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A Study on Selecting Criteria of Working Fluid in Loop Heat Pipes with a Circular Plate Type Evaporator

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ABSTRACT: In these days, the necessity of thermal management has become significant due to the increased heat dissipation and higher heat density of electronic equipment and/or parts released. A loop heat pipe(LHP) has been payed closer attention to the potential candidate of an electronic cooling. As of the LHP with a circular plate type evaporator developed, this study focused on its operating characteristics on the steady state in accordance with charging different working fluid. The relationship between working fluid and operating characteristics is discussed.

Key words: Loop Heat Pipe (LHP), Working fluid, and Pentane

Nomenclature

h_{fg} : latent heat of vaporization, [J/kg]
Q : heat load, [W]
R : thermal resistance, [℃/W]
T : temperature, [℃, K]
P_c : maximum capillary pressure generated by the wick on liquid, [Pa]

Greek Symbols

σ : surface tension, [N/m]
ρ : density, [kg/m^3]
μ : viscosity, [Pa s]
ϵ : porosity, [%]

Subscripts

a : ambient
cc : compensation chamber
c : condenser
e : evaporator
eff : effective
g : gravity
j : junction
l : liquid
t : total
v : vapor
w : wick

1. Introduction

The rapid growth of electronic chips has resulted in the release of the high performance electronic products with increasing their heat dissipation simultaneously. For instance, the waste heat released by the central processing unit (CPU) for a desktop, laptop and server PC is 80W to 130W. Besides, the heat density of the parts has become as small as 1 to 4cm^2. So that, it is required to maintain the chip surface temperature below 100 ^{o}C. In order to

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keep out of the high temperature of chip sets, some technologies such as fin−plate heat sink, two−phase heat transfer like heat pipes and vapor chambers, or liquid cooling and thermoelectric coolers had been developed to solve these problems. However, they are not completely satisfied with recent demands including heat transfer capability, reliability or cost issues. The development of loop heat pipes (LHPs) has come up as a potential candidate to meet these challenging needs [1].

LHPs are the two-phase heat transport devices utilizing the evaporation and condensation of working fluid to transfer heat, and the capillary forces developed in fine porous wicks to circulate the fluid. In comparison with conventional heat pipes, the LHPs have not only all the advantages of them but also additionally furnish reliable operation over long distance at any orientation in the gravity field. They have proven themselves as high coefficient heat transport devices for the spacecraft thermal systems. The principle structure of the LHP consists of the evaporator section including compensation chamber, vapor removal channel and wick, a condenser section, vapor line, and liquid line. The working fluid vaporizes when heat load is applied in evaporator section due to increasing temperature in saturation pressure. This vapor flows along the vapor line from evaporator to condenser to release heat in over there due to decreasing temperature. The wick circulates working fluid from condenser to evaporator side. Therefore, among several LHP design parameters, the thermo−physical properties of working fluid also play an important role in performance of the LHP. The working fluid depend mainly upon properties such as surface tension, vapor pressure, and latent heat as well as maybe toxic, flammable factors because of their harmful for human and environment. Water, acetone, ethanol, methanol, ammonia, and etc. have been examined as the working fluids.

The main purpose of this study is to obtain operating characteristics on the steady state in accordance with charging different working fluid. Water and pentane were chosen as working fluid carried out on the circular shape plate evaporator LHP.

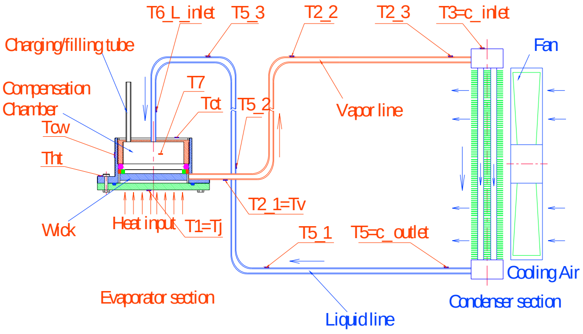

Fig. 1 Schematic of LHP system

2. Experimental description and method

The structure of the LHP in this experiment is shown in Fig. 1 including evaporator section, finned condenser, vapor line, and liquid line. For the evaporator structure, the circular shape was fabricated from stainless steel with diameter 43mm. The circular copper plate, 3.25mm of thickness, was used to connect with the heat source. The sintered bronze powder − flat plate − groove wick, 42mm of diameter, 5mm of thickness, 1−3 m of pore diameter, and 65 % of porosity was made as capillary pump. The vapor and liquid lines with 1/12" of out diameter and 1/8" in diameter were used to connect between evaporator and condenser parts.

