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Capillary Limit of a Multiple-Evaporator and Multiple-Condenser Miniature Loop Heat Pipe

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An experimental investigation was conducted on the capillary limit of a miniature loop heat pipe with multiple evaporators and multiple condensers. Tests were conducted under various operating conditions: 1) heat load to one evaporator only; 2) even heat loads to both evaporators; 3) no temperature control of either compensation chamber; 4) controlling the temperature of one or both compensation chambers using thermoelectric devices; 5) placing the loop in a horizontal position with evaporators and compensation chambers on the same plane; and 6) placing the loop in a vertical position with evaporators above the compensation chambers. The physical processes that lead to evaporator deprime and the recovery from the deprime were examined. The effect of the gravity on the capillary limit and the compensation chamber temperature was also investigated.

Nomenclature

G_{COND}	=	thermal conductance between the evaporator and the compensation chamber, W/K
$\dot{m}$	=	mass flow rate, kg/s
Q_L	=	heat load to evaporator, W
Q_{LEAK}	=	heat leak, W
r_P	=	radius of curvature of the wick at the vapor/liquid interface, *m*
T_{CC}	=	compensation chamber temperature, K
T_{EVAP}	=	vapor temperature in the evaporator, K
ΔP_{BAY}	=	pressure drop across the bayonet section, Pa
ΔP_{CAP}	=	capillary pressure, Pa
ΔP_{COND}	=	pressure drop across the condenser section, Pa
ΔP_G	=	gravity head, Pa
ΔP_{GRV}	=	pressure drop across the evaporator groove section, Pa
ΔP_{LIQ}	=	pressure drop across the liquid line section, Pa
ΔP_{TOT}	=	total pressure drop in loop heat pipe, Pa
ΔP_{VAP}	=	pressure drop across the vapor line section, Pa
ΔP_{WICK}	=	pressure drop across the wick section, Pa
λ	=	latent heat of vaporization of the working fluid, J/kg
Δv	=	difference in specific volume between the liquid and vapor phases, m^3/kg
σ	=	liquid surface tension force, N/m
θ	=	contact angle between the liquid and solid, deg

Subscripts

c	=	condenser
e	=	evaporator
i	=	index
t	=	total

I. Introduction

LOOP heat pipes (LHPs) which use surface tension forces developed in fine porous wicks to circulate fluid can transfer large heat loads over long distances with small temperature differences and no external pumping power. An LHP consists of an evaporator, a condenser, a compensation chamber (CC), and vapor and liquid lines. The evaporator is made with an integral CC with a bayonet and a secondary wick connecting these two elements. The CC saturation temperature determines the loop operating temperature. Because the CC is physically adjacent to the evaporator and is located in the path of the fluid circulation, its temperature is a function of the loop operating conditions such as the evaporator heat load, condenser sink temperature, and ambient temperature [1]. Under normal operation, the overall pressure drop in the loop must not exceed the capillary pumping capability of the primary wick. In addition, thermodynamic constraints require that the temperature difference between the evaporator and the CC match the corresponding pressure drop across the primary wick [2,3].

For multiple heat sources or a heat source with a large thermal footprint that needs to be cooled, an LHP with multiple evaporators will be very desirable. The feasibility of a multiple-evaporator LHP has been demonstrated [4–8]. There are several challenges for such a system. A simple thermodynamic analysis shows that, under most conditions, only one of the CCs will contain two-phase fluid and control the loop operating temperature. All other CCs will be completely liquid filled [1,6]. This characteristic has been experimentally verified through extensive testing of an LHP with two evaporators and two condensers [6–8]. Test results also show that control of the loop operating temperature can switch from one CC to another as the operating conditions change. Other issues such as interactions between individual CCs, temperature stability, and the loop's adaptability to rapid power and sink-temperature cycle were also investigated.

Recently a miniature LHP (MLHP) with two evaporators and two condensers has been developed at the NASA Goddard Space Flight Center, and a comprehensive test program has been executed to characterize the MLHP's thermal performance including startup,

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capillary limit, power cycle, sink-temperature cycle, and gravity effect. This paper focuses on the capillary limit of the MLHP. The capillary limit test was conducted by gradually increasing the heat load to the evaporators until a temperature excursion occurred in one of the evaporators. The loop behavior after the capillary limit was exceeded, and the ability of the loop to recover from a deprime were also examined. Experimental tests were conducted under various operating conditions: 1) heat load to one evaporator only; 2) even heat loads to both evaporators; 3) no temperature control of either CC; 4) controlling the temperature of one or both CCs using thermoelectric devices; 5) placing the loop in a horizontal position with evaporators and CCs on the same plane; and 6) placing the loop in a vertical position with evaporators above CCs.

In the following sections, a theoretical background will be given first. This will be followed by descriptions of the test article and test setup. Experimental results will then be discussed in detail, including the physical processes that lead to evaporator deprime and recovery from deprime. Some issues related to the heat transport limit of an LHP will also be addressed.

