Thermal Vacuum Testing of a Small Loop Heat Pipe with a PTFE Wick for Spacecraft Thermal Control

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A loop heat pipe (LHP) is a two-phase heat transfer device that utilizes the evaporation and condensation of a working fluid to transfer heat, and the capillary forces developed in fine porous wicks to circulate the fluid. LHPs have been gaining increased acceptance for spacecraft missions, and recently, small LHPs on the order of a few hundred watts have been investigated for this purpose. In this study, a 100W class small LHP with a polytetrafluoroethylene wick as the primary wick was designed and fabricated for thermal vacuum testing. The LHP has a thermoelectric converter (TEC) to control the loop operating temperature. The thermal vacuum test was conducted to evaluate the LHP’s thermal performance under a space-simulated environment such as ultra-high-vacuum, and black body radiation, except for a gravitational effect. The loop showed large thermal hysteresis before and after the large and small head loads. The TEC was able to control the loop operating temperature with a small amount of electrical power.

Key Words: Loop Heat Pipe, PTFE Wick, Spacecraft Thermal Control, Thermal Vacuum Test

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>A</td>
<td>surface area</td>
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<tr>
<td>C_p</td>
<td>specific heat</td>
</tr>
<tr>
<td>D_i</td>
<td>inner diameter</td>
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<tr>
<td>D_o</td>
<td>outer diameter</td>
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<tr>
<td>h_{CC-d}</td>
<td>heat transfer coefficient</td>
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<tr>
<td>r</td>
<td>radius</td>
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<tr>
<td>ΔP</td>
<td>pressure drop</td>
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<tr>
<td>P_{cap}</td>
<td>capillary force</td>
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<tr>
<td>ΔP_{tot}</td>
<td>total pressure drop</td>
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<tr>
<td>k_{eff}</td>
<td>effective thermal conductivity</td>
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<tr>
<td>K</td>
<td>permeability</td>
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<tr>
<td>L</td>
<td>length</td>
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<tr>
<td>m</td>
<td>mass flow rate</td>
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<tr>
<td>Q_{sub}</td>
<td>amount of liquid subcooling</td>
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<tr>
<td>Q_{h}</td>
<td>heat leak through the wick</td>
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<tr>
<td>Q_{CC-d}</td>
<td>heat exchange between CC and ambient</td>
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<tr>
<td>Q_{Load}</td>
<td>applied power</td>
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<tr>
<td>T_{amb}</td>
<td>ambient temperature</td>
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<tr>
<td>T_{rad}</td>
<td>radiator temperature</td>
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<tr>
<td>T</td>
<td>temperature</td>
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Subscripts

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>Evap</td>
<td>evaporator</td>
</tr>
<tr>
<td>CC</td>
<td>compensation chamber</td>
</tr>
<tr>
<td>Cond</td>
<td>condenser</td>
</tr>
<tr>
<td>Vap</td>
<td>vapor line</td>
</tr>
<tr>
<td>Liq</td>
<td>liquid line</td>
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<tr>
<td>Wick</td>
<td>wick</td>
</tr>
<tr>
<td>Groove</td>
<td>groove</td>
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<tr>
<td>Grav</td>
<td>gravity</td>
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1. Introduction

Loop heat pipes (LHP), which work based on capillary force and thermal energy generated from a cooling target, are gaining great acceptance as energy-saving cooling devices, especially in the field of space engineering1,2. In future space missions such as lunar landings and interplanetary missions, power sources will be strictly limited, and thermal control with an LHP will be an effective method of conserving power for these missions. LHP performance is strongly dependent on its wick performance, as well as the properties of the working fluid and the loop configuration. So far, stainless steel, nickel, and titanium have been used as wick materials in space applications because they have good chemical compatibility with various working fluids and low thermal conductivity. However, the smaller the LHP evaporator size, the larger the heat leak across the wick becomes, and thus new kinds of wicks with much lower thermal conductivity will be required. The purpose of this study was to develop a high-performance...
small LHP with a plastic wick for space applications. Several plastic wicks have been under investigation at Nagoya University, JAXA, and Tohoku University. As one of our scientific approaches, LHPs with polytetrafluoroethylene (PTFE) wicks are being studied. The current wick performance made of the porous metals for the stage-of-art LHPs in Space field are around 1–3 μm. There are several reports on plastic wicks, but the development of wicks with small pore size and high permeability proved difficult. On the other hand, in this research, PTFE wick with pore radii of 0.8–2.2 μm could be developed, and tests were conducted in an atmospheric condition to evaluate the basic performance of the LHP by changing power to the evaporator. The test results showed good thermal performance of the PTFE wick.

