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Investigation of Startup Behaviors of a Loop Heat Pipe

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The results of ground experiments on startup behaviors of a loop heat pipe are presented. One objective is to investigate the effects of working conditions on the startup of loop heat pipes. Startup behaviors as functions of various parameters, including the vapor/liquid distribution in the evaporator, startup heat load, sink temperature, and adverse elevation are described and explained. The physical process of startup is described, and the explanation that pressure transfer leads to the saturated temperature rise in the compensation chamber during startup is discussed. The other objective is to investigate the effect of startup on the steady-state operation of loop heat pipes. Test results indicate that evaporation inside the wick tends to occur at low startup heat loads when the evaporator, including the vapor grooves and the evaporator core, is flooded with liquid. Some peculiar phenomena, including evaporation inside the wick, temperature oscillation at the condenser inlet, and reverse flow equilibrium, which lead to higher operating temperatures, were observed during startup.

Nomenclature

C_p	=	specific heat at constant pressure, $J/kg \cdot K$
d	=	differential
E	=	thermodynamic energy, J
L	=	length, m
$\dot{m}$	=	mass flow rate, kg/s
p	=	pressure, Pa
Q	=	heat load or power, W
Q_{app}	=	heat applied to evaporator
Q_{HL}	=	heat leak from evaporator to compensation chamber
Q_{loop}	=	heat absorbed by liquid evaporation in evaporator
Q_{sub}	=	amount of liquid subcooling at inlet of compensation chamber
r	=	latent heat of working fluid, J/kg
T	=	temperature, K
$(UA/L)_{\mathrm{liq\text{-}amb}}$	=	thermal conductance per unit length from fluid to ambience, $W/m \cdot K$
v	=	specific volume, m^3/kg
Δ	=	difference
τ	=	time, s

Subscripts

amb	=	ambience
CC	=	compensation chamber
fg	=	difference of thermal properties between saturated fluid and gas
in	=	inlet of pipe
liq	=	liquid
out	=	outlet of pipe
pipe	=	liquid or vapor pipe
sat	=	saturation or saturated
sub	=	liquid subcooling at inlet of compensation chamber
vap	=	vapor

Introduction

LOOP heat pipes (LHPs) are two-phase heat transfer devices that utilize the evaporation and condensation of a working fluid to transfer heat. An LHP consists of an evaporator, a condenser, a compensation chamber (CC), and vapor and liquid lines. Figure 1 shows the detailed structure of the evaporator and the CC of an LHP. The basic principles of an LHP are introduced in Refs. 1 and 2. When an LHP starts to work, liquid is vaporized in the vapor grooves, and the menisci formed in the wick develop capillary pressure to push the vapor through the vapor line to the condenser, where heat is rejected and the fluid condenses. Then liquid is pushed back through the liquid line to the evaporator. It is necessary for a successful startup that evaporation occur in the vapor grooves instead of in the liquid core, which will initiate the circulation of the working fluid in the correct direction.

Whenever heat loads are applied to the evaporator, LHPs are able to self-start without preconditioning. However, self-start does not imply an instant start or a quick start.[1] Sometimes this process may take a long time. During the startup, the evaporator temperature may rise above the final steady-state operating temperature and in some cases may even exceed the maximum allowable temperature. It is important to characterize the startup and its effect on the steady-state operation of LHPs and to take measures to improve the reliability of startup. One objective of this paper is to investigate the effects of different working conditions on the startup of LHPs. This paper describes and explains the physical process of four startup cases depending on the two-phase state in the evaporator and summarizes the startup difficulty by using the three startup parameters, namely, the temperature overshoot (the evaporator temperature rise during the startup), the liquid superheat in the vapor grooves (the temperature difference between the liquid in the vapor grooves and the saturated vapor or liquid in the CC), and the startup time (from the beginning to the moment that the evaporator temperature starts to drop). Efforts are also made to investigate the effects of startup heat

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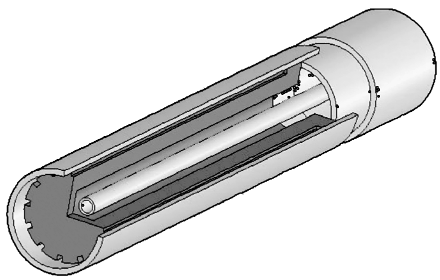

Fig. 1 Construction of evaporator and CC.

load, sink temperature, and adverse elevation on the startup performance of LHPs. Although Refs. 1–3 present investigations on startup characteristics of LHPs, in this paper comprehensive tests were conducted to validate the explanation and analysis, and more detailed startup physical processes, more reasonable explanations, and some new phenomena of startup behaviors are presented.

The other objective of this paper is to investigate the effects of startup on the steady-state operation of LHPs. Although startups in different situations experience different evaporator temperature overshoots, liquid superheats, and startup time, they ultimately should not affect the steady-state operation of LHPs. However, startups are not always successful in tests if a successful one has been defined to have the ability to lead LHPs to operate at normal temperatures. Actually, some startups lead to higher steady-state operating temperatures, and some even fail to start. Cheung et al.[4] have observed temperature hysteresis, which refers to the operating temperature change for seemingly identical operating conditions. Temperature hysteresis takes place at low-power operation after a sudden power decrease, and it is known that the two-phase dynamics in the evaporator core has a great influence on the heat leak from the evaporator to the CC. As a consequence, the steady-state temperatures can be shifted. Cheung et al.[4] also describes the phenomenon that high liquid superheat during startup leads to higher operating temperatures. It is hypothesized that the higher operating temperatures are caused by the increased heat leak after the penetration of vapor into the evaporator core at high superheat startups. Furthermore, in the tests reported in this paper, it is found that evaporation inside the wick tends to occur at low startup heat loads when the evaporator, including the vapor grooves and the evaporator core, is flooded with liquid. Some peculiar phenomena during startup were observed, including evaporation inside the wick, temperature oscillation at the condenser inlet, and reverse flow equilibrium. Test results indicate that such startups lead to higher steady-state operating temperatures. Descriptions and explanations of these phenomena are presented.

The major results and conclusions presented in this paper can be reproduced. However, the peculiar phenomenon of temperature rise of the CC with step increment appeared only twice during the tests that took about three months.

Test Setup

The test device is an ammonia/stainless-steel LHP with a nickel wick built by the Ural Branch of the Russian Academy of Sciences. Table 1 shows the geometric parameters of the components, where o.d. and i.d. represent the outside and inside diameters, respectively. No secondary wick is used for the ground test, and so the test LHP has just a bayonet in the evaporator core.

The condenser line was mounted on an aluminum cold plate with imbedded coolant channels, which were cooled by a low constant temperature trough. Heat input to the evaporator was provided by film heaters, which were attached directly on the evaporator symmetrically. The entire loop was thermally insulated with sponge, which had the thermal resistant of about 10 K/W per unit length, to reduce the parasitic heat loss. There were 20 copper/constantan (type T) thermocouples (TCs) used to monitor the temperature profiles of the loop. Figure 2 shows the TC locations. In addition, TC 19 was for ambient temperature and TC 20 for sink temperature. A data acquisition system, which consisted of a data logger linked to a personal computer and IMPview software, was used to display and save the data.