II. Theoretical Background

In this section, physical processes involved in the operation of loop heat pipes with multiple evaporators and multiple condensers will be discussed so that the experimental results can be better understood. Figure 1 depicts the flow schematic and the corresponding pressure drop diagram for a typical LHP with two parallel evaporators and two parallel condensers. The capillary pressure that each wick is able to sustain can be expressed as

$$\Delta P_{\mathrm{CAP},i} = \frac{2\sigma\cos\theta_i}{r_{P,i}} \qquad (i = 1, 2) \tag{1}$$

As the heat load is applied to the evaporators, fluid flow will be established, and a pressure drop will be present in each component of the LHP. The mass flow rate through each evaporator can be calculated as

$$\dot{m}_i = \frac{Q_{L,i}}{\lambda_i} \qquad (i = 1, 2) \tag{2}$$

The total mass flow rate through the transport lines and the condenser is $\dot{m}_t = \dot{m}_{c,1} + \dot{m}_{c,2}$. For the LHP to function properly, each evaporator must be able to sustain the total pressure drop imposed upon its wick:

$$\Delta P_{\mathrm{CAP},i} \geq \Delta P_{\mathrm{TOT},i} \qquad (i = 1, 2) \tag{3}$$

The equality sign holds true when the capillary limit is reached. The total pressure drop that the primary wick has to sustain is the sum of the pressure drops in the bayonet, primary wick, vapor grooves, vapor line, condenser, liquid line, and that due to the gravity head, that is

$$\Delta P_{\mathrm{TOT},i} = \Delta P_{\mathrm{BAY},i} + \Delta P_{\mathrm{WICK},i} + \Delta P_{\mathrm{GRV},i} + \Delta P_{\mathrm{VAP}} + \Delta P_{\mathrm{LIQ}} + \Delta P_{\mathrm{COND}} + \Delta P_{G,i} \qquad (i = 1, 2) \tag{4}$$

In Fig. 1, it is assumed that evaporator 2 (E2) receives a higher heat load than evaporator 1 (E1). The pressure drop from the vapor line (point 5) to the liquid line (point 12) via condensers is common to both evaporators and is a function of the total heat load applied to the two evaporators. The pressure drops from the outer diameter of the E1 wick (point 1) to the vapor line (point 5) and from the liquid line (point 12) to the inner diameter of the E1 wick (point 14) are dependent upon the heat load to the E1 only. Likewise the pressure drops from the outer diameter of the E2 wick (point 3) to the vapor line (point 5) and from the liquid line (point 12) to the inner diameter of the E2 wick (point 16) are dependent upon the heat load to the E2 only. Thus the total pressure drop imposed upon each evaporator is a function of the total heat load as well as the heat load distribution between the two evaporators.

When the heat load is applied to only one of the evaporators, the evaporator receiving no heat load actually works as a condenser. Figure 2 shows the pressure drop diagram when E1 receives no external heat load [9]. The flow from the liquid line (point 5) to the vapor line (point 12) via E1 outlet (point 2) is in the reverse direction. Consequently the pressure drop that the E1 wick has to sustain could be much smaller than that shown in Fig. 1. The exact amount of heat dissipation through E1 is determined by the conservation laws of mass, momentum, and energy among the two condensers and E1, and is a function of many factors, including the heat load, line sizes, condenser sink temperatures, and ambient temperature. As the heat load to E2 increases, the pressure drops imposed upon both evaporators also increases. Whenever the pressure drop exceeds the capillary limit of either evaporator, vapor penetration will occur in the weaker evaporator first.

Because both the CC and the evaporator contain two-phase fluid, there exists a relationship between the temperature difference and the pressure difference of these components, as expressed by the Clausius–Clapeyron equation [10].

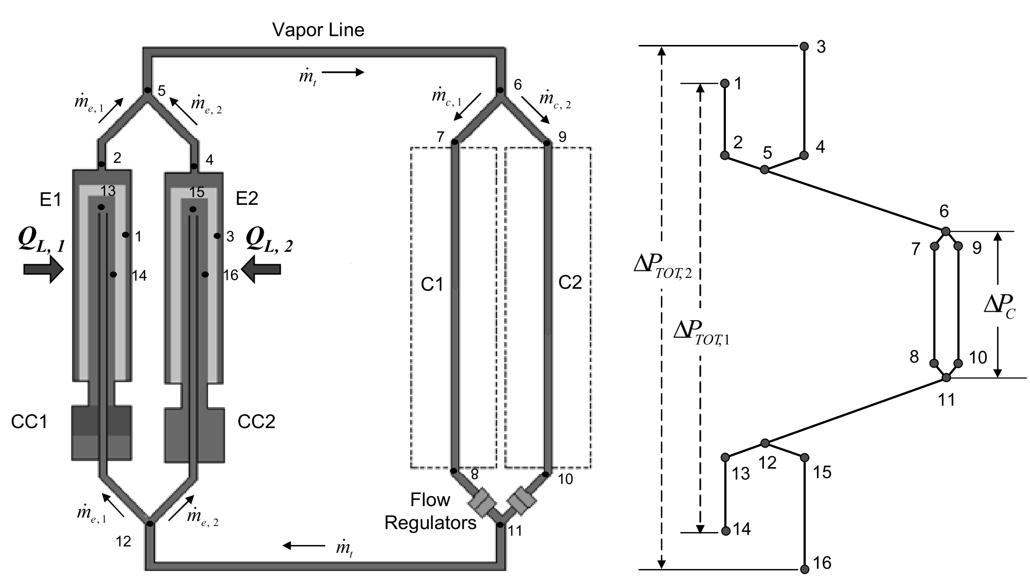

Fig. 1 Flow schematic and corresponding pressure drop diagram for an LHP with two evaporators and two condensers with heat load to both evaporators.