In the condenser section, for rejection of the latent heat transported from evaporator section, a fin−and−tube−type condenser was installed in the end of vapor line. It consists of 8 tubes, 3/16"of diameter with 100mm in length and 20mm in thickness, 4 tubes for each row. It was provided with external fins measuring 85x20mm^2 with the thickness of 0.2mm and fin pitch of 0.5mm. This condenser was cooled by a fan giving the velocity 3.624 m/s.

The LHP was designed on the basis of the maximum capillary pressure generated by the porous structure on the working fluid, which depends on the surface tension coefficient of the liquid working fluid and mean effective pore radius of the wick structure.

$$\Delta P_{\max} = \frac{2\sigma_l}{r_{eff}} \tag{1}$$

This value must be greater than or equal to the total pressure loss occurred by flowing the working fluid inside the loop. This is necessary condition for the startup and continuous operation of the LHP and it can be formulated in mathematical form as follows:

$$\Delta P_{\max} = \Delta P_w + \Delta P_l + \Delta P_v + \Delta P_{gr} + \Delta P_g \tag{2}$$

In addition, the operating characteristics of LHP are involved on the diagram of its working cycle that is presented in Fig. 2 The point 1 on the saturation line determines the vapor parameters P_1, T_1above the evaporating surface of the wick menisci in the evaporation zone, and the section 1-2 corresponds to the vapor motion in the system of vapor−removal channels into the vapor line. Since the vapor motion here proceeds along the hot wall of the evaporator, a decrease in its pressure is accompanied by a slight superheat. The vapor motion in the vapor line (section 2-3) ideally may be considered close to isothermal.

Pressure losses in the LHP condenser are usually negligible. The working fluid here is condensed and in section 4-5 supercooled. Further its motion along the liquid line in the diagram is shown as isothermal, though in many actual cases it may be accompanied by considerable heating or cooling owing to the heat exchange with the surrounding medium. As a result, a liquid with parameters T_6, P_6 enters the compensation chamber.

Simultaneously here comes part of the heat flow supplied to the evaporator, at the expense of which the working fluid is heated to the temperature T_7. The section 7-8 corresponds to the liquid filtration through the wick into the evaporation zone. On this way the liquid may prove to be superheated, but its boiling−up does not take place owing to the short duration of its being in such a state. The point 8 determines the state of the working fluid in the vicinity of the evaporating menisci, and the pressure drop $P_{1\text{-}8}$ corresponds to the value of total pressure losses in all the sections of the working−fluid circulation.

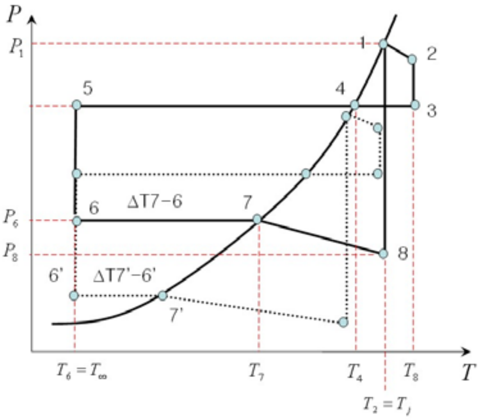

Fig. 2 The P − T diagram of LHP

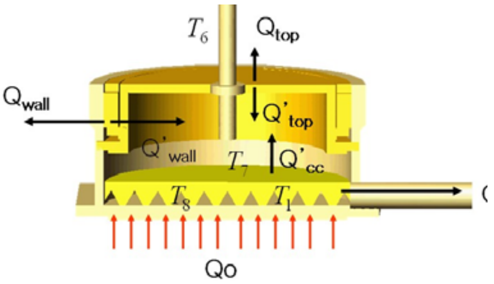

Fig. 3 The schematic of thermal model evaporator section

To compare the characteristics of two kinds of working fluid, some parameters focused on evaporator section have been calculated. In this schematic of the cross section of evaporator and thermal schematic modeled as shown in Fig.3, the thermal parameters ar able to be formulated in mathematical by below equations.

