Journal of Enhanced Heat Transfer, 16(2) 161–170 (2009)
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Design and Experimental Research of a Flat-Plate Type CPL with a Porous Wick in the Condenser

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A new type of a copper-methanol capillary pumped loop (CPL) with a flat-plate type of evaporator and a flat-plate type of condenser with a porous element is designed, fabricated, and tested. The multi-layer stainless steel meshes and copper meshes are adopted as a capillary structure of the evaporator and condenser, respectively. The start-up, unstable operation characteristics, and heat transfer performance are investigated experimentally. The experimental results show that when heat load is lower than 450 W the system presents good start-up and continuous operation performance. The experimental results also indicate that the CPL has a good heat transfer characteristic with thermal resistance lower than 0.026°C/W.

Key words: capillary pump loop; evaporator; cobndenser; flat-plate type; experiment

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1. INTRODUCTION

A capillary pumped loop (CPL) is a passive two-phase thermal transport device that utilizes latent heat of liquid vaporization and surface tension forces to circulate working fluid and accomplish heat transfer between a heat source and a heat sink. The major components of CPL include an evaporator, a condenser, a reservoir and a vapor/liquid line. During normal operation, heat is applied to the evaporator, which makes the working fluid evaporate. The working vapor then carries the energy down the vapor line to the condenser where the vapor is condensed, the thermal energy in the vapor is released and removed out by the heat sink. Then the condensed fluid is transferred down the liquid line to the evaporator by the capillary pumping force caused by the vaporization/condensation interface.

CPL was first invented in the mid-1960s by F. J. Stenger at NASA/Lewis, but received special attention in the late 1970s [Bazzo & Riehl, 2003]. Since then, CPL has been extensively studied experimentally and theoretically [Krotiuk, 1997; Pettigrew et al., 2001]. The start-up difficulty and pressure oscillation during CPL operation are the main problems that plague the application of CPL. To solve a start-up problem, a time-consuming pre-heating procedure is usually adopted to "prime" the evaporator by applying power to the reservoir [Butler et al., 2002; LaClair & Mudawar, 2000]. The pre-heating process can cause largest liquid velocities that the CPL will ever encounter, and most of the liquid with large velocity must pass through nearly the entire CPL pipeline to arrive at the reservoir. When the pressure difference required to perform this purge exceeds the capillary limit of the wick, vapor bubbles from the vapor side will certainly be injected through the wick into the liquid core and cause depriming, which is responsible for more than 90% of CPL depriming [Hoang, 1997]. To enhance the possibility of start-up success, the CPL system generally utilizes starter heaters and a starter pump for clearing liquid in the vapor line, nevertheless, sometimes CPL still cannot start up successfully [Butler et al., 2002; LaClair & Mudawar, 2000].

Pressure oscillation is another issue of concern and have been intensively investigated due to its deleterious effects on CPL operation [Hoang & Ku, 1996; Ku, 1994; Muraoka et al., 1998]. Cottschlich and Richter [1991] developed a thermal power loop with a porous wick in a condenser, and their research indicates that this structure can reduce temperature oscillations. Muraoka et al. [1998] designed a capillary pumped loop with the disk-type evaporator and condenser, and the condenser contained a porous structure. They performed theoretical and experimental studies for the system, and their study showed that these structures can reduce, even eliminate, pressure oscillations, while greatly improving operational performance of CPL. Qu et al. [2006] conducted an analytical investigation to understand the internal fluid flow and heat transfer in the condenser with a porous wick of the CPL system. A mathematical model is proposed and used to describe the flow and heat transfer, especially interfacial transport phenomena. The liquid–vapor interface of the condensed liquid film was numerically determined. Some factors, such as vapor pressure, wall subcooling, wall heat load, and thermofluid properties, have significant influence on the heat transfer process and the shape of the liquid–vapor interface. Meanwhile, an experimental facility was set up and an experimental investigation was completed. Theoretical prediction and experimental results show quite good consistence. However, their CPL system has no reservoir, which causes the poor thermal controlling capability.

