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Investigation of Effects of Auxiliary Measures for Startup of Loop Heat Pipes

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Experiments have been conducted to investigate the feasibility of two active measures, namely, local heating on the evaporator and cooling the compensation chamber, with a thermoelectric cooler. A new passive auxiliary measure, which utilizes the latent heat of phase change material to maintain the compensation chamber temperature at the melting point to help estimate the required superheat, has been presented for the first time. The effects of the three auxiliary measures on startups and the steady-state operation have been investigated. Test results indicate that the three auxiliary measures are helpful in establishing the required liquid superheat and reducing the evaporator temperature overshoot, as well as in lessening the startup time. Local heating on the evaporator near the evaporator outlet will not affect the steady-state operation, whereas local heating on the evaporator near the compensation chamber will lead to either sustaining operating temperature rises or higher operating temperatures. The continued operation of the thermoelectric cooler will decrease the operating temperatures at low heat loads and the typical V operating temperature curve will be changed to a nearly linear one. At too low heat loads, the phase change material can only delay the evaporator temperature rise and gain time for the loop heat pipe to wait for the arrival of a higher heat load, which is more favorable for startups.

Nomenclature

A	=	area, m^2
$C_{p,\mathrm{liq}}$	=	specific heat at constant pressure, J/kg · K
D_i	=	inner wick diameter, m
D_o	=	outer wick diameter, m
h	=	specific enthalpy, J/kg
L_A	=	length from locally heated zone to point A, m
L_{HZ}	=	length of locally heated zone on evaporator wall, m
L_{VG}	=	length of vapor grooves, m
$m_{\mathrm{PCM\_liq}}$	=	mass of liquid phase change material, kg
$\dot{m}$	=	mass flow rate, kg/s
P_{TEC}	=	input power of thermoelectric cooler, W
p	=	pressure, Pa
Q	=	heat load, W
$\dot{Q}_{\mathrm{HL}}$	=	heat leak from evaporator to compensation chamber (CC) W
$\dot{Q}_{\mathrm{loop}}$	=	heat absorbed by liquid evaporation in evaporator, W
$Q_{\mathrm{PCM\_liq}}$	=	amount of latent heat of liquid phase change material, J
$\dot{Q}_{\mathrm{sub}}$	=	liquid subcooling at inlet of CC, W
R_{axi}	=	thermal resistance in axial direction of wick, K/W
R_{rad}	=	thermal resistance in radial direction of wick, K/W
r	=	latent heat of phase change material, J/kg
T	=	temperature, K
U	=	thermal conductance per unit area, W/m^2 · K
v	=	specific volume, m^3/kg
Δ	=	difference
λ_{eff}	=	effective thermal conductivity of wick, W/m · K

Subscripts

abs	=	heat load absorbed by thermoelectric cooler
amb	=	ambience
cond	=	condenser
fg	=	difference of thermal properties between saturated fluid and gas
in	=	inlet of component
liq	=	liquid
out	=	outlet of component
rej	=	heat load rejected by thermoelectric cooler
sat	=	saturation or saturated
sink	=	heat sink
sub	=	liquid subcooling at inlet of compensation chamber
vap	=	vapor

I. Introduction

THE loop heat pipe (LHP) was invented in the U.S.S.R. in 1972 (Ref. 1) and is a valuable and promising two-phase heat transfer device for the spacecraft thermal control system. However, there are still some problems hindering the practical application of the LHP, one of which is the startup problem. Although the startup of the LHP is a complicated dynamic process and many factors are involved, there are only four startup situations,[2] that depend on the liquid/vapor composition in the evaporator core and vapor grooves. The startup situation when liquid floods the vapor grooves and vapor exists in the liquid core is the most difficult. When the vapor grooves are flooded with liquid, a liquid superheat is required to initiate the nucleate boiling, whereas the existence of vapor in the evaporator core will lead to the high heat leak from the evaporator to the compensation chamber (CC). Therefore, at low startup heat loads, the temperature difference between the evaporator and the CC, namely, the liquid superheat, is difficult to attain, and the liquid nucleate boiling is difficult to reach. In this situation, the startup will last a long time, and the evaporator temperature will rise above the final steady-state temperature and even exceed the allowable

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temperature. Furthermore, Zhang et al.[3] pointed out that some unstable phenomena, including the reverse flow, the evaporation inside the primary wick, and the temperature oscillation, occurred during startups at low heat loads and that such startups would lead to higher steady-state operating temperatures, namely, the temperature hysteresis. To solve the startup problem, measures must be taken to ensure that the evaporator temperature rise will not exceed the allowable value and that the startup will not influence the steady-state operating temperatures.

A successful startup requires that the liquid superheat in the vapor grooves be obtained quickly before the evaporator temperature rises to a high value. Because the working fluid in the CC is in a two-phase saturated state, the liquid superheat is actually the temperature difference between the evaporator and the CC. Two active auxiliary measures have been suggested to solve the startup problem: one is to apply an extra heat load on the evaporator, and the other is to cool the CC by using a thermoelectric cooler (TEC).[4] These two measures are both helpful to establish the temperature difference between the evaporator and the CC. However, they both require active control and extra energy. In this paper, a new passive auxiliary measure for the LHP startup, which utilizes the latent heat of phase change material (PCM) to maintain the CC temperature at the melting point and which will help establish the temperature difference between the evaporator and the CC, has been presented for the first time. Because no, or few, test results on the active auxiliary measures have been reported in previous literature, extensive experiments are conducted to investigate and analyze the effects of the three auxiliary measures on startups and the steady-state operation. The effects of the thermal capacity of instruments attached to the evaporator on the assisted startup are also discussed.

II. Test Setup

The test device was an ammonia–stainless-steel LHP with a nickel wick. Table 1 shows the geometric parameters of the components. No secondary wick was used for the ground test, and so the test LHP had only a bayonet in the evaporator core.

The condenser line was mounted on an aluminum cold plate with imbedded coolant channels, which was cooled by a low constant temperature trough. Heat input to the evaporator was provided by film heaters that were attached directly to the evaporator symmetrically. The entire loop was thermally insulated with sponge, which had a thermal resistance of about 10 K/W per unit length, to reduce the parasitic heat loss. Detailed descriptions of the three auxiliary measures, including the local heating on the evaporator, cooling the CC by the TEC, and fixing PCM containers on the CC, are provided hereafter.

A. Local Heating on Evaporator

The test results in Refs. 3–5 indicated that the difficult startups were always observed at low heat loads, whereas the LHP started up easily at high heat loads. A high heat load is favorable to establish the temperature difference between the evaporator and the CC quickly. The results suggested an effective method to solve the startup problem, namely, assistant heating on the evaporator. In this paper, an improved auxiliary measure, namely, applying an extra low heat load on a small area of the local evaporator zone, was taken. Because the extra heat load on the evaporator was low, this measure would not affect the final steady-state temperatures much in the operation after startups. However, the low heat load on an even smaller area could cause a higher heat flux, and this would be helpful to establish the liquid superheat in the vapor grooves to initiate the nucleate boiling during startups.