$$Q_{incc} = Q'_{wall} + Q'_{top} + Q'_{cc} = \dot{m} c_p (T_7 - T_6) \tag{3}$$

$$Q_{hfg} = \dot{m} h_{fg} \tag{4}$$

Substituting equation (4) into equation (3) gives an expression as below.

$$\Delta T_{7-6} \frac{Q_{incc}}{Q_{hfg}} = \frac{h_{fg}}{c_p} \tag{5}$$

The thermal characteristic of the LHP is calculated on the basis of the junction temperature, saturated vapor temperature, and maximum heat capacity. The total thermal resistance is calculated on the basis of relations given below.

$$R_t = \frac{T_j - T_a}{Q_o} \qquad (6)$$

In equation (5), there is difference between compensation chamber temperature and liquid inlet temperature. That is, a function of h_{fg}/c_{pl} ratio of equation (5) is the thermo−physical properties of working fluid. With the same evaporator structure and heat input that mean the Q_{incc}/Q_{hfg} ratio is constant, ΔT_{7-6} will be depending on h_{fg}/c_{pl}. When ΔT_{7-6} is decreased because of decreasing of h_{fg}/c_{pl} ratio the compensation chamber temperature will be move up on the saturation line like Fig. 4 and thus, the total thermal resistance of the system will be increased, conversely, the total thermal resistance will be decreased. Through the table of thermo−physical properties of working fluid, the h_{fg}/c_{pl} ratio can be calculated.[4] The graph depicts the h_{fg}/c_{pl} ratio relating to vapor temperature is shown in Fig. 4 In this graph, the h_{fg}/c_{pl} ratio of water is greater than pentane's with the same vapor temperature respectively.

Basically, the performance of the LHP depends mainly upon the properties of working fluid such as, surface tension, latent heat, and constant pressure specific heat. The surface tension and latent heat among these properties are affected to the high capillary pressure and the high heat transfer capability in accordance with what working fluid applies. Thus, it is easy not only circulate working fluid from condenser to evaporator but also transport high amount of heat dissipation from heat source. However, as mentioned above, the h_{fg}/c_{pl} ratio must also be evaluated. Therefore, in this study, the properties of water and pentane were taken into consideration since both were used in the same LHP as working fluid.

In this experiment, to measure temperature, thermocouples, with an accuracy of 0.1 ^{o}C, were installed at different locations along the external surface. Liquid and vapor pressure were probed by a pressure transmitter.