The capillary force, which is derived from the meniscus formed at the liquid–vapor surface of a porous wick in the evaporator, is the only force to drive CPL, so the evaporator is the most important part, and thorough understanding of the operational mechanisms of the CPL evaporator is crucial to design the CPL system. Cao and Faghri [1994] presented a conjugate analysis of a flat-plate type evaporator, in which the porous wick is liquid saturated, and liquid evaporation takes place only on the face of the porous wick. The research of Demidov and Yatsenko [1994] presented that evaporation could take place within the porous wick, and their mathematical model was derived on the basis of Darcy's law. Figus at al. [1999] studied the heat and mass transfer in a porous wick of the evaporator in both the continuum (Darcy) model and the pore network model. LaClair and Mudawar [2000] prosecute a thermal transient research in a fully flooded cylindrical capillary evaporator subject to uniform heat flux prior to the initiation of boiling, and the results were obtained by adopting the Green function method. Damronglerd and Zhang [2006] performed a numerical study of transient fluid flow and heat transfer in a porous medium with partial heating and evaporation on the upper surface. The dependence of the saturation temperature on pressure was accounted for by using the Clausius–Clapeyron equation. A new kind of boundary conditions was applied in order to reduce restrictions used in an analytical solution and to study changes in velocity and temperature distributions before reaching the steady-state evaporation. The results showed that the maximum velocity occurred near the end of the heated plate. Also, the temperature near the heated surface was higher than the saturation temperature representing superheated liquid. Huang et al. [2005] performed theoretical analysis and numerical study of phase-change heat transfer in the porous wick of the CPL evaporator which is similar to Muraoka's one. Liu et al. [2007] used a numerical method to simulate the flow and heat transfer in the porous wick of the CPL evaporator under different heat fluxes for different geometrical data. The calculation results indicate that the heat flux and geometrical data, such as the fin width and porous wick thickness, exert a great influence on flow and heat transfer of the evaporator. The study demonstrates that the field synergy principle can be applied to optimize the evaporator configuration and analyze heat transfer enhancement in the CPL evaporator.

The objective of this paper is conducting an experimental investigation of a new flat-plate type CPL, including start-up, steady and transient performance. The results obtained in the investigation will be helpful for understanding the operation performance for the flat-plate type CPL and their potentials to be used as thermal management devices. Based on the performance experiment, the CPL heat transfer characteristic is also analyzed.

2. THE NEW FLAT-PLATE TYPE CPL AND EXPERIMENTAL SETUP

Figure 1 schematically shows the conception model of a new flat-plate CPL. For the CPL under investigation, both the evaporator and condenser are designed as a flat-plate type with a porous wick, and a reservoir is used to control and adjust operation temperature of the system. Figure 2 schematically shows the detailed structure of the evaporator. There is a cross channel for liquid supply in the evaporator inlet for reducing

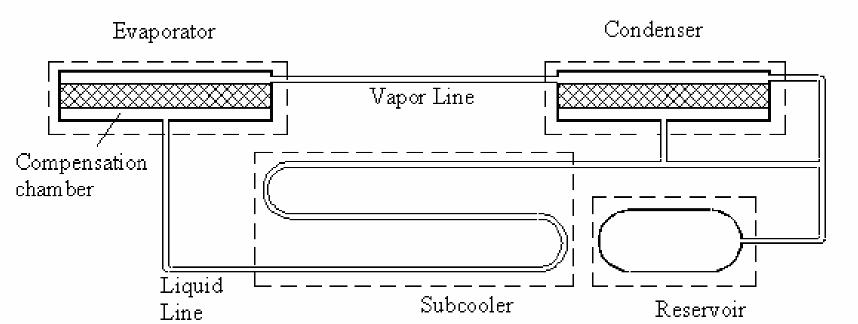


FIGURE 1. Schematic diagram of the flat-plate type CPL under investigation.

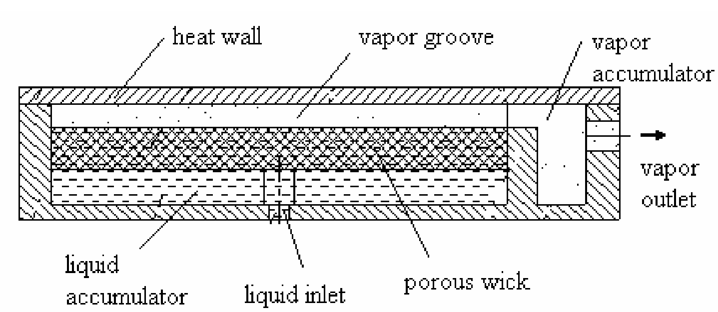


FIGURE 2. Schematic of the flat-plate type evaporator.