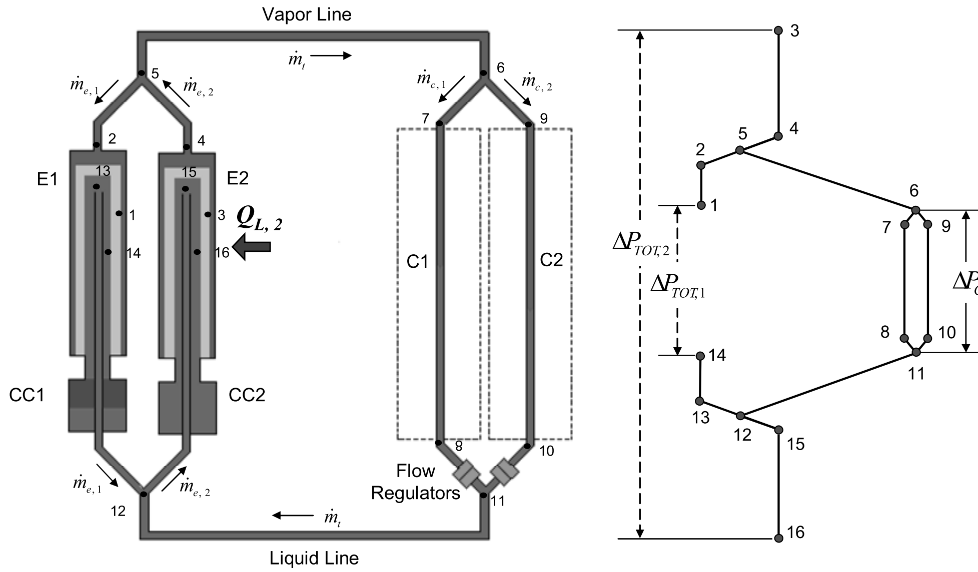

Fig. 2 Flow schematic and corresponding pressure drop diagram for an LHP with two evaporators and two condensers with heat load to E2 only.

$$\Delta P_{\mathrm{TOT},i} - \Delta P_{\mathrm{WICK},i} = \frac{\lambda(T_{\mathrm{EVAP},i} - T_{\mathrm{CC},i})}{T_{\mathrm{CC},i}\Delta v_i} \qquad (i = 1, 2) \qquad (5)$$

Thus for a given pressure drop in the loop, a corresponding temperature difference exists between the evaporator and the CC. Such a temperature difference affects the heat leak from the evaporator to the CC, which can be expressed as

$$Q_{\mathrm{LEAK},i} = G_{\mathrm{COND},i}(T_{\mathrm{EVAP},i} - T_{\mathrm{CC},i}) \qquad (i = 1, 2) \qquad (6)$$

The heat leak is a critical component in determining the CC saturation temperature, which in turn governs the loop operating temperature. The thermal conductance, $G_{\mathrm{COND},i}$, is highly dependent upon the vapor void fraction in the evaporator core. If the evaporator core is completely filled with liquid, the heat leak is transmitted by heat conduction through the evaporator shell, and the thermal conductance is usually very small. If vapor exists in the evaporator core, the evaporator core becomes part of the CC. Heat can be transferred from the evaporator to the CC by conduction through the primary wick. The thermal conductance becomes large due to the short heat transfer path. Moreover the higher the vapor void fraction, the larger the heat leak [2,11]. Because the pore sizes are not uniform, r_P in Eq. (1) refers to the largest (and hence the weakest) pore of the wick. As the weaker pores fail, vapor will penetrate through the wick and reach the evaporator core. Because the evaporator core can tolerate vapor, the stronger wick pores can continue to pump liquid and the loop can continue to work. However vapor penetration results in a higher heat leak to the CC, and hence a higher operating temperature. Two events follow after vapor penetrates the wick. Because the surface tension decreases with an increasing temperature, more and more pores will fail, leading to more vapor penetration and an ever-increasing operating temperature. On the other hand, the viscosity of the fluid decreases with increasing temperature, leading to a smaller total pressure drop. Consequently a new steady state could be reached at a higher operating temperature if the capillary limit (maximum heat load) is not exceeded by too much. One indication that the capillary limit is exceeded is a rapid increase of the temperature difference between the evaporator and the CC due to decreasing thermal conductance. Another indication is that the temperature of the CC connected to the failing evaporator will rise rapidly and begin to control the loop operating temperature regardless of which CC has been in control before vapor penetration.

The heat transport capability of a capillary two-phase thermal system is measured by the maximum heat load it can carry without exceeding the allowable temperature. For some two-phase systems such as capillary pump loops, the maximum heat load is reached when the total pressure drop is equal to the capillary limit because any further increase in the heat load will lead to blockage of the liquid flow and hence a temperature excursion of the evaporator. This is not true for LHPs. The LHP can reach a new steady state even after the capillary limit is exceeded at a given set point temperature as described previously. Thus the concept of a heat transport limit becomes more ambiguous. Both the capillary limit and the pressure drop are functions of the temperature, and there are many factors that can affect the LHP temperature. When multiple parallel evaporators are present, the transport limit is also a function of the heat load distribution among the evaporators, and thus there is a small range of values for the maximum heat load. The effect of various parameters on the capillary limit of an LHP with two evaporators is the subject of this investigation.

III. Test Article and Test Setup

Figure 3 shows a picture of the MLHP test article, which consists of two parallel evaporators, two parallel condensers, a common vapor transport line, and a common liquid return line. Each evaporator has its own integral CC. The main features of this MLHP include: 1) 9.65-mm o.d. evaporators; 2) titanium primary wicks with pore sizes of approximately 1.5 μm; 3) stainless steel (SS) vapor and liquid transport lines having 2.38 and 1.59 mm o.d., respectively; 4) aluminum condensers having 2.38 mm o.d.; 5) and a thermoelectric converter (TEC) attached to each CC. A flow regulator made of a capillary wick with a 10 μm pore radius is installed downstream of each condenser. The flow regulator prevents vapor from penetrating the wick before both condensers are fully used, and hence serves to balance the flows between the two condensers. The loop is charged with 29 g of anhydrous ammonia.