The final goal of this study is to apply the small LHP with PTFE wick to space applications. In order to apply the LHP to space application, it is required that the thermal performance is demonstrated under a space-simulated environment, such as microgravity, ultra-high-vacuum, and black body radiation. However, it is difficult to simulate such environment at the same time above grand. From this reason, usually, thermal vacuum tests, in which ultra-high-vacuum and black body radiation below 150 K are simulated, are conducted. Although thermal vacuum test is essential to evaluate the thermal characteristics of the loop, so far, quite a few papers have reported on the thermal vacuum test of the LHPs. There was no report on the thermal vacuum test of LHP with plastic wicks. In this paper, the test results of thermal vacuum test for a LHP with a PTFE wick is reported. The tests include the power step-up and step-down test, power cycle test, temperature control test, and sink cycle test. The start-up failure due to insufficient liquid link in an evaporator, and large temperature hysteresis are also reported.

Note that since the thermal vacuum test was conducted above ground, the thermal performance of the LHP under microgravity might be different. Especially, the two-phase distribution in the loop is different from that under gravitational environment, and, as a result, thermal performance such as heat leak discussed later might be different. The authors would emphasize that all the tests reported in this paper were conducted under gravitational environment.

2. The PTFE LHP Concept

2.1. Normal operating principal of a LHP

Figure 1 illustrates the schematic of a typical LHP, which consists of an evaporator, a compensation chamber (CC), a vapor transport line, a condenser, and a liquid return line. A proper amount of the working fluid was charged in the LHP. Only the evaporator has a wick structure, and the rest of the loop components are made of smooth wall tubing. Normal operation of the LHP occurs as follows: Heat applied to the evaporator vaporizes the liquid. The vapor generated in the evaporator is transported to the condenser through the vapor transport line. The vapor is condensed and subcooled in the condenser. The subcooled liquid is returned to the CC through the liquid return line. The CC stores the excess liquid. The CC controls the loop operating temperature, since the CC is always saturated.

The detailed operation of the LHP is thermally and hydrodynamically explained as follows: When the evaporator is heated, menisci are formed at vapor/liquid interfaces to develop the capillary pressure, \( P_{\text{cap}} \), to transport the fluid around the loop, written as

\[
P_{\text{cap}} = \frac{2 \sigma \cos \theta_{\text{wick}}}{r}
\]

where \( \sigma \) is the surface tension of the working fluid, \( r \) is the radius of the curvature of the meniscus, which cannot be smaller than the effective pore radius of the wick, and \( \theta_{\text{wick}} \) is the contact angle between the working liquid and wick wall. To operate the loop, the capillary pressure in the wick must be able to sustain the total system pressure drop, \( \Delta P_{\text{tot}} \), which is the sum of the pressure drops in the wick, vapor grooves, vapor line, condenser, liquid line, and gravity head, i.e.,

\[
P_{\text{cap}} \geq \Delta P_{\text{tot}} = \Delta P_{\text{w}} + \Delta P_{\text{vapor}} + \Delta P_{\text{g}} + \Delta P_{\text{cond}} + \Delta P_{\text{liq}} + \Delta P_{\text{grav}}
\]

The pressure drop in the wick is expressed as

\[
\Delta P_{\text{w}} = \frac{\mu_{\text{L}}}{\rho_{\text{L}}} \ln \left( \frac{D_{\text{in}}}{D_{\text{out}}} \right) \rho_{\text{L}} \kappa
\]

where \( \mu_{\text{L}} \) and \( \rho_{\text{L}} \) are the viscosity and the density of the fluid in the liquid phase, respectively, \( D_{\text{in}} \) and \( D_{\text{out}} \) are the inner and outer diameter of the wick, respectively, \( \kappa \) is the length and the permeability of the wick, respectively, and \( \rho_{\text{L}} \) is the mass flow rate across the wick.

