

**Investigation of the Temperature Hysteresis Phenomenon of a Loop Heat Pipe**

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**Abstract**

The temperature hysteresis phenomenon of a Loop Heat Pipe (LHP) was experimentally investigated. The temperature hysteresis was identified by the fact that the operating temperature depends upon not only the imposed power but also the previous history of the power variation. The temperature hysteresis could impose limitations on the LHP applications since the LHP may exhibit different steady-state operating temperatures at a given power input even when the condenser sink temperature remains unchanged. In order to obtain insight to this phenomenon, a LHP was tested at different elevations and tilts by using an elaborated power profile. A hypothesis was suggested to explain the temperature hysteresis. This hypothesis explains well the experimental observations. Results of this study provide a better understanding of the performance characteristics of the LHPs.

**Nomenclature**

$h$	effective height of liquid column, m
$g$	gravitational acceleration, $m/s^2$
$\dot{Q}_{HL}$	heat leak or back conduction, W
$UA$	overall thermal conductance, $W/m^2.K$
$\Delta P_w$	pressure drop across wick, Pa
$\Delta P_{GR}$	pressure drop due to gravity, Pa
$\Delta T_{SAT}$	difference in LHP operating temperature or hysteresis, K
$\Delta T_w$	temperature difference across wick, K
$(\partial T / \partial P)_{SAT}$	slope of the vapor-pressure curve
$\rho$	density, $kg/m^3$

**1. Introduction**

Loop Heat Pipes (LHPs) offer very efficient and unique design alternatives for the thermal control problems encountered in space and ground applications. A LHP is a passive two-phase thermal control device, which utilizes the latent heat of vaporization of a working fluid to transfer heat, and the surface tension forces formed in a fine-pore wick to circulate the working fluid. As shown in Fig. 1, most LHP designs consist of an evaporator, a reservoir or compensation chamber, vapor and liquid transport lines, and a condenser. Unlike the heat pipes, the LHP needs a wick structure only in the evaporator section. This wick is referred as the primary wick. In many LHPs, a secondary wick is also used between the evaporator and the compensation chamber to ensure that the primary wick is wet at all times.

The evaporator is the heat absorbing element of the LHP. Due to the intimate contact between the evaporator and the compensation chamber, part of the applied heat is transferred back from the evaporator to the compensation chamber. This back conduction or heat leak is strongly dependent on the two-phase

flow dynamics and heat transfer between the evaporator and the compensation chamber. The LHP starts when the temperature difference between evaporator and the compensation chamber is high enough to provide the required superheat for boiling. After the start-up, the vapor is pushed to the vapor lines due to the high pressure on the evaporative side of the wick. The capillary meniscus inside the wick adjusts itself to match the total pressure drop in the loop. The vapor is then condensed in the condenser, and later subcooled. The subcooled liquid returns to the evaporator by passing through the compensation chamber. The steady-state operation of the LHP requires that the subcooling match the heat leak and the heat exchange between the compensation chamber and ambient. A detailed thermodynamic analysis of the operation of the two-phase heat transfer devices can be found in Ku (1994).

The LHP was invented in the former Soviet Union in the early 1970s and the first patent was issued to Maidanik et al. (1985). The successful operation of the LHP in micro-gravity was first demonstrated aboard a Russian spacecraft Granat in 1989 (Maidanik et al., 1991). Since then the LHPs have extensively been tested on the ground. Brief surveys of the earlier experimental studies on the LHPs can be found in Dickey and Peterson (1994), Wirsh and Thomas (1996), and Kaya et al. (1999). In spite of the extensive ground testing, only a limited number of flight tests has been performed. These were two successful flight experiments of the American Loop Heat Pipe (ALPHA) on the STS-83 and STS-94 missions in April and July 1997 (Lashley et al., 1998), respectively, and another successful flight test of a Russian LHP on the STS-87 mission in November 1997 (Bienert, 1998).

