1.36 and 1.68 for groove and mesh types, respectively. Moreover, gravity has a larger effect on the groove type heat pipes [3].

As conventional heat pipes, the capillary structures FPHPs are also mainly mesh, groove, and porous. Various techniques have been developed to fabricate them, such as diffusion wire bonding [4,5] for meshes, diamond cut [6] for microgrooves, etching for special grooves [7,8]. These methods generally can make smaller heat pipes and are suitable for cooling of electronic components. Mesh and groove capillary structures are popular for their relative simplicity in manufacturing and reasonable performance. In this regard, Lefevre et al. [9] showed that the maximal heat flux obtainable was 2 W/cm^2 with a thermal resistance of 0.035 K/W for T_{sat} of 70 °C for their grooved Cu-methanol flat heat pipe. The results of 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 the FPHP. Hence the filling ratio and vapor space thickness should be optimized. Lips et al. [12] showed the importance of liquid–vapor interface and the need for its accurate assessment. Moreover, Lips et al. [13] showed that dry out was not due to boiling but was a classical capillary limit. Lefèvre et al. [14] investigated a series of six different FPHPs made of either silicon or copper with different capillary structures and sizes. The results showed that 80 W of heat transfer could be achieved and new laws were in need for better understanding of the mechanisms involved. Lefèvre et al. [15] showed that the thermal performance FPHPs of the CuSn 325 square screen mesh and coarse screen mesh associated with rectangular grooves in methanol was not significantly

* Corresponding author. Tel.: +886 6 2757575 63342; fax: +886 6 2766549.
E-mail address: shun.chen0727@gmail.com (J.-S. Chen).

http://dx.doi.org/10.1016/j.ijheatmasstransfer.2014.06.029
0017-9310/

Nomenclature

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

different. Chien and Shih [16] studied the effect of mesh size, filling volume ratio, and inclination angle on the thermal resistance of their Cu-water FPHP. The results showed that a larger mesh size would yield a lower thermal resistance and the inclination angle had a larger effect on the condenser than on the evaporator. Hu and Jia [17] investigated the start-up performance of pulsating FPHPs. The results indicated that both heating power and filling ratio were important factors for the start-up of the pulsating FPHP. Wang et al. [18] presented the length effect of the evaporation and condensation sections on the thermal performance of their sintered porous FPHP. The results showed that the FPHP would dry out at a lower heating power with an increased condensation section length and would perform better when the condensation section length approached the evaporation section length. For FPHP applications in electronic cooling, Boukhanouf et al. [19] showed that the thermal spreading resistance of their Cu-water porous FPHP was about 40 times smaller than that of a solid copper block. This was considerably larger than that obtained by Lefèvre et al. [10]. For start-up and shut-down transient characteristics, Wang and Vafai [20,21] investigated the transient characteristics of their FPHP during start-up and shut-down operations. The results showed that the wick in the evaporator section created the largest thermal resistance; whereas the wick in the condenser section was second. Sonan et al. [22] predicted the transient performance of a FPHP by coupling the wall heat transfer calculation to the fluid flow in both vapor and liquid phases and demonstrated that the FPHP clearly worked as a very good thermal spreader. Harmand et al. [23] studied the transient thermal performance of the heat pipe used to cool electronic components in a starter–alternator and demonstrated that local hot points could be avoided even in complex and confined geometries.

The above studies on FPHPs clearly indicates two key aspects of FPHPs. One is that they are good heat spreaders; the other is that their performance depends strongly on the geometry, capillary structure, and liquid filling ratios. The capillary structures reported for FPHPs in the literature mainly focused on sheets of meshes or grooves on one side. Hence, for practical considerations, this study examined the features of FPHPs with grooves on both sides made by Al extrusion because this technique could be readily adopted for mass production. In addition, the effect of leakage was also explored so that practical utilization of FPHPs could be properly addressed. All the Al FPHPs investigated had the same size and were filled with acetone. The main parameter investigated was the liquid filling ratio. The thermal performance was characterized by the thermal resistance, the effective thermal conductivity, and the maximum heat transport capability Q_{max}. Details are presented in the following sections.

2. FPHP fabrication

The FPHP was fabricated as follows. Firstly, structural forming by extrusion of aluminum 6061 was performed to make the FPHP with internal channels and associated capillary grooves. The extrusion process was tuned so that no cleaning was required for the FPHP. Secondly, the extruded FPHP was cut into a desired length and sealed at one end. Thirdly, acetone (chemical pure, 99.87% pure) was filled into the grooves to the required amount by a syringe with a volumetric scale. Fourthly, air in the FPHP was evacuated until the pressure was about 80 torrs. At that moment, the other end of the FPHP was sealed so that no extra outgassing was needed, thus completing the fabrication process. Lastly, the fabricated FPHP was set in a hot water tank with temperature of about 100 °C for reliability screening tests including leakage proof and quickness of heat transfer before characterizing the thermal performance.

A typical fabricated FPHP is shown in Fig. 1(a) with a marked A–A cross sectional cut. The A–A cross-sectional cut is depicted in Fig. 1(b) which illustrates that the FPHP had fourteen channels across its width. Each channel had sixteen capillary grooves. The aspect ratio of each capillary groove was 2. The related sizes are summarized in Table 1.

The liquid filling ratio (ϕ_i) is defined by Eq. (1):

$$\phi_i \equiv \frac{V_i}{V_o} \times 100\% \tag{1}$$

The channel volume V_o was fixed to 0.597 cm^3 and ϕ_i was varied from 5% to 50% with an interval of 5%. That is, V_i was changed from 0.03 to 0.3 cm^3 with an increment of 0.03 cm^3 for each different ϕ_i test. Each FPHP had a specific ϕ_i.

3. Experimental approach

A schematic diagram of the FPHP experimental set-up is shown in Fig. 2, (a) for the overall side view of the setup whereas (b) for the top view of the FPHP. The FPHP was placed horizontally on the heating and cooling modules. The FPHP was enclosed inside a temperature controlled space with a size of 1200 mm (length) by 500 mm (width) by 400 mm (height). The temperature inside the controlled space was automatically controlled at 40 ± 2°C by a cooling fan equipped with a temperature sensor. The contact thermal resistance between the FPHP and the heating module was reduced by both thermal grease and an insulation board under a pressure of 196 KN/m^2 as shown in Fig. 2(a); similarly for the cooling module arrangement. The heating module was made of a Cu block (2 mm thick) which was heated by a resistance heater (0.25 mm thick) to provide input heating powers to the FPHP. The input heating power P_i was varied from 5 to 60 W with an increment of 5 W by adjusting the needed current and voltage to the heating element. The cooling module was made of an Al 6061 block which was cooled by a water cooling system. The water cooling system delivered cooling water (25 ± 1 °C) at a fixed flow rate of 1200 ± 20 cm^3/min. The evaporation and condensation sections had the same size of 50 ± 1 mm (width) and 30 ± 1 mm (length). K-type thermocouples with an uncertainty of about ±1 °C were used to measure the FPHP wall