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Operating Characteristics of Loop Heat Pipes

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ABSTRACT

The loop heat pipe (LHP) was invented in Russia in the early 1980's. It 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. The LHP is known for its high pumping capability and robust operation because it uses fine-pored metal wicks and the integral evaporator/hydro-accumulator design. It has gained rapid acceptance in recent years as a thermal control device in space applications. This paper presents an overview of the LHP operation. The physical processes and the thermal-hydraulic behaviors of the LHP are first described. Operating characteristics as functions of various parameters including the heat load, sink temperature, ambient temperature, and elevation are presented. Peculiar behaviors in LHP operation such as temperature hysteresis and temperature overshoot during start-up are explained. Issues related to multiple-evaporator LHP's are also addressed.

INTRODUCTION

Loop heat pipes (LHP's) are two-phase heat transfer devices that utilize the evaporation and condensation of a working fluid to transfer heat, and the capillary forces developed in fine porous wicks to circulate the fluid. The LHP was first developed in the former Soviet Union in the early 1980's [1], about the same time that the capillary pumped loop (CPL) was developed in the United States [2-4]. The LHP is known for its high pumping capability and robust operation because it uses fine-pored metal wicks and the integral evaporator/hydro-accumulator design. The LHP technology is rapidly gaining acceptance in aerospace community. It is the baseline design for thermal control of several spacecraft, including NASA's GLAS, EOS-Chemistry and GOES spacecraft, ESA's ATLID, CNES' STENTOR, RKA's OBZOR, and several commercial satellites [5-9].

Numerous LHP papers have been published since the mid-1980's. Most papers presented test results and discussions on certain specific aspects of the LHP operation. LHP's and CPL's show many similarities in their operating principles and performance characteristics. However, they also display significant differences in many aspects of their operation. Some of the LHP behaviors may seem strange or mysterious, even to experienced CPL practitioners. A comparison of some aspects of both devices has previously been published [10]. The main purpose of this paper is to present a comprehensive description of the operating principles and thermal-hydraulic behaviors of LHP's. Operating principles will be given first, followed by a description of the thermal-hydraulics involved in LHP operation. Operating characteristics and important parameters affecting the LHP operation will then be described in detail. Peculiar behaviors of the LHP, including temperature overshoot during start-up and temperature hysteresis, will be explained. For simplicity, most discussions will focus upon LHP's with a single evaporator and a single condenser, but devices with multiple evaporators and condensers will also be discussed. For phenomena that are still not well understood, educated guesses or speculations will be proposed. It should be emphasized that opinions expressed in this paper are the author's own, and do not represent an official position of NASA.

OPERATING PRINCIPLES OF LHP

Figures 1 shows the flow schematic of an LHP. It consists of an evaporator, a condenser, a compensation chamber, and vapor and liquid transport lines. Only the evaporator and the compensation contain wicks; the rest of the loop is made of smooth wall tubing. The wick in the evaporator is made with fine pores for purpose of developing a capillary pressure to circulate fluid around the loop, while the wick in the compensation chamber is made with larger pores for purpose of managing fluid ingress and egress. The operating principle of the LHP is as follows. As heat is applied to the evaporator, liquid is vaporized and the menisci formed at the liquid/vapor interface in the evaporator wick develop capillary forces to push the vapor through the vapor line to the condenser. Vapor condenses in the condenser and the capillary forces continue to push liquid back to the evaporator. The waste heat from the heat source provides the driving force for the circulation of the working fluid and no external pumping power is required. The two-phase compensation chamber stores excess liquid and controls the operating temperature of the loop.

In order for the loop to continue to function, the wick in the evaporator must develop a capillary pressure to overcome the total pressure drop in the loop. One of the advantages of a capillary loop is that the meniscus in the evaporator wick will automatically adjust its radius of curvature such that the resulting capillary pressure is equal to the total system pressure drop. The total pressure drop in the system is the sum of frictional pressure drops in the evaporator grooves, the vapor line, the condenser, the liquid line, and the evaporator wick, plus any static pressure drop due to gravity:

$$\Delta P_{tot} = \Delta P_{groove} + \Delta P_{vap} + \Delta P_{con} + \Delta P_{liq} + \Delta P_w + \Delta P_g \qquad (1)$$

The capillary pressure rise that the wick can develop is given by

$$\Delta P_{cap} = 2\sigma \cos\theta / R \qquad (2)$$

where σ is the surface tension of the working fluid, R is the radius of curvature of the meniscus in the wick, and θ is the contact angle between the liquid and the wick. As the heat load to the evaporator increases, so will the mass flow rate and the total pressure drop in the system. In response, the radius of curvature of the meniscus decreases so as to provide a higher capillary pressure that matches the total system pressure drop. The radius of curvature will continue to decrease with increasing heat loads until it is equal to the pore radius of the wick, R_p. Under this condition, the wick has reached its maximum capillary pumping capability:

$$\Delta P_{cap,\,max} = 2\sigma \cos\theta / R_p \qquad (3)$$

Further increase of the heat load will lead to vapor penetration through the wick and system deprime. Thus, under normal operation, the following condition must be satisfied at all times:

$$\Delta P_{tot} \leq \Delta P_{cap} \qquad (4)$$

The compensation chamber is located close to the evaporator. In fact, the compensation chamber is usually made as an integral part of the evaporator, and a secondary wick is used to connect the two elements. Liquid returning from the condenser always flows through the compensation chamber before it reaches the evaporator. The secondary wick provides a liquid link between the compensation chamber and the evaporator so that the evaporator will always be replenished with liquid. There are two major advantages of such a design. First, the loop can be started by directly applying power to the evaporator without the need of pre-conditioning. Second, the evaporator is tolerant of vapor bubbles in its liquid core. Because the primary wick is made of metal powder with a high thermal conductivity, liquid evaporation usually takes place inside the evaporator core and vapor bubbles are present there in most operation. To prevent vapor bubbles from accumulating inside the evaporator core, the secondary wick design incorporates vapor arteries which allow vapor bubbles to vent to the compensation chamber.

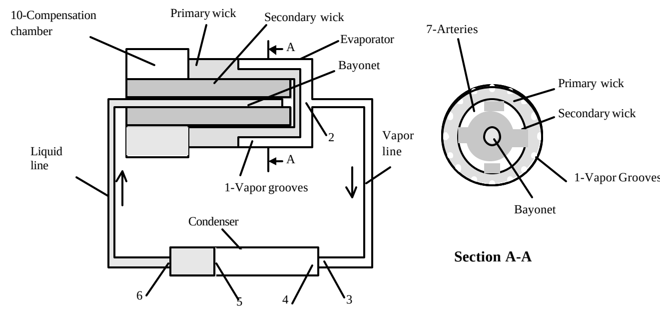


Figure 1. Flow Schematic of an LHP

Regardless whether or not vapor bubbles are present, the evaporator core can be considered as an extension of the compensation chamber, and both have the same absolute pressure during steady operation.

THERMOHYDRAULICS OF LHP

Since the compensation chamber is in line with the fluid returning from the condenser, another saturation state which represents the compensation chamber now exists on the liquid return line. Because the two saturation states are thermodynamically related, the following condition must be satisfied for an LHP:

$$\Delta P_{tot} - \Delta P_w = (dP/dT)\,(T_e - T_{cc}) \qquad (5)$$

where T_e is the saturation temperature of the vapor inside the evaporator grooves, T_{cc} is the saturation temperature of the fluid in the compensation chamber, and dP/dT is the slope of the pressure-temperature saturation line at T_{cc}. This equation states that, for a given pressure differential between the evaporator and the compensation chamber, a corresponding difference in the saturation temperatures must also exist between the two elements so as to generate exactly the same pressure differential.