Under steady state operational conditions, the energy balance of the LHP is

\[
Q_{\text{cc}} = Q_{\text{in}} - Q_{\text{cc-a}}
\]

where \( Q_{\text{cc}} \) is the heat exchange between the CC and the returning liquid, \( Q_{\text{in}} \) is the conductive heat leak from the evaporator and the CC through the wick and the liquid core, and \( Q_{\text{cc-a}} \) is the heat exchange between the CC and the ambient. The LHP operating temperature adjusts itself such that the condenser generates enough subcooling to match the heat leak and the heat exchange with the environment. \( Q_{\text{cc}} \) is obtained by the following equation:

\[
Q_{\text{cc}} = m C_{\text{s}} (T_{\text{cc}} - T_{\text{a}})
\]

where \( m \) and \( C_{\text{s}} \) are the mass flow rate and the heat capacity of the fluid, respectively, and \( T_{\text{cc}} \) and \( T_{\text{a}} \) are the temperature of the CC and the inlet of the CC, respectively. \( Q_{\text{in}} \) is obtained by the following equation:
where \( k_{\text{eff}} \) is the effective thermal conductivity of the wick, \( T_{\text{Evap}} \) is the saturation temperature of the vapor inside the evaporator grooves, and \( T_{\text{CC}} \) is the saturation temperature of the fluid in the CC. This temperature difference is caused by the total system pressure drop, excluding the pressure drop in the wick. This can be obtained from the Clausius-Clapeyron equation, as follows:

\[
T_{\text{Evap}} - T_{\text{CC}} = \left( \frac{\partial T}{\partial P} \right)_{\text{Sat}} \cdot (\Delta P_{\text{Total}} - \Delta P_{\text{Wick}})
\]  

(7)

The heat exchange between the CC and the environment can be written as follows:

\[
Q_{\text{CC-\text{Amb}}} = h_{\text{CC-\text{Amb}}} (T_{\text{CC}} - T_{\text{Amb}})
\]

(8)

The CC temperature is determined by this energy balance. Generally, \( Q_{\text{CC-\text{Amb}}} \) is smaller compared to \( Q_{\text{NL}} \) and negligible, and eq. (4) can be treated as follows:

\[
Q_{\text{SC}} = Q_{\text{NL}}
\]

(9)

### 2.2. Temperature control

The loop operating temperature is determined by the saturation temperature in the CC. By heating or cooling the CC, the overall loop temperature can be controlled. A detailed explanation can be seen in Ref. (9). In this study, a thermoelectric converter (TEC) was used for the temperature control.

### 2.3. Plastic wick

The requirements for plastic porous materials as primary wick are high heat resistance, and good chemical compatibility and wettability with candidate working fluids, as well as the requirements for conventional metallic wicks such as small pore size, large porosity, and large permeability. Metal wicks with low thermal conductivities such as those made of nickel, titanium, and stainless steel have been used as LHP primary pumps. Ceramic wicks have also been under investigation to achieve less heat leak\(^{(12)}\). However, the requirement for thinner, high-performance small LHPs has meant that much lower thermal conductivity is required to reduce the heat leak. Thus, plastics are being investigated as high-performance wicks. Historically, plastic wicks have been used in capillary pumped loop technology. However, the development of wicks with small pore size and high permeability was difficult. For example, ultra high molecular weight polyethylene wicks were used as a primary pump of the capillary pumped loop for TERRA satellite which was launched in 1999. The size of the pore radius was 15\(\mu\)m\(^{(10)}\).

In a previous study, a PTFE wick with the following performance was developed \(^{(6)}\):

- Pore radius: 0.8 ~ 2.2 \(\mu\)m
- Porosity: 27 ~ 50%

In the LHP fabricated in the present study for the thermal vacuum test, a wick with the pore radius of 1.2 \(\mu\)m and porosity of 34% was selected. The permeability of the wick was estimated to be about 3.3 \(\times\) 10\(^{-17}\) m\(^{2}\), which was calculated from the pore radius and porosity.