The LHP shares many design similarities with the Capillary Pumped Loop (CPL). The basic difference between these two devices is the location of the reservoir. This simple difference has an important effect on the thermodynamics of the system. Both technologies offer extensive design flexibility. A comprehensive discussion on the comparison of the CPLs and LHPs can be found in Nikitkin and Cullimore (1998). The devices exhibiting both CPL and LHP characteristics are also proposed (Hoang et al., 1997).

The LHPs have steadily been gaining acceptance in recent years due to their simplicity and robustness. They are currently baselined for the thermal control systems of NASA's Goescience Laser Altimeter System (GLAS) (Douglas et al., 1999), some future DoD spacecraft, and next generation large communication satellites. Since the transport lines of the LHPs can be made flexible, the LHP enables to develop innovative technologies such as deployable radiators (Lashley et al., 1998). Due to the available high capillary head, applications on the ground and even under high-g conditions are feasible. Specific applications in aircraft industry include avionics cooling (Gernert et al., 1996) and anti-icing system for the engine cowl using waste engine heat (Phillips and Gernert, 1998). The terrestrial applications include roof top solar installations and the cooling of remote communication sheds by transporting the heat into the ground, and the cooling of semiconductor chips and reactors.

Increased interest in the use of LHPs has resulted in the need for a better understanding of their thermal performance. The main purpose of this paper is to present the thermal characteristics of a Loop Heat Pipe (LHP) with a specific emphasis on the temperature hysteresis phenomenon. It was previously observed that the LHP may exhibit different thermal characteristics under seemingly identical operational conditions. In other words, the steady-state operating temperature of the LHP depends on not only the imposed heat load but also on the previous history of the heat load cycle. It should be noted that the two-phase flow structure inside the LHP is obviously expected to be different to result in the so-called temperature hysteresis. The temperature hysteresis has not been addressed adequately in open literature. To the authors' best knowledge, the temperature hysteresis was first reported by Wolf and Bienert (1994). They observed the temperature hysteresis only at high elevation operation, and did not suggest any explanation for the phenomenon. In a more elaborated study (Cheung et al., 1998), the hysteresis was reported even at zero elevation. In this work, an explanation for the hysteresis was suggested. The present work complements this previous work to explain the temperature hysteresis phenomenon. The importance of the two-phase flow structure in the evaporator and the compensation chamber was emphasized. The temperature hysteresis may be problematic for applications with a wide range of operating power since the loop operating temperatures can vary according to the heat load variation pattern. Accordingly, the LHP may not maintain the instrument temperature within specified design limits.

## **2. Experimental Setup and Instrumentation**

An extensive test program was conducted at NASA Goddard Space Flight Center (GSFC) on a research LHP designed for the Naval Research Laboratory (NRL). This LHP was fabricated by Dynatherm Corporation, Hunt Valley, Maryland. A schematic of the LHP is illustrated in Fig. 2. The LHP had a cylindrical evaporator of 25.4 mm in diameter and 305 mm in length. The compensation chamber diameter

was the same as the evaporator diameter, and it was 127 mm long. A sintered nickel wick with an effective pore radius of less than 1.2 micron was used. The liquid and vapor transport lines were made of stainless steel tubes of 4.8 mm outer diameter. Each transport line had an overall length of about 1524 mm. The condenser was made up of a stainless steel tube of 4.8 mm outer diameter with an overall length of 2032 mm. It had three separate sections linked in series. The condenser line was attached to the coolant loop by small aluminum saddles. To minimize the thermal contact resistance between the condenser and coolant tubes, thermal epoxy filling was used. The working fluid of the LHP was ammonia.

To monitor the temperature profiles of the LHP and environment, the system was instrumented with 45 copper/constantan (type T) thermocouples. These thermocouples were placed at critical locations of the LHP. The uncertainty of the thermocouple readings was estimated to be  $\pm 0.5$  K.

Heat input was provided by film heaters, which can provide up to 700 W of electrical power. An aluminum saddle was attached to the top of the evaporator to provide mounting surface for the electrical heater. The aluminum saddle also ensures that heat could spread evenly on the upper side of evaporator. Another tape heater was attached to the compensation chamber to control the set-point temperature for the set-point temperature control tests. Relays and variacs were used to control the heater power. The uncertainty in measuring the power input was estimated to be less than 2%.