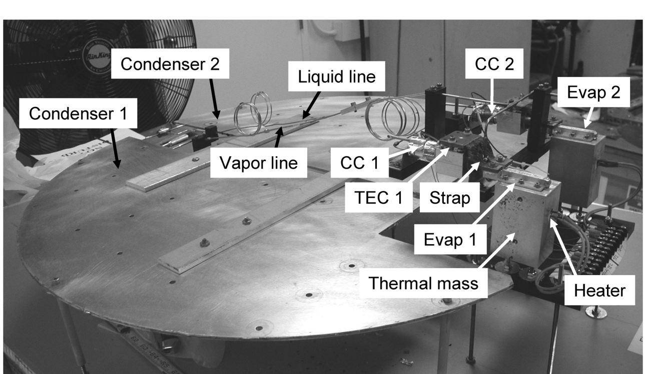

Fig. 3 Photograph of MLHP.

Table 1 Summary of design parameters

Evaporator	Aluminum 6061 9.65 mm o.d. × 52 mm · L
Primary wick	Titanium 6.35 mm o.d. × 3.18 mm i.d. 1.5 μm pore size 0.2 E–13 m^2 permeability
CC	SS 304L 22.2 mm o.d. × 72.4 mm · L
Vapor line	2.38 mm o.d.
Liquid line	1.59 mm o.d.
Condenser	2.38 mm o.d. × 2540 mm · L
Working fluid	Ammonia 29 g

Table 1 shows the design parameters of the main components. A 400 g aluminum mass was attached to each evaporator to simulate the instrument mass. A cartridge heater was attached to each thermal mass to provide heat loads between 1 and 150 W per evaporator. Power was manually controlled with an error of ±5%. A TEC was attached to each CC. A copper thermal strap was used to connect the rear side of the TEC to the evaporator. Each condenser was attached to a cold plate, and each plate was cooled by a separate chiller. MLHP was covered with foam insulation to minimize heat exchange between the MLHP and atmosphere. Figure 4 shows a schematic of the test loop with thermocouple locations. The loop temperatures were monitored by 72 thermocouples. A data acquisition system consisting of a data logger, a personal computer, a CRT monitor, and Labview software was used to monitor and store data. The data were updated on the monitor and stored in the computer every second.

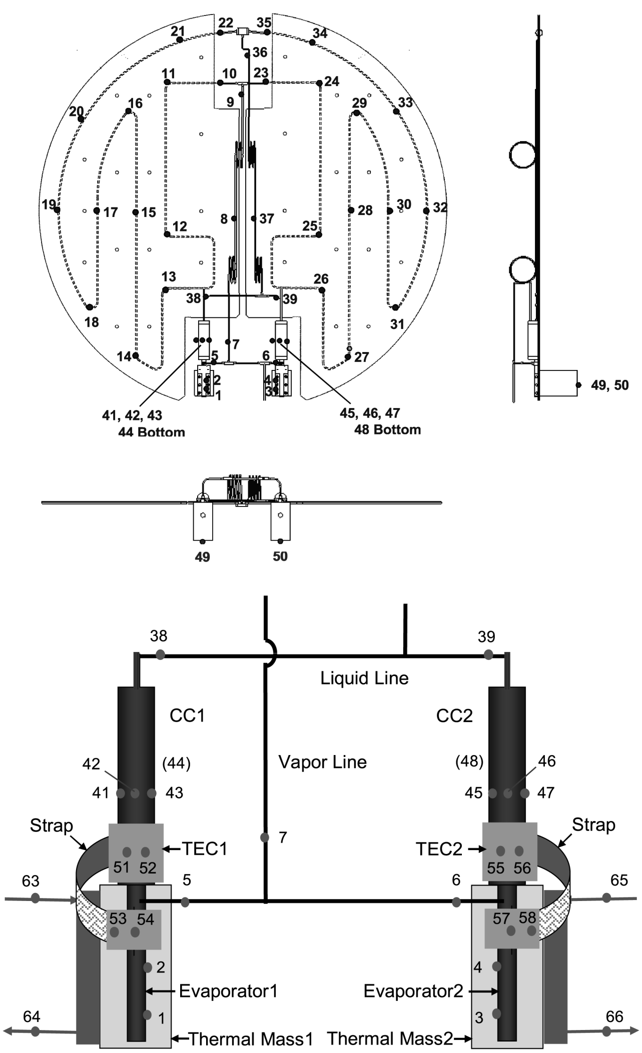

Fig. 4 Thermocouple locations for the MLHP.

IV. Test Results

More than 80 capillary limit tests were conducted in a laboratory environment. In some tests, the CC temperatures were not controlled, that is, each CC was allowed to reach its natural equilibrium temperature under the given test condition. This is referred to as the natural operating temperature of the LHP. In other tests, one or both of the CCs were kept at a fixed temperature between 303 and 313 K using the TECs. The sink temperatures of condenser 1 and condenser 2 (C1/C2) were varied between 253/253 and 283/293 K. Power profiles included a heat load to one evaporator only and even heat loads to both evaporators. To evaluate the effects of gravity on the LHP capillary limit, tests were conducted under horizontal and vertical configurations. In the horizontal configuration, the evaporators and CC were on the same horizontal plane. In the vertical configuration, the evaporators were directly above the CCs. In most tests, the system heat load was raised to a higher level after the capillary limit had been exceeded so as to demonstrate that the loop could reach another steady state at a higher temperature. Recovery of the evaporator was verified by reducing the heat load to the evaporators near the end of each test. In this paper, several typical test results will be presented, as listed in Table 2. Included in the table for each test are the power profile, condenser sink temperatures, and whether or not the CC temperatures were actively controlled.

In an LHP with multiple evaporators, only one of the CCs will contain two-phase fluid and control the loop operating temperature. All other CCs will be filled with liquid [12–14]. In this test program, there were four thermocouples attached to each CC. It was observed throughout the test program that the CC containing two-phase fluid displayed a uniform temperature whereas the liquid-filled CC displayed nonuniform, subcooled temperatures.