A thermodynamic analysis of a capillary two-phase system can help the understanding of thermal and hydraulic processes in the LHP operation [11]. Figure 2 shows a pressure versus temperature diagram during steady operation of an LHP. The numbers in the diagram correspond to the physical locations shown in Figure 1. The vapor generated at the outer diameter of the evaporator wick (point 1) is at a saturation state. As the vapor flows along the vapor grooves, it becomes superheated at the exit of the evaporator (point 2) due to heating and a decrease in the absolute pressure. Assuming the vapor line is perfectly insulated, the temperature of the vapor will remain unchanged. Since the pressure continues to drop along the way, the vapor becomes more and more superheated relative to the local saturation pressure until it reaches the entrance of the condenser (point 3). The vapor gives out its sensible heat and begins to condense inside the condenser (point 4). The vapor condensation takes place along the saturation line where both the pressure and the temperature decrease. At point 5, the vapor condensation is completed, and the liquid continues to be subcooled inside the condenser until it exits at point 6. The subcooled liquid flows in the liquid line, where its temperature may increase or

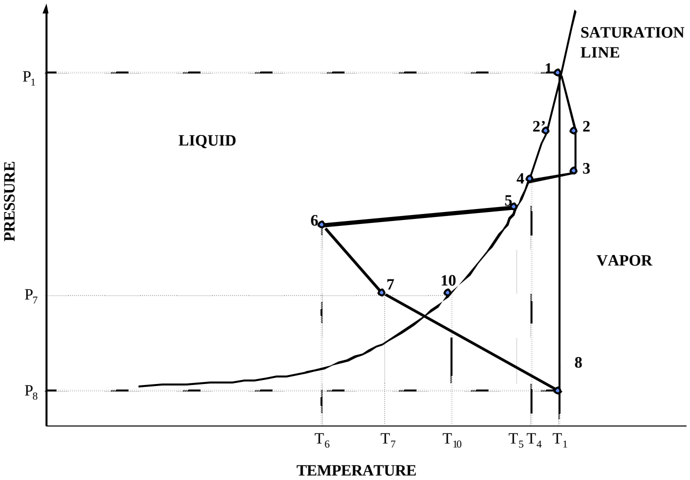


Figure 2. Pressure versus Temperature Diagram of an LHP

decrease, depending on whether the liquid loses or gains heat from ambient. As the liquid reaches the evaporator core (point 7), its pressure is at P_7 and its temperature at T_7. Since there is no flow between the compensation chamber and the evaporator core during steady state, the saturation pressure P_{10} in the compensation chamber must be equal to P_7.

From thermodynamic states shown in Figure 2, the requirement expressed by equation (5) can be written as:

$$P_1 - P_{10} = (dP/dT)\,(T_1 - T_{10}) \qquad (6)$$

The evaporator core temperature T_7 can be subcooled and deviate from T_{10}. In most cases, however, liquid evaporation also takes place inside the evaporator core. Under such a condition, points 7 and point 10 become the same, and the evaporator core is at the same temperature as the compensation chamber. Whether or not the evaporator core contains vapor bubbles affects the heat leak from the evaporator to the compensation chamber, and has a significant impact on the loop operation. The presence of vapor bubbles shortens the heat flow path and significantly increases the heat leak.

The derivative dP/dT in equation (6) can be related to physical properties of the working fluid by using the Clausius-Clapeyron equation:

$$dP/dT = \lambda / (T_{10}\,\Delta v) \qquad (7)$$

where Δv is the difference in the vapor and liquid specific volumes and λ is latent heat of vaporization of the working fluid. Equation (6) clearly illustrates that the pressure drop between the evaporator and the compensation chamber due to flow losses must be supported by the difference in the saturation temperatures. The meniscus in the evaporator wick merely works as a vapor/liquid separator which prevents a vapor back flow. When external conditions change, the compensation temperature and the loop operating temperature will both move along the saturation line in search for a new equilibrium state that will satisfy equations (1) through (5). This is how the LHP operating temperature is determined.

The coupling between the pressure drop and the temperature drop across the evaporator wick is responsible for many of the peculiar behaviors found in LHP operation. It will be discussed later that equation (5) has serious implications in the loop operating temperature at low heat loads as well as at high elevations in ground testing. It may impose a low power limit for LHP operation, or result in temperature overshoot during start-up. The only exception to equation (5) is when the compensation chamber is completely filled with liquid. Under such a condition, the compensation chamber and the evaporator core are subcooled, and both point 7 and point 10 in Figure 4 move horizontally to the left.

COMPENSATION CHAMBER SIZING AND FLUID INVENTORY

The compensation chamber is a critical component of the LHP and its design must be considered very carefully. The size of the compensation chamber and the fluid inventory affect the overall heat conductance, the low power start-up, and other performance characteristics. The fluid inventory in the LHP is a function of the operating temperature range and the volumes of the compensation chamber and other elements. Even though there is no theoretical upper limit for the compensation chamber volume, space and weight constraints require the volume to be optimized. In addition, the compensation chamber must satisfy the minimum volume requirement. The compensation chamber volume must be able to accommodate at least the liquid swing volume (and density changes) between the hot case and the cold case of the loop operation. In the hot case, a maximum heat load is applied to the evaporator and the condenser sink temperature is at its maximum. In the cold case, no heat load is applied to the evaporator and the condenser sink temperature is at its minimum. The LHP is usually charged such that some liquid is left in the compensation chamber when the rest of the loop is completely flooded under the cold case, and some vapor space is available in the compensation chamber when the condenser is fully utilized under the hot case. Sizing of the compensation chamber and the loop fluid inventory are usually considered concurrently. The general approach is described below.

The fluid inventory must satisfy the following relation under the cold case:

$$M = \rho_{l,c}\,(V_{loop} + \beta\, V_{cc}) + \rho_{v,c}\,(1-\beta) V_{cc} \qquad (8)$$

where M is the desired fluid inventory in the LHP, V_{loop} is the loop total volume excluding the compensation chamber, V_{cc} is the compensation chamber volume, β is the fraction of compensation chamber volume occupied by the liquid, and $\rho_{l,c}$ and $\rho_{v,c}$ are liquid density and vapor density, respectively, of the working fluid at the minimum temperature. The same fluid inventory must also satisfy the following relation under the hot case:

$$M = \rho_{l,H}\,[V_{liq} + V_{pw} + V_{sw} + (1-\alpha)\,V_{cc}] + \rho_{v,H}\,(V_{gr} + V_{vap} + V_{con} + \alpha\, V_{cc}) \qquad (9)$$

where V_{liq} is the volume of the liquid line; V_{pw} , and V_{sw} are the void volumes of the primary wick and the secondary wick, respectively; V_{gr} , V_{vap} , and V_{con}, are volumes of the evaporator grooves, the vapor line and the condenser, respectively; $\rho_{l,H}$ is the liquid density and $\rho_{v,H}$ is the vapor density of the working fluid; and α is the void fraction of compensation chamber volume. Note that $V_{loop} = V_{liq} + V_{pw} + V_{sw} + V_{gr} + V_{vap} + V_{con}$. Values of α and β are selected at the designer's discretion. A careful selection of these two values will yield an optimal compensation chamber volume. Once values of α and β are determined, the compensation chamber volume and the fluid inventory can be calculated from the above equations. Finally, the fluid charge must be checked against the upper limit allowed by the loop. An upper limit exists because the loop must be able to contain all the liquid volume at the maximum non-operating temperature in order to prevent bursting due to the hydrostatic pressure. Thus the following constraint applies:

$$M \le \rho_{l,max} (V_{loop} + V_{cc}) \qquad (10)$$

where $\rho_{l,max}$ is the liquid density at the maximum non-operating temperature. Equation (10) can often become the driver in the compensation chamber sizing.

LHP OPERATING CHARACTERISTICS

<u>Loop Operating Temperature</u>

As stated previously, the compensation chamber in an LHP is located near the evaporator and is in line with the fluid returning from the condenser. Thus, the compensation chamber temperature is related to the evaporator temperature and the enthalpy of the returning fluid. Although it still controls the loop operating temperature, the compensation chamber temperature itself is a function of the evaporator heat load, the condenser sink temperature and the temperature surrounding the liquid return line. In order to explain the complex phenomenon involved in the determination of the LHP operating temperature, a simplified thermal network of the LHP model is presented in Figure 3. The original diagram was presented by Bienert and Wolf [13] with more detailed accounts of heat transfer between various elements. The thermodynamic state of the fluid and the heat input/output at each location is clearly defined. In this diagram, Q is the heat transfer rate, T is the temperature, m is the fluid mass flow rate, h is the enthalpy, and q is the vapor quality. The subscripts for various elements are self-explanatory.

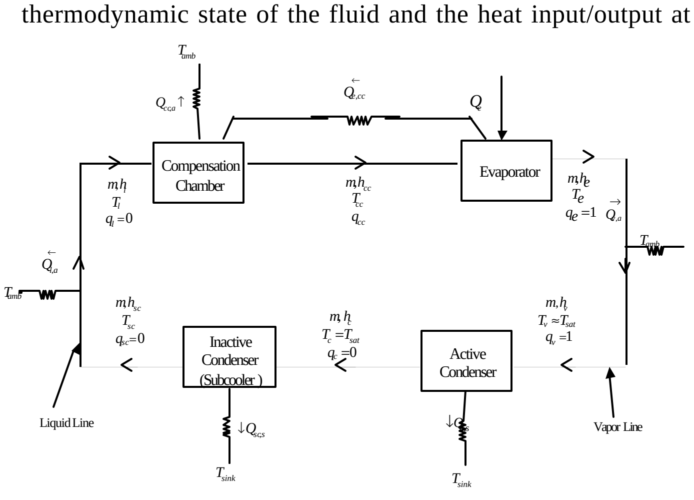


Figure 3. Simplified Thermal Network of the LHP Model.