The condenser tubes were cooled by a refrigerator with a 1.5 kW cooling capacity. A mixture of ethylene glycol (60%) / water (40%) was used as coolant fluid. The cooling system maintained the sink temperature within  $\pm 1$  K of the chosen sink temperature. Some high frequency oscillations of the sink temperature were observed. However, the effect of these fluctuations on the system performance was negligible since the induced fluctuations in the working fluid temperatures due to these fluctuations were less than  $\pm 0.25$  K. A 15 mm-thick Armaflex material was used to insulate the entire LHP.

Data were displayed in real time and stored for later analysis by a computerized data acquisition system. Sampling rate of data was 10 kHz with a resolution of 16-bit and an analog-to-digital converter system accuracy of 0.6%.

### 3. Discussion of Results

The LHP was studied at various orientations (tilt and elevation) to investigate the temperature hysteresis characteristics. In this paper, the tilt refers to the orientation of the loop when it is rotated along its longitudinal axis. In other words, a positive tilt means that the compensation chamber is above the evaporator. The elevation refers to the rotation along the transversal axis. A positive elevation means that the condenser is above the evaporator and the compensation chamber. For the elevation tests, the evaporator and the compensation chamber are leveled within  $\pm 2.5$  mm in the same horizontal plane. All the tests were conducted with a sink temperature of 263 K. The power input was cycled from low power to high power and back.

The choice of the power cycles, shown in Table 1, was based on the previous work performed on the same LHP (Cheung et al., 1998) so as to compare the results. In this earlier work, the LHP was leveled horizontally within  $\pm 2.5$  mm such that the evaporator, the compensation chamber, and the condenser were in the same horizontal plane. It was shown that during a power cycling test, when a moderate power step change (less than 100 W) was applied, the loop operating temperatures were very consistent at a given power level. However, after applying a higher power step change (e.g. 50 W/400 W/50 W), it was found that the operating temperatures were higher than their initial steady-state values for the same input power. The results showed that there was a consistent trend of increasing operating temperatures for power levels below 200 W when the power was cycled from high to low. The operating temperature, however, was very stable at power levels above 200 W regardless of the power step. As a result of these findings, it was concluded that a moderate power step change (less than 100 W) was not sufficient to alter the two-phase flow characteristics inside the evaporator core to cause major changes in the operating temperatures. It was also found that a larger power step change (e.g. 50 W/700 W/50 W) augmented the temperature hysteresis.

In this present study, the temperature hysteresis phenomenon was further investigated at the different orientations of the LHP. The experimental results are presented on the LHP performance curve (operating temperature versus input power). Figure 3 shows the results obtained for an elevation of 1550 mm (vertical distance between the axis of the evaporator and the lowest point of the condenser). It could be seen in Fig. 3 that when the power was decreased from 400 W with a step change of 100 W (Cycle I and II), a large hysteresis was observed at power levels of 200 W and lower. Accordingly, a 100-W power step change could no longer be considered as moderate when the LHP was at elevation.

Figure 4 shows a summary of the results obtained during Cycle II for different elevations when the power was decreased in steps from 300 W to 50 W. When the LHP was horizontal no hysteresis was observed within this power range. Higher elevations resulted in larger hysteresis. The loop temperatures were very stable for power levels higher than 300 W regardless of the LHP orientation.

In the light of all these observations, the following mechanism is suggested for the temperature hysteresis. When the input power is decreased, the vapor-liquid interface will move towards the inlet of the condenser since less condenser area is required to reject the applied heat. Some liquid will be drawn from the compensation chamber back to the condenser. As shown in Fig. 1, this will be accomplished by secondary wick structure between the evaporator and the compensation chamber. In the case of a rapid power turn down, the required pressure drop may exceed the capillary pumping limit of the secondary wick. It is hypothesized that this will cause a partial dry-out in the secondary wick. In other words, an elongated vapor bubble or a number of elongated bubbles would be trapped inside the secondary wick as well as in the vapor removal channels of the secondary wick. This will bring additional vapor-liquid interface or better thermal coupling between the evaporator and the compensation chamber, therefore increasing the heat leak. The heat leak can be compensated only by the incoming subcooled liquid and the heat exchange between the compensation chamber and ambient. A larger heat leak will therefore result in higher operating temperatures, explaining the temperature hysteresis.