Table 1 Geometric parameters of test LHP

Components	Dimensions
Length of evaporator × o.d./i.d., mm	$\Phi 20.4/18 \times 175$
Length of vapor line × o.d./i.d., mm	$\Phi 3/2.2 \times 2600$
Length of liquid line × o.d./i.d., mm	$\Phi 3/2.2 \times 2200$
Length of condenser × o.d./i.d., mm	$\Phi 3/2.2 \times 2000$
Volume of CC, ml	17
Charge of working fluid, g	27
Maximum radius of wick, μm	1.5
Permeability of wick, m^2	$>5 \times 10^{-14}$
Length of wick × o.d./i.d., mm	$18/6 \times 148$
Height × width of vapor grooves, mm	1×1

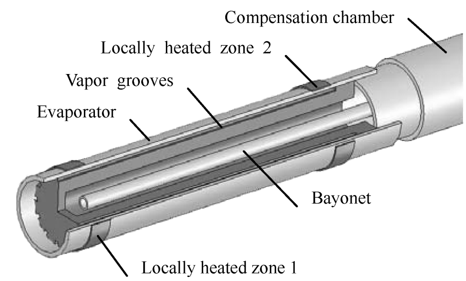


Fig. 1 Local heating on evaporator.

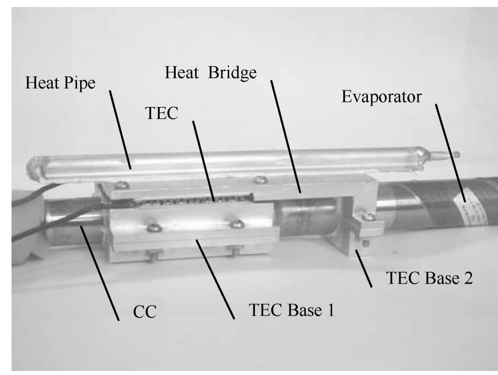
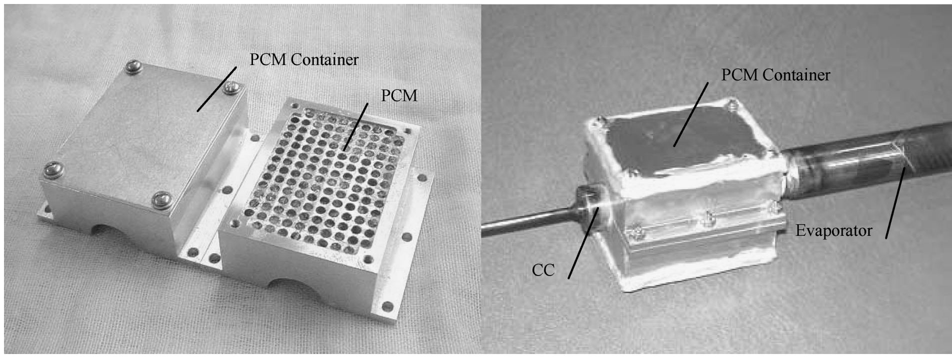


Fig. 2 Cooling CC by using TEC.

An evaporator base for accumulating heat is often fixed on the evaporator to provide the cylindrical evaporator with a plate to contact the heat instrument even. The locally heated zone must be separated from the evaporator base because heat will be transferred to the instruments during the startup if the assistant heat load is applied directly on the evaporator base. Therefore, the locally heated zones can be on the two ends of the evaporator. In the tests, the locally heated zone 1 was near the evaporator outlet, and the locally heated zone 2 was near the CC, as shown in Fig. 1. The areas of the two film heaters on the heated zones were 6 cm^2 each.

B. Cooling CC by TEC

As shown in Fig. 2, the bottom cold side of the TEC was attached to the TEC base 1 on the CC, and the upper hot side was attached to a heat bridge, which was also one of the two bases on the evaporator (the TEC base 2). When the working fluid in the CC was cooled by the TEC, the heat rejected from the hot side was conducted to the TEC base 2 on the evaporator though the heat bridge. This design helped to establish the temperature difference of the local evaporator zone and helped the CC to initiate the local nucleate boiling. The accessories of the TEC system had been designed with the following features: 1) To reduce the thermal resistance between the cold side of the TEC and the CC, a larger TEC base 1 was preferred to increase the contact area between the base and the CC. 2) A conventional heat pipe was glued onto the heat bridge to reduce the thermal resistance between the hot side of the TEC and the evaporator, which was the highest and the primary one among all of the thermal resistances. 3) The TEC base 2 must be separated from the evaporator base to prevent the heat from being transferred to the instruments, and a smaller TEC base 2 on the evaporator was preferred to reduce the contact area. Although a smaller base would increase the thermal resistance, a smaller area could achieve a higher heat flux on the evaporator, which would be helpful to the startup of the LHP. A smaller TEC base would leave a larger effective area of the evaporator base at which to couple the evaporator and the heat instruments together. The dimensions (length × width × height) of



a) PCM containers with inner holes **b) Assembly of PCM containers and LHP**

Fig. 3 Fixing PCM containers on CC.

the TEC were $40 \times 40 \times 4\ \mathrm{mm}^3$, and the allowable maximal voltage, current, and temperature difference between the hot side and cold side were 12V, 6A, and 50°C, respectively.

C. Fixing PCM Containers on CC

Because the phase change process is isothermal, the PCM can absorb the heat leak from the evaporator to the CC and maintain the CC at a constant value for some time during the startup. Because the liquid temperature in the vapor grooves will still rise due to the heat applied to the evaporator, the temperature difference between the evaporator and the CC can be established to initiate the nucleate boiling. The phase change device consisted of an external housing and the PCM, which could maintain the CC temperature at the PCM melting point with no moving parts, moving fluids, or power input. The melting point of the PCM must be higher than the initial ambient temperature. In other words, the PCM should be in the solid state before startups, otherwise the PCM would be not able to maintain the CC at its melting point when absorbing heat from the CC during startups. The octadecane (molecular formula $C_{18}H_{38}$), which is a kind of paraffin PCM with a purity of 99% and a melting range from 26 to 29°C, was chosen in the tests. When the ambient temperature was lower than about 25°C, startup tests could be conducted. The paraffin PCM has a high specific latent heat, but a low thermal conductivity, which is unfavorable for the PCM to absorb the heat leak effectively. Special aluminum containers with many inner holes full of PCM were designed to improve the PCM thermal transfer performance, as shown in Fig. 3a. In the tests, the two PCM containers were assembled on the LHP CC, as shown in Fig. 3b.