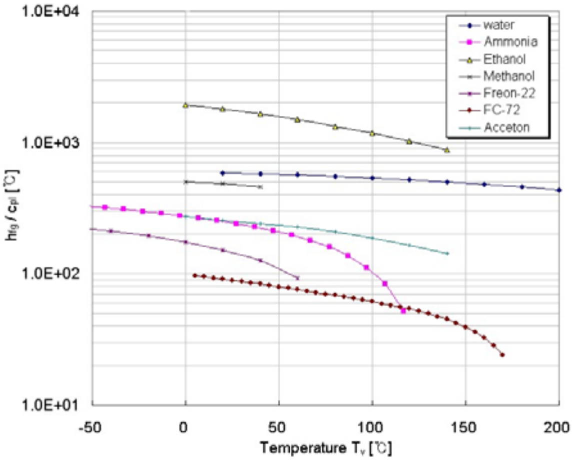

Fig. 4 The ratio h_{fg}/c_{pl} vs vapor temperature

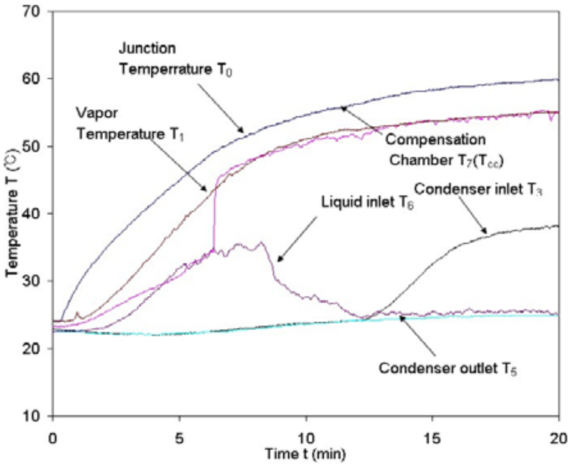

Fig. 5 Startup of LHP at 15W heat load with water

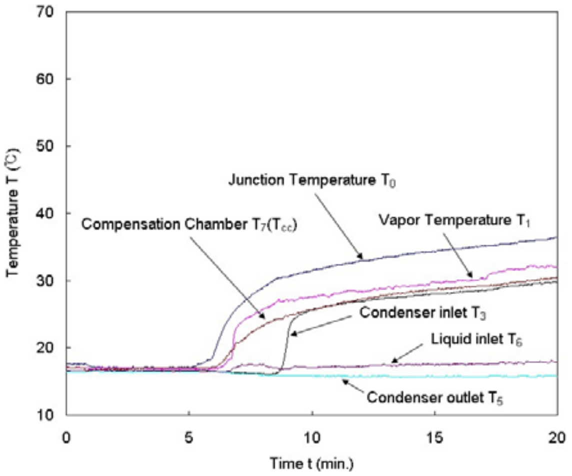

Fig. 6 Startup of LHP at 15W heat load with pentane

The pointed locations for temperature and pressure measurement as shown above in Fig. 1 were monitored via data acquisition software. The bottom surface of the evaporator was supplied to the heat load Q_0 between 15W and 60W in increments of 15W.

Through one inserted thermocouple in the compensation chamber(CC), the temperature was measured, and then the internal CC temperature was assumed as T_7.

3. Results and discussion

The startup of the LHP is a very complicated phenomenon affected by a large number of factors that include pre-startup states in the evaporator, charged liquid inventory, value of applied heat load, heat leaks across the wick, and orientation of the loop. The successful startup is when the liquid-vapor interface has already been inside the evaporator zone. However, if the evaporator is flooded with liquid and an interface is inside the compensation chamber, the startup will be the most difficult and needs a high heat flow rate to initiate fluid motion. In this case, from applying different working fluid, the startup of the LHP with the same heat load, 15W were shown in the Fig. 5 and 6. In both cases, the evaporator temperature was similar to or higher than compensation chamber temperature. The given results showed the startup was successful however the junction temperature in case for water was higher than for pentane. This phenomenon is able to explain that the boiling temperature of pentane is smaller than water's. The amount of heat was transferred earlier in the case of pentane from heat source. The Figure 7 pointed out the average temperature of two cases corresponding to power input. The results also illustrated that the average temperature in the water case was higher than in the pentane case for the same level of heat input in the next steps. In the maximum heat input, 60W, the average vapor temperature of water was 81.7 °C while the average vapor temperature of pentane was 61.4°C. The average compensation chamber temperature T_7 of water was 78.1 °C and 54.3 °C in pentane case. And then, the average liquid inlet temperature T_6 of water was 27.9 °C comparing with 24.1 °C of pentane. These data proved that the difference between T_7 and T_6 of LHP with water as working fluid was greater than with pentane.