The heat leak from the evaporator to the compensation chamber can be determined by the following equation:

$$\dot{Q}_{HL} = (UA) \Delta T_w \quad (1)$$

The temperature difference across the wick structure  $\Delta T_w$  can be obtained from the Clausius-Clapeyron relation. Alternatively, it can directly be calculated from the slope of the vapor-pressure curve,  $(\partial T / \partial P)_{SAT}$ :

$$\Delta T_w = \left( \frac{\partial T}{\partial P} \right)_{SAT} (\Delta P_w + \Delta P_{GR}) \quad (2)$$

The pressure drop across the wick,  $\Delta P_w$  consists of the frictional pressure drops in the vapor and liquid lines, the condenser, and the evaporator. The pressure drop in the evaporator contains those in the bayonet tube and the vapor grooves, excluding the pressure drop inside the wick structure. The pressure difference due to the gravitational forces,  $\Delta P_{GR}$  is taken into account only when the loop is positioned with an elevation against gravity and it can be calculated as follows:

$$\Delta P_{GR} = \rho g h \quad (3)$$

By substituting Eqs. (2) and (3) into Eq. (1), one will obtain the following equation:

$$\dot{Q}_{HL} = (UA) \left( \frac{\partial T}{\partial P} \right)_{SAT} (\Delta P_w + \rho g h) \quad (4)$$

The difference in the amount of heat leak between two consecutive cycles with a temperature hysteresis (e.g. between Cycle I and Cycle II) for a given power input, sink and ambient temperature can be determined by the following equation:

$$\dot{Q}_{HL} \Big|_I^{II} = \left[ (UA) \left( \frac{\partial T}{\partial P} \right)_{SAT} \Delta P_w \right] \Big|_I^{II} + \left[ (UA) \left( \frac{\partial T}{\partial P} \right)_{SAT} \right] \Big|_I^{II} (\rho g h) \quad (5)$$

Thus, according the above hypothesis, the partial dry-out will increase the thermal conductance, (UA) to result in a higher heat leak, therefore higher saturation temperature. The slope of the vapor-pressure curve and the pressure drop across the wick depend on the LHP saturation temperature. For a given elevation, the pressure drop due to gravity is also a function the LHP saturation temperature. This is because the density varies with the saturation temperature, and the effective height of the liquid column in the transport lines, h depends on the location of the vapor-liquid interface in the condenser, therefore, on the saturation temperature. However, this dependence is weaker than the other terms. For the sake of simplicity, the pressure drop due to gravity can be assumed constant. Equation (5) thus explains why the temperature hysteresis was larger when the LHP elevation was increased, as it is shown in Fig. 4. Higher pressure drop

due to gravity will result in higher difference in heat leak, thus larger temperature hysteresis. It should be noted that the gravity head is imposed upon the primary wick. The secondary wick needs to overcome only the pressure difference between the evaporator and the compensation chamber.

Table 2 summarizes the calculated pressure drop values at an input power of 100 W as a function of elevation for the NRL LHP for the purpose of comparison.

Similarly, according to Eq. (5), a smaller power increase may cause a noticeable temperature hysteresis if a sufficiently high gravitational head is imposed on the loop. No attempt was made to determine experimentally this minimum power increment for a given loop elevation. Wolf and Bienert (1994) observed a temperature hysteresis only at high elevation. This is because they used a small power step of approximately 25 W. The hypothesis proposed in this study thus explains why the temperature hysteresis in their study was limited to only high elevation operation.

This hypothesis is also consistent with the observation reported in Cheung et al. (1998) that a larger power step change augments the temperature hysteresis. This is because a larger power step change will be more stressful on the secondary wick, causing a larger portion of the wick to dry out, therefore, contributing to a higher heat leak.