All of the tests must start at the most difficult startup situation, which is characterized by that in which the liquid filled vapor grooves and the liquid core have vapor inside, to investigate the feasibility and effects of the auxiliary startup measures. The most difficult startup situation can be obtained as follows. First, heat the compensation chamber for several minutes to increase the saturated temperature and pressure. The high pressure in the CC will push the working fluid to flood the external loop and the vapor grooves. Second, lay the loop vertically with the CC above the evaporator for about 12 h. The vertical position will keep the liquid flooding the vapor grooves due to gravity and the temperature of the whole loop will decrease to the ambient temperature gradually. The preceding operation ensures that the vapor grooves are flooded with liquid. Third, the startup tests can be conducted in a horizontal position, and the vapor/liquid interface exists in the liquid core as determined by the charge quantity of the working fluid. Thus, the vapor/liquid distribution in the evaporator can be obtained that ensures even the most difficult startups.

Copper/constantan (type T) thermocouples (TCs) were used to monitor the temperature profiles of the loop. The denotations and locations of the thermocouples are explained hereafter: TC cond_in and TC cond_out were at the condenser inlet and outlet, respectively. TC vap was at the evaporator outlet (vapor line inlet). TC CC_in was at the CC inlet (liquid line outlet). TC CC denoted the temperature of the saturated vapor zone of the CC, whereas TC CC 1 and TC CC 2 were on the vapor zone and the liquid zone of the CC, respectively. TC evap denoted the evaporator temperature. In the tests of startups with local heating measures, TC evap 1 and TC evap 2 were on the locally heated zone and the nonheated zone of the evaporator, respectively. In the tests of startups with the TEC cooling measure, TC evap 1 and TC evap 2 were near the TEC base 2 and on the unheated zone of the evaporator, respectively. TC base 2 was on the flange of the TC base 2. TC PCM 1 and TC PCM 2 were on the top and bottom PCM container walls, respectively. TC sink was for the sink temperature, and TC amb was for the ambience temperature.

III. Test Results and Discussions

Three parameters concerning the startup of the LHP were the startup time, the temperature overshoot, and the liquid superheat. When the vapor grooves were flooded with liquid, a liquid superheat was required to initiate the nucleate boiling in the vapor grooves. Because the working fluid in the CC was in a two-phase saturated state and no temperatures of the working fluid were measured in the tests, the liquid superheat could be defined as the wall temperature difference between the evaporator and the CC. The temperature overshoot was the evaporator temperature rise during the startup. A high-temperature overshoot might exceed the allowable temperature limit for the instruments. The startup time was the time duration from the beginning when the evaporator temperature increased due to thermal load to the moment when the evaporator temperature started to drop. The applications of the LHP showed a low-temperature overshoot and a short startup time.

A. Startups without Auxiliary Measures

The characteristics of the startup without auxiliary measures were investigated first, and then a comparison could be drawn to show the effects of the auxiliary measures. The four different vapor/liquid distributions were obtained as follows. In situation 1, the CC was heated and the loop was laid vertically with the CC above the evaporator for more than 12 h. Then, the vapor grooves and the evaporator core would be flooded with liquid. In situation 2, the LHP was started as described in situation 1 to operate for nearly 1 h and was then shut down. Vapor would exist in the vapor groove, whereas the evaporator core would be still filled with liquid. In situation 3, the CC was heated and the loop was laid vertically with the CC above the evaporator for more than 12 h. As a result, the tests could be conducted in a horizontal position with the CC and the evaporator on the same horizontal plane. Liquid would flood the vapor grooves, whereas the interface of the vapor/liquid would exist in the evaporator core, which was determined by the charge quantity of the working fluid. In case 4, the LHP was started as described in situation 3 to operate for nearly 1 h and was then shut down. Vapor would exist in the vapor groove and the vapor/liquid interface would still exist in the evaporator core.

Table 2 lists the four startup cases (not in the order as described) depending on the vapor/liquid distribution in the evaporator at a 6-W startup heat load. The item of two-phase distribution in Table 2 represents the two-phase states in the vapor grooves and the

Table 2 Summary of 6-W startups without auxiliary measures

Two-phase distribution	Startup time, min	Temperature overshoot, °C	Liquid superheat, °C	Steady-state temperature above ambient, °C
Vapor/liquid	1	1.3	0	1.9
Vapor/vapor	2.2	2.5	0	2.1
Liquid/liquid	5.5	12	8.5	1.5
Liquid/vapor	80	42	2.3	2.2

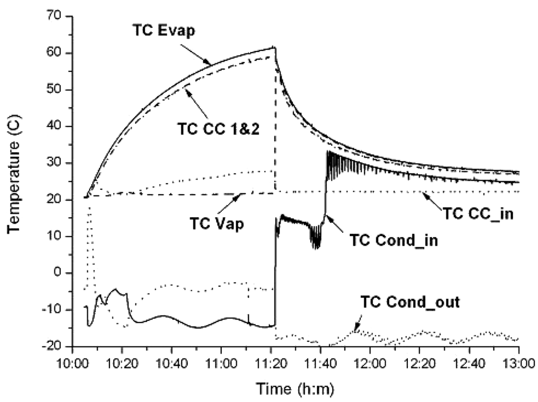


Fig. 4 Most difficult startup situation.

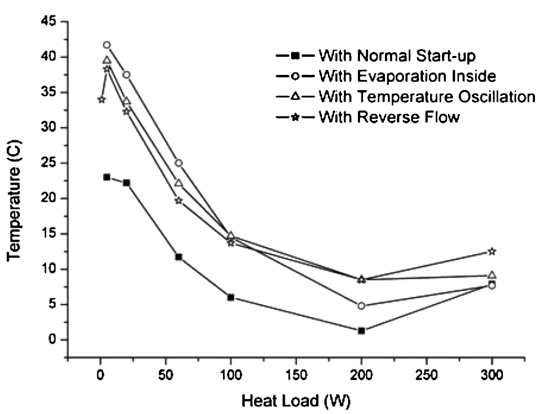


Fig. 5 Effect of unstable startups on operations.

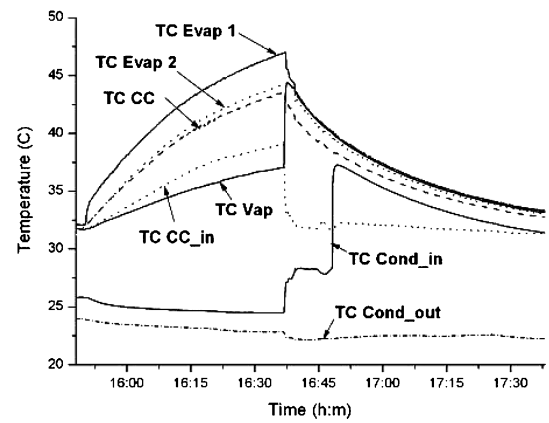


Fig. 6 Startup at 0.5-W/cm^2 local heat flux (zone 1 heating).