The compensation chamber can exchange energy with the evaporator, the environment, and the liquid returning from the condenser. Part of the heat applied to the evaporator goes directly to the vaporization of the liquid, and the other part leaks to the compensation chamber. Thus,

$$Q_e = Q_{e,cc} + Q_{e,vap} \qquad (11)$$

$$Q_{e,vap} = m\lambda \qquad (12)$$

$$Q_{e,cc} = G_{e,cc} (T_e - T_{cc}) \qquad (13)$$

where m is the mass flow rate, and $G_{e,cc}$ is the thermal conductance between the evaporator and the compensation chamber. When the evaporator core is completely filled with liquid, heat transfer between the compensation chamber and the evaporator is by heat conduction through the hermetic case, and $G_{e,cc}$ is usually small. However, if vapor is present in the liquid core, additional heat is transmitted by conduction through the primary wick and then by evaporation and condensation through the vapor arteries inside the secondary wick, much like a heat pipe. Such a vapor connection almost always exists and is the primary heat transfer mechanism between the compensation chamber and the evaporator. The heat load to the evaporator directly impacts the heat leak to the compensation chamber.

Heat exchange between the compensation chamber and its environment is by radiation or convection. Active heating or cooling of the compensation chamber is also possible. The enthalpy of the fluid returning from the condenser affects the compensation chamber temperature by fluid mixing. Under steady state, the heat leak from the evaporator to the compensation chamber must be balanced by the subcooled liquid returning from the condenser, assuming the heat exchange between the compensation chamber and the environment is negligible. Therefore,

$$Q_{e,cc} = mC_p \Delta T = mC_p (T_{cc} - T_{in}) \qquad (14)$$

where C_p is the liquid specific heat, ΔT is the liquid subcooling and T_{in} is the liquid temperature at the entrance to the compensation chamber. The liquid exiting the condenser section will exchange heat with its surroundings as it flows along the liquid line. The temperature difference can be expressed as:

$$T_{in} - T_{sc} = Q_{l,a} / (mC_p) \qquad (15)$$

where T_{sc} is the temperature of the liquid leaving the subcooler, and $Q_{l,a}$ is the heat leak to the liquid from the surroundings. Note that the mass flow rate is a function of the heat load to the evaporator. In most LHP operation the sink temperature is lower than the ambient. At a low heat load, the mass flow rate is small and the liquid will usually have a long residence time in the liquid line, resulting in a high heat gain and low subcooling. Consequently, T_{in} is close to the ambient temperature. At a high heat load, the mass flow rate is high and the heat gain is low. Thus, T_{in} is close to T_{sc}. Effects of the heat gain along the liquid line at low and high powers will be referred to frequently in the following discussions. Because of the cross-link of the energy balance in the condenser, the liquid line, the evaporator, and the compensation chamber, an LHP usually requires a much longer time to achieve a steady state than a CPL.

Figure 4 depicts the operating temperature as a function of the evaporator heat load during the test of the NRL LHP [14], which used ammonia as the working fluid. This figure is typical of LHP operating temperatures when the condenser sink temperature is lower than the ambient temperature. This test was conducted at a condenser sink temperature of 0 ^{0}C and an ambient temperature of 23 ^{0}C. The operating temperature as a function of the heat load can be explained as follows. At low heat loads, the condenser is only partially utilized for vapor condensation. Liquid exiting the condenser is close to the sink temperature. As the liquid moves along the liquid line, its temperature rose to near the ambient temperature as it enters the compensation chamber due to heat leaks from the surroundings. The decreased subcooling is compensated for by an increase in the compensation chamber temperature in order to balance the heat leak from the evaporator. From equation (14), a substantial increase in the compensation chamber temperature may be needed because of a very low mass flow rate. As the heat load increases, the mass flow rate also increases and the heat gain of the returning liquid decreases. Because of increases in both the liquid subcooling and the mass flow rate, the compensation chamber temperature decreases. This trend continues until the condenser is fully utilized and the compensation chamber temperature reaches a minimum. The loop operates under a variable conductance mode over this power range. In this region, the primary factor determining the compensation chamber temperature is the heat gain of the returning liquid.

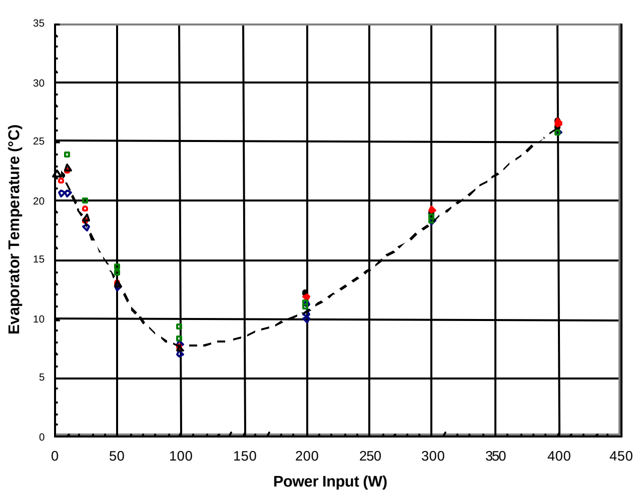


Figure 4. Operating Temperature of NRL LHP

As the heat load continues to increase, the condenser can no longer dissipate the excess energy. Thus, warmer fluid will flow back to the compensation chamber, forcing its temperature to increase until the condenser regains its heat dissipation capability. At equilibrium, the condenser remains fully utilized. The operating temperature increases almost linearly with the heat load, and the loop operates under a constant conductance mode. In this region, subcooling of the liquid exiting the condenser dominates the loop operating temperature. In theory, vapor can return to the compensation chamber in steady LHP operation as long as the compensation chamber can dissipate the additional energy. Under this condition, the compensation chamber simply works as the condenser. Such a heat dissipation by the compensation chamber is not practical in real applications. Thus, the fluid returning to the compensation chamber is usually subcooled liquid.

Figure 5 shows experimental data in testing the GLAS prototype LHP using ammonia as the working fluid [9]. The ambient temperature was 297K and two condenser sink temperatures were used. In the high power region, a higher sink temperature yields a higher operating temperature. A higher sink temperature means a warmer liquid is returning to the compensation chamber. The compensation chamber has to raise its saturation temperature in order to compensate for the reduced subcooling. In the low power region, the two curves began to merge regardless of the condenser sink temperature. Even though the colder sink provides a colder liquid at the condenser exit, parasitic heat gains along the liquid line heat the liquid to the same temperature anyway because slow liquid motion allows a sufficient time for heat transfer. Since a lower sink temperature provides higher liquid subcooling to the compensation chamber, the minimum operating temperature occurs at a higher power with a lower sink temperature. Experimental data on operating temperature as a function of the sink temperature abound [13, 16, 17].

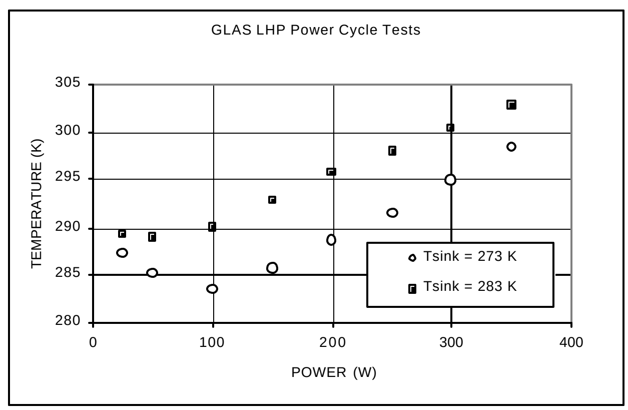


Figure 5 Operating Temperature of GLAS LHP

Likewise, the ambient temperature affects liquid enthalpy returning to the compensation chamber. When the sink temperature is lower than ambient, liquid gains heat from ambient along the liquid line. The larger the temperature difference, the higher the heat gain. Moreover, the effect of the ambient temperature on the loop operating temperature becomes more pronounced at low powers. Because the liquid line can not be perfectly insulated, it is very difficult to achieve steady LHP operation at low powers even if the ambient temperature fluctuates only a few degrees in ambient tests. When the ambient is colder than the condenser sink, the liquid line becomes the subcooler section of the condenser, and the loop operating temperature will continue to decrease with decreasing powers throughout the entire power range.