Table 1 Geometric parameters of the test LHP

Components	Dimensions
o.d./i.d. × length of evaporator, mm	$\Phi 18/16 \times 175$
o.d./i.d. × length of vapor line, mm	$\Phi 3/2.2 \times 2500$
o.d./i.d. × length of liquid line, mm	$\Phi 3/2.2 \times 2800$
o.d./i.d. × length of condenser, mm	$\Phi 3/2.2 \times 2000$
o.d./i.d. × length of bayonet, mm	$\Phi 3/2.2 \times 230$
Volume of CC, ml	20
Charge of working fluid, g	29.9
Maximum radius, μm	1.1
Porosity, %	55
Wick permeability, m^2	$>5 \times 10^{-14}$
o.d./i.d. × length, mm	$16/8 \times 125$
Height × width of grooves, mm	1×1

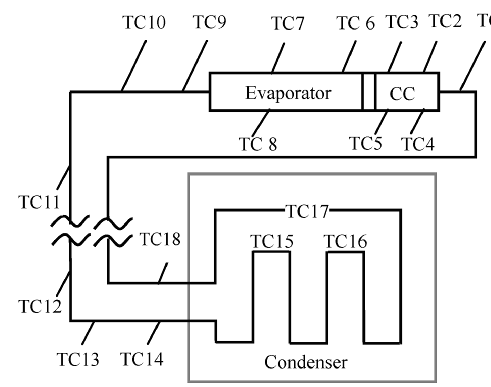

Fig. 2 LHP schematic with thermocouple locations.

Test Results and Discussion

The test results and discussion are presented in two sections: the effects of working conditions on the startup of LHPs and the effects of startup on the steady-state operation of LHPs.

Effects of Working Conditions on the Startup

The effects of working conditions, including the vapor/liquid distribution in the evaporator, sink temperature, startup heat load, and adverse elevation on the startup of LHPs are presented in the following four subsections.

Vapor/Liquid Distribution in the Evaporator

The four startup cases of the LHP depend on the two-phase compositions in the evaporator core and evaporator grooves.[1] The four different vapor/liquid distributions can be obtained as follows. In case 1, the loop is laid vertically with the CC above the evaporator, as shown in Fig. 3a, for more than 12 h. Then the vapor grooves and the evaporator core will be flooded with liquid due to the gravity. In case 2, the LHP is started as described for case 1 to operate for nearly 1 h, and the loop is shut down. Then vapor will exist in the vapor groove, whereas the evaporator core will be still filled with liquid. In case 3, the loop is laid horizontally with the CC and the evaporator on the same horizontal plane, as shown in Fig. 3b, for more than 12 h. Because the CC has a larger thermal capacity than the vapor grooves, the temperature of the CC decreases more slowly than the evaporator after shutdown. Thus, vapor will show a tendency to exist in the CC rather than in the vapor grooves, and liquid will fill the vapor groove. In a horizontal position, the interface of the vapor/liquid will exist in the evaporator core as decided by the charge quantity of the working fluid. In case 4 the LHP is started as described for case 3 to operate for nearly 1 h and the loop is shut down. Then vapor will exist in the vapor groove, whereas the vapor/liquid interface still exists in the evaporator core. The following four startup cases were all conducted with a 5-W heat load, a sink temperature of 16 ± 2°C, and an ambient temperature of 22 ± 2°C.

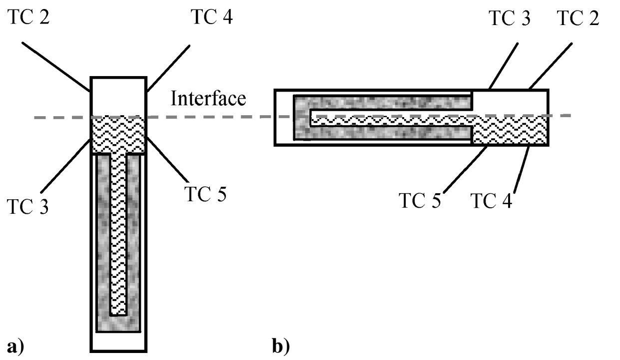

Fig. 3 Position of evaporator and CC in tests: a) vertical position and b) horizontal position.

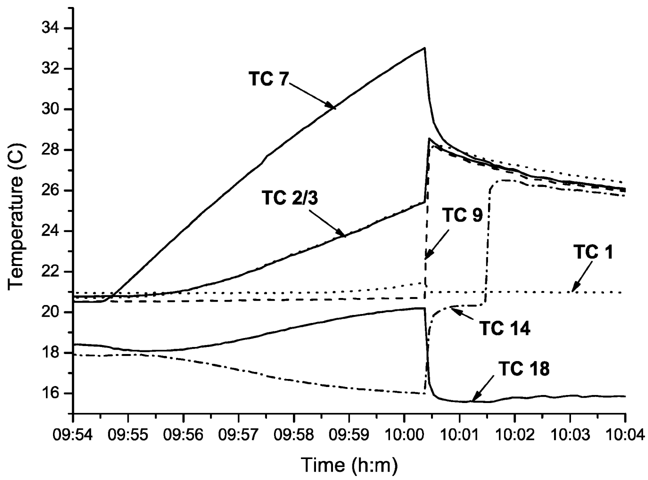

Fig. 4 Startup case 1.

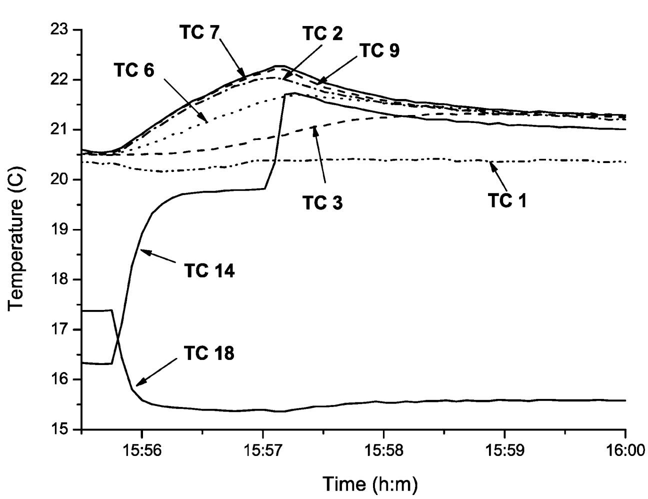

Fig. 5 Startup case 2.

When the LHP is laid vertically as shown in Fig. 3a, startup cases 1 and 2 can be obtained. These two cases of startup have the same characteristics in that the evaporator core is filled with liquid and that the heat leak from the evaporator to the CC by the heat transfer modes of case conduction and fluid convection is small. In case 1, the vapor grooves are flooded with liquid, and Fig. 4 shows the temperature profiles of this startup case. In case 2, vapor exists in the vapor groove, and Fig. 5 shows the temperature profiles.

In case 1, shown in Fig. 4, as the heat was applied on the evaporator, TC 7 on the evaporator rose to acquire the required superheat, whereas TCs 2 and 3, respectively, on the vapor and liquid zone of the CC (Fig. 3) rose a little for the small heat leak across the wick, which was driven mainly by the case conduction and the fluid convection. The liquid superheat increased gradually until the nucleating boiling was initiated. Then TC 7 on the evaporator dropped sharply for the evaporation of fluid, whereas TC 9 at the outlet of the evaporator rose immediately, indicating that the vapor generated in the vapor grooves had been pushed to the vapor line. At the same time, the TCs 2 and 3 on the CC wall rose together. Because at the incipience of boiling vapor could penetrate the wick into evaporator core if the superheat was high enough,[4] the phase change heat transfer between the evaporator core and the CC caused by the existence of vapor in evaporator core would lead to the sharp temperature increase of TCs 2 and 3. This explanation was proved to be true by the temperature rise of TC 6, TC 2, and TC 3 together after the nucleate boiling. In addition, test results indicate that condensation of the vapor in the CC due to the pressure increase, which is caused by the nucleating boiling of liquid in the vapor grooves, can also lead to the temperature increase of the CC. This will be discussed and explained in detail in startup case 2. After the vapor was generated in the vapor grooves, TC 14 at the condenser inlet rose, whereas TC 18 at condenser outlet dropped, indicating that the fluid began to circulate. The subcooled liquid at ambient temperature returned to the evaporator to cool the CC, and then TC 7 on the evaporator and TCs 2 and 3 on the CC decreased gradually until the loop reached equilibrium.