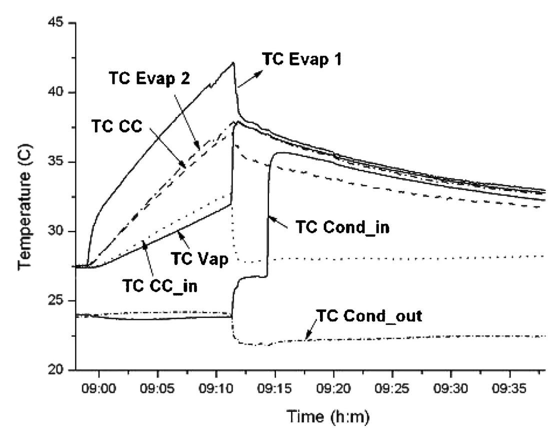


Fig. 7 Startup at 1-W/cm^2 local heat flux (zone 1 heating).

evaporator core, respectively. For example, "liquid/vapor" denotes that liquid filled the vapor grooves, whereas vapor existed in the evaporator core. From the top down in Table 2, the startup required a longer time, experienced a larger temperature overshoot, and appeared to be more difficult to achieve.

Test results indicate that the startup situation, in which liquid flooded the vapor grooves and vapor existed in the liquid core, was the most difficult. Figure 4 shows the temperature profiles. Because the vapor grooves were flooded with liquid, a liquid superheat was required to initiate the nucleate boiling. The heat leak from the evaporator to the CC was high due to the existence of vapor in the liquid core. Therefore, the temperature difference between the evaporator and the CC was difficult to attain. Both temperatures continued rising. About 80 min later, the nucleate boiling occurred in the vapor grooves (indicated by the sharp temperature rise of the vapor line inlet of TC vap). The evaporator temperature overshoot was over 42°C, which might have exceeded the allowable value. Furthermore, it is stated in Ref. 3 that when the evaporator was flooded with liquid, some peculiar unstable phenomena, including reverse flow, evaporation inside the primary wick, and temperature oscillation, occurred during startups at low heat loads. Startups with such unstable phenomena would lead to higher steady-state operating temperatures, which was the temperature hysteresis of the LHP, as shown in Fig. 5 (Ref. 3).

Test results show that when a LHP started at low heat loads without auxiliary measures, the evaporator temperature overshoot might be large and exceed the allowable value. Furthermore, some unstable startups would affect the steady-state operation of the LHP, leading to higher operating temperatures. Auxiliary measures must be taken to solve these problems.

B. Local Heating on Evaporator

In this section, effects of the local heat flux and its location on the startup, as well as the steady-state operation, have been investigated. The effects of the thermal capacity of the instruments attached to the evaporator on the assisted startup are discussed.

1. Heating on Zone 1 near Evaporator Outlet

As shown in Fig. 1, the locally heated zone 1 was near the evaporator outlet with a 6-cm^2 area. Three heat fluxes were applied in the tests. Figures 6–8 are the startup temperature profiles at a heat flux of 0.5, 1, and 1.5 W/cm^2, respectively. The temperature difference between the evaporator and the CC was small all along during the 6-W startup without auxiliary measures in Fig. 4. However, when the local zone 1 was heated, as shown in Figs. 6–8, the temperature difference between the local zone on the evaporator and the CC increased significantly. Furthermore, the temperature difference between the locally heated zone and the CC increased more quickly at a higher assistant heat flux. Although the heat leak from the locally heated zone to the CC was still high for the existence of vapor in the

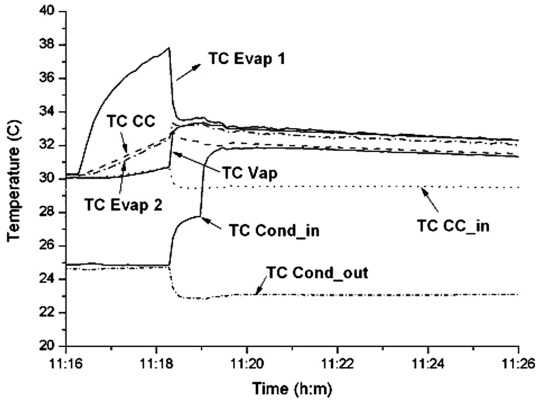


Fig. 8 Startup at 1.5-W/cm^2 local heat flux (zone 1 heating).

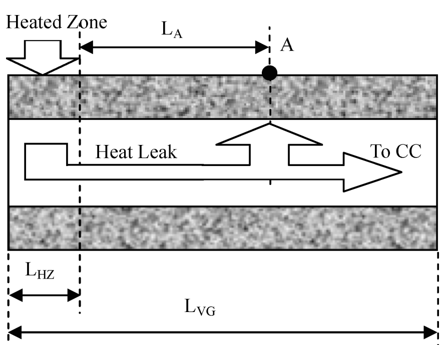


Fig. 9 Thermal resistances of primary wick.

liquid core, the temperature of the liquid in the locally heated vapor grooves increased faster than that of the working fluid in the CC for the high local heat flux, and the local liquid superheat was easily established to initiate the nucleate boiling in the local vapor grooves. The local nucleate boiling finally led to the liquid evaporation in the entirely of the vapor grooves.

Note that although the TC evap 2, which was on the unheated zone of the evaporator, was near the locally heated zone and far away from the CC, the temperature of TC evap 2 was lower than the temperature of the locally heated zone, but close to the saturated temperature of the working fluid in the CC, as shown in Figs. 6–8. The thermal analysis of the wick will be described to explain the foregoing phenomenon as follows. As shown in Fig. 9, the ratio of the axial thermal resistance of the ammonia-soaked wick from the locally heated zone to point A on the unheated zone, to the radial thermal resistance of the ammonia-soaked wick inside the unheated zone can be calculated as in Eq. (1). The ratio of the length of the unheated vapor grooves to the outer wick radius is at least seven, and then the result is that the axial thermal resistance is much higher than the radial one. The heat leak from the locally heated zone to the unheated zone through the evaporator wall and the soaked wick in the axial direction is very small for the high thermal resistance. On the other hand, because the radial thermal resistance of the unheated wick is low, part of the heat leak from the heated zone to the CC can be conducted to the unheated zone of the evaporator wall outside the wick, and, as a result, the temperature difference across the wick would be small. The temperature difference along the liquid core and CC is also very small because of the highly conductive heat link between the liquid core and the CC. Thus, the temperature profiles of the unheated zone on the evaporator were close to that of the CC during startups and lower than the locally heated zone 1.

$$\frac{R_{\text{axi}}}{R_{\text{rad}}} = \frac{4L_A \big/ \lambda_{\text{eff}}\pi\left(D_o^2 - D_i^2\right)}{\ell n(D_o/D_i)/2\pi\lambda_{\text{eff}}(L_{\text{VG}} - L_{\text{HZ}})}$$
$$= \frac{8.193L_A(L_{\text{VG}} - L_{\text{HZ}})}{D_o^2} > 57.348\frac{L_A}{D_o} \tag{1}$$