### 3. Design and Fabrication of the PTFE LHP

Table 1 lists the LHP specifications for the present work. Figure 2 shows the laboratory setup for the LHP with temperature measurement points. The evaporator was made of stainless steel tubing 12 mm in O.D. \(\times\) 77 mm long. The CC was made of a stainless steel tube 20 mm in O.D. \(\times\) 40 mm long. The vapor line and liquid line, each 900 mm long, were made of stainless steel tubes 4.8 mm and 3.2 mm in O.D., respectively. The condenser was made of a stainless steel tube 3.2 mm in O.D. \(\times\) 3000 mm long. The condenser tube was connected to an aluminum plate. The condenser was embedded in a radiator panel made of aluminum alloy 5052. The surface of the radiator panel was painted with Z-306 black paint to enhance the radiative heat exchange. The differences from the test article for the ambient testing\(^{(3)}\) are as follows:

- Heat in the condenser was finally rejected via thermal radiation, while heat was rejected via a water chiller in the ambient testing. Since thermal conductance by radiation was smaller than that by conduction, the length of the condenser was longer.
- The volume of the reservoir and the amount of the fluid became larger due to the extension of the condenser length.

<table>
<thead>
<tr>
<th>Table 1. LHP specifications.</th>
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<tr>
<td><strong>Evaporator</strong></td>
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<tr>
<td><strong>CC</strong></td>
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<td><strong>Vapor line</strong></td>
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<td><strong>Liquid line</strong></td>
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<td><strong>Condenser</strong></td>
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<td><strong>Fluid</strong></td>
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\[\text{Fig. 2. LHP for thermal vacuum test.}\]
Flexible hoses with 1/8 inch O.D. were installed at the vapor line and the liquid line to make a deployable radiator configuration.

The radiator was $500 \times 600$ mm. The size was calculated so that about 120W of heat could be rejected from the radiator surface when the average temperature of the radiator surface was about 50°C and fin efficiency was 0.8. In the test, since the heat load to the evaporator of the loop was less than 100W, the amount of heat rejection was controlled by controlling the temperature with a sheet heater attached to the rear side of the radiator.

During the test, the condenser was set 50 mm above the evaporator/CC section as shown in Fig. 2 (b). The hydrostatic loss was calculated by:

$$\Delta P_{\text{h}} = \rho gh$$

where, $h$ is the difference in height between the evaporator and the condenser. The calculated value of the hydrostatic loss was about 340 ~ 400 Pa over the temperature range from 0 to 100°C. Since capillary force, generated by the 1.2 μm-PTFE-wick and acetone, is about 21900 ~ 41500 Pa, the effect of the hydrostatic loss can be negligible.

Figure 3 shows a photo of the evaporator and the CC section. A TEC module was attached to the CC through the copper block. A thermal strap made of piled graphite sheets was attached to the rear side of the TEC. The other side of the strap was attached to the evaporator through the copper block. The loop was charged with about 20 grams of acetone. The amount of working fluid needed to charge the LHP was determined by the simultaneous equations in cold and hot cases proposed by Ku1), as functions of temperature range and total volume of the LHP. A 200-gram aluminum heater block was attached to the evaporator to simulate the instrument mass. Four cartridge heaters were inserted into the thermal mass to provide heat loads to the evaporator. Forty T-type thermocouples were used to monitor the temperatures.

4. Test Setup and Test Conditions

4.1. Test setup

The thermal vacuum test was conducted at ISAS/JAXA. The test system was composed of a thermal vacuum chamber, a measurement and power control unit, and a data acquisition system. The chamber had a cylindrical profile with its axis horizontal to the ground. The chamber size was 1.1 m in axial length and 1.0 m in inside diameter, and it was equipped with a cold shroud wall that was controlled to liquid nitrogen temperature levels throughout the test. The shroud wall was coated with black paint with an emittance of approximately 0.9. Throughout the test, vacuum levels within the chamber were maintained at a pressure of 10-6 torr. A data acquisition system consisting of a personal computer, a display monitor, and a data logger were used to display and store test data every four seconds. The temperatures of the LHP were recorded throughout the entire test at four-second intervals. In the thermal vacuum test, the loop was covered with multilayer insulation, except the radiator, to prevent radiative heat leak from the test article to the inner wall of the chamber. Figure 4 shows a photo of the test setup in the thermal vacuum chamber.

4.2. Test conditions

Table 2 lists the test conditions for the thermal vacuum test. A startup test was conducted first, and then a power step-up and step-down test was conducted to evaluate the basic heat transport capability of the LHP. A power cycling test was conducted to evaluate the repeatability and the response of the loop to the large power change. A temperature control test of the loop was also conducted, and last a sink cycle test was conducted to demonstrate the adaptability of the loop to changes in the thermal environment.