In case 2, shown in Fig. 5, as the heat load was applied to the evaporator, TC 9 at the inlet of the vapor line rose immediately, indicating that vapor was generated due to the existence of the vapor/liquid interface in the vapor grooves. The temperature rise of TC 14 at the condenser inlet and drop of TC 18 at the condenser outlet indicated that the loop had started to circulate. Although the return liquid had begun to cool the CC, both TC 7 on the evaporator and TC 2 on the CC still rose until their temperature difference increased to a certain value. Then the two temperatures decreased to equilibrium.

Note that the main cause of the saturated temperature rise of the CC is not the heat leak from the evaporator to the CC, but the pressure change in the CC. The locations of TC 3, TC 7, and TC 2 are shown in Fig. 3a. TC 3 was on the liquid zone of the CC, whereas TC 2 was on the vapor zone. TC 3 was between TC 7 and TC 2. TC 3 did not rise together with TC 2 and TC 7 after the nucleate boiling. If the heat leak from the evaporator to the CC led to the temperature rise of TC 2 on the CC, the temperature value of TC 3 should have been lower than TC 7 and higher than TC 2. However, TC 2, which was located farther away from TC 7 than TC 3, was higher than TC 3 throughout the startup. This phenomenon proves that the heat leak from the evaporator to the CC should not be the main cause of the saturated temperature rise in the CC. This can be explained as follows. In a time of $\mathrm{d}\tau$, the energy equation of the vapor in the CC is given as

$$\frac{\mathrm{d}E_{\mathrm{CC,vap}}}{\mathrm{d}\tau} = \mathrm{d}Q_{\mathrm{CC,vap}} - p\,\mathrm{d}V_{\mathrm{CC,vap}} \tag{1}$$

The item $-p\,\mathrm{d}V_{\mathrm{CC,vap}}$ in Eq. (1) is the CC energy change caused by the compression or expansion of the vapor in the CC. In a steady-state analysis, because the saturated pressure in the CC is in equilibrium and remains constant, no volume change of the vapor in the CC will occur, and $-p\,\mathrm{d}V_{\mathrm{CC,vap}}$ will be zero. However, in a transient state analysis, the vapor fluid in the CC may expand or be compressed, and $-p\,\mathrm{d}V_{\mathrm{CC,vap}}$ in Eq. (1) must be considered. During the startup shown in Fig. 5, vapor is generated in the vapor grooves, and the vapor temperature increases at the beginning. The saturated temperature rise of the vapor results in the saturated pressure rise in the vapor groove. The increasing pressure is transferred through the external loop to the CC, leading to an absolute pressure rise in the whole loop. At the increased pressure, the vapor in the CC is compressed and becomes superheated, and so part of the vapor condenses. The latent heat, which is produced by the condensed vapor, then heats up the liquid, leading the working fluid in the CC to be saturated again at a higher saturated temperature corresponding to the increased saturated pressure. Then a conclusion can be drawn that the pressure rise that leads to the condensation of working fluid in the CC is the main cause of the saturated temperature rise in the CC during the startup when liquid floods the evaporator core.

The saturated temperature difference across the wick is related to the pressure difference in accordance with the Clausis–Clapeyron relation, namely, Eq. (2) (see Ref. 2), where ΔT is the saturated temperature difference and Δp is the saturated pressure drop and where vapor is pushed into the condenser after the required temperature difference is obtained. Note that Eq. (2) is used only for a simplified analysis to explain the results for the working fluid and is actually not pure. Then the temperatures of the CC and the evaporator begin to decrease until they reach equilibrium. The temperature drop between the CC and the evaporator is decided by the negative value of Eq. (3):

$$\Delta p = \frac{h_{\mathrm{fg}}}{T_{\mathrm{sat}} v_{\mathrm{fg}}} \Delta T \tag{2}$$

$$\frac{\mathrm{d}E_{\mathrm{CC}}}{\mathrm{d}\tau} = Q_{\mathrm{HL}} - Q_{\mathrm{sub}} = Q_{\mathrm{HL}} - \dot{m} C_{p,\mathrm{liq}} \Delta T_{\mathrm{sub}} \tag{3}$$

When the LHP is laid horizontally as shown in Fig. 3b, startup cases 3 and 4 can be obtained. These two cases of startup have the same characteristic that vapor exists in the evaporator core and the heat leak from the evaporator to the CC is high enough for a phase change to take place in the evaporator core. The heat transfer coefficient of evaporation/condensation is often an order of magnitude greater than convection/conduction.[3] In case 3, the vapor grooves are flooded with liquid, and Fig. 6 shows the temperature profiles of this startup case. In case 4, vapor exists in the vapor groove, and Fig. 5 shows the temperature profiles.

In case 3, shown in Fig. 6, because the vapor grooves were flooded with liquid, a liquid superheat was required to initiate the nucleate boiling. The link of the evaporator core and the CC was like a traditional heat pipe for the existence of vapor in the evaporator core, and so the heat leak from the evaporator to the CC was high, and the liquid superheat required in the vapor grooves was difficult to attain. In this startup, vapor was generated 40 min later. Then the loop started to circulate, and the temperatures of the CC and the evaporator decreased until they arrived at equilibrium. However, note that the liquid superheat in the vapor grooves at the incipience of nucleate boiling was small. It seems that nucleate boiling would take place at a small liquid superheat at high temperature.

In case 4, shown in Fig. 7, vapor existed in both the vapor grooves and the evaporator core. As the heat was applied, the temperature of TC 9 at the evaporator outlet (namely, the vapor line inlet) rose immediately, indicating that vapor was generated, and the loop started to circulate. At the same time, the high heat leak caused by the existence of vapor in the evaporator core led to the temperature rise of the CC. After the temperature difference between TC 7 on the evaporator and TC 2 on the CC had increased to a maximum value,

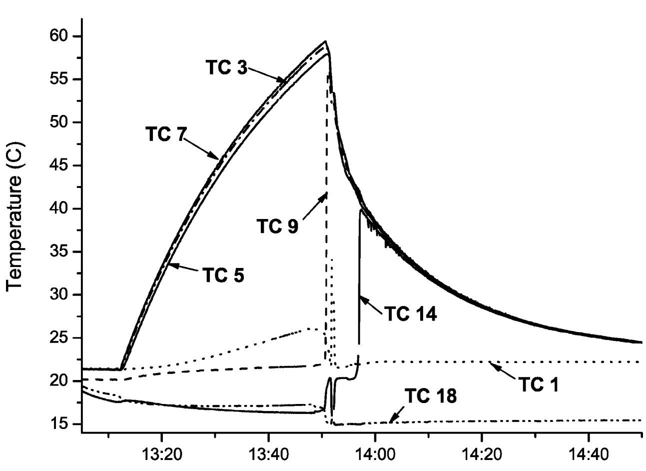

Fig. 6 Startup case 3.

Table 2 Startups with a heat load of 5 W, adverse elevation

Two-phase distribution	Start time, min		Overshoot, K		Superheat, K	
	0 m	2 m	0 m	2 m	0 m	2 m
Liquid/vapor	40	13	39	18	1.3	0.5
Liquid/liquid	6	3.2	12	3.5	7.6	2.3
Vapor/vapor	2.2	2.7	2.5	2.8	0	0
Vapor/liquid	1.6	2.4	1.8	2.7	0	0

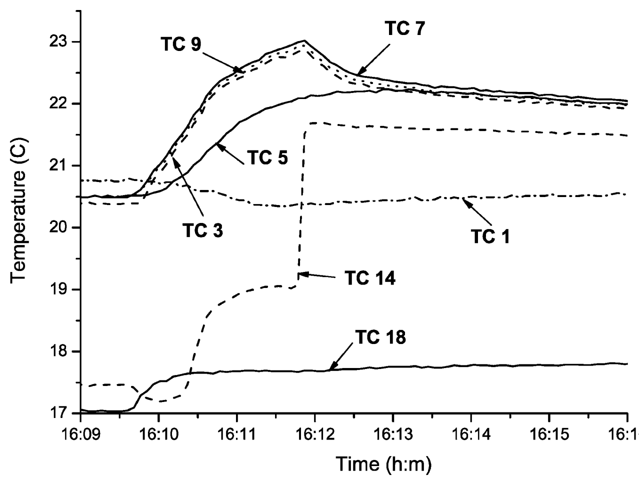

Fig. 7 Startup case 4.