Based on the preceding analysis of the temperature profile of the unheated evaporator zone, two important conclusions for the startup in the practical applications can be drawn. The first one is that a large thermal capacity of the instruments attached to the unheated evaporator zone would be favorable for the startups assisted by the measure of local heating on zone 1. This can be explained as follows. In applications, instruments will be fixed on the unheated evaporator zone, and the heat transfer between the unheated evaporator zone and the attached instruments will affect the startup. Because the CC temperature, which is close to the unheated evaporator zone, is higher than the initial ambient temperature, heat will be conducted from the unheated evaporator zone to the instruments during startups. Actually, the heat conducted to the instruments is first transferred from the inside liquid core to the outside unheated evaporator zone, which is part of the heat leak from the locally heated zone to the CC, as shown in Fig. 9. Because part of the heat leak is conducted to the instruments, the heat leak from the heated zone to the CC is reduced, and the CC temperature will rise more slowly. Therefore, the instruments attached to the evaporator are helpful to establish the temperature difference between the locally heated evaporator zone and the CC, and a large thermal capacity of the instruments is favorable for the startup assisted by local heating on zone 1. The second conclusion is that the temperatures of the instruments attached to the evaporator would not exceed the allowable value if the locally heated zone temperature is below the allowable value. Although part of the heat leak would be transferred to the instruments, the temperature of the unheated zone (or instruments) on the evaporator would be close to the CC temperature and lower than the temperature of the locally heated zone during startups. A suitable heat flux could ensure that the temperature rises of the heated evaporator zone and the CC would not exceed the allowable value and that the high temperatures should not damage the instruments during the startup.

After the nucleate boiling in the vapor grooves, the temperatures of the evaporator and the CC began to drop until equilibrium. The final steady-state operating temperatures approximated the ambient temperatures, as shown in Figs. 6–8. Note that the ambient temperatures were close to the initial temperatures because the startups only lasted a short time. These results were in accordance with those of the startups without auxiliary measures and indicated that local heating on the evaporator near the evaporator outlet would not influence the steady-state operation of the LHP.

Table 3 lists the results of the effects of the assistant local heat fluxes on the startups. As the heat fluxes increased, the evaporator temperature overshoot decreased, the startup lasted a shorter time, and the startup appeared to be easier. The high heat flux is obviously helpful to the startup. Test results indicate that local heating on zone 1 is a feasible auxiliary measure that helps to establish the temperature difference between the local evaporator zone and the CC and the decrease of the temperature overshoot and the startup time. Furthermore, the auxiliary measure will not influence the steady-state operation of the LHP, and a large thermal capacity of the instruments attached to the evaporator is also favorable for these assisted startups. According to the applications requirements, applying a suitable heat flux on the evaporator near the evaporator outlet can completely solve the startup problems.

Note that the liquid superheats of the startups at different assisted heat fluxes were not the same as shown in Table 3. Actually, the superheats were not the same even at the same heat fluxes. The degree of the superheat required to initiate the nucleate boiling is influenced by many factors, such as the existence of the noncondensable gas, the size and distribution of the bubbles in the vapor grooves and on the inside evaporator wall, the purity of the working fluid, and so on. It is difficult to predict it in the tests.

2. *Heating on Zone 2 near CC*

As shown in Fig. 1, the locally heated zone 2 was near the CC with a 6-cm^2 area. Figs. 10 and 11 are the startup temperature profiles at a heat flux of 0.5 and 1 W/cm^2, respectively. Test results indicate that the local heating on the local zone 2 was helpful in establishing the temperature difference between the local evaporator zone and

Table 3 Effects of auxiliary measures on startups

Extra energy, W	Heat load, W	Time, min	Temperature overshoot, °C	Liquid superheat, °C	Steady-state temperature above ambient, °C
			No measures		
——	6	80	42	2.3	2.2
			Local heating		
3 (0.5W/cm^2)	——	47	18	3	1.1
6 (1.0W/cm^2)	——	12.4	14.7	4.2	1.7
9 (1.5W/cm^2)	——	2.1	7.9	5.5	1.9
			TEC cooling		
4.8	——	13.9	7.1	6	−7.2
			PCM		
——	3	195	21	1.9	2.8
——	6	20	12.9	3.1	2.3
——	12	9.8	16	5.8	2.4

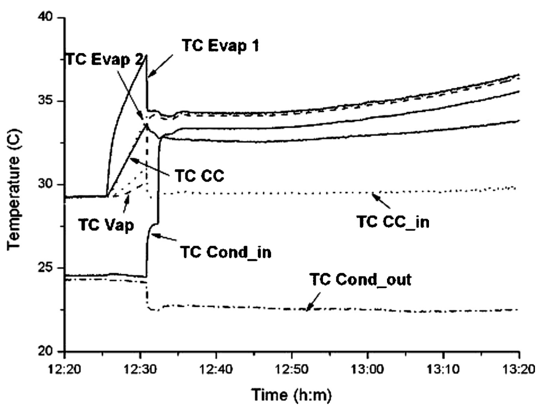


Fig. 10 Sustaining temperature increase.

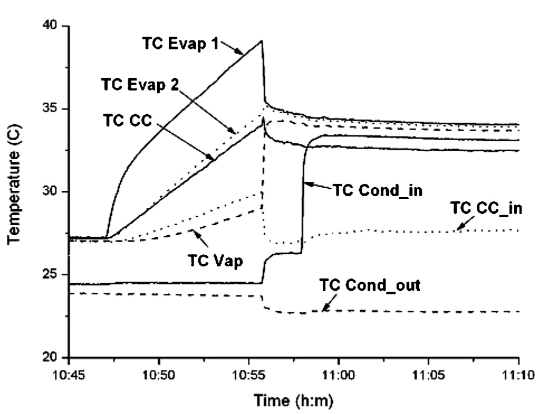


Fig. 11 Operating at higher temperature.

the CC as well as in reducing the startup time and the evaporator temperature overshoot. However, local heating near the CC affected the steady-state operating temperatures after the startup. Test results indicate that local heating on the evaporator near the CC would lead to either a sustaining operating temperature rising (shown in Fig. 10) or to a higher operating temperature after the startup. (As shown in Fig. 11, the steady-state temperature was about 7.0°C above the ambient temperature, whereas it was only about 2.2 °C for startups at the same heat load with no auxiliary measures.)