A. No Active Control of CC Temperatures

1. *Heat Load to E1 Only*

Figure 5 illustrates the loop temperatures in a capillary limit test where the heat load was applied to evaporator 1 (E1) only. The condenser 1 and condenser 2 (C1/C2) temperatures were kept at 273/273 K. Because evaporator 2 (E2) received no heat load, E2 worked as a condenser. Under such a condition, compensation chamber 2 (CC2) would always control the loop operating temperature before the loop reaching its capillary limit [11,12]. This was experimentally verified in this test for E1/E2 heat loads between 20/0 and 80/0 W. As shown in Fig. 6, in this power range the CC2 temperatures (TC45 to TC48) were uniform and were higher than the compensation chamber 1 (CC1) temperatures (TC41 to TC44) which were subcooled and nonuniform. The capillary limit of E1 was exceeded at 100/0 W as evidenced by four accompanying events [15]. First the CC1 temperatures (TC41 to TC44) became uniform and exceeded the CC2 temperatures (TC45 to TC48), which became subcooled and spread. This suggested that vapor had penetrated through the E1 wick, and CC1 began to control the loop operating temperature. Second immediately following vapor penetration, cold liquid was pushed from the E1 inlet to the E2 inlet along the liquid line, causing the E2 inlet temperature TC39 to drop temporarily. Third the CC1 temperature increased rapidly for a modest power increase. Fourth the temperature difference between E1 and CC1 also increased rapidly for a modest power increase due to a decreasing thermal conductance after the vapor penetration. Nevertheless the loop continued to function at a higher operating temperature. The loop also approached another steady temperature as the heat load further increased to 110/0 W. The loop recovered as the heat load was reduced to 60/0 W. However the CC1 temperature was about 10 K higher than that at 60/0 W before the vapor penetration, indicating a residual effect from the earlier vapor penetration [16,17].

2. *Heat Load to E2 Only*

Figure 7 shows the loop temperatures in a capillary limit test where the heat load was applied to evaporator 2 (E2) only. The C1/C2 sink temperatures were kept at 273/273 K. This was the same test as the one shown in Fig. 5 except that power was applied to E2 only. Similar

Table 2 Summary of the capillary limit tests

Date	Power application	Tilt	CC1/CC2 set point, K	C1/C2 sink, K	Tests performed
5 July 2005	E1 only	Horizontal	--/--	273/273	E1/E2: 20/0, 40/0, 60/0, 80/0, 90/0, 100/0, 110/0, 120/0, 60/0
8 June 2005	E2 only	Horizontal	--/--	273/273	E1/E2: 0/10, 0/20, 0/40, 0/60, 0/80, 0/90, 0/100, 0/110, 0/120
6 July 2005	Both	Horizontal	--/--	273/273	E1/E2: 20/20, 30/30, 40/40, 50/50, 60/60, 70/70, 75/75, 80/80, 50/50
8 Nov. 2005	E1 only	Horizontal	303/--	273/273	E1/E2: 5/0, 20/0, 40/0, 60/0, 70/0, 80/0, 90/0, 100/0, 110/0, 120/0, 0/0
29 Sept. 2005	Both	Horizontal	303/303	273/273	E1/E2: 20/20, 30/30, 40/40, 50/50, 55/55, 60/60, 65/65, 70/70, 75/75, 80/80, 50/50
6 Sept. 2005	E1 only	Vertical	--/--	273/273	E1/E2: 5/0, 20/0, 40/0, 60/0, 80/0, 90/0, 95/0, 100/0, 5/0
7 Sept. 2005	Both	Vertical	--/--	273/273	E1/E2: 5/5, 10/10, 20/20, 40/40, 60/60, 70/70, 75/75, 5/5

phenomena were observed again. Between 0/10 and 0/110 W, E1/CC1 worked as a condenser, and CC1 controlled the loop operating temperature, whereas CC2 was liquid filled. When a heat load of 0/120 W was applied, E2's temperature rose rapidly. About 30 min after 120/0 W was applied, CC2's temperature began to rise rapidly, and E2's temperature rose at a much higher rate. At this point, vapor penetration occurred, and cold liquid was pushed from the E2 inlet to the E1 inlet along the liquid line, causing the E1 inlet temperature TC38 to drop temporarily. Finally the E2 temperature exceeded the maximum temperature set for this test, and the heat load was removed. There is a difference in capillary heat transport limitation for E1 heat load only in Fig. 5 and E2 heat load only in Fig. 7 due to a slight difference in the pore size and permeability of the two primary wicks. Another noticeable difference between E1 and E2 is that, at the same heat loads, E1 temperature was always higher than E2. This was most likely caused by the relatively new technology in manufacturing the miniature LHPs.

3. *Even Heat Load to Both Evaporators*

Figure 8 shows the loop temperatures in a capillary limit test where an even heat load was applied to both evaporators. The C1/C2 sink temperatures were kept at 273/273 K. Throughout the test, the E1 temperature was always higher than the E2 temperature. The evaporator temperatures began to rise rapidly at 70/70 W for a modest power increase. The CC1 temperature also rose and began to control the loop operating temperature. After the capillary limit was exceeded at the given operating temperature, however, the loop continued to operate steadily at higher operating temperatures with 75/75 and 80/80 W. When the heat load was reduced to 50/50 W, the loop recovered from dryout. However the CC1 and CC2 temperatures were 12 and 9 K higher than those at 50/50 W before the vapor penetration, indicating a residual effect from the earlier vapor penetration. Pressure drop was calculated by using computer code and the value in the liquid line when the 70/70 W was applied was estimated to be about 820 Pa, whereas the pressure drop in the vapor line was 13,000 Pa.