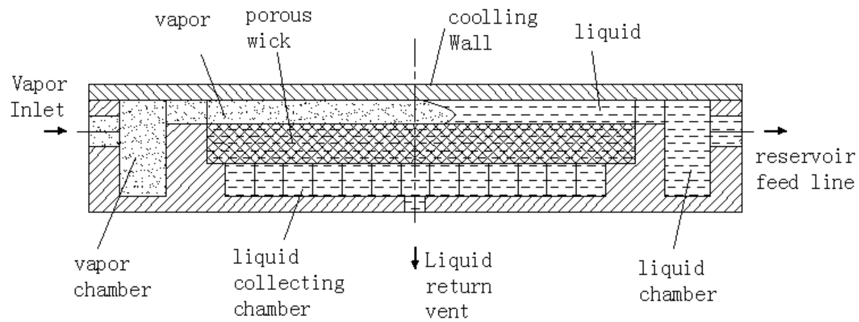


FIGURE 3. Schematic of the flat-plate type condenser.

TABLE 1. The Geometric Characteristics of the New Flat Plate Type of CPL

Evaporator	Length, mm		240	Condenser	Length, mm		260
	Width, mm		140		Width, mm		140
	Vapor groove	Number	25		Vapor groove	Number	25
		Width, mm	3			Width, mm	3
		Height, mm	4			Height, mm	4
Evaporator porous wick	Permeability, m^2		6.524×10^{-12}	Condenser porous wick	Permeability, m^2		2.024×10^{-11}
	Porosity, %		59.41		Porosity, %		52.85
Vapor line	Outer diameter, mm		10	Reservoir	Diameter, mm		60
	Inner diameter, mm		8				120
Liquid line	Outer diameter, mm		6				
	Inner diameter, mm		4.5				

the probability of a dry-out point caused by strong evaporation in the case of high heat flux, and therefore to improve the stability of the system. Also, a plane condenser with the porous wick is designed, which has three ports for vapor-pipe inlet, liquid-pipe outlet, and adjusting pipe connected to the reservoir and the detailed structure of the condenser is schematically shown in Figure 3. The above design is significantly different from the traditional CPL. Since such a design can reduce the flow resistance, fluid can be easily pressed into the reservoir, it is expected that the start-up performance and the adjustment capability of the present CPL system can be improved efficiently. The geometric characteristics of CPL are presented in Table 1.

The experimental setup is schematically shown in Figure 4. Thirty five T-type thermocouples are used to monitor the temperature changes throughout CPL, of which T1 to T9 are used to monitor the evaporator temperature, T13 to T20 are used to monitor the condenser temperature, T10 is fixed into the vapor line interior, so it can be seen as vaporizing temperature T_{evp}. T25 lies in the condenser outlet, T35 is used to monitor the subcooled temperature, T36 is the reservoir temperature. Before the test all the thermocouples are calibrated with a measurement error less than $\pm 0.04^{o}$C. Three pressure transducers are used to measure the pressures of the evaporator outlet, condenser outlet, and reservoir, respectively. High-purity methanol (99.95%) was used as the working fluid in this research, and the charging inventory of the working fluid is 1.02 kg (about 60%). In order to keep the system operation at a low temperature, the system must work with a certain vacuum pressure. A plane heater is used to simulate the heat load of the evaporator and two electrical heaters are used to control the temperature in the reservoir. A three-horse power 35% glycol-water refrigeration system is chosen as a heat sink of the condenser. In order to reduce the heat loss from the loop to the ambient the whole loop is thermally