When the local zone 2 near the CC was heated, heat would be easy to be transferred to the CC, leading to a higher heat leak from the evaporator to the CC. The increased heat leak changed the energy equilibrium of the CC, and finally increased the steady-state operating temperatures. When the heat exchange between the CC and the ambient temperature is neglected, the equation of the energy conservation of the CC can be obtained as

$$\dot{Q}_{\mathrm{HL}} = \dot{Q}_{\mathrm{sub}} = \dot{m}C_{p,\mathrm{liq}}(T_{\mathrm{CC\_sat}} - T_{\mathrm{CC\_in}}) \tag{2}$$

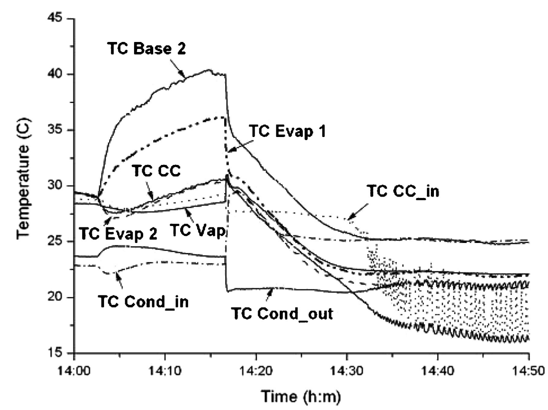


Fig. 12 Startup with TEC system.

According to Eq. (2), the higher heat leak from the evaporator to the CC will lead to the saturated temperature increase in the CC. Because the saturated temperature difference is related to the pressure difference in accordance with the Clausis–Clapeyron relation (3), the temperature difference between the evaporator and the CC is very small. When the heat leak is increased, the temperature of both the CC and the evaporator will increase until Eq. (2) is satisfied again. If the heat leak could not be balanced by the returned subcooled liquid, the temperatures of the evaporator and the CC would keep rising,

$$\Delta T = (T_{\mathrm{sat}} v_{\mathrm{fg}} / h_{\mathrm{fg}}) \Delta p \tag{3}$$

Test results indicate that although local heating on zone 2 near the CC was helpful in establishing the temperature difference between the local evaporator zone and the CC and reducing the temperature overshoot and the startup time, it increased the final steady-state operating temperatures and deteriorated the heat transfer performance of the LHP. The locally heated zone 2 is not ideal for assistant heating.

C. Cooling CC by TEC

As shown in Fig. 2, during the startup, the bottom cold side of the TEC absorbed heat, cooling the CC, and the heat rejected from the hot side of the TEC was conducted to the local zone on the evaporator through a heat bridge, heating the local evaporator zone. The sum of the heat absorbed by the cold side of the TEC and the TEC input power is equal to the amount of the heat that should be rejected from the hot side of the TEC, and the equation of the energy conservation is given as

$$\dot{Q}_{\mathrm{TEC\_rej}} = \dot{Q}_{\mathrm{TEC\_abs}} + P_{\mathrm{TEC}} \tag{4}$$

Figure 12 shows the temperature profiles of the startup assisted by the TEC system without extra heat load on the evaporator. The TEC input power during the startup was 4.8 W (4V×1.2A). When the TEC was operating, the CC temperature kept decreasing first, whereas TC evap 1 on the evaporator near the TEC base 2 increased for the heat conducted from the TEC hot side through the heat bridge. As time went on, the temperature difference between the TC evap 1 and TC CC increased. When the CC temperature got to the lowest degree, the temperature difference between the TC evap 1 and TC CC (namely, the local liquid superheat) increased to 4°C, but the local nucleate boiling was not yet initiated. Because vapor existed in the liquid core during the startup, the heat leak from the evaporator to the CC could not be neglected. When the heat leak could not be balanced by the cooling of the TEC cold side, the CC temperature began to rise. However, the temperature difference between TC evap 1 and TC CC still kept increasing. When the TC evap 1 increased from 29 to 36°C, the local liquid superheat reached 5°C, and the local nucleate boiling in the vapor grooves inside the TEC base 2 was initiated. After vapor was generated in the local vapor grooves near the CC, the local pressure rose sharply, and the remaining liquid in the vapor grooves would be pushed to the evaporator outlet. The final phase change mode was the surface evaporation in the vapor grooves outside the wick.

During the startup, TC base 2 was higher than TC evap 1 in transfering the heat from the TC TEC base to the evaporator. After the nucleate boiling, the temperature difference between the TEC base 2 and the locally heated evaporator zone was still about 4°C, which was too large. Measures can be taken to reduce the contact thermal resistance.

Note that, during the startup, TC evap 1 continued rising because it absorbed the heat from the TEC hot side, whereas TC evap 2 on the unheated evaporator zone, which was only 1 cm away from TC evap 1, was much lower than TC evap 1, but close to the CC temperature all along. This can be explained by the fact that the axial thermal resistance of the ammonia-soaked wick from TC TEC base 2 to TC evap 2 on the unheated zone is much higher than the radial thermal resistance of the ammonia-soaked wick inside the unheated zone, which is discussed in the preceding subsection "Heating on Zone 1 Near the Evaporator Outlet." Before the nucleate boiling, TC evap 2 was lower than TC evap 1 and close to the TC CC, whereas after the nucleate boiling vapor was generated in the vapor grooves. Both TC evap 1 near the TEC base 2 and TC evap 2 on the unheated evaporator were at saturated vapor temperatures. In practical applications, the instruments will be attached to the unheated evaporator zone. As stated earlier, heat would be conducted from the unheated evaporator zone to the instruments, which is part of the heat leak from the evaporator under the TEC base 2 to the CC. Then, the heat leak to the CC would be decreased, and the CC temperature rise would be slower. Therefore, a large thermal capacity of the instruments would also be favorable for the TEC-assisted startup.

When no auxiliary measures are taken, the final steady-state operating temperatures after startups were close to the ambient temperature at low heat loads, as shown in Fig. 13. When the TEC system was used the TEC would continue cooling the CC after startups, and the operating temperatures would be lower than those at startups without auxiliary measures. If the TEC continued operating, the energy equation of the CC, namely, Eq. (2), could be rewritten as

$$\dot{Q}_{\mathrm{HL}} = \dot{Q}_{\mathrm{sub}} + \dot{Q}_{\mathrm{TEC\_abs}} = \dot{m} C_{\mathrm{p\_liq}} (T_{\mathrm{CC\_sat}} - T_{\mathrm{CC\_in}}) + \dot{Q}_{\mathrm{TEC\_abs}} \quad (5)$$

Because the liquid core was flooded with the low-conductive liquid after the startup, most of the heat, which was conducted to the evaporator through the heat bridge, was absorbed by the evaporation of the liquid working fluid inside the evaporator, and less was leaked back to the CC. Therefore, the heat leak increase caused by the TEC system on the left-hand side of Eq. (5) could be neglected in a simplified analysis. When Eq. (5) was compared with Eq. (2), there was extra item of the heat absorbed by the TEC cold side, $Q_{\mathrm{TEC\_abs}}$, on the right-hand side of Eq. (5), which was the reason that the saturated temperature in the CC finally decreased. The CC temperature finally dropped below the ambient temperature (Fig. 12). As shown in Fig. 13, test results indicate that, at the heat loads in which the CC was in a saturated state, the steady-state operating temperatures were all lower than those without the TEC on the CC.