B. Active Control of CC Temperatures

The CC temperature can be controlled at a fixed set point that is higher than the loop's natural operating temperature for a given test condition by using a TEC. When one of the CCs is controlled at the desired temperature, the other CC will be liquid filled. With the loop

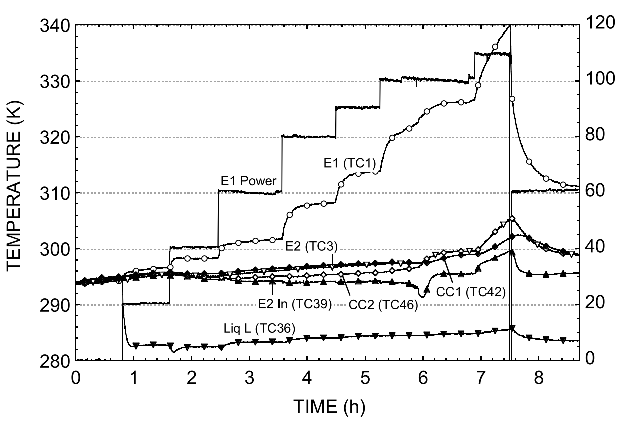

Fig. 5 Loop temperatures with heat load to E1 only.

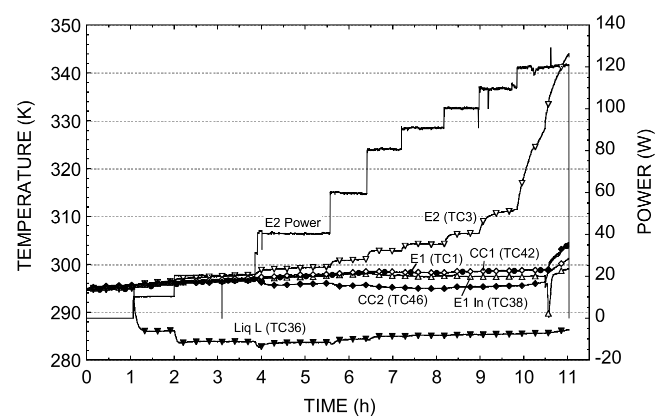

Fig. 7 Loop temperatures with heat load to E2 only.

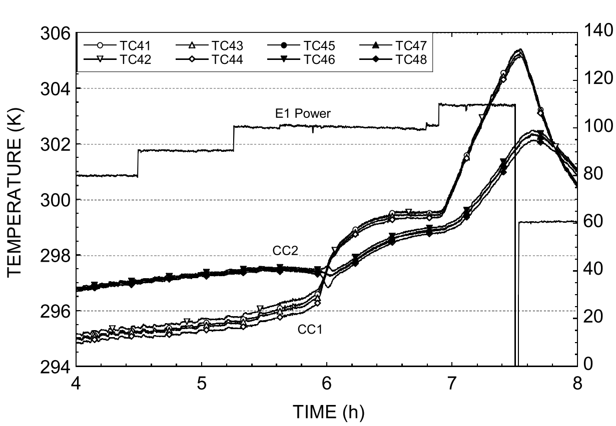

Fig. 6 CC temperatures with heat load to E1 only.

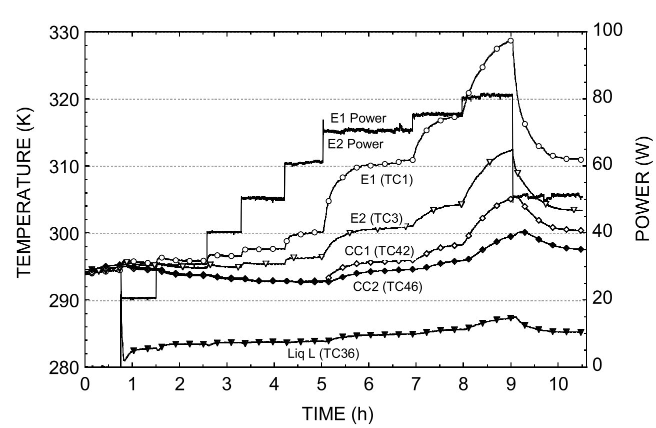

Fig. 8 Loop temperatures with heat load to both evaporators.

operating at a constant temperature, the surface tension force and the capillary limit of each evaporator as expressed in Eq. (1) are fixed.

1. Heat Load to E1 Only

Figure 9 shows the loop temperatures in a test where the CC1 temperature was controlled at 303 K, and the heat load was applied to E1 only. The CC1 temperature could be maintained at 303 K for heat loads between 5/0 and 110/0 W. As E1 reached its capillary limit at 110/0 W, vapor penetrated through the E1 wick, and the E1 temperature rose. In Fig. 5, the CC1 temperature (i.e., the natural operating temperature) exceeded 305 K at 110/0 W. In Fig. 9, the CC1 temperature was kept at 303 K through TEC cooling. This is evidenced by the decreasing TEC control heater power shown in Fig. 10 as E1 approached and then exceeded its capillary limit. In the figure, the TEC power was assigned a positive value when the TEC was heating the CC and a negative value when the TEC was cooling the CC.

2. Even Heat Load to Both Evaporators

Figure 11 shows the loop temperature in the test where CC1/CC2 were controlled at 303 K, and even heat loads were applied to both E1 and E2. Throughout the test, CC1/CC2 were able to control the loop operating temperature at 303 K. Figure 12 illustrates the TEC power. As the heat load was increased to 70/70 W, TEC1 and TEC2 power began to decrease. At 80/80 W, TEC2 control switched from heating to cooling, indicating that E2 had exceeded its capillary limit. The loop completely recovered as the heat load was reduced to 50/50 W. However the E1 and E2 temperatures were 3 and 2 K higher than those at 50/50 W before the vapor penetration. In addition, the TEC power was also smaller than that at the same heat load to the evaporators. This indicated that a higher heat leak had occurred after the previous event of vapor penetration.