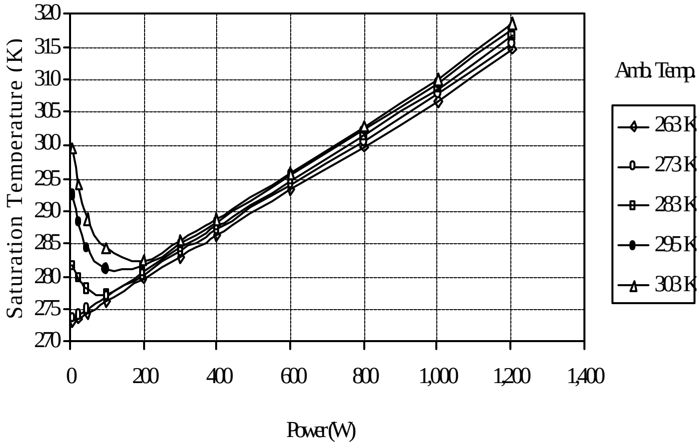


Figure 6 GLAS LHP Temperature Predictions

The same is true if the liquid line is perfectly insulated. Figure 6 presents the effect of the ambient temperature on the loop operating temperature as predicted by a computer model [18]. In this figure, the sink temperature is fixed at 273K while the ambient temperature varies between 263K and 303K. When the ambient temperature is equal to or lower than the sink temperature, the operating temperature decreases with decreasing power over the entire power range.

<u>Operating Temperature Control</u>

The previous section discussed the LHP operating temperature as a function of the evaporator power, sink temperature, and heat gain of the liquid line. The compensation chamber saturation temperature was left to reach its own equilibrium temperature. In some space applications, the instrument requires the evaporator temperature to be maintained within a very narrow range. This can be accomplished within certain limitations by actively controlling the compensation chamber saturation temperature.

Active control of the compensation chamber temperature can be illustrated by examining the two temperature curves shown in Figure 7. The two curves represent typical operating temperatures as a function of evaporator powers at two sink temperatures. It is assumed that the condenser is colder than ambient in either case. One can

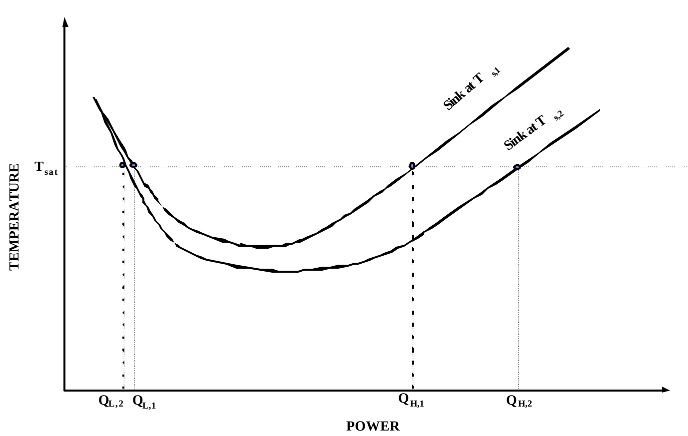


Figure 7 Operating Temperatures at Different Sink Temperatures

draw a horizontal line representing the desired compensation chamber saturation temperature, T_{sat}. This line intersects the upper curve at two points having power inputs of Q_{L1} and Q_{H1}, respectively. For power inputs between Q_{L1} and Q_{H1} , the saturation temperature can be controlled at T_{sat} by providing additional heat to the compensation chamber. The required heater power is represented by ($mC_p\Delta T$), where m is the fluid mass flow rate, C_p is the liquid specific heat, and ΔT is the difference between T_{sat} and the natural equilibrium temperature of the compensation chamber without the external power.

The power input Q_{H1} represents the maximum condenser heat dissipating capability at the saturation temperature T_{sat} and the sink temperature T_{s1}. For power inputs greater than Q_{H1}, the compensation chamber is heated above the controller's set point by the warmer returning fluid, essentially deactivating the controller. In order to maintain the saturation at T_{sat}, either the radiator size has to increase, or the sink temperature has to decrease. Both have the same effect of providing sufficient subcooling to the compensation chamber to maintain its equilibrium temperature. The latter is represented by the lower curve in Figure 7. In theory, active cooling can also be provided to maintain the compensation chamber saturation temperature. However, such a proposition is impractical because the compensation chamber then has to function as a condenser to reject the additional heat in excess of the condenser's capability.

For evaporator powers lower than Q_{L1}, reduced liquid subcooling due to higher heat gains forces the compensation chamber to raise its equilibrium temperature above T_{sat}, again deactivating the controller. As shown in Figure 7, lowering the condenser sink temperature will have little effect on the compensation chamber temperature because the heat gain of the returning fluid dominates the process. It is feasible to use a thermoelectric cooler to maintain the compensation chamber temperature in this region because the required cooling will be small. However, this has not be demonstrated.

Note that external power required to maintain the compensation chamber at the set point, $mC_p\Delta T$, varies with the evaporator power and the sink temperature. The maximum heater power does not necessarily correspond to the lowest temperatures on these curves because the mass flow rate depends on the evaporator power. An LHP with an evaporator power of 500W and a subcooling of 10K will require an external power of 20W to maintain the compensation chamber at 298K using ammonia as the working fluid. The required power increases with evaporator heat load for the same subcooling. These values are high compared to those in a CPL.

Maintaining the compensation chamber temperature at a higher set point by using an external heater is equivalent to raising the subcooling of the liquid in the condenser and the liquid line; the additional subcooling is in turn compensated by the additional power. By doing so, the overall conductance of the device degrades because the condenser becomes underutilized and the temperature difference between the evaporator and the condenser increases for the same heat load. The decrease of overall thermal conductance may be secondary or inconsequential in some applications when tight operating temperature control is the primary requirement. Because the compensation chamber temperature is affected by many factors, the operating temperature in an LHP usually fluctuates more than that in a CPL under the same operating conditions. Precise control of the loop operating temperature is more difficult in an LHP than in a CPL.

Figure 8 shows the temperature profiles during the GLAS LHP testing [9]. The compensation chamber was kept at 298K by an electrical heater, the ambient temperature was 296K, the condenser sink temperature was varied between 263K and 283K, and the evaporator power varied between 100W and 300W. It can be seen that the evaporator and vapor line temperatures were in tandem with the compensation chamber temperature at all times. The compensation chamber temperature was controlled at 298K except when the sink temperature was raised to 283K and the evaporator power increased to 300W. Under that condition, the condenser reached its maximum capacity, and the compensation chamber temperature was forced to increase to 303K. The compensation chamber heater power was deactivated during this period. As the sink temperature lowered to 263K subsequently, the compensation chamber temperature dropped to 298K and was maintained by the control heater again.

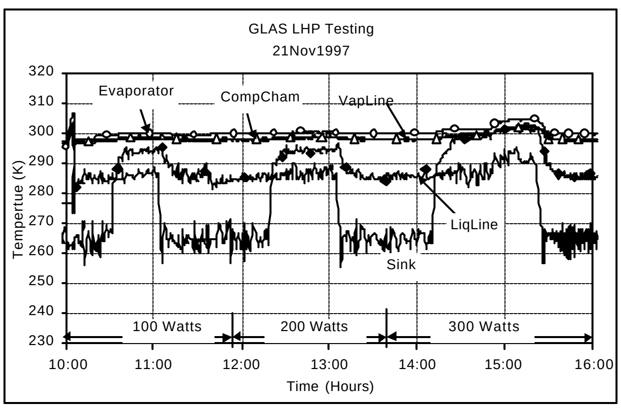


Figure 8 GLAS LHP Operation with Various Power and Sink Conditions

References 19 and 20 also presented test results on active control of the compensation chamber temperature. In those tests, the compensation chamber temperature was set to different values while the sink temperature and the evaporator power profile remained the same. As expected, the higher the compensation chamber set point temperature, the higher the evaporator power over which the loop operating temperature could be controlled because the condenser heat dissipation capability increased with an increasing vapor temperature.

Start-up

Start-up represents perhaps the most complex transient phenomenon in the LHP operation. On one hand, the secondary wick between the compensation chamber and the evaporator ensures that the evaporator is always replenished with liquid and the loop can be started simply by applying power to the evaporator without the need of a lengthy pre-conditioning process. Such a self-start feature is one of the attractions of LHP's. Tests indicate that the LHP can start successfully even with the liquid line initially at a high superheated state [21]. One the other hand, self-start does not necessarily imply instant or quick start. Start-up of an LHP is a function of the compensation chamber and evaporator construction, initial conditions inside the evaporator, and even the operation immediately prior to the start-up. In particular, the initial states of the working fluid across the primary wick in the evaporator play a vital role during the start-up transient. Some peculiar behaviors during start-up include temperature overshoot and the existence of a minimum power requirement. Furthermore, the way an LHP starts can have residual effects in subsequent operation.

There are four possible situations of the liquid/vapor states inside the evaporator/compensation chamber prior to start-up [22], as shown in Figure 9. Also shown in the figure are temperature profiles of the compensation chamber and evaporator in each situation from the moment power is applied to the evaporator. In all cases it is assumed that prior to applying power to the evaporator the entire loop is in equilibrium with ambient except for the condenser, which is colder. It needs to be emphasized that the temperature profiles subsequent to start-up are functions of the heat leak from the evaporator to the compensation chamber and the initial liquid line temperature, and the profiles shown in the figure are only pertinent to the conditions described above. The focus here is the system behavior prior to liquid evaporation or nucleate boiling.