When no auxiliary measures were taken, the typical V operating temperature curve could be observed when the sink temperature was lower than the ambient temperature. As Fig. 13 shows, as the heat loads increased, the steady-state operating temperatures (the upper temperature curve without the TEC) would first drop and then continue rising. This can be explained as follows. At low heat loads, the CC is in a saturated two-phase state, and the operating temperatures are determined by the thermal equilibrium in the CC. As the heat load increases, the subcooling of return liquid increases, thus, the utilized heat transfer area of the condenser will increase and the operating temperatures will decrease due to the thermal equilibrium in the saturated CC. After the CC is filled with liquid, the condenser is fully utilized, and the operating temperatures begin to continue increasing as the heat load increases.

However, if the TEC continued operating after the startup, the operating temperatures were always lower than those without the TEC when the CC was in a saturated state. As shown in Fig. 13, the upper typical V operating temperature curve is for the operation without the TEC, whereas the nearly linear one is for the operation with the TEC system. After startup, the TEC continued working and decreased both the saturated temperature and pressure in the CC, thus, the liquid was pushed into the CC. The CC would always be flooded with liquid, and the condenser was fully utilized at any heat loads. When the CC is flooded with liquid, or in the subcooled state, the operating temperatures will be determined by the effective two-phase heat transfer area in the condenser instead of the thermal equilibrium in the CC. Furthermore, the LHP with the TEC system will always operate under the constant conductance mode at any heat load. According to Eq. (3), the temperature difference between the evaporator and the condenser is small. When the heat leak from the evaporator to the CC across the primary wick is neglected, the operating temperatures with the TEC system operating after startups can be calculated as

$$\dot{Q}_{\mathrm{loop}} = U_{\mathrm{cond}} A_{\mathrm{cond}} (T_{\mathrm{cond}} - T_{\mathrm{sink}}) \Rightarrow T_{\mathrm{cond}} = T_{\mathrm{sink}} + Q_{\mathrm{loop}} / U_{\mathrm{cond}} A_{\mathrm{cond}} \quad (6)$$

As Table 3 shows, the TEC system was obviously helpful in establishing the temperature difference between the local evaporator zone and the CC and reducing the temperature overshoot and the startup time. Furthermore, if the TEC continued operating after the startup, the condenser would be fully utilized at any heat loads. Because the temperature difference between the evaporator and the condenser was small and the condenser was always fully utilized, the LHP with the TEC system operated more efficiently, like a conventional heat pipe.

The temperature oscillation at the CC inlet was often observed after the TEC-assisted startups in the tests. However, the evaporator temperature did not oscillate very much, as shown in Fig. 12. Because the cooling effect provide by the TEC on the CC is determined by the temperature difference between the hot and cold sides of the

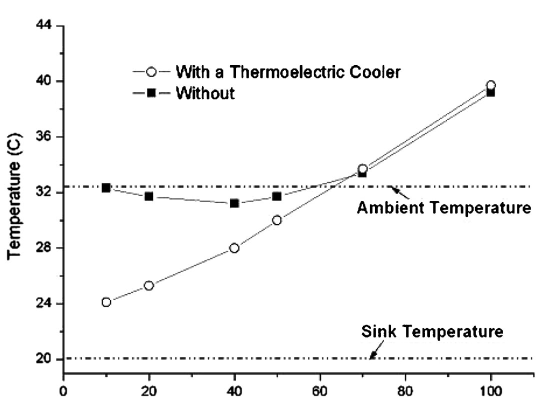


Fig. 13 Effect of TEC on steady-state operation.

TEC, the cooling of the TEC on the CC will always vary with the temperature differences between the CC and the local evaporator zone inside the TEC base 2. Therefore, the temperature oscillation might be caused by the unstable cooling amount of the TEC on the CC. Further experiments and theoretical investigation are needed to explain this temperature oscillation.

D. Fixing PCM Containers on CC

In this section, the feasibility of the auxiliary measure of fixing PCM containers on the CC will be investigated, as will its effects on the steady-state operation of the LHP. Furthermore, the effects of different heat loads on the PCM-assisted startup are also investigated.

Figure 14 shows the temperature profiles of a 6-W startup with the PCM containers on the CC. Note that, as shown in Fig. 4, the temperature difference was small all along during the startup without the PCM container at the same heat loads. However, in the PCM assisted startup, as the 6-W heat load was applied to the evaporator, the temperature difference between the evaporator and the CC increased gradually for the large thermal capacity of the PCM containers on the CC. When the CC temperature increased to the PCM melting point, the solid PCM began to melt, absorbing heat from the CC, and the CC temperature was maintained near the melting point. At the same time, both the evaporator temperature and the temperature difference between the evaporator and the CC still increased, but more slowly than before. About 10 min after the CC temperature reached the PCM melting point, the temperature difference between the evaporator and the CC required to initiate the nucleate boiling was obtained, and vapor was generated in the vapor grooves. Compared with the 6-W startup without PCM containers on the CC (Fig. 4), both the temperature overshoot and the startup time decreased, and the liquid superheat was much easier to obtain in the startups with PCM containers on the CC.

Figure 15 shows the temperature profiles of a 12-W startup with the PCM containers on the CC. As the heat load was increased from 6 to 12 W, the temperature difference between the evaporator and the CC increased more quickly. When the CC temperature rose to the PCM melting point and was maintained at a constant value, the required temperature difference between the evaporator and the CC was obtained immediately. Test results indicate that, for the PCM assisted startups, a higher startup heat load led to a shorter startup time but resulted in a higher evaporator temperature overshoot.

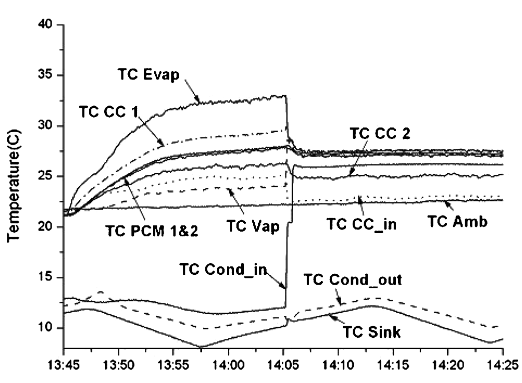


Fig. 14 6-W startup assisted by PCM.

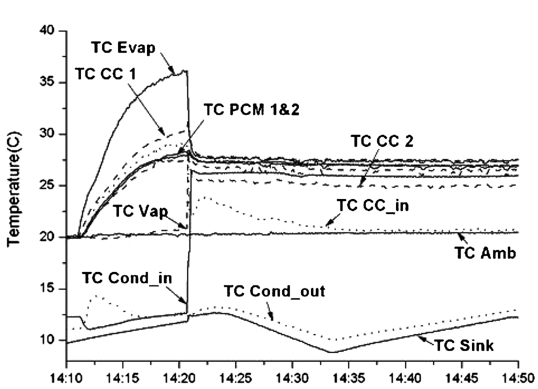


Fig. 15 12-W startup assisted by PCM.