TC 7 and TC 2 began to decrease and finally reached equilibrium. The temperature drop of the CC and the evaporator was decided by the negative value of Eq. (3).

In Fig. 7, the temperature of TC 14 should not have been lower than TC 9 and TC 7 because the heat loss of the vapor line in the situation was due less to the low working temperature, and the working fluid at the condenser inlet should be saturated (be either vapor or a two-phase fluid). More analysis is required to give a reasonable explanation.

Table 2 lists the four startup cases depending on the vapor/liquid distribution in the evaporator. The item of two-phase distribution in Table 2 represents the two-phase states, respectively, in the vapor grooves and the evaporator core. For example, liquid/vapor denotes that liquid fills the vapor grooves, whereas vapor exists in the evaporator core. From the top down in Table 2, the startup requires a shorter time, experiences a smaller temperature overshoot, and appears to be easier to achieve.

In the tests, the ultimate steady-state temperatures of the four startup cases approximated to each other. Note that the four cases themselves have no effect on the steady-state operation of LHPs.

Sink Temperature

Some tests were conducted to investigate the effect of sink temperature on the startup. Results indicate that the sink temperature does not affect the startup at low heat loads when the liquid line is not thermally insulated absolutely. This can be explained as follows. When the heat load is low, the mass flow rate of fluid is very small:

$$Q_{\mathrm{loop}} = r \times \dot{m} \tag{4}$$

When the axial conductance of the working fluid and pipe are neglected, the energy equation of liquid line can be given as

$$-\dot{m} C_{p,\mathrm{liq}} \frac{\mathrm{d}T}{\mathrm{d}Z} = \left(\frac{UA}{L}\right)_{\mathrm{liq\text{-}amb}} (T - T_{\mathrm{amb}}) \tag{5}$$

Assume that $(UA/L)_{\mathrm{liq\text{-}amb}}$ is constant. Then the expression of the temperature of liquid into the CC, namely, the outlet temperature of liquid fluid, can be obtained from Eq. (5) as

$$T_{\mathrm{out}} = T_{\mathrm{amb}} + (T_{\mathrm{in}} - T_{\mathrm{amb}}) \exp\left\{-\frac{(UA/L)_{\mathrm{liq\text{-}amb}}}{\dot{m}C_{p,\mathrm{liq}}} L_{\mathrm{pipe}}\right\} \tag{6}$$

The sink temperature is usually lower than the ambient temperature. According to Eq. (6), lower mass flow rate leads to a higher fluid temperature into the CC. When the startup heat load is low, the mass flow rate of working fluid needed to reject the heat from the evaporator is small. Although the fluid has been cooled to be close to the sink temperature in the condenser, because the liquid has moved so slowly along the liquid line, its temperature rose to be close to the ambient temperature in the returning liquid line due to the heat leak from the ambience. That means, at a low startup heat load, the sink temperature will not affect the temperature of the returning liquid at the inlet of the CC if the liquid line is not offered an absolute thermal insulation and, ultimately, will not affect the CC temperature or the startup of the LHP.

Startup Heat Load

Tests were conducted to investigate the effects of heat load on the startup. The LHP was settled as described in case 3 before the startup to ensure that the most difficult startup condition would occur. Figures 6, 8, and 9 show the startup temperature profiles at a heat load of 5, 20, and 200 W, respectively.

Comparisons of startups at various heat loads are presented in Table 3. It is obvious that the high heat loads help the startup of the LHP. At a higher heat load, the required liquid superheat in the vapor grooves is achieved more quickly, and it is easier to initiate nucleating boiling.

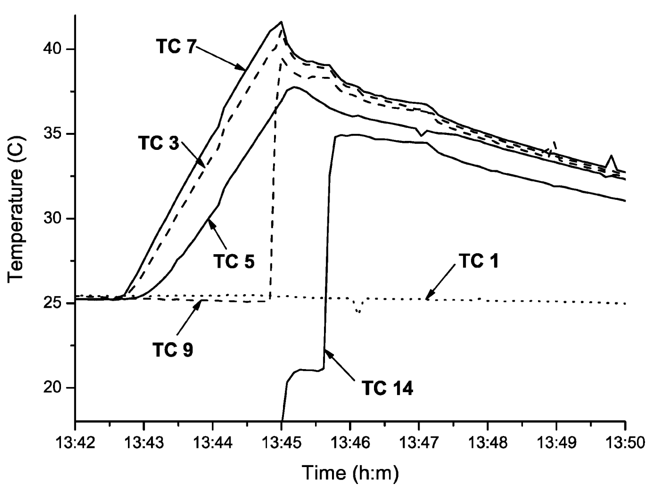

Fig. 8 Startup at 20 W.

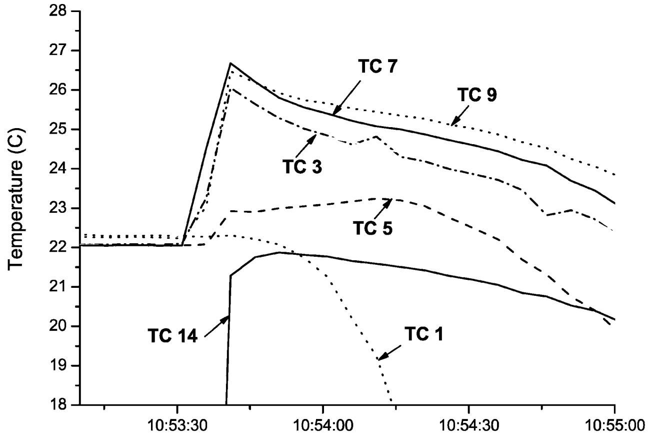

Fig. 9 Startup at 200 W.

Table 3 Effect of heat load on startup

Power, W	Time, min	Overshoot, K	Superheat, K
5	40	39	1.3
20	2.5	17	1.8
200	0.33	4.5	1.5

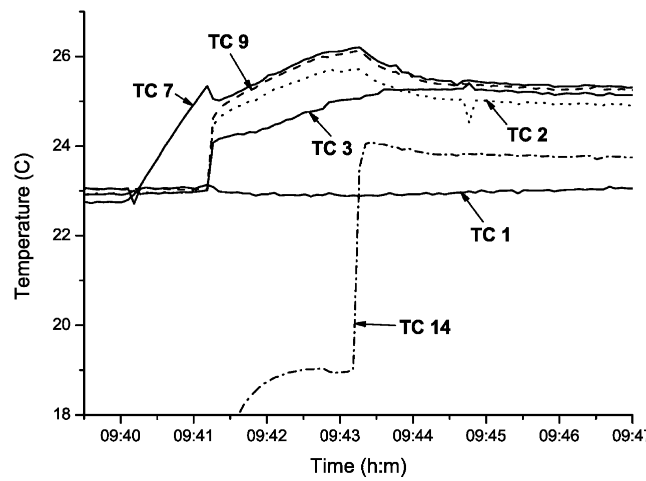

Fig. 10 Case 1, 2-m adverse elevation for 5-W startup.

In Table 3, it appears that the liquid superheat required to initiate the nucleate boiling is independent of the startup heat load. In fact, because too many factors influence the degree of incipient superheat, there is no way to predict the superheat degree to be experienced in any unit in any given circumstance.[5] However, it is found that a high superheat is always needed when the evaporator is flooded with liquid, whereas a very low superheat is required when the vapor grooves are filled with liquid and vapor exists in the evaporator core. It seems that nucleate boiling is easier to initiate with small liquid superheat at high temperatures, whereas a high superheat is required at low temperatures. Further efforts are needed to explain why a low superheat can initiate nucleate boiling at high temperatures.