There are three major factors that affect the LHP start-up. First, if the evaporator grooves are completely liquid filled, a superheat is required to initiate nucleate boiling. If the vapor grooves already contain vapor, liquid will evaporate as soon as power is applied without any superheat requirement. Second, if the evaporator core is liquid filled, the

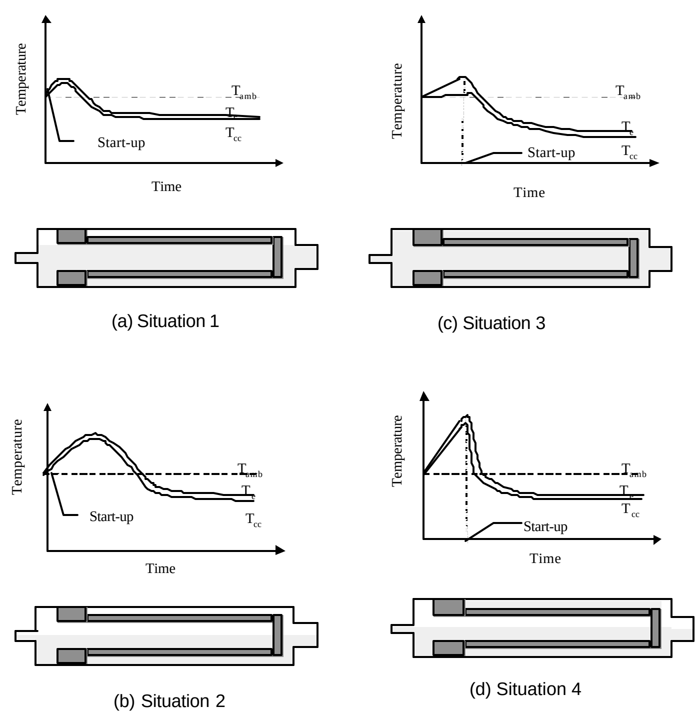


Figure 9 LHP Start-up Scenarios

heat leak from the evaporator to the compensation chamber is minimal since heat is transmitted by conduction through the evaporator shell. However, if vapor is present in the evaporator core, heat is transmitted through the primary wick and reaches the vapor immediately. The evaporator core becomes a vapor extension of the compensation chamber. Heat leak therefore increases substantially. Third, the evaporator power also affects the start-up through interactions with the other two factors.

In situation 1, vapor phase exists in the vapor grooves at the outer diameter of the wick, and liquid completely fills the evaporator core. This is the most benign case for LHP start-up. As the heat load is applied to the evaporator, liquid will start evaporating immediately since no liquid superheat is required. Because the evaporator core is filled with liquid, the heat leak from the evaporator to the compensation chamber is small. The cold liquid returning from the condenser will not reach the compensation chamber until some time after start-up. Then the compensation chamber and the evaporator temperatures decrease, and eventually the system reaches equilibrium.

In situation 2, vapor exists in both the vapor grooves and the evaporator core. Start-up under such a condition is similar to that in situation 1, where liquid evaporation starts as soon as a heat load is applied to evaporator. However, because vapor also exists in the evaporator core, a much larger heat leak from the evaporator to the compensation chamber is realized. Until the cold liquid from the condenser can compensate for the heat leak, both the compensation chamber and evaporator temperatures will continue to rise.

In situation 3, both the vapor grooves and the evaporator core are flooded with liquid. The saturation state exists elsewhere in the loop where liquid and vapor phases coexist at the ambient temperature. As a heat load is applied to the evaporator, the liquid in the vapor grooves must be superheated before nucleation can be initiated. Since the heat leak from the evaporator to the compensation chamber is small, the compensation chamber temperature remains virtually unchanged as the evaporator is being heated. At the boiling incipience, vapor could penetrate the evaporator wick if the superheat is high enough. Vapor can also be generated inside the evaporator core after the boiling inception due to the high thermal conductivity of the metal wick. Immediately after boiling, the evaporator temperature drops sharply and the compensation chamber temperature begins to control the loop operating temperature. The system eventually reaches equilibrium. This situation is closest to fully flooded CPL start-up.

Situation 4 presents the most difficult condition for LHP start-up. The compensation chamber is at the saturation state. The evaporator grooves are filled with liquid while the evaporator core contains vapor. As the heat load is applied to the evaporator, a liquid superheat is required to initiate nucleation in the evaporator grooves. However, the heat leak from the evaporator to the compensation chamber also raises the compensation chamber temperature. If the temperature of the liquid inside the vapor grooves can rise at a faster rate than the compensation chamber, the required superheat can be achieved and boiling will start. Otherwise, the required liquid superheat for nucleate boiling will never be achieved and the loop will not start. The evaporator and the compensation chamber will eventually be heated to such a high temperature that the applied heat load is dissipated to the surroundings by convection and/or radiation. Thus, there exists a minimum heat load below which the LHP will not start. Moreover, even if the applied heat load is large enough to start the loop, the temperature at the boiling incipience may exceed the maximum allowable temperature. Such a temperature overshoot must be considered in the LHP design.

Figures 10 and 11 depict the start-up temperatures during GLAS LHP tests when the compensation chamber and the evaporator were on the same horizontal plane [9]. As a heat load of 100W was applied to the evaporator, both the compensation chamber and evaporator temperatures rose together as shown in Figure 10. When the superheat reached about 3 K, nucleate boiling began and both temperatures

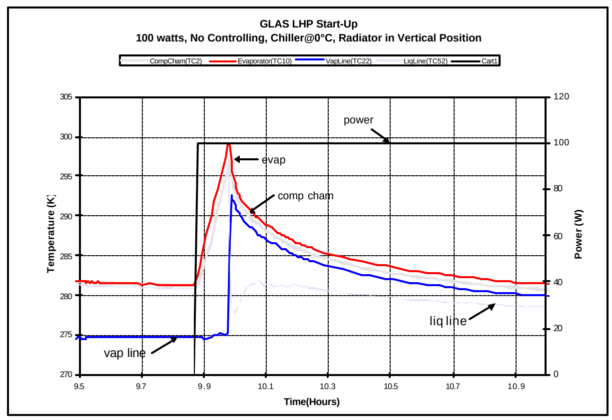


Figure 10 GLAS LHP Start-up – Case 1

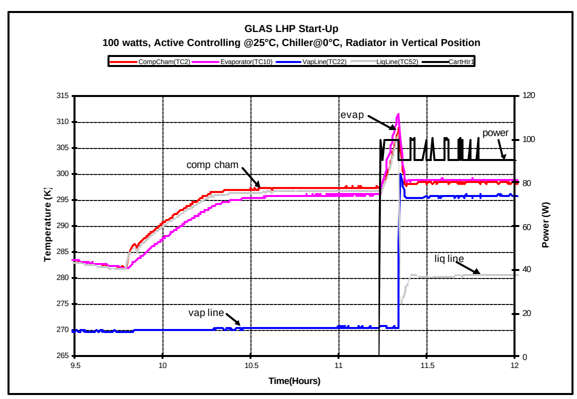


Figure 11 GLAS LHP Start-up – Case 2

dropped sharply. In Figure 11, power was applied to the compensation chamber to keep its temperature at 298K prior to start-up. While the compensation chamber was being heated, the evaporator temperature also rose in tandem with the compensation chamber. As a heat of 100W load was applied to the evaporator, both evaporator and compensation chamber temperatures rose again, similar to those in Figure 10 after a heat load was applied to the evaporator. These two tests strongly indicated a very large thermal conductance between the compensation chamber and the evaporator, similar to situation 4 discussed above. The quick response of the evaporator to the compensation chamber temperature, and vice versa, means the heat transfer is through a vapor connection. Such a temperature overshoot during start-up was also reported in other studies [14, 23, 24].

Tests of the GLAS LHP were also conducted by rotating the loop by 90 degrees such that the compensation chamber and the evaporator were vertical with the compensation chamber above the evaporator [9]. Figure 12 shows that as power was applied to the evaporator the compensation chamber temperature remained constant. When the evaporator temperature rose about 3 K higher than the compensation chamber, boiling started. In Figure 13, the start-up procedure was identical to that shown in Figure 11, where the compensation chamber was controlled at 298 K prior to start-up. During the heat-up of the compensation chamber, the evaporator temperature remained unchanged. Then as the evaporator was heated, the compensation chamber temperature remained unchanged. Boiling occurred at a superheat of 3 K. The initial condition with vertical compensation chamber and evaporator corresponds to that shown in situation 3, and the start-up was similar to that demonstrated by a fully flooded CPL.