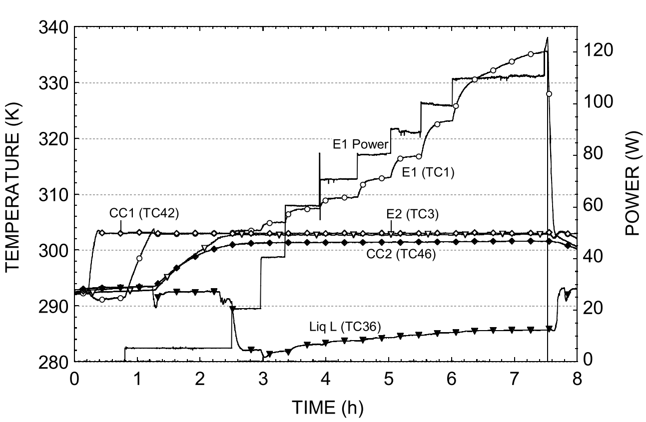

Fig. 9 Loop temperatures with heat load to E1 only with CC active control.

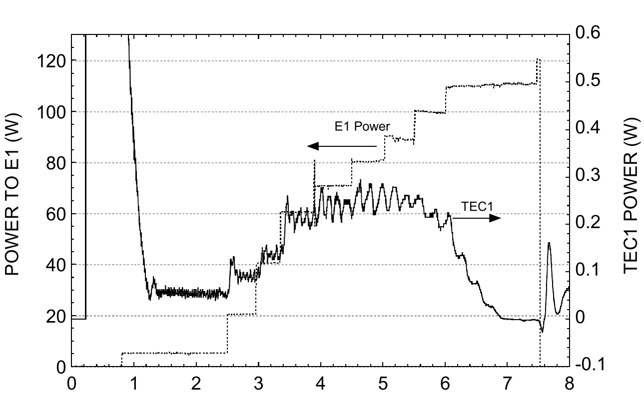

Fig. 10 TEC power with heat load to E1 only.

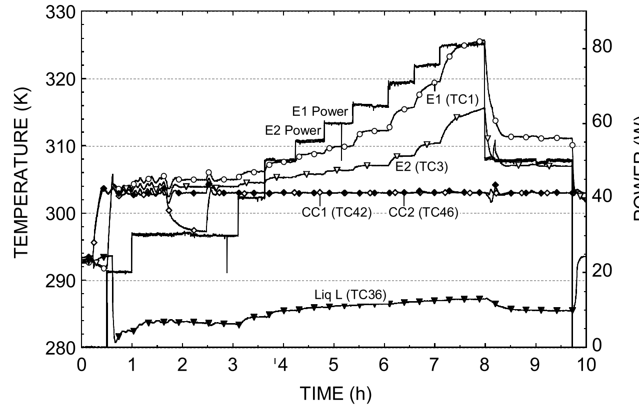

Fig. 11 Loop temperatures with heat load to both evaporators and active control of CC temperatures.

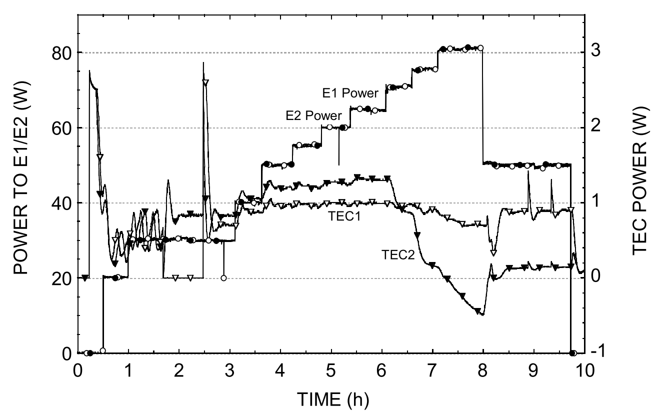

Fig. 12 TEC powers with heat load to both evaporators.

C. Gravity Effects

To evaluate the gravity effect on the LHP operating temperature and heat transport limit, the MLHP was also tested in a vertical position where the evaporators were above the CCs. The LHP natural operating temperature would increase in the vertical configuration, especially at low power, for the following reason. As the pressure difference across the evaporator wick increased due to gravity head, the difference in saturation temperatures also increased, as dictated by Eq. (5). This would lead to an increasing heat leak from the evaporator to the CC. Because the enthalpy of the liquid entering the compensation chamber did not change, the only way to compensate for the increasing heat leak was by increasing the CC temperature. Gravity would also affect the LHP heat transport capability, that is, the LHP heat transport capability would decrease in the vertical configuration due to an additional gravity head [see Eqs. (3) and (4)].

1. Heat Load to E1 Only

Figure 13 shows the loop temperatures in a test where the LHP was in a vertical configuration, and the heat load was applied to E1 only. The C1/C2 sink temperatures were kept at 273/273 K. This test was the same test as shown in Fig. 5 except for the different test configuration. At low power, CC1 controlled the loop operating temperature. At the heat load of 60/0 W, CC2 began to control the loop operating temperature. Above 80/0 W, operating temperature control was switched from CC2 to CC1 again, and the E1 temperature rose rapidly, indicating that E1 had reached its capillary limit. The loop could operate steadily at a higher saturation temperature with 90/0 W. At 100/0 W, the E1 temperature suddenly rose and began a temperature excursion. When the heat load was reduced to 5/0 W, the loop recovered from dryout. However the CC1 and CC2 temperatures were a few degrees higher than at 5/0 W before vapor penetration. Figure 14 shows the CC temperatures as a function of the applied power in horizontal and

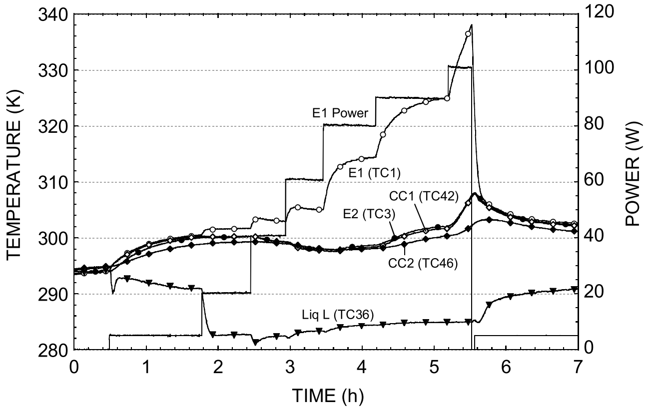

Fig. 13 Loop temperatures with heat load to E1 only in vertical configuration.