The test results prove that an effective way to assist the startups of case 3 is by the application of an extra low heat load on a small area of the evaporator. Because a low heat load on a small area will cause a high heat flux, it will help to initiate nucleate boiling in the vapor grooves during startups, but will not affect the steady-state temperatures much during operation.

Adverse Elevation

An adverse elevation means that the evaporator is above the condenser. Because the saturated pressure drop is related to the saturated temperature difference across the wick in accordance with the Clausius–Clapeyron relation, the additional pressure at adverse elevations required to pump the return liquid fluid against gravity will increase the temperature difference across the wick required for startup, hence delaying startup (see Ref. 3).

However, not all of the startups at adverse elevations lasted a longer time and experienced a larger temperature overshoot in the test program. Test results indicate that the effect of elevation is to increase the temperature overshoot and startup time when vapor exists in the vapor grooves, but to decrease the temperature overshoot, liquid superheat, and startup time when the vapor grooves are filled with liquid (Table 2). Figures 10 and 11 show the temperature profiles of two 5-W startups with a 2-m adverse elevation, respectively, settled overnight at the positions shown in Figs. 3a and 3b. The vapor grooves were considered to be filled with liquid. Compared with the 5-W startups at no adverse elevation (Figs. 4 and 6), the liquid superheat, the temperature overshoot, and the startup time were all decreased. The test results suggest that the two-phase state in the evaporator before start for the loop will be changed due to the buoyancy effect in the 1-g environment. When vapor exists in the

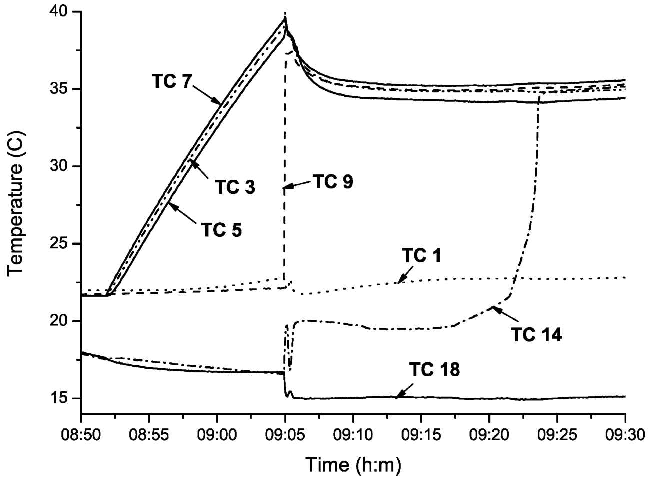

Fig. 11 Case 3, 2-m adverse elevation for 5-W startup.

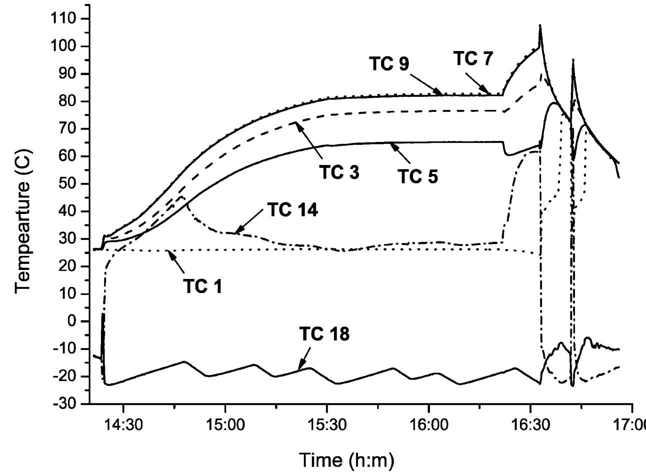

Fig. 12 Startup with evaporator above CC at 20 W.

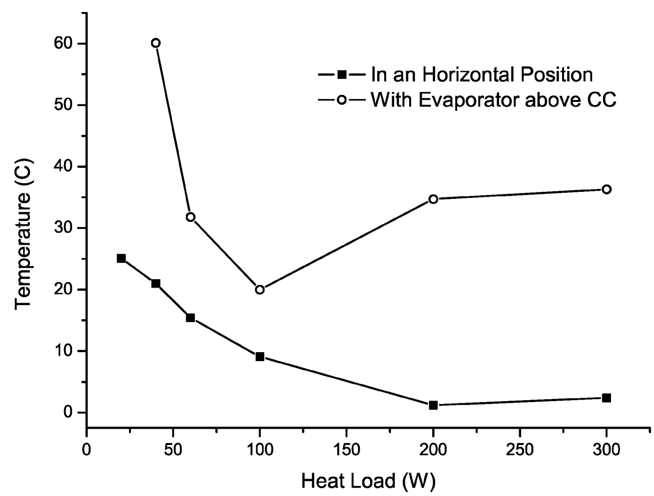

Fig. 13 Operating temperatures with 40-W startup heat load.

vapor grooves, the two-phase state will not change, and the effect of adverse elevations is only to increase the pressure drop and delay the startup, whereas when the vapor grooves are initially filled with liquid, the vapor bubbles will ascend toward the evaporator. The presence of vapor in the vapor grooves will promote the initiation of nucleate boiling at a lower superheat.

In conclusion, adverse elevations have two-sided effects on the startup. On the one hand, the additional pressure drop caused by the adverse elevation will delay the startup. On the other hand, the presence of vapor bubbles due to the buoyancy effect will promote evaporation. Moreover, a peculiar startup phenomenon was observed in the tests at adverse elevations, shown in Fig. 10. This startup was composed of two startup cases. At the beginning, the temperature profiles were in accordance with those of the startup case 1. After the nucleate boiling, the temperature profiles changed in accordance with those of the startup case 2. The temperatures of the evaporator and the CC did not decrease, but increased for about 2 min and then fell subsequently to reach equilibrium. Note that the temperature of TC 2 on the vapor zone of the CC rose quickly with TC 7 on the evaporator, whereas TC 3 on the liquid zone of the CC rose slowly. This phenomenon again proves that the pressure rise that leads to the condensation of working fluid in the CC is the main cause of the saturated temperature rise in the CC during the startup when liquid floods the evaporator core.

Effects of Startup on Operation of LHPs

The effects of startup on the steady-state operation of LHPs are discussed in the following four subsections. Test results indicate that an unfavorable relative position between the evaporator and the CC leads to a failed startup and very high operating temperatures and that the startups with evaporation inside the wick, temperature oscillation, and reverse flow equilibrium shift the steady-state operating temperatures.

Relative Position of Evaporator and CC

In the 1-g environment, the orientation of the evaporator and the CC influences the startup and steady-state operation of LHPs. When the CC is above the evaporator, liquid will fill the evaporator core and even the vapor grooves. Then the startup cases 1 and 2 will occur. When the CC and the evaporator are on the horizontal plane, vapor will exist in the evaporator core. Then the cases 3 and 4 will occur. However, when the evaporator is above the CC, startup will become very difficult.