Figure 5 represents the power cycling results for an elevation of 2080 mm. This position corresponds to the maximum elevation of the LHP. The response of the LHP was very similar to the results obtained at the previous elevation. The same explanations are valid.

In order to verify the above hypothesis, the LHP was also tested at the maximum tilt of  $90^\circ$ . In this position, the compensation chamber was located above the evaporator, and the entire loop was in a vertical plane. The results of this orientation are presented in Fig. 6. In this position, no hysteresis was expected since a partial dry-out was unlikely due to the favorable gravitational head. One may argue that the temperatures at low powers were not as stable as at high powers and some hysteresis was still noticeable. However, two important factors must be taken into account. First, at low powers, the LHP is much more sensible to the changes in the ambient temperature. Second, the waiting time to reach the steady state was much longer for low powers because of the reduced mass flow rates (e.g. 7 hours at 10 W versus less than 1 hour at 400 W). Due to the long waiting times, the complete power cycle at this orientation took approximately 10 days. As a result, it was difficult to have a constant ambient temperature during the testing especially at low powers. Slight changes in operating temperatures especially at low powers should thus be anticipated. The NRL LHP is currently under investigation in a thermal vacuum chamber. In vacuum, the influence of the ambient temperature fluctuations on the operating temperature can be minimized. It is expected the results of the vacuum tests will result in a better understanding of the role of the heat exchange between the LHP and ambient.

An important point, which deserves some discussion, is the highly consistent operating temperatures at high powers (above 200 W in the horizontal position and above 300 W at elevation). This was observed throughout all the cycling tests regardless of the orientation and the nature of the power cycle. This could be explained by the fact that above these power limits, the condenser was fully utilized. Therefore, all the excess liquid was pushed to the compensation chamber. The compensation chamber was hard filled with liquid. It should be also noted that the LHP tested in this study was slightly overcharged. Therefore, the primary and secondary wick were wetted at all times and no hysteresis was observed. This is also consistent with the hypothesis proposed in this study. Furthermore, during a power increase, the returning fluid going through the bayonet tube to the compensation chamber will flush the vapor slugs trapped in the wick structure, especially at high mass flux rates associated with high powers.

#### **4. Summary and Conclusions**

The temperature hysteresis phenomenon of a LHP was experimentally investigated. The temperature hysteresis can be problematic because the steady-state operating temperature may be different at the same power input, depending upon the past history of the power variation. Therefore, the LHP may not satisfy the temperature requirements and maintain the instrument temperature within specified design limits if large changes of power input occur during the mission. It is suggested that the temperature hysteresis was caused by the partial dry-out of the secondary wick when the capillary pumping limit of the secondary wick was exceeded due to a rapid power decrease. The partial dry-out would in turn increase the heat leak from the evaporator to the compensation chamber, leading to higher operating temperatures. To verify this hypothesis, the LHP was tested at different orientations (elevation and tilt). It was shown that the experimental observations were consistent with the suggested hypothesis. The hypothesis explains why a higher elevation or a larger power step down would cause a larger temperature hysteresis, which were not

explained in the earlier studies. No hysteresis was observed when the LHP was oriented such that the compensation chamber was above the evaporator due to the favorable gravitational head imposed on the secondary wick. It was also found that the operating temperatures were very stable at high powers. This was explained by the fact that at high powers the compensation chamber was hard filled with liquid, therefore, the primary and secondary wicks always were wet. The effects of the heat exchange between the compensation chamber and the ambient were more noticeable at low powers.

Although the hypothesis proposed in this paper explains well the temperature hysteresis, further research is required in this area. Additional research areas are: LHP testing in vacuum environment; flow visualization tests using see-through evaporators; hysteresis and secondary-wick pore size relation, and mathematical modeling of the hysteresis mechanism.

The effects of the temperature hysteresis should be taken into account in the LHP design analysis and the experimental verification program. The LHP should be designed such that the operating limits of the application are within the stable power range of the LHP without temperature hysteresis.