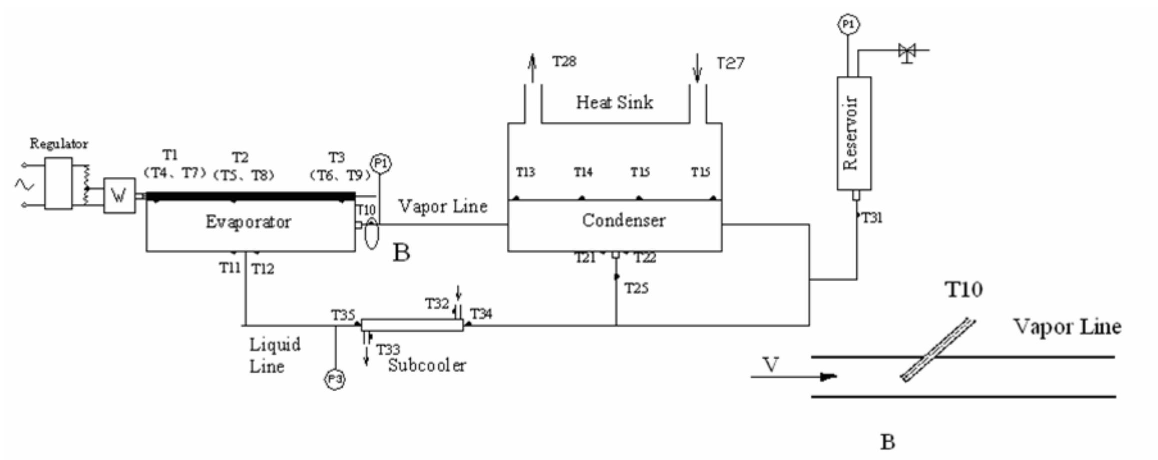


FIGURE 4. Schematic diagram of the CPL experimental setup.

insulated with thermal insulation material, and with heat balance calculation between the heat load and the heat sink the heat loss is less than 5%.

All instruments and thermocouples are connected to a Keithley 2700 data acquisition system, where a computer monitors the temperature and pressure changes. Based on this experimental system, many operation performances such as reliability, temperature controlling, and start-up can be tested and verified. The experimental test is performed to investigate the thermal performance of CPL. All test is carried out at the room temperature about 23^{o}C. The experimental tests were divided into two parts. The first one is the start-up test, where CPL starts from room temperature conditions and reaches the steady state. In such a test, the time to reach the steady state is looked forward to be short, therefore the time to reach the steady state was carefully observed. The second one is the heat load profile test, where CPL is exposed to the power change applied to the capillary evaporator. In such a test, the CPL capability to follow the changes of the heat load was examined.

3. EXPERIMENTAL RESULTS AND DISCUSSION

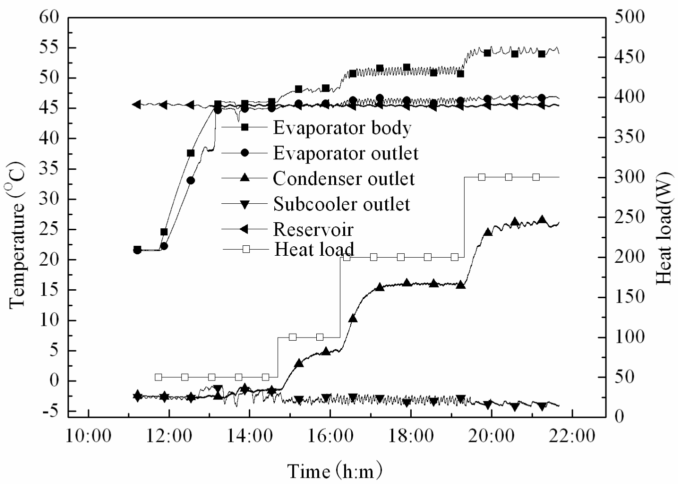
3.1 Performance Test for CPL

The performance tests for CPL include start-up tests and heat profile tests. The CPL start-up tests were the most critical for evaluating the CPL reliability. For most CPL systems there is a strong tendency of depriming with lower or higher heat load. Figure 5 shows the transient response of the temperatures for the new type of CPL during the start-up and unstable operation process with different heat loads. As shown in Figures 5a–5f, CPL presents a good start-up performance under different heat loads.

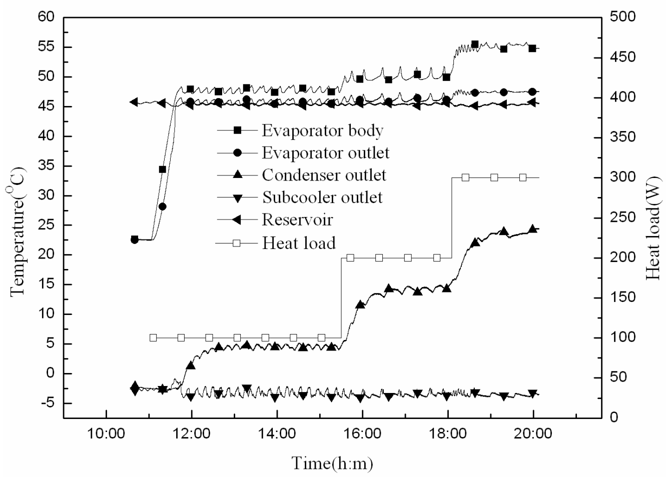
It can be seen from Figure 5 that, with the heat load applied to the evaporator, the temperatures of the evaporator body and evaporator outlet increase gradually, but temperatures of the remaining parts of CPL show little change. With the time going on, when the temperature of the evaporator rises to the saturated temperature corresponding with the system pressure, the liquid begins to vaporize. The resultant vapor pushes the liquid of the vapor line, so the evaporator outlet temperature rises quickly. Hereafter the temperature goes to stable, and the resultant vapor begins to enter the condenser, as a result, the condenser outlet temperature rises. At this time the CPL starts up successfully. From Figure 5 it can also be observed that the larger the start-up heat load, the quicker the start-up process.