5. Results and Discussion

5.1. Power step-up and step-down

Figure 5 shows the temperature profile of the LHP during the power step-up and step-down tests. At first, the loop did not start under normal power application, that is, when heat...
was applied to the evaporator. There are two possible reasons for this. One is that the condenser temperature was very low at the beginning of the test, so the condenser was filled with liquid and the amount of liquid in the CC was insufficient. The second reason is that usually the charge amount was decided so that if 10-15% of the liquid remained in the CC even whole loop, except the CC was filled with liquid in the cold case. However, there was no secondary wick in this LHP that would have linked the liquid between the CC and evaporator core. As a result, the remaining liquid stayed at the bottom of the CC and the wick did not fill with liquid sufficiently.

To avoid this problem, the following procedure was conducted before the startup:
- Turn on the heater attached to the radiator panel to increase the radiator temperature to 15 °C.
- Turn on the TEC and heat the CC so that the liquid in the CC was ejected to the wick side.

When the power of 20W was applied to the evaporator after these two steps were taken, the loop could start.

Figure 6 shows the evaporator and the CC temperature of the LHP during the power step-up and step-down test. After start-up, the power was increased from 25W to 60W. The loop operated stably without any capillary limit or oscillations. However, the overall temperature became relatively high. Above the 55W power application, the evaporator temperature exceeded 100 °C. In the step-down mode, the power decreased from 60W to 10W. The loop still operated even at 10W. After that, the power was increased to 40W again. It was observed there were temperature hystereses in which the loop reached lower temperatures than it had with the first and the second 40W power applications.

Figure 6 shows the evaporator and the CC temperature of the loop as a function of the applied power in steady state. The heat leak calculated from eqs. (4) and (9) is also shown in the plot. It was confirmed that there were two different temperature hystereses. Comparing the temperatures reached in the step-down test with those in the step-up test, we see that the temperatures of the evaporator and the CC in the step-down test were higher. In the case of 40W power applications, for example, the temperature of the evaporator and the CC in the step-down test were about 9 °C and 11 °C higher than those in the step-up test, respectively. It appeared that this temperature hysteresis was due to the capillary hysteresis in the wick. Since the wick has an irregular capillary structure, the evaporating liquid front in the wick might have been different before and after the higher power applications. In this result, the heat leak from the evaporator to the CC and the thermal resistance between the wick and the evaporator case increased, and also both the evaporator and the CC temperatures increased.

Comparing the temperatures reached in the evaporator and the CC when 40W was applied during the step-down test with those reached when 40W was applied after the power step-down test, the temperature differences became larger. The temperatures of the evaporator and the CC in the step-down test were about 13 °C and 36 °C higher, respectively, than those of the final 40W power application. The calculated results of the heat leak from the evaporator to the CC when the first, second, and the third 40W power applications occurred were 3.3W, 3.7W, and 2.6W respectively. Note in the calculation of the heat leak using eq. (5), the temperature dependence of the specific heat was taken into account. However, the specific heat changes from among 2161 ~ 2189 J/kgK. It was clear that the difference of the heat leak is mainly dependent on the temperature difference between the CC and the inlet of the CC. When the third 40W power application occurred, the heat leak was 25 ~ 30% smaller than before. The reasons for such a difference will be discussed in the next section.

5.2. Power cycle

Figure 7 shows the temperature profile of the LHP during the power cycle test. It was confirmed that the loop showed good responsiveness and good repeatability between 20W and 60W cycles. However, there was a 40K difference between the 40W power applications before and after the power cycle.

Figure 8 shows the temperature distribution of the LHP when 40W was applied before and after the power cycle test. It was confirmed that the CC, the evaporator, and the vapor line parts showed large temperature differences before and after the power cycle test. The condenser and the liquid line, however, showed almost the same temperature between before and after the power cycle test. The reason for such a temperature difference nevertheless the same powers.
were applied to the evaporator is considered as follows:

![Temperature profile of the LHP during the power cycle test.](image)

**Fig. 7.** Temperature profile of the LHP during the power cycle test.

![Temperature distribution of the LHP when 40W was applied before and after the power cycle test.](image)