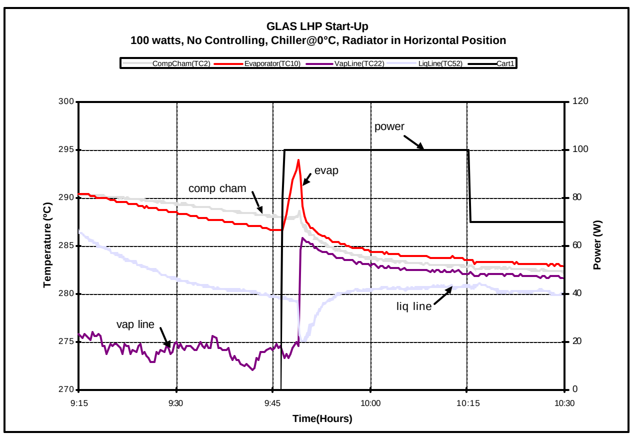


Figure 12 GLAS LHP Start-up – Case 3

Another parameter affecting LHP start-up is the superheat at the boiling inception. In the NRL LHP tests, some start-ups required 10 K superheat and some needed only

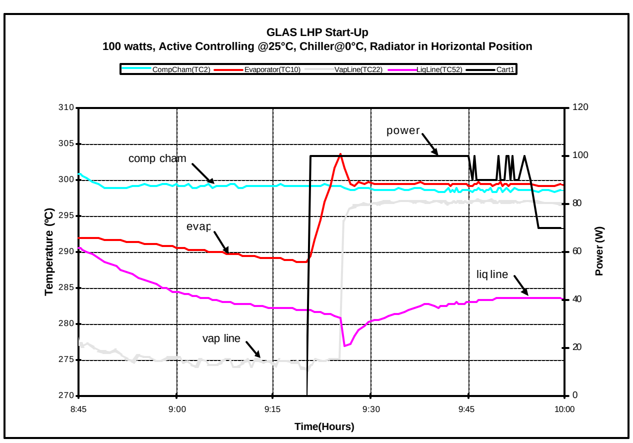


Figure 13 GLAS LHP Start-up – Case 4

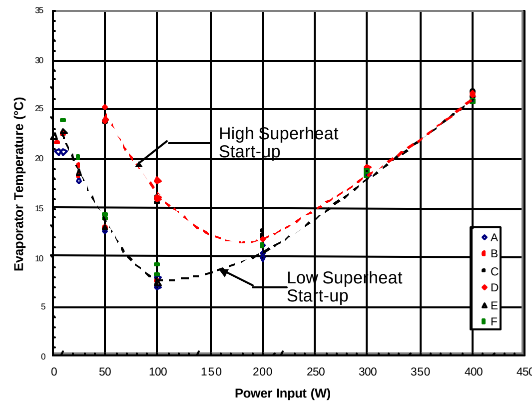


Figure 14 Operating Temperatures of NRL LHP with High and Low Superheats

1 K superheat, all under seemingly identical conditions, i.e., sink temperature at -10 ^{o}C and evaporator power at 50 W. The corresponding steady state temperatures after the start-up were 23.9 ^{o}C and 13.3 ^{o}C, respectively. Moreover, the difference in the start-up steady temperatures carried to subsequent tests for powers of 200 W or smaller, as shown in Figure 14 [14]. For powers of 300 W or higher, the temperature difference disappeared. It is possible that the initial condition prior to start-up discussed earlier affected the boiling superheat. It appeared that the high superheat of 10 ^{o}C caused the vapor to penetrate the primary wick and change the two-phase flow structure inside the evaporator, thereby increasing the heat leak from the evaporator to the compensation chamber. The residual effect of start-up was important only at low powers because the heat gain of the returning liquid dominated the loop operating temperature at low powers. At high powers, some of the bubbles inside the evaporator core might have been collapsed due to a large value of subcooling ($mC_p\Delta T$), eliminating the residual effect of the start-up.

Temperature Hysteresis

Results from testing several LHP's indicated that the LHP operating temperature also depended on whether the evaporator power was increasing or decreasing. Even though the test conditions, i.e. the evaporator power, the sink temperature and the ambient temperature were all the same, the loop operated at different temperatures at different times. Such a disparity in the loop operating temperature for otherwise the same test conditions is termed "hysteresis".

Figure 15 shows the operating temperature of the NRL LHP with the loop in a horizontal plane over a 72-hour period of continuous operation [14]. The loop started with 10W and the evaporator temperature stabilized at 24 ^{o}C after 5 hours. The evaporator power was then stepped up and down between 50W and 400W. With moderate power changes (<100W), the evaporator temperatures repeated consistently throughout the up and down cycles over the entire power range. However, following a power cycle of 50W/400W/50W, the evaporator steady state temperature increased by 4 ^{o}C at 50W. Subsequent power increase from 50W to 100W/200W/300W/400W/100W confirmed that there was a consistent up-shift trend in the evaporator temperature for powers of 200W or less.

When the heat load to the evaporator decreases, liquid will be injected from the compensation chamber through the evaporator core to the condenser in order to reduce the area for condensation. With a moderate power decrease, the secondary wick ensures only liquid is expelled. With a large power decrease, however, the pressure head required to expel liquid may exceed the capillary limit of the secondary wick. As a result, vapor is also expelled from the compensation chamber and accumulates in the evaporator core. As the vapor volume inside the evaporator core increases, so do the area for liquid evaporation and the heat leak from the evaporator to the compensation chamber. This is hypothesized as the physical process in the power turn down from 400W to 50W and the subsequent power cycle. Hysteresis is seen only at low powers because the liquid

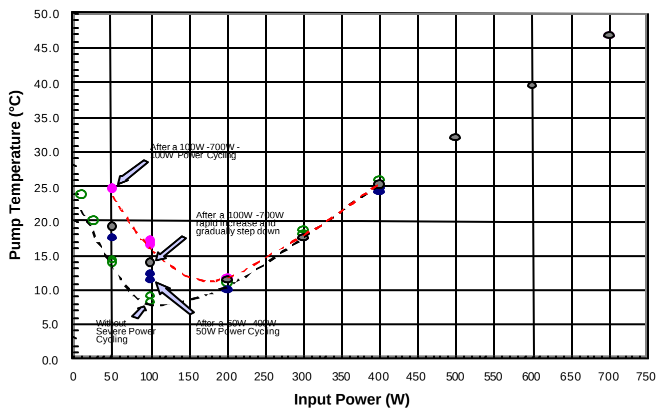
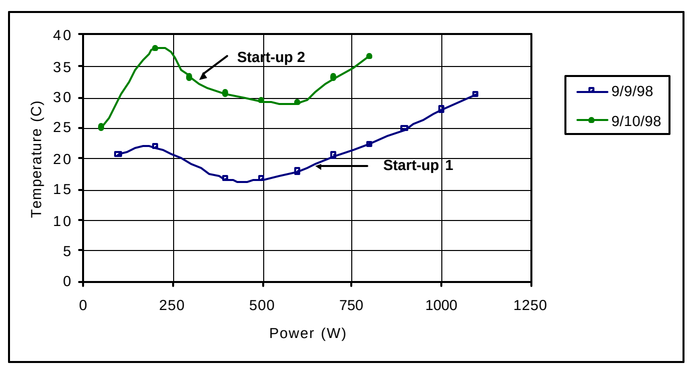


Figure 15 Temperature Hystereses in NRL LHP Operation

subcooling term ($mC_p\Delta T$), which will compensate for the increased heat leak from the evaporator to the compensation chamber, can only be increased by a larger ΔT through the rise of the compensation chamber temperature. At high powers, increased liquid subcooling due to large mass flow rates collapses excess vapor bubbles inside the evaporator core and hence eliminates temperature hysteresis.

The test shown in Figure 15 continued with a rapid power change of 100W/700W/100W, followed by a modest power cycle of 50W/100W/200W/100W. The loop operating temperatures were found to be even higher at powers of 200W or lower. A large power step down clearly augmented the hysteresis effect because more vapor bubbles were accumulated in the evaporator core. Finally, the evaporator power was increased to 700W and then gradually decreased to 600W/500W/400W/300W/ 200W/100W/50W. The purpose of this test was to reduce the dynamics of fluid expulsion and thereby reduce the possibility of injecting vapor from the compensation chamber to the evaporator. Figure 15 shows that the operating temperatures were significantly reduced for powers of 200W and below. This part of the test seemed to indicate a reduction in the vapor volume inside the evaporator core after the loop went through the low operating temperature region. Such a postulate is supported by other test data reported by Wolf and Bienert [25]. In their tests, the operating temperature of the LHP decreased as the heat load gradually increased from 10W to 25W/50W/75W/100W/150W. The heat load was then decreased in reverse order. At the same power level, the operating temperature was lower than the previous value for powers below 100W. Furthermore, their tests showed that the hysteresis occurred only at high adverse elevations. With the evaporator at a higher position, the condenser will be filled even faster when power steps down, which in turn demands the compensation chamber to expel fluid at such a high flow rate that the secondary wick can not sustain the resulting pressure drop.