During the start-up process the largest pressure difference in the system would occur. In order to start up successfully for CPL, the traditional CPL must be fully flooded by heating the reservoir. The resultant vapor must displace the liquid from the vapor line and move it into the reservoir, which may lead to pressure surge. For CPL successful start-up the resultant pressure surge is not too large for avoiding evaporator depriming.

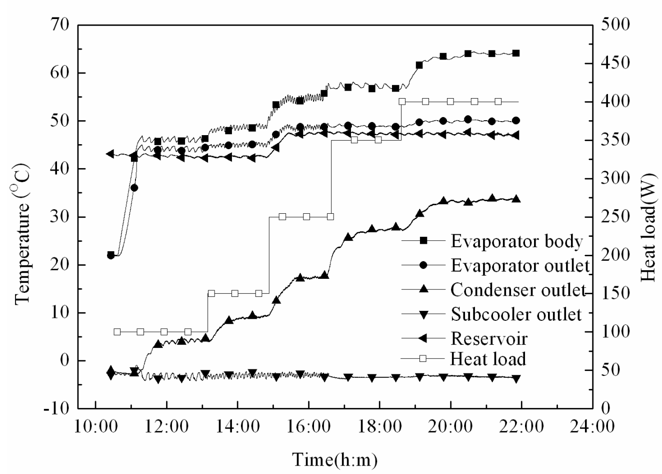
In a traditional CPL there is only one fluid passage between the reservoir and the loop, so the flow resistance during start-up is great and these can result in