Figure 12 shows the temperature profiles of a startup with the evaporator 6 mm above the CC. The sink temperature and the ambient temperature were -20°C and 25°C, respectively. When a 20-W heat load was applied, the final steady-state temperature exceeded 80°C. During the startup, vapor got into the condenser at the beginning, but receded from the condenser back into the vapor line about one-half an hour later. In this unfavorable position, liquid filled the CC first due to the gravity, and vapor existed in both the vapor grooves and the evaporator core. After the heat load was applied, vapor was generated on both sides of the wick, and the vapor generated inside the wick tended to stay or ascend to the top region of the evaporator core due to gravity. Then evaporation inside the wick continued for the existence of the vapor/liquid interface, leading to a high heat leak from the evaporator to the CC. Although the loop started, the mass flow rate was small for the high heat leak, which is explained in Eq. (7), and the return subcooled liquid was not able to condense the vapor inside the wick effectively. Evaporation in both sides of the wick prevented the loop from developing the temperature difference between the evaporator and the CC required to push the vapor into the condenser, which was related to the pressure difference in accordance with the Clausis–Clapeyron relation. High heat leak and a small mass flow rate finally led to a very high steady-state temperature.

$$Q_{\mathrm{app}} = Q_{\mathrm{HL}} + Q_{\mathrm{loop}} = Q_{\mathrm{HL}} + r \times \dot{m} \tag{7}$$

Even though the heat load was increased to 60 and 100 W later, the evaporator temperature continued rising, and the loop did not appear to reach equilibrium. The test was halted because of safety considerations. It was a failed startup. This test was carried out again with identical operating conditions, except for the startup heat load. As shown in Fig. 13, when the startup heat load was increased directly to 40 W, the loop started to work, but the steady-state temperature was still higher. Note that a larger mass flow rate at a higher heat load is helpful to the startup in this unfavorable position because the larger mass flow rate is able to condense the vapor inside the wick more effectively and reduce the heat leak. In the application of the spacecraft thermal control system, a secondary wick will be fixed between the evaporator core and the CC to assist the liquid supply, and no such startup problem will occur in the mirogravity environment. The dual-CC LHP is also introduced in Ref. 6 to solve the startup and operation problems given an unfavorable position for terrestrial application.

Evaporation Inside the Wick

In several startup tests, evaporation occurred inside the wick when the evaporator, including the vapor grooves and the evaporator core, was filled with liquid. One of these startups with a 2-m adverse elevation at 5 W is to be described. All tests in the following text were conducted with a sink temperature of about $-20°C$. The loop was laid vertically, the CC above the evaporator, and settled overnight before startup. Otherwise, liquid would have flooded the evaporator.

As Fig. 14 shows, before the nucleate boiling in the vapor grooves, the CC temperatures of TCs 2 and 3 rose together with stepwise increments. The same temperature trends and values of TCs 2 and 3 indicated that phase change heat transfer had occurred inside the evaporator core and that vapor had been generated inside the wick. It is obvious that the vapor was not from the vapor grooves because no vapor had been generated outside the wick. (TC 9 at the vapor line inlet does not increase, but rather stays constant all along.) Then the vapor generated inside the wick ascended to the CC and condensed, where it was cooled by the cold liquid. The link between the evaporator core and the CC was like a traditional heat pipe. Furthermore, it is hypothesized that the vapor bubbles were being generated and growing when the temperatures of TCs 2 and 3 kept constant, whereas the vapor bubbles were ascending and condensed, releasing the latent heat when the temperatures of TCs 2 and 3 increased sharply. The stepwise temperature rise of the CC describes the physical process of growing, ascending, and condensation of vapor bubbles inside the evaporator core and the CC.

The evaporation inside the wick can be explained as follows. When the liquid flooded the evaporator, a liquid superheat is required to initiate the nucleate boiling in the vapor grooves. Because the heat is transferred from the outside case to inside the evaporator core, the liquid temperature in the vapor grooves is always higher than that inside the evaporator core, and evaporation should have tended to take place in the vapor grooves rather than inside the wick. However, sometimes vapor or noncondensable gas may be trapped on the inside surface of the wick in the preceding operation, making the inside wick surface a more preferable place for nucleate boiling or surface evaporation, even if the liquid inside the evaporator core has a lower temperature than that in the vapor grooves. Although such a startup phenomenon has taken place only twice during the past tests of three months, measures must be taken in LHP design to avoid this inside evaporation.

Reverse flow was observed during the startup. As Fig. 14 shows, the condenser outlet temperature of TC 18 was higher than the condenser inlet temperature of TC 14, which indicating that fluid at ambient temperature was flowing from the liquid line into the condenser, whereas fluid at sink temperature was flowing from the condenser into the vapor line. This is the reverse flow to be described and discussed in detail later. Although the heat leak was high and the CC temperature rose continually, the liquid superheat in the vapor grooves was increasing. In this startup, when nucleate boiling occurred at a superheat of 6.7 K, reverse flow disappeared at once, and the loop started to circulate.

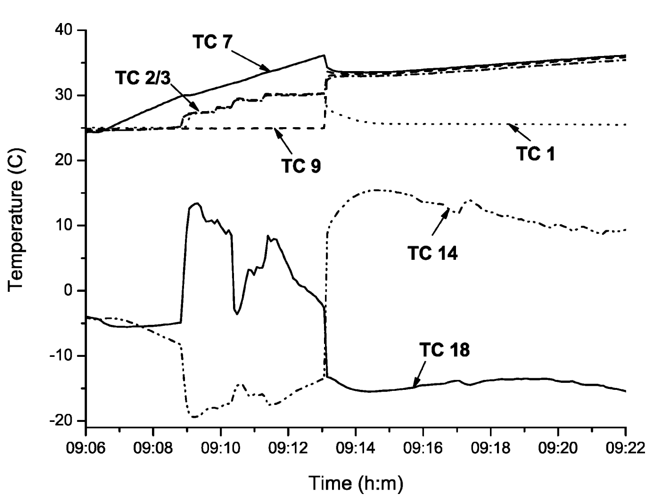

Fig. 14 Startup with evaporation inside wick.

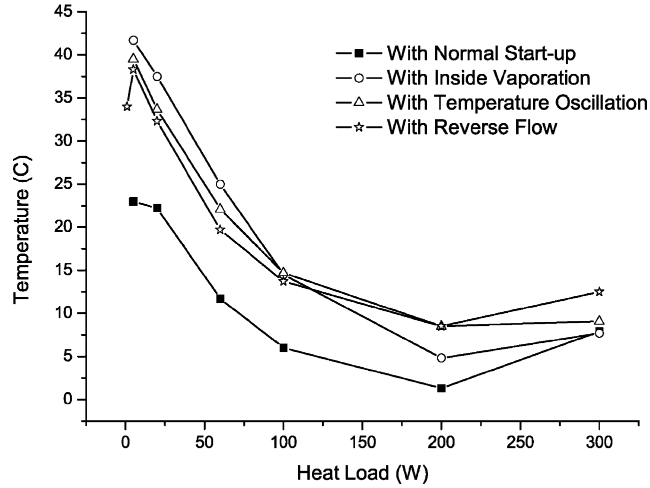

Fig. 15 Comparison of operating temperatures with different startup situations.

Furthermore, higher operating temperatures were observed after such a startup with evaporation inside the wick. After the nucleate boiling in the vapor grooves, working fluid started to circulate. However, vapor was not pushed into the condenser, and the condenser actually did not continue because the temperatures of TCs 13 and 14 near the condenser inlet did not increase to the saturated vapor temperature. This was caused by the evaporation on both sides of the wick. Because the evaporation also took place inside the wick, the heat leak was very large. The pressure difference between the evaporator and the CC was too small to push the vapor into the condenser, and so the temperatures of TCs 9 and 2 continued rising until the LHP achieved equilibrium. The LHP finally operated at a higher steady-state temperature. As Fig. 15 shows, the operating temperatures of the LHP after a startup with evaporation inside the wick were higher than normal at low heat loads when the CC was not filled with liquid. After the CC was filled with subcooled liquid, no evaporation would continue inside the wick, and the LHP would then work normally.

Temperature Oscillation

When the loop was laid vertically, with the CC above the evaporator, and settled overnight before startup, the temperature oscillation phenomenon was observed several times. Figure 16 shows the temperature profiles of a 5-W startup with the evaporator flooded with liquid at a 2-m adverse elevation. After the nucleate boiling, the fluid started to circulate, but TC 7 on the evaporator and TC 2 on the CC still kept rising to reach equilibrium. The sustained temperature rise of the evaporator and the CC indicated that the heat leak from the evaporator to the CC was higher than normal and could not be balanced by the subcooled liquid from the condenser. The increased heat leak was caused by the evaporation and condensation of the fluid in the evaporator core and the CC for the heat leak by the heat transfer mode of the thermal conduction through a hermetic case, and the conduction and convection of the working fluid would have been low.