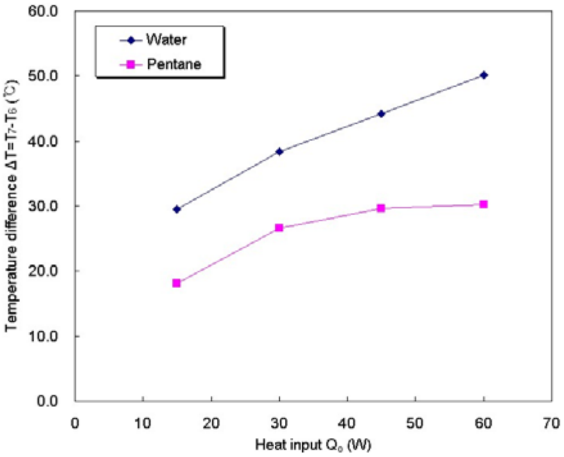

Fig. 7 ΔT_{7-6} vs. heat input

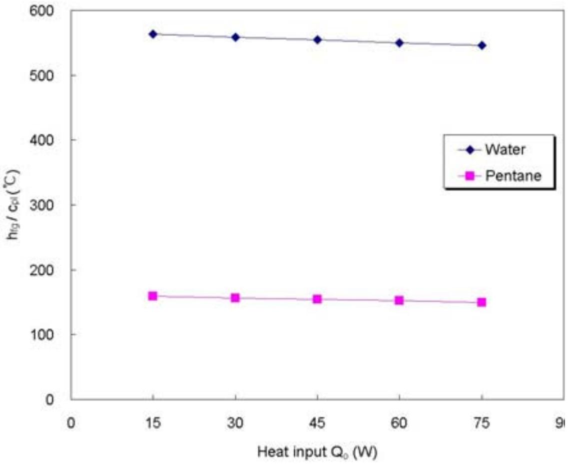

Fig. 8 the values of h_{fg}/c_{pl} vs. heat input

From equation (5), because of the same initial conditions, the h_{fg}/c_{pl} ratio of pentane was smaller than the h_{fg}/c_{pl} ratio of water and the experimental results confirmed this statement. In Fig. 7 and 8, the difference between T_7 and T_6 and the values of h_{fg}/c_{pl} were plotted versus applied power. The values of ΔT_{7-6} were from 18.1 °C to 30.2 °C for pentane and from 29.5 °C to 50.2 °C for water with the range of values of heat input from 15W to 60W and the same range of heat input, the values of h_{fg}/c_{pl} ratio were from 162.1 (1/°C) to 151.6 (1/°C) for pentane and from 564 (1/°C) to 548 (1/°C) for water.

The graph in Fig. 9 illustrated the total thermal resistance R_t in two cases with the same range of heat input. With water, the value range of R_t was from 2.42 °C/W to 1.03 °C/W and with pentane, was from 1.98 °C/W to 1.01 °C/W. The ambient temperature in for two cases was between 20 °C and 25.5 °C. So, the total thermal resistance decreased when the heat input increased.

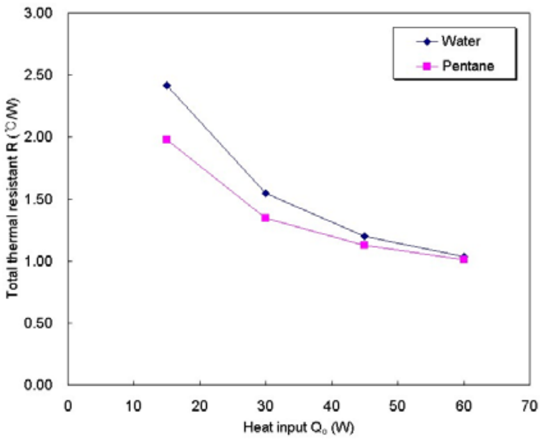

Fig 9. The total thermal resistance vs. heat input

4. Conclusion

The LHP with a circular plate type evaporator operated with pentane and water, respectively. In steady state, the vapor temperature could reach to 81.7 °C for water and 61.4 °C for pentane at 60W heat input. With forced air cooling of the condenser, the device was able to transfer a maximum heat flux as high as 10.3 W/cm^2 for water and 6.2 W/cm^2 for pentane over a distance of up to 500mm from heat source to condenser. For heat load in the range of 15W to 60W at 22.7°C of average ambient temperature. The total thermal resistance of the LHP was 2.42°C/W to 1.03 °C/W for water and 1.98°C/W to 1.01 °C/W for pentane. Compared with pentane, water was definitely superior thermal characteristics. However, the total thermal resistance of the LHP with pentane was lower than one with water. Consequently, in order to design the LHP to apply the electronic cooling in respect of lower heat input parts such as CPU or the other chips together with even lower total thermal resistance, pentane may be a good choice in spite of low capillary pressure and the small amount of heat transfer rate.

Acknowledgement

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