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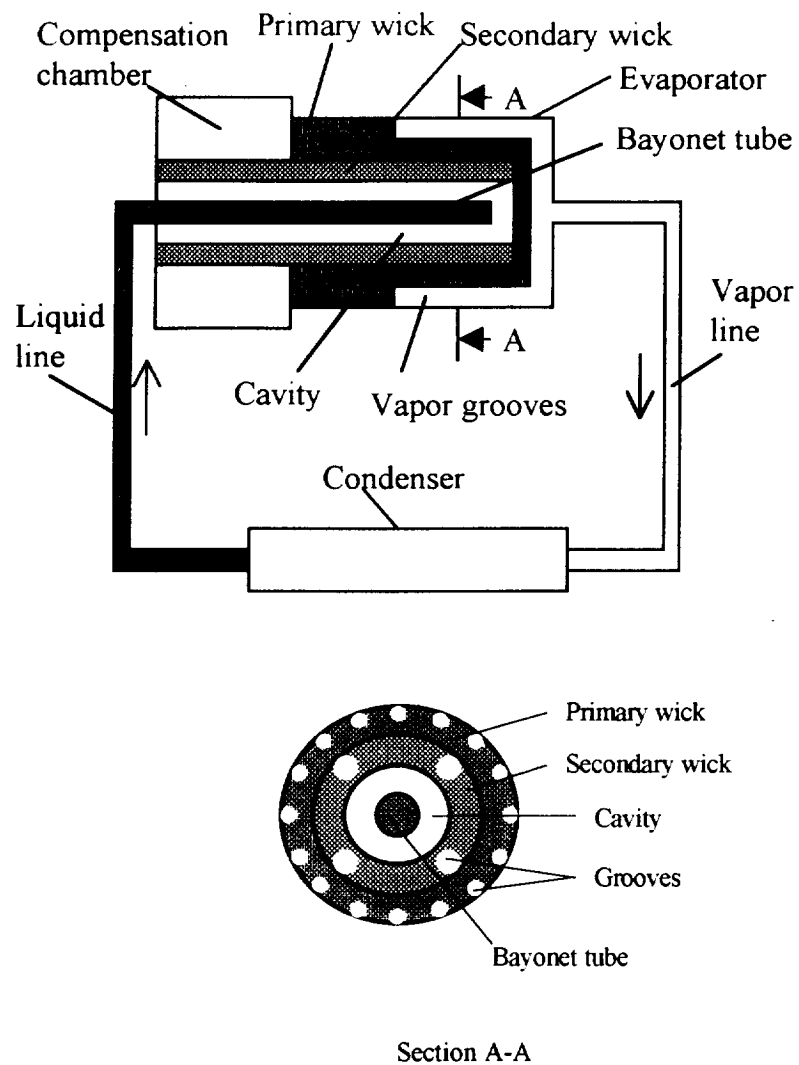


Fig. 1 Schematic of a LHP and the cross-section of the evaporator (Not drawn in scale).

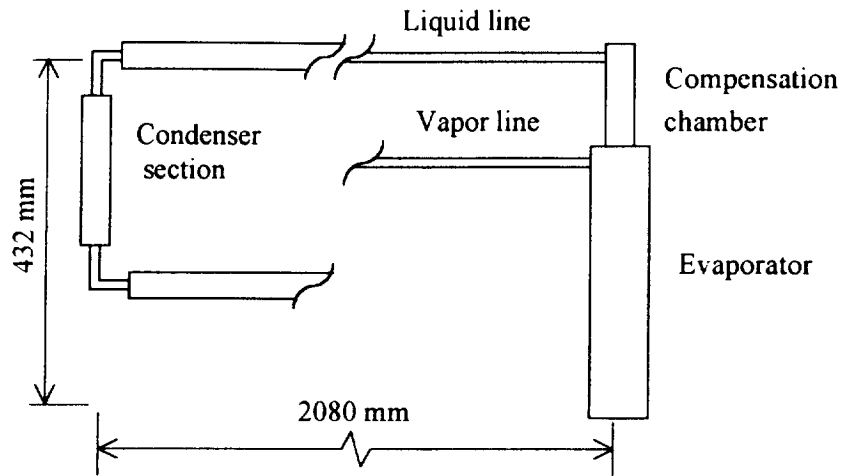


Fig. 2 Schematic of the NRL LHP.