The detailed temperature profiles of startup, shown in Fig. 17, must be noted. Before TC 7 on the evaporator reached the peak value, no evaporation occurred inside the wick, and the heat leak was low. During this process, TCs 2 and 3 remained nearly constant. When the liquid superheat in the vapor grooves increased to 4 K, nucleate

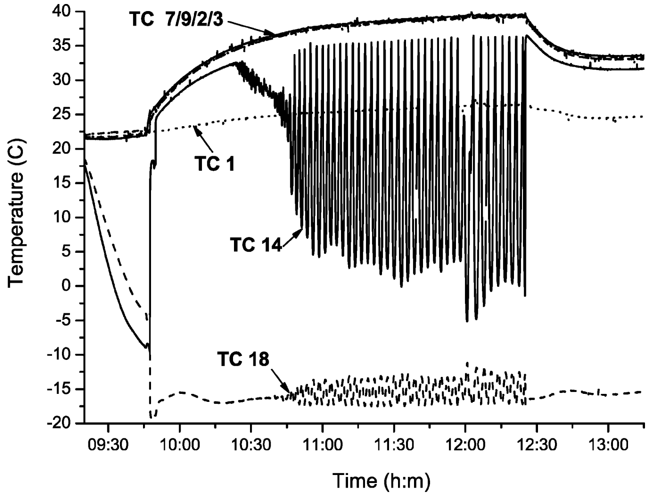

Fig. 16 Temperature oscillation after startup.

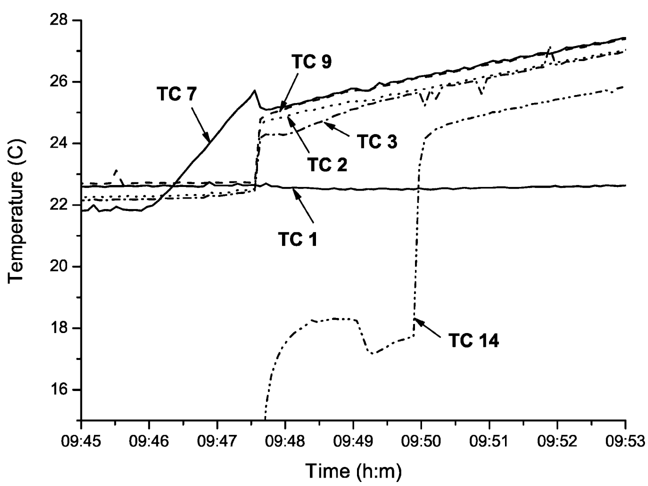

Fig. 17 Startup followed by temperature oscillation.

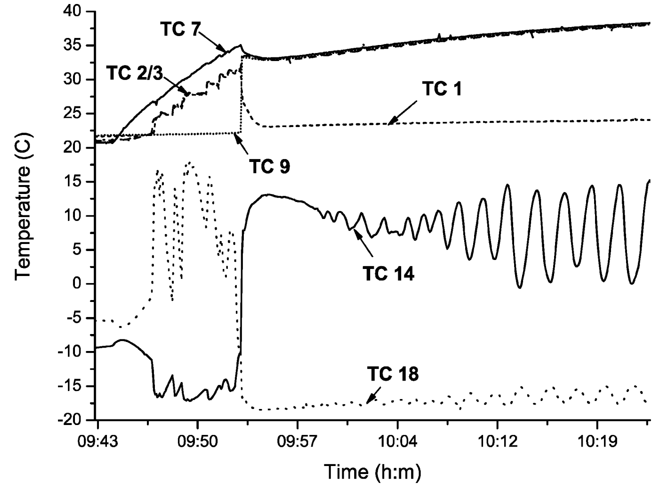

Fig. 18 Temperature oscillation caused by evaporation inside wick.

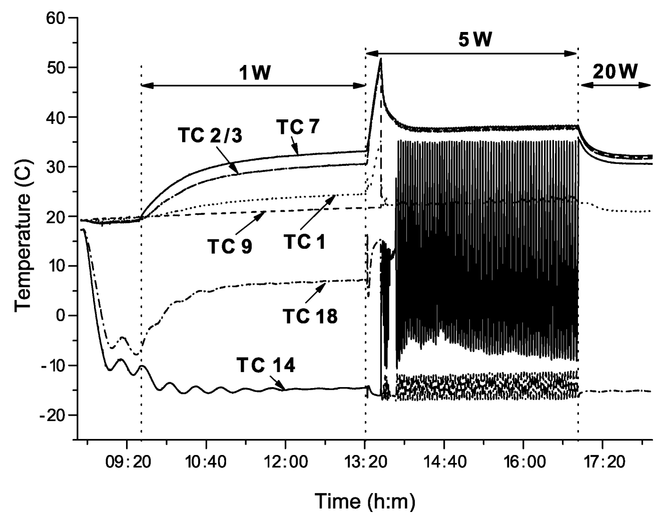

Fig. 19 Temperature oscillation after reverse flow equilibrium.

boiling took place. The temperature rise of TC 9 indicated that vapor was generated. It is hypothesized that the vapor pressure was so much higher than that inside the wick that vapor might penetrate the wick into the evaporator core. Then evaporation would take place inside the evaporator core, leading to the higher heat leak from the evaporator to the CC. The vapor or gas might be trapped on the inside surface of the wick, and so the penetration had a residual effect on the later operation of LHP. As Fig. 15 shows, the steady-state temperatures were higher at heat loads when the CC was not filled with liquid.

Ku et al.[7] have also described and analyzed the temperature oscillations at the condenser outlet in a miniature LHP. It is found when the heat load is high enough, vapor is pushed outside the condenser, and temperature oscillations take place at the condenser outlet. That is characterized by the vapor/liquid interface's moving back and forth around the condenser outlet. Actually, if the charge mass of the working fluid has been adequate enough to ensure that part of the condenser acts as a subcooler at any heat loads, temperature oscillations at the condenser outlet can be avoided.

However, in the tests reported in this paper, temperature oscillation took place at the condenser inlet. After the LHP started to work, vapor was pushed into the condenser, which was proved by the temperature rise of TC 14 near TC 7 on the evaporator. However, TC 14 dropped one-half an hour later, indicating that the pressure difference between the evaporator and the CC was not large enough to push the vapor into the condenser for the high heat leak. As Fig. 16 shows, the temperatures of TCs 14 and 18, respectively, at the inlet and outlet of the condenser, soon began to oscillate. Actually, TC 7 on the evaporator and TC 2 on the CC were also oscillating in a smaller temperature range. When TC 14 rose to the wave crest (the maximum peak value), namely, the saturated temperature, vapor was pushed into the condenser, and the vapor/liquid interface was inside the condenser. When TC 14 dropped to the wave trough (the minimum peak value), namely, the subcooled liquid temperature, the vapor/liquid interface receded from the condenser to the vapor line. The temperature oscillation was characterized by the vapor/liquid interface's moving back and forth around the condenser inlet in this example.

Another two examples of temperature oscillation were observed in the tests. One was caused by the evaporation inside the wick, with temperature profiles shown in Fig. 18. The temperature rise of TCs 2 and 3 with a step increment indicated that evaporation indeed occurred. The other temperature oscillation, with temperature profiles shown in Fig. 19, was observed after reverse flow equilibrium, which will be discussed in the following text. However, when the temperature oscillation occurred, vapor could be pushed into the condenser by increasing the heat load, and then the temperature oscillation would disappear. Such startups with evaporation inside the wick had a resident effect on the steady-state operation, leading to higher operating temperatures at low heat loads, as shown in Fig. 15.