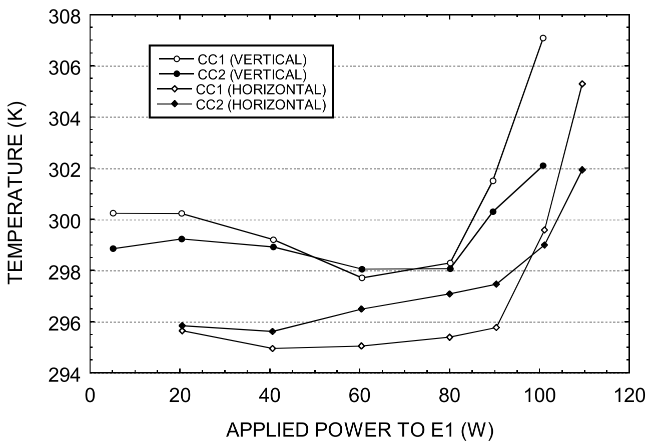

Fig. 14 CC Temperature as a function of heat load in horizontal and vertical configurations.

vertical configurations. The evaporator temperature was higher under the vertical configuration than under the horizontal configuration at all heat loads. In addition, the heat transport capability under the vertical configuration was about 20 W lower than under the horizontal configuration.

2. Even Heat Load to Both Evaporators

Figure 15 shows the loop temperature in the test where even heat loads were applied to both E1 and E2. Throughout the test, CC1 controlled the loop operating temperature. At 60/60 W, the E1 temperature began to rise rapidly, an indication that the loop had reached its capillary limit. At 75/75 W, the E1 temperature exceeded

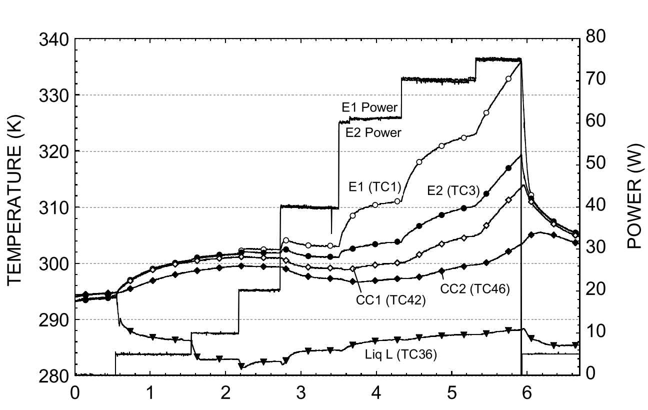

Fig. 15 Loop temperatures with heat load to both evaporators in vertical configuration.

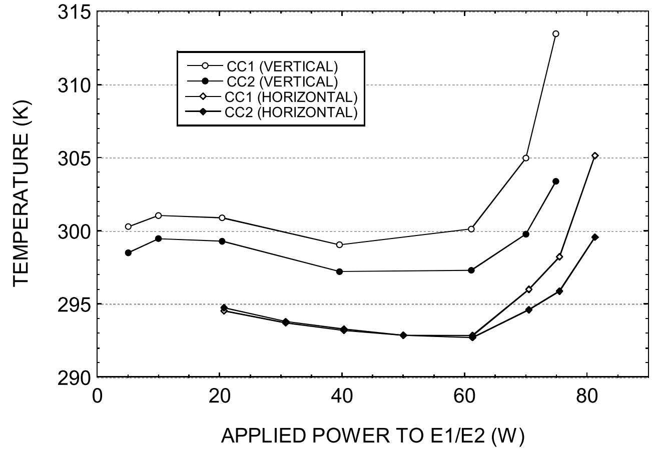

Fig. 16 CC Temperature as a function of heat load in horizontal and vertical configurations.

the maximum allowable temperature set for this test. When the heat load was reduced to 5/5 W, the loop recovered from dryout. The CC1 and CC2 temperatures were also slightly higher than those at 5/5 W before vapor penetration. Figure 16 shows the CC temperature as a function of the applied power in the horizontal and vertical configurations. Compared with the test performed under the horizontal configuration shown in Fig. 8, the evaporator temperature was higher and the capillary limit was about 20 W lower under the vertical configuration.

V. Summary

Several high-power tests were conducted to characterize the capillary limit of a miniature LHP with two evaporators and two condensers. Heat load was applied to one evaporator and to both evaporators, and the physical process in each case was discussed. Test results showed that when the capillary limit was exceeded at a given operating temperature, the loop could continue to operate at a slightly higher heat load and reach a new steady-state at a higher operating temperature. Tests were also conducted with and without active control of the CC set point temperature. In addition, test results showed that TECs were very effective in controlling the CC temperatures. In cases where the CC temperatures were actively controlled, either one or both CCs were controlled using TECs. Under the vertical configuration, the natural operating temperature of the loop was higher than that under the horizontal configuration at the same heat load. Furthermore the heat transport capability decreased in the vertical configuration due to an addition pressure drop imposed by the liquid head.

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