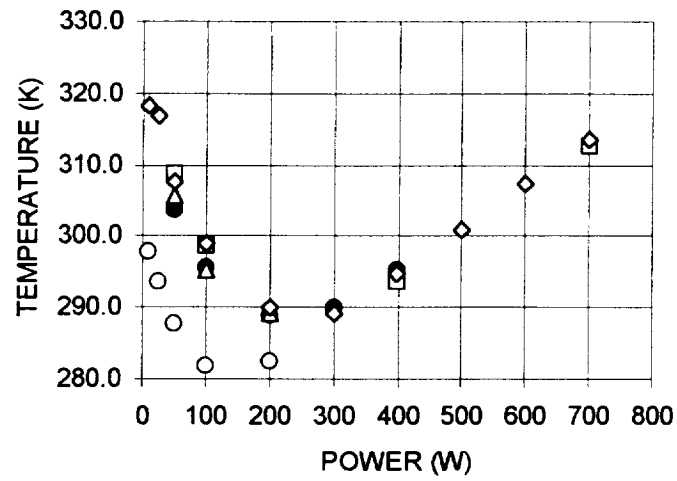


Fig. 3 Steady-state operating temperatures at an elevation of 1550 mm (O, Cycle I; ●, Cycle II; Δ, Cycle III; □, Cycle IV; and ◇, Cycle V).



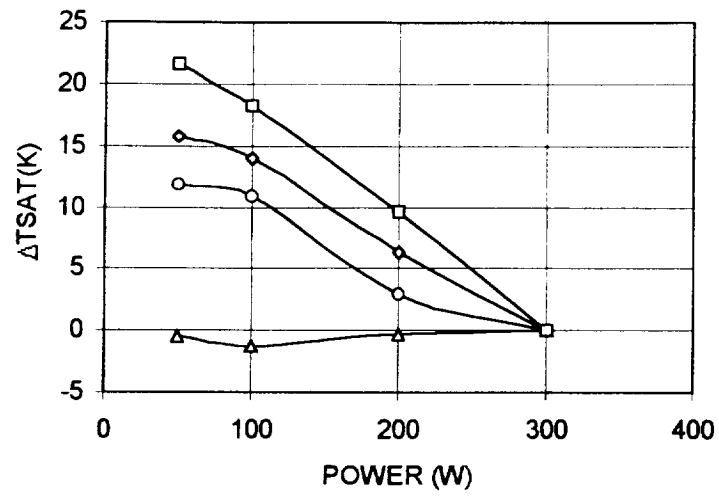


Fig. 4 Hysteresis during Cycle II for different elevations (Δ, zero elevation; O, 660 mm; ◊, 1550 mm; ◻, 2080mm).

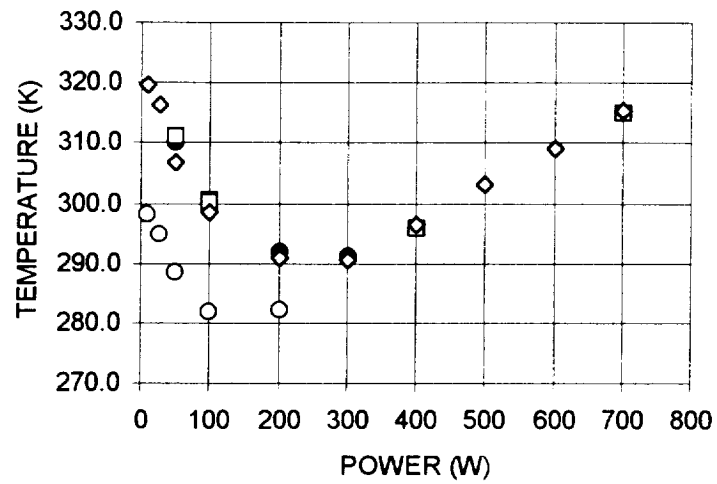


Fig. 5 Steady-state operating temperatures at an elevation of 2080 mm (O, Cycle I; ●, Cycle II; Δ, Cycle III; ◻, Cycle IV; and ◊, Cycle V).