Reverse Flow

Reverse flow often occurred when the vapor grooves were filled with liquid. Before boiling, part of the heat load heats up the liquid in the vapor grooves, and the remainder goes into the evaporator core across the wick. The heat leak from the evaporator to the CC caused by conduction, convection, or two-phase heat exchange increases the saturated temperature of the CC, which leads to the corresponding saturated pressure rise, whereas the pressure of superheated liquid in the vapor grooves does not increase. The high pressure in the CC will push the working fluid out to flood the loop, much like the prestartup of capillary pumped loops. The reverse flow will cease after the loop is full of liquid. However, when evaporation occurs inside the wick, menisci will be formed at the liquid/vapor surface inside the wick. The menisci will develop a capillary pressure that will circulate the fluid and lead to a sustained reverse flow.

As explained, evaporation inside the wick can lead to reverse flow. In Figs. 14 and 18, as the heat load was applied to the evaporator, TC 18 at the condenser outlet rose, whereas TC 14 at the condenser inlet dropped, which indicated that the reverse flow was prevailing. The capillary pressure provided by the menisci formed inside the wick pushed the fluid to flow in reverse. The fluid flowed through the bayonet to the liquid line and then passed the condenser and the vapor line, finally returning to the vapor grooves. In such a state, vapor was generated inside the wick, and the loop was supplied by the return liquid in the vapor grooves.

Reverse flow often takes place before the nucleate boiling is initiated. Thus, the duration that reverse flow lasts is decided by the time required to initiate the nucleate boiling. Because the initiation of nucleate boiling is more difficult and requires a longer time for a lower startup heat load, reverse flow lasts a longer time. In the tests, reverse flow lasted only about 10 s in a 20-W startup with a 2.5-m adverse elevation shown in Fig. 20 and lasted for 5 min in the 5-W startup with no adverse elevation shown in Fig. 4. Furthermore, it was found that when the heat load was too low to establish the liquid superheat in the vapor grooves, reverse flow continued and even achieved equilibrium.

Figure 21 shows the temperature profiles of a startup with a 2.5-m adverse elevation at 1 W. After the heat load was applied, the liquid superheat increased gradually. However, the required value was never achieved at this low heat load, and vapor was not generated in the vapor grooves. As time went on, the heat leak across the wick led to evaporation inside the wick. Menisci formed at the liquid/vapor surface inside the wick and developed a capillary pressure to circulate the fluid, leading to a sustained reverse flow. When the reverse flow achieved equilibrium, part of the heat load applied to the evaporator was dissipated to the ambience through the liquid pipe line, and the remainder of the heat load was balanced by the subcooled liquid in the vapor grooves from the vapor line. The condenser did not act because the vapor did not reach the condenser, and the liquid flowing into the vapor grooves was also heated up to the ambient temperature. Thus, the final steady-state operating temperature was higher than normal.

When the heat load was increased to achieve the required superheat, the reverse flow disappeared, and the fluid started to circulate in the right direction. Figure 15 shows the effect of adverse flow equilibrium on the steady-state operating temperatures. The operating

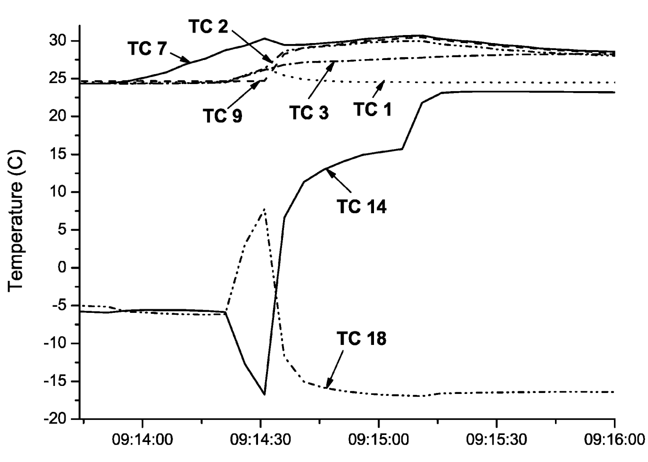

Fig. 20 Temperature profiles of startup at 20 W.

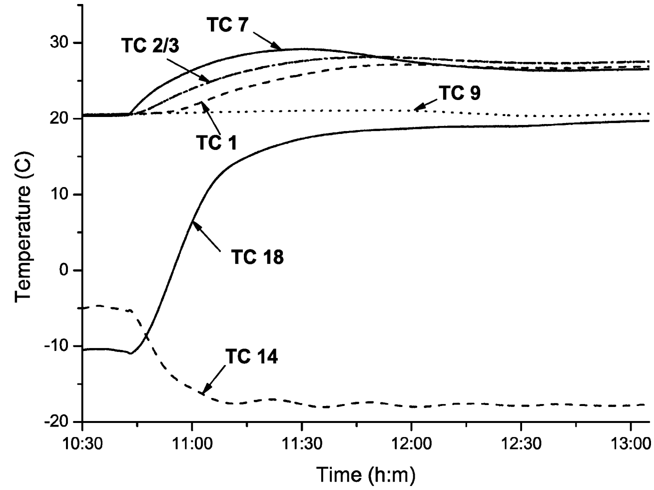

Fig. 21 Reverse flow equilibrium at 1 W.

temperatures were higher at such heat loads when the CC was not flooded with liquid because of the inside evaporation after reverse flow equilibrium as discussed earlier.

The phenomenon that the reverse flow equilibrium at 1 W was followed by a temperature oscillation at 5 W was also observed. As shown in Fig. 19, when the heat load was increased from 1 to 5 W, the reverse flow disappeared, and the working fluid started to flow in the forward direction, which was indicated by the temperature rise of TC 14 and temperature drop of TC 18. It is hypothesized that the vapor or gas might be trapped inside the wick after the inside evaporation and that the evaporation continued when the heat load rose to 5 W, then the temperature oscillation occurred.

Conclusions

Extensive ground experiments have been conducted to investigate the startup behaviors of a loop heat pipe. It is found that not only the heat leak from the evaporator to the CC, but also the pressure rise that leads to the condensation of working fluid in the CC, will cause the temperature rise in the CC during the startup. In some cases, pressure change is the main cause of the temperature change in the CC. In ground tests, the effects of elevation increases the temperature overshoot and the startup time when vapor exists in the vapor grooves, whereas elevation decreases the temperature overshoot and the startup time when the vapor grooves are filled with liquid. Test results indicate that high startup heat loads are helpful to the startup and that the sink temperature does not affect the startup when the liquid line is not absolutely thermally insulated.

Although test results indicate that the four startup cases depending on the vapor/liquid state in the evaporator have no effect on the operation of LHPs, evaporation inside the wick tends to occur at low startup heat loads when the evaporator is flooded with liquid. The evaporation inside the wick is proved by the temperature rise of both the vapor and liquid zones on the CC with stepwise increments. When the evaporator is flooded with liquid, some peculiar phenomena, including evaporation inside the wick, temperature oscillation, and reverse flow equilibrium, occur at low startup heat loads, which lead to higher operating temperatures. A bidiathermancy wick, which possesses an outside layer with high conductivity to vaporize the liquid and an inside layer with low conductivity to help establish the temperature difference between the evaporator and the CC, namely, the liquid superheat in the vapor grooves, is suggested by authors to avoid evaporation inside the wick.

The conclusion concerning temperature oscillation can be drawn that it will occur at very low and high heat loads. At low heat loads, the vapor/liquid interface will not be overwhelmingly pushed into the condenser for the conceivable evaporation inside the wick, and temperature oscillation will take place at the condenser inlet. In contrast, at high heat loads, vapor can not be condensed absolutely in the condenser, and temperature oscillation will occur at the condenser outlet. Temperature oscillation at the condenser outlet can be avoided by increasing the working fluid charge mass to ensure that part of the condenser acts as a subcooler.

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