Another type of temperature hysteresis was observed during the test of the Switch Box LHP, which was designed for terrestrial applications with the compensation chamber above the evaporator in a vertical configuration. This LHP used ammonia as the working fluid. There is no secondary wick in the design since gravitational force will pull liquid to fill the evaporator core. Tests were conducted with the evaporator and the compensation chamber located 4 meters above the condenser in an upright position. Figure 16 shows the evaporator temperatures at various powers in two heat transport tests. The lower curve represents the first test where the LHP deprimed at 1200W. The same test was repeated the very next day and temperatures are represented by the upper curve. The evaporator temperatures were higher in the second test than in the first test at all power levels. Apparently there was a residual effect from the deprime of the LHP in the first test. One will expect the evaporator core to be filled with liquid at all times by the gravitational force. However, vapor bubbles might have been trapped in the primary wick after the first test. With the evaporator temperature above 338K at deprime, the loop was left to cool down overnight in ambient. The evaporator temperature was still several degrees higher than ambient the next morning prior to the start of the second test. Apparently, the tiny bubbles entrapped in the wick were not completely collapsed. The temperature hysteresis was also reported in other studies [14, 24, 25].



Start-up 1 - 100 W start-up; chiller at -10 C
Start-up 2 - 50 W start-up; chiller at -10 C (Start-up after de-prime)

Figure 16 Operating Temperatures of Switch Box LHP

In summary, the temperature hysteresis is most likely caused by a change in the structure of the liquid/vapor mixture inside the evaporator core. Because of this change, the heat leak from the evaporator to the compensation chamber also changes. Consequently, more than one set of temperatures T_1 and T_{10} can satisfy equation (6).

Loop Deprime or Shutdown

During the survival mode of space instrument operation, little or no power is available to the instrument. The LHP is required to shut down during those periods; otherwise heat will be continuously transmitted to the sink and the instrument temperature may drop too low. The traditional method to shut down a CPL is to apply a heat load to the liquid line to boil off the liquid. The evaporator will deprime after it is vapor-filled. Once the loop deprimes, the heat load to the liquid line can be removed. Thus, the required shutdown energy is small. Unfortunately, this scheme will not work for the LHP because the evaporator is vapor tolerant. Adding heat to the liquid line will only increase the loop operating temperature and will not deprime the loop.

The only viable method (other than using a mechanical valve) to shut down an LHP is to remove the heat load from the evaporator and maintain the compensation chamber temperature above the evaporator temperature for the entire shutdown period. As soon as the evaporator temperature drops below the compensation chamber set point, the evaporator will be flooded with liquid, effecting a loop shutdown. Since there is no flow in the loop, the compensation chamber temperature will not be affected by the sink temperature. For reliability, a control heater with a thermostat can be installed on the compensation chamber. The thermostat can be set a few degrees higher than the minimum allowable for the evaporator. If the evaporator temperature ever rises above the compensation chamber set point again, the loop will start and operate at the compensation chamber set point temperature.

Effect of Elevation and Tilt

The heat leak from the evaporator to the compensation chamber has profound effects on the loop operating temperature. Such a heat leak depends on whether or not vapor exists inside the evaporator core, which in turn is a strong function of the elevation and tilt in one-G environments. An adverse elevation means the evaporator is above the condenser and an adverse tilt means the evaporator is above the compensation chamber.

The LHP operating temperature increases with increasing adverse elevations at low powers in ground operation. Tests of an ammonia LHP at two adverse elevations of 0.91 m and 2.74 m indicated a temperature difference of 8K at a heat load of 25 W [25]. The difference in operating temperatures decreased with an increasing heat load and eventually disappeared at powers of 200 W or higher.

The increase of the operating temperature with the elevation can be explained as follows. As the pressure difference across the evaporator wick increases due to gravity head, the difference in saturation temperatures must also increase, as dictated by equation (5). Since the liquid enthalpy entering the compensation chamber does not change, the only way to satisfy the increasing pressure drop is by an increase in the evaporator vapor temperature. However, as the vapor temperature increases, the heat leak from the evaporator to the compensation chamber also increases. The compensation chamber temperature must then increase in order to provide enough liquid subcooling to compensate for the increased heat leak. A higher compensation chamber temperature requires an even higher evaporator vapor temperature. Such reciprocal effects accumulate quickly. In general, the effect of increasing pressure drop due to increasing heat load can be compensated for by an increasing liquid subcooling, $mC_p\Delta T$, through the increase of the mass flow rate. However, the effect due to hydrostatic pressure, which is superimposed upon the frictional pressure drop, is independent of flow rate and can only be overcome by an increase in the loop operating temperature. This effect is therefore most pronounced at low powers. At high powers, the subcooling of the returning liquid is sufficient to balance the increased heat leak, and the increase in the loop operating temperature due to higher elevations becomes hardly noticeable.

This effect can be illustrated by a theoretical analysis. Consider an ammonia LHP operating at 300K in a horizontal position with a pressure gradient of 1000Pa across the evaporator wick due to frictional losses. The corresponding temperature difference in the saturation temperatures is 0.033K. The thermal conductivity is estimated to be about 4 W/m-K across a nickel wick with 70% porosity. A heat leak of 0.15 W from the evaporator to the compensation chamber can be expected based on heat conduction across a wick with inner and outer diameters of 11 mm and 21 mm, respectively, and a length of 125 mm. If the evaporator is now raised 2 meters above the condenser, the pressure gradient becomes 1000+12000=13000Pa, with a corresponding temperature difference of 0.43K and a heat leak of 2.0W. Assume the evaporator power is 25W, then the mass flow rate will be 2.15E-5 kg/s. The increase in subcooling required to offset the 1.85W increase in heat leak is 18K. In other words, the temperature difference across the LHP would have increased 18K simply by raising the evaporator 2 meters! Ground test data shows that the effect of elevation on the loop operating temperature is usually lower than predicted. In the above calculation, it is assumed the entire inner surface of the wick is available for liquid evaporation. Depending on the structure of the liquid/vapor mixture inside the evaporator core, the actual liquid evaporation area could be much less, thus providing a smaller temperature increase.

The elevation affects not only the loop operating temperature but also the start-up transient with low powers. More experimental data on ground tests with various combinations of the elevation and tilt can be found in the literature [24, 27, 28]

Effect of Evaporator Mass

Most LHP tests in the laboratory were conducted by attaching a heater directly to the evaporator itself. Because of its relatively small mass, the evaporator will respond quickly to any heat load change. In space applications, the instrument attached to the evaporator usually has a much larger thermal mass, and the increased mass will help damp out fast transients due to power changes. Consequently, the temperature hysteresis may be reduced or eliminated. On the other hand, a large evaporator thermal mass implies that the loop will most likely go through the low power operation where the loop operating temperature is more difficult to control. In addition, a larger thermal mass will make the start-up more difficult. It could increase the start-up time, or increase the temperature overshoot, or prevent a successful start-up. The actual outcome depends upon the initial conditions of the compensation chamber and the evaporator prior to start-up, and the specific design configuration.

Tests of an LHP with large thermal masses have been conducted and demonstrated successful operation [23]. The loop could be started successfully with heat loads as low as 3W. However, a temperature overshoot of 5 to 10 ^{o}C was observed if the compensated chamber was pre-heated.

Effect of NCG

Non-condensable gases (NCG's) can be generated for a number of reasons, including cleaning of the envelope and wicks, purity of the working fluid, and chemical reactions between the working fluid and the envelope materials. Ammonia has been shown to be compatible with aluminum and stainless steel during the development of heat pipes and CPL's. Similar cleaning procedures for heat pipes and CPL's were used for the fabrication and assembly of LHP's. Nickel and titanium have been used in the former Soviet Union for years with no adverse effects.

The generation of NCG can be investigated analytically based on past experience with heat pipes. The amount of NCG generated is a function of the amount of the working fluid, the surface area of the materials in contact, the operating conditions, and the time period of exposure. The NCG inventory can be projected at the end of life of the LHP service.

There are several destinations of the NCG generated in the LHP: 1) The NCG can collect at the compensation chamber where the flow is stagnant. 2) The NCG can accumulate in the condenser where the temperature is lowest. 3) The NCG can be absorbed and circulate with the working fluid around the loop. 4) The NCG can be absorbed by the envelope or wick materials. The NCG will affect LHP performance only if it appears as gas bubbles. In the condenser section, higher solubility will likely drive the gas into the solution. Some of the remaining NCG will block part of the condenser and reduce the system overall thermal conductance. The rest will flow to the compensation chamber and the evaporator. Because the evaporator core contains vapor arteries, the NCG will be allowed to vent to the compensation chamber. The NCG in the compensation chamber will increase the subcooling requirement for the liquid returning from the condenser since the liquid within the compensation chamber itself must be slightly subcooled in order to coexist with a mixture of working fluid vapor and NCG at the same temperature and pressure. The net effect is an increase of the loop operating temperature and a decrease of the overall thermal conductance.