**Fig. 8.** Temperature distribution of the LHP when 40W was applied before and after the power cycle test.

When 40W was applied to the evaporator for the first time, the wick did not fully get wet. Although there was still fluid in the CC, there was no secondary wick to transport liquid to the primary wick, as discussed above. When the power was applied to the evaporator, the evaporation started on the wet part of the wick surface. In the case of normal operation, the vapor went to the vapor line. However, in this case, the wick did not get fully wet, and part of the vapor penetrated to the evaporator core side through the wick. Finally, the vapor reached the CC. This phenomenon can be observed by the temperature increase of the CC. Since the permeability of the wick was very small, the pressure on the surface of the wick remained higher than that in the core, even though there was a vapor link. Therefore, the vapor that went to the vapor line could reach the condenser and be returned to the evaporator inlet. The returned subcooled liquid was directly supplied to the tip of the evaporator core from the bayonet tube. As a result, the loop could continue to work. Once the power decreased to 20W, the mass flow rate decreased suddenly. The liquid remaining in the evaporator core permeated the whole wick. At this point, normal operation was established. In the normal operation, the heat leak from the evaporator to the CC was only thermal conductance across the wick. Therefore, the temperatures of the evaporator and the CC when 40W was applied after the power cycle test became lower. These heat leaks calculated from eqs. (4) and (9) were 4.6W and 2.9W, respectively, and there was about a 37% difference.

### 5.3. CC temperature control

Figure 9 shows the temperature profile of the LHP during the CC temperature control test. The power of 40W was applied to the evaporator throughout the test, and the CC temperature was controlled with the TEC module. The CC set point was changed from 30°C to 60°C by 5°C steps. It was observed that the loop was controlled well without any drying out. The accuracy of the temperature was within almost 0.1°C. TEC power was controlled with PID control. Since PID parameters were not optimized, the TEC powers at the first reaction were very high. However, when the temperature reached steady state, the required TEC power was 6W at the most. It was also observed that the evaporator temperature changed about 15°C, while the CC temperature was changed about 30°C. The LHP operates based on the differential pressure between the evaporator and the CC expressed by eq. (7). If the heat load applied to the evaporator is constant, the evaporator temperature will increase as the CC temperature increases, which is controlled by the TEC, to keep the same temperature difference and therefore the same differential pressure. However, the test result showed that the temperature difference between the evaporator and the CC became smaller as the CC temperature increased. Instead of that, the temperature of the condenser inlet (TC13 and TC14) decreased. This means that the vapor-liquid front shifted to the inlet of the condenser from the middle. Since the differential pressure between the evaporator and the CC decreased, to compensate for this decreasing, a two-phase region in the condenser decreased for lower $\Delta P_{\text{cond}}$. It is considered that the certain amount of the evaporator power might be absorbed by the TEC module through the thermal strap. Further tests and redesign of the TEC module are required.

![Temperature profile of the LHP and TEC power when the CC temperature was changed.](image)

**Fig. 9.** Temperature profile of the LHP and TEC power when the CC temperature was changed.

### 5.4. Sink cycle

Figure 10 shows the test result of the sink cycle test. The power of 50W was applied to the evaporator throughout the test, and the CC temperature was controlled to be 40°C. The
sink temperature was changed to fall from 15°C to -10°C. In an actual situation in a spacecraft, the sink temperature will shift, but in the test, the shroud wall was a constant temperature, since it was filled with liquid nitrogen. For this reason, the test was conducted by changing the radiator temperature with the heater attached to the radiator. It was observed that the evaporator temperature was almost constant, even though the sink temperature was changed by 25°C. The required TEC power increased as the condenser temperature increased. This can be explained as follows: When the radiator temperature decreased, two-phase region in the condenser decreased. As a result, the subcooled region increased, and therefore the heat leak increased as expressed by eq. (9). In order to compensate $Q_{HLQ}$, much TEC power was required.

The required power to maintain the CC temperature at 40°C was 0.1–3.1 W. Good temperature controllability of the loop by the TEC module with a small amount of electrical power was demonstrated.

6. Conclusions
A small LHP with a PTFE wick for spacecraft thermal control was fabricated, and a thermal vacuum test was conducted. The results are summarized as follows:
- The loop could not start by normal operation. This was due to insufficient liquid link between the CC and the evaporator since there were no secondary wick and low radiator temperature. By heating both the radiator and the CC before start up, the loop could start.
- After the loop started, the LHP could operate at power levels up to 60 W. The loop showed two kinds of temperature hystereses before and after the large and small head loads.
- A TEC could control the loop operating temperature with a small amount of electrical power even when the radiator temperature changed. It was also found that redesign of the TEC module was required for evaporator temperature control.

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References