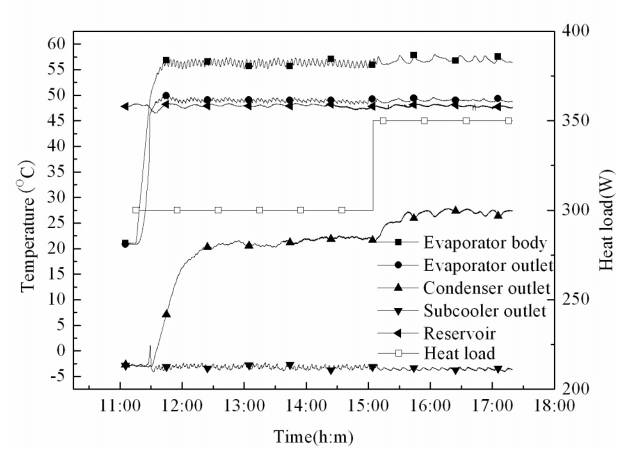
a) Q: 50-100-200-300 W



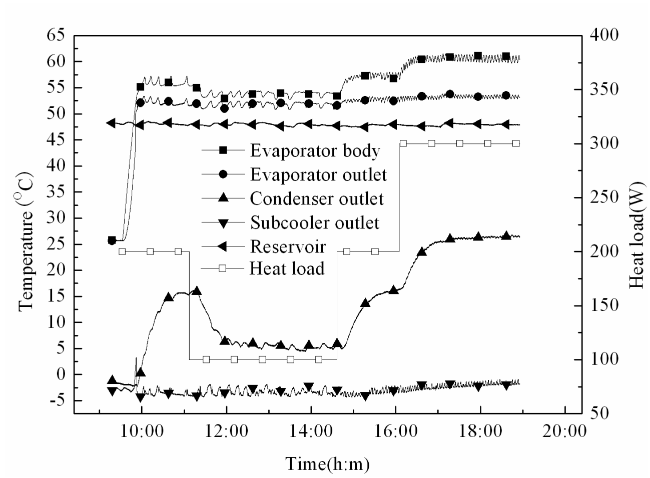
b) Q: 100-200-300 W



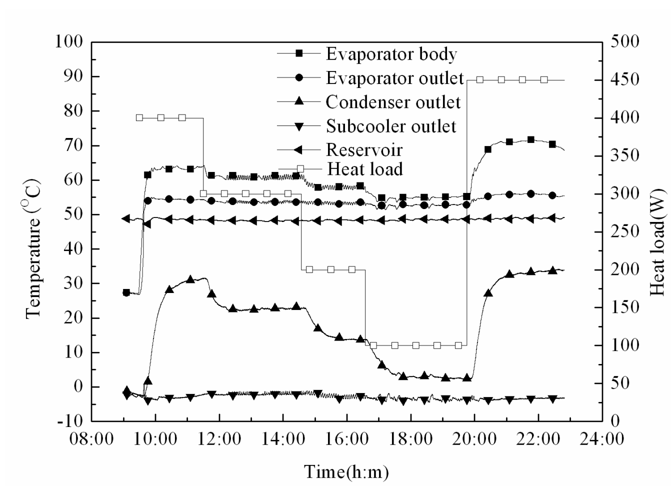
c) Q: 100-150-250-350-400 W



d) Q: 100-150-250-350-400 W



e) Q: 200- 100-200-3400 W



f) Q: 400-300-200-450 W

FIGURE 5. CPL performance tests at different heat loads.

start-up difficulty, even failure. Aiming to improve the start-up performance, we change both the components construction and system configuration in the present CPL. Besides the reservoir, a liquid compensation chamber is designed near to the capillary wick in the evaporator to store fluid. Such a design makes the wick always be filled with liquid. Also, a flat-plate type of the condenser with a porous wick is designed and adopted in our CPL. A vapor collection chamber lies at the inlet of the condenser. There are 17 parallel channels in the flat-plate condenser, when vapor flows into the condenser, it can flow in all channels nearly equally by means of distribution of the vapor collection chamber. Besides one vapor inlet, the flat-plate capillary condenser has two outlet ports, one is from the condenser to the liquid line and another is from the condenser to the reservoir, by which the reservoir can make fluid exchange within the loop. And this is the major difference between our flat-plate type CPL and a traditional CPL. When the operation conditions change, for example, as shown in Figure 5a, the heat load increases, which can produce more vapor, in order to condense more vapor a larger condensation surface is needed. Therefore, some liquid is expelled out of the condenser through the reservoir feed line, after the expelled fluid flows out of the condenser, one part flows into the reservoir and another flows to the liquid line. Such a design cannot only reduce the flow resistance between the loop and the reservoir, but ensures sufficient liquid supply to the evaporator. When the heat load decreases, the fluid flow phenomena are similar to those in the case of heat load increasing, but the flow directions are opposite. The porous wick in the condenser generates a stable physical interface between the liquid and vapor phases inside the CPL, which reduces or even eliminates the difficulties associated with the start-up procedure and pressure oscillations.

From Figure 5 it can be observed that if the reservoir temperature is controlled precisely, once the CPL starts up successfully, with the increasing heat load, the evaporator body temperature increases slightly and with the decreasing heat load, the evaporator body temperature also decreases slightly, but the vapor temperature (evaporator outlet temperature) remains stable. But if the reservoir temperature is changed, the vapor temperature is also changed, as shown in Figure 6c. The reason for this is that when the system works normally, the system and the reservoir are in the equilibrium state both hydrodynamically and thermodynamically. Because the fluid of the reservoir is in the vapor/liquid equilibrium state, if the reservoir temperature maintains stable, the reservoir pressure can also maintain stable, so the system pressure can remain stable so as to the evaporation temperature. If the reservoir temperature is changed, the system temperature can also be changed.

From Figure 5 it can also be observed that with low heat load the temperature difference between the evaporator body and the evaporator outlet is small, but with the larger heat loads the temperature difference is obviously larger. The reason for this can be explained as follows. When CPL operates at a low heat load, the resulted vapor can flow out the evaporator quickly, and with a larger heat load, more fluid is vaporized, but the resulted vapor cannot flow out the evaporator