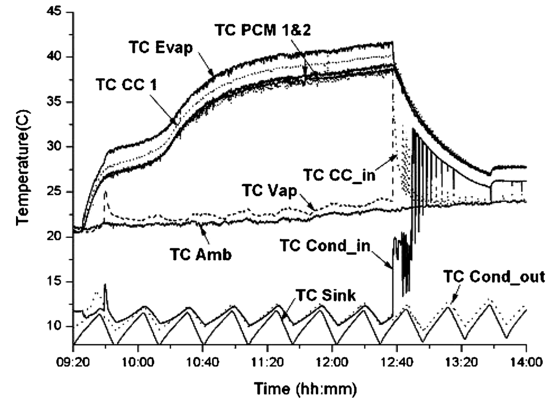


Fig. 16 3-W startup assisted by PCM.

Figure 16 shows the temperature profiles of a 3-W startup with the PCM containers on the CC. As the heat load was decreased to 3 W, the temperature difference between the evaporator and the CC increased more slowly and was not large enough to initiate the nucleate boiling even when the CC temperature was maintained at the PCM melting point. Note that TC CC1 also stayed for a time at the PCM melting point, but it could not be easily recognized because the time abscissa of Fig. 16 has a larger scale than Figs. 14 and 15. After all of the PCM melted, the CC temperature could not be maintained near the melting point any longer, and both the evaporator and CC temperatures began to rise again. About 3 h later, nucleate boiling with a small superheat occurred at about 41°C. Although the LHP did not start up when the CC temperature was maintained at the PCM melting point, the PCM delayed the evaporator temperature rise. In practical applications, delaying the evaporator temperature rise is valuable because it allows time for the LHP to wait for the coming of high heat loads, at which the startup will be much easier.

During the PCM assisted startups, part, or all, of the PCM melted and absorbed the heat leak from the evaporator to the CC, maintaining the CC temperature near the melting point. After the startup, the absorbed heat could not be rejected immediately, and the PCM would still maintain the temperature of the CC and the evaporator near the melting point for some time, as shown in Figs. 14–16. The special phenomenon of the PCM subcooling during its freezing process after the startup is also shown in Fig. 16. The temperatures of the evaporator and the CC would be near the melting point until the latent heat of the melted PCM in the containers was balanced by the subcooling of the returned liquid, then the returned liquid would begin to absorb the sensible heat of the PCM, and the temperatures of the evaporator and the CC would drop to the final steady-state operating temperature. As Eq. (7) shows, because the subcooling of the returned liquid at low heat loads was low, the time τ at which the LHP operated near the melting point, which was, namely, the time at which the PCM rejected its latent heat, would be very long, and the tested result was over 4 h in the 12-W startup. As the heat load increased, the returned liquid subcooling would increase, and the time that the LHP operated near the melting point would decrease

$$Q_{\mathrm{PCM}} = m_{\mathrm{PCM\_liq}}\, r_{\mathrm{PCM}} = \int_0^{\tau} \dot{Q}_{\mathrm{sub}}\,\mathrm{d}t \tag{7}$$

After vapor was generated in the vapor grooves, the outside evaporator and the CC were both in the saturated state. Then, the evaporator temperature was related to the CC temperature according to the Clausis–Clapeyron relation, and their temperature difference was very small.

As is stated earlier, the PCM containers on the CC will influence the operation for some time after startup. However, when the CC is in the two-phase state, the PCM containers on the CC can maintain the operating temperatures of the LHP within a small range, and it appears that fixing the phase change device on the CC is a good passive method to control the temperature of the instruments attached to the LHP evaporator within the desirable small range. If the heat source (instrument) produces intermittent heat loads, the LHP will be in a cycle of startup, operation, and shutdown, and the PCM will not only help the startups, but also realize the thermal control within a small range for some time. During the startup, the PCM will absorb heat to assist the startup or delay the temperature rise of the CC and the evaporator, whereas after the startup, the latent heat of the PCM will be rejected to the CC to maintain the CC and the evaporator temperatures near the melting point. In addition, if fluctuant heat loads are applied on the LHP evaporator, the phase change device on the CC can also realize the precise thermal control within a small temperature range and improve the reliability of the LHP. At low heat loads when the CC is not flooded with liquid, the temperatures of the evaporator and the CC might drop below the PCM melting point, and the PCM will reject its latent heat to maintain the CC temperatures (namely, the LHP operating temperatures) near the melting point. In contrast, at high heat loads when the CC is filled with liquid, the temperatures of the evaporator and the CC might increase above the PCM melting point, and the PCM will absorb the heat from the CC to delay the CC temperature rise.

Table 3 lists the test results of the PCM-assisted startups at different heat loads. Test results indicate that the PCM containers on the CC are obviously helpful to establish the temperature difference between the evaporator and the CC and reduce the evaporator temperature overshoot and the startup time. Furthermore, the startup will last a longer time at a lower heat load. When the heat load is too low to establish the required the liquid superheat to initiate the nucleate boiling, the PCM can only delay the temperature rise of the evaporator and the CC and gain additional time for the LHP to wait for the coming of high heat loads. In addition to the ability to assist in the startup of the LHP, the PCM containers on the CC can realize the precise thermal control of the operating temperature and improve the reliability of the LHP.

IV. Conclusions

The two active auxiliary measures, including local heating on the evaporator and cooling of the CC by a TEC and the passive auxiliary measure of fixing PCM containers on the CC, are helpful in establishing the temperature difference between the evaporator, or local evaporator zone, and the CC, namely, the required liquid superheat, and reduce the evaporator temperature overshoot as well as the startup time.

Test results indicate that local heating near the evaporator outlet will not affect the steady-state operation of the LHP, whereas local heating near the CC will lead to either a sustained operating temperature rise or a higher operating temperature after startup. The continued operation of the TEC after startup will decrease the operating temperatures at the heat loads when the CC is not filled with liquid, and the typical V operating temperature curve will be changed to a nearly linear one. The thermal capacity of the instruments attached to the unheated evaporator zone is favorable for the two active assisted startups. The two active auxiliary measures for startups are simple and feasible, but require extra power input and a control system.

Fixing PCM containers on the CC is a complete passive auxiliary measure with no moving parts, control systems, or power input. This method can realize the precise thermal control of the operating temperature and improve the reliability of the LHP. However, at too low heat loads, the PCM cannot help to initiate the nucleate boiling immediately, and can only delay the evaporator temperature rise and gain additional time for the LHP to wait for the coming of higher heat loads, which are more favorable for startups. Furthermore, the PCM containers on the CC will still maintain the operating temperatures of the LHP within a very small range for some time after the startup.

Acknowledgment

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