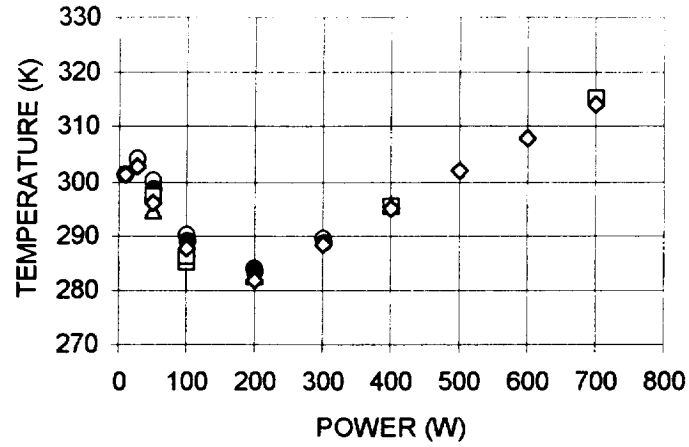


Fig. 6 Steady-state operating temperatures at a tilt of  $90^\circ$  (O, Cycle I; •, Cycle II;  $\Delta$ , Cycle III;  $\square$ , Cycle IV; and  $\diamond$ , Cycle V).

Table 1 Summary of the power profile for test cycles.

Cycle number	Cycle symbol	Power profile (W)
I	o	10-25-50-100-200-300
II	•	400-300-200-100-50
III	$\Delta$	400-50-100-200
IV	$\square$	400-700-100-50-100
V	$\diamond$	700-600-500-400-300-200-100-50-25-10

Table 2 Calculated pressure drop across the wick structure and due to the gravity at 100 W.

Elevation (mm)	$\Delta P_w$ (Pa)	$\Delta P_{GR}$ (Pa)
0	390.2	0
660	353.3	4059.7
1550	318.6	9477.1
2080	303.2	12555.2

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From: "James R. Frost" <James.R.Frost.1@gsfc.nasa.gov>  
Subject: Review of Presentation/Publication Packages  
Cc: Jettis.R.Hedgpeth.1@gsfc.nasa.gov

Reference(s):

Investigation of the Temperature Hysteresis Phenomenon of ALoop Heat Pipe,  
Jentung,

This morning I received your request(s) on a NASA 1676 form (NASA Scientific and Technical Document Availability Authorization (DAA)) for export control approval on the above referenced paper intended for public disclosure. As this paper provides specific technical information related to remote sensing satellites and/or related instruments, disclosing this technology would fall under the jurisdiction of the U.S. State Department, International Traffic In Arms Regulations (ITAR). As such, I would like to ask you to review and if appropriate, sign the applicable exemption on the attached NASA STI Public Disclosure (Export Control) Checklist. As you will see from this attachment, this checklist is in "draft" form, and we still expect to make minor changes, incorporating input from activities like your own before making this checklist final. We are confident that this checklist, as a replacement for the "draft" Public Domain Declaration will better share the concerns of the GSFC Export Control office, and will lead to more expedient Export Control approvals for GSFC Technical Documents in the near future.

Thanks for your assistance on this presentation, and we look forward to your input on this "draft" checklist. The Export Control office fax number is X6-1774 if you would prefer to fax a response back in lieu of responding via e-mail.

Thanks,

Jim Frost



Export\_Control\_checklist.wd.rev1.doc

2/17/99

Jim:

I have signed under Exempt<sup>m</sup> II. The test article was also described in an early paper, "Mathematical Modeling of Loop Heat Pipes," approved for publication by your office in November, 1998. Please contact me if you need further information. Thanks.

*Jentung*

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