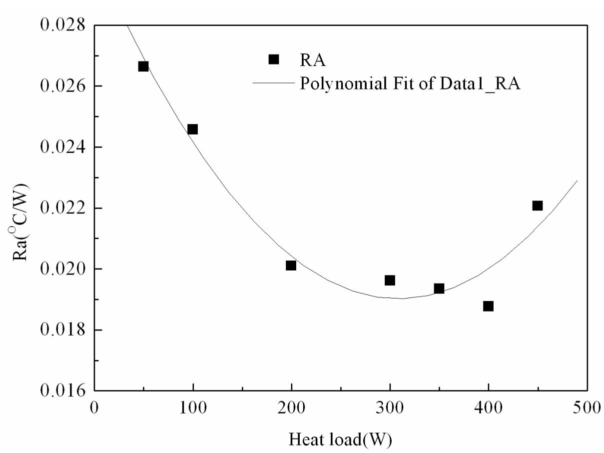


FIGURE 6. Heat resistance variety of the flat-plate type CPL with heat load.

quickly, which forms a vapor film in the evaporator channels and leads to a larger thermal resistance between the evaporator body and the porous wick surface. Therefore, the temperature difference between the evaporator body and vapor (evaporator outlet) is large with a large heat load. We can deduce that if the vapor flow channel is optimized to reduce the flow resistance, the temperature difference between the evaporator body and vapor can also be reduced, and the CPL running performance can be improved, as pointed by Liu et al. [2007].

From the results observed in the tests, the present CPL design shows a good thermal behavior for both start-up and heat load profiles with heat load lower than 450 W (heat flux is about 11,500 W/m^2). The temperature of the capillary evaporator body is very stable without any overshooting at any time during its operation. The proposed system is proved to be able to operate for several hours continuously with very little variation under its operation conditions.

3.2 Heat Transfer Performance

Since CPL will be utilized in cooling of electronic devices, the effective heat transfer performance is important. The thermal resistance of the evaporator can be defined as

$$\mathrm{Ra} = \frac{\Delta T}{Q}$$

where Ra is the thermal resistance of the evaporator, Q is the heat load applied on the evaporator, and $\Delta T = T_{\mathrm{e}} - T_{\mathrm{vap}}$, T_{e} is the evaporator body temperature which can be obtained by averaging the temperature of T1 to T9, and T_{vap} is the temperature of the evaporator outlet.

The result presented in Figure 6 shows the variation of thermal resistance with the heat load applied on the evaporator. From the figure it can be observed that with an increase of the imposed heat load, the thermal resistance decreases to a minimum value and then increases afterwards. This phenomenon can be explained according to the visualized experiment of Liao and Zhao [1999]. At a certain value of the imposed heat load (for example Q = 200 W), nucleate boiling took place at the porous wick surface close to the heated surface of the evaporator. The capillary force, which is derived from the meniscus formed at the liquid–vapor interface of the porous wick in the evaporator, drove the liquid fluid to flow, as a result, a number of small dispersed bubbles were continuously generated, so the heat transfer coefficient was small and the thermal resistance was large. As the imposed heat load was increased further, both the number of the acting bubbles and capillary force were increased. And resulted bubbles can flow into the vapor line timely; therefore, the heat transfer coefficient increased, as a consequence, the thermal resistance decreased. When the imposed heat load was further increased, the bubbles continued to increase, if they could not flow out the evaporator timely, they would combine to a dry vapor film at the interface between the heated surface and the porous wick. Under this situation, heat was transferred to the evaporating front within the porous structure via the vapor film. The lower thermal conductivity of the vapor film leads to a smaller heat transfer rate. This is why the thermal resistance begins to increase beyond a certain value of heat load. From the figure of evaporator resistance a conclusion can be drawn that although CPL presents good work capability during normal operation, its heat transfer capability is limited to some extent, for example, less than 450 W. When CPL is used in high heat load situation, the heat transfer capability limitation should be taken into account for insuring the CPL normal operation.

CONCLUSIONS

In this study, the start-up characteristics and transient performance of a new CPL with a flat-plate type of the evaporator and condenser was tested. The experimental results shows that the CPL has good start-up, transient and steady-state performance characteristics with the heat load less than 450 W. As the heat load increases, the temperature difference between the evaporator body and vapor also increases. Heat transfer capability is also investigated in this paper. The study presents that the heat load has an effect on the heat transfer capability, and the evaporator thermal resistance is less than 0.026$^{\mathrm{o}}$C/W when heat load is less than 450 W. The optimal heat load was found to be 320 W for the given experimental condition, and corresponding thermal resistance is 0.019$^{\mathrm{o}}$C/W.

ACKNOWLEDGMENT

This work is supported by the National Key Basic Research Development Program of China (No. 2007CB206900).

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