An experimental study of effects of NCG on the LHP operation has been conducted by Nikitkin and Bienert [29], using ammonia as the working fluid. Tests were performed by injecting hydrogen gas into the loop through the vapor line. The amounts of hydrogen gas injected ranged from 1/3 to 10 times of the projected end-of-life inventory. Test results indicated that the effects of the NCG on LHP performance were minimal and no LHP failure was experienced during any test. In large quantities, NCG increased the start-up time and the operating temperature. Measured effects of NCG were considerably smaller than theoretically predicted. It appeared that gas was absorbed by the working fluid and by the envelope and wick materials.

Propylene is beginning to gain interest as the working fluid in applications where the condenser temperature is below 193K. NCG generation and its effects on the propylene loop operation need further investigation.

Effects of Convection

In ambient testing of the LHP, convection between ambient and the LHP components may play important roles. Effects due to convection can be eliminated in vacuum. The NRL LHP was tested in a thermal vacuum chamber for the sole purpose of investigating the effects of convective heat transfer on the loop performance. Instead of simulating space environments, all test conditions were set as close as possible to those in ambient tests. The thermal vacuum tests showed very similar results compared to the ambient tests in all aspects. However, the loop operating temperature was higher than its ambient counterpart at the same evaporator heat load and sink temperature once the operating temperature exceeded the surrounding temperature. The results indicated that free convection in ambient tests helped the compensation chamber to dissipate heat, thereby reducing the liquid subcooling requirement and resulting in lower operating temperature. The general "V" shaped temperature curve shown in Figure 7 had a larger slope at low and high powers when the loop was tested in vacuum. A corollary is that the power range over which the operating temperature can be actively controlled becomes narrower in vacuum environments.

Multiple Evaporators and Multiple Condensers

Discussions thus far have been focused upon LHP's with a single evaporator and a single condenser. There are several advantages of an LHP with multiple evaporators and multiple condensers. Such a system can provide a higher heat transport capability, and can be used to cool multiple heat sources or a heat source with large thermal footprints. Multiple evaporators also facilitate heat load sharing among evaporators, and multiple condensers can be placed at different locations on the spacecraft, therefore providing design and operation flexibility. Feasibility of such systems has been demonstrated [6, 30, 31]. Nevertheless, multiple components, especially multiple evaporators, do add complexities to the design and operation of an LHP.

Unlike in a CPL, no flow regulators are needed for multiple condensers in an LHP since the evaporator can tolerate vapor bubbles. However, the subcooling of the liquid returning to the compensation chamber is a function of how each condenser is utilized. Thus, it is possible to have a whole new set of temperature hysteresis, depending on how the flow is distributed among the condensers. Such a possibility has not been experimentally investigated. Adding flow regulators to the condensers may reduce, but will not eliminate, such temperature hystereses. Multiple evaporators can be plumbed in parallel with a common compensation chamber [6]. This configuration makes the LHP operation closely resemble that of a CPL. Because of the limited capillary pumping capability of the secondary wick, the configuration with a common compensation chamber is confined to evaporators that are close to one another. Another approach is to plumb individual evaporators in parallel, each having its own compensation chamber. LHP's with two evaporators and one condenser have been demonstrated to operate properly with power inputs ranging from 100W to 500W and with evaporators at different elevations [30]. Nevertheless, as the number of evaporators increases, the loop could exhibit some peculiar behaviors when compared to the single evaporator device. Some possible scenarios are analyzed below.

Recall that equation (5) must be satisfied in LHP operation unless the compensation chamber is completely filled with liquid. In a multiple-evaporator LHP, each compensation chamber would reach its own equilibrium and operate at its own saturation temperature if other evaporators were absent. However, since all evaporators are connected through the common vapor line and there can be only one operating temperature for the entire loop, the compensation chamber with the highest vapor temperature will prevail. All other evaporators are forced to operate at this higher temperature. This may result in excess subcooling unless the additional heat leak from the evaporator to the compensation chamber can compensate for the difference. Thus, it is possible that all compensation chambers except the one that controls the loop operating temperature will be hard filled with liquid. This can be illustrated by Figure 17 where an LHP with two evaporators and one condenser is shown. In order for both compensation chambers to contain vapor and liquid phases, the following conditions must be satisfied:

$$P_{1,E} - P_{1,C} = (dP/dT)\,(T_{1,E} - T_{1,C}) \qquad (16)$$
$$P_{2,E} - P_{2,C} = (dP/dT)\,(T_{2,E} - T_{2,C}) \qquad (17)$$
$$P_{1,C} - P_{2,C} = (dP/dT)\,(T_{1,C} - T_{2,C}) \qquad (18)$$
$$P_{1,E} - P_{2,E} = (dP/dT)\,(T_{1,E} - T_{2,E}) \qquad (19)$$

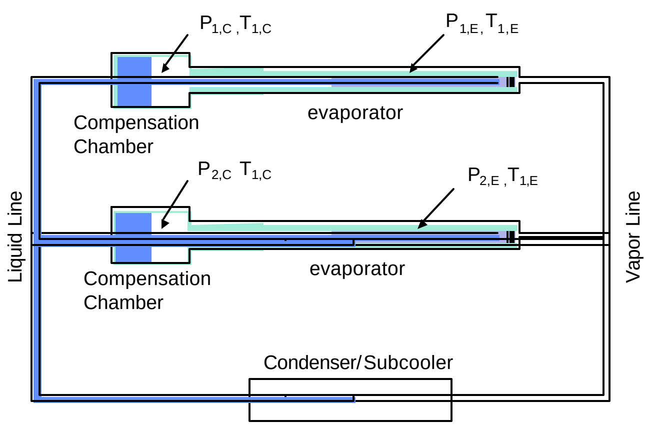


Figure 17. Schematic of an LHP with Two Evaporators

Equation (19) is automatically satisfied regardless which evaporator controls the loop operating temperature because the capillary forces will maintain the pressure balance. The other three equations, however, may not be satisfied simultaneously. This will become more evident when the number of evaporators increases, as will be elaborated on next.

For an LHP that has N evaporators and compensation chambers, the following thermodynamic relations must be satisfied in order for all compensation chambers to have two-phase fluid:

Between each individual evaporator and compensation chamber:

$$P_{i,E} - P_{i,C} = (dP/dT)\,(T_{i,E} - T_{i,C}) \qquad i = 1, N \qquad (20)$$

Between any pairs of compensation chambers:

$$P_{i,C} - P_{j,C} = (dP/dT)\,(T_{i,C} - T_{j,C}) \qquad i = 1, N \quad j = 1, N, \quad i \neq j \qquad (21)$$

Between any pairs of evaporators:

$$P_{i,E} - P_{j,E} = (dP/dT)\,(T_{i,E} - T_{j,E}) \qquad i = 1, N \quad j = 1, N, \quad i \neq j \qquad (22)$$

The pressure drop in any segment of the loop is fixed once the heat load distribution among all evaporators is fixed. All evaporators will operate at a nearly constant temperature with the pressure drops among them being sustained by the capillary forces. Because of external heat loads, the N(N-1)/2 relations expressed by equation (22) are automatically satisfied. However, it is difficult to satisfy the N(N+1)/2 relations expressed by equations (20) and (21) simultaneously. The difficulty arises because the pressure drop due to flow losses among any two elements is determined by the mass flow rate between them, but the temperature difference is determined by the prevailing thermal conditions which are independent of the pressure drop. In fact, it is very likely that only one evaporator/compensation chamber can satisfy equation (20), and all other compensation chambers become liquid-filled.

The operation of a multiple-evaporator LHP becomes even more complex when body forces are present. Body forces not only change the pressure drops between loop elements but also affect the equilibrium temperature of any compensation chamber with a low power to its evaporator. It becomes even harder for all two-phase elements to satisfy equations (20) and (21) simultaneously. This is why some of the compensation chambers will inevitably be flooded with liquid. Flooding of the compensation chamber will also occur when the evaporator is working as a condenser under the heat sharing mode. In a multiple-evaporator system, any number of the evaporators can work as condensers; the only requirement is that at least one evaporator has to receive net heat input.

Because individual compensation chamber temperature is a function of the power input to the respective evaporator, liquid redistribution must occur each time there is a change in the power distribution among evaporators. Such a process may lead to unstable operation of the entire system. The worst case is when one evaporator has a very low heat load while others have much higher heat loads. The evaporator with a low heat load will actually control the loop operating temperature. When the heat load distribution changes, as will happen when instruments are turned on and off alternately, flow redistribution due to fast transients may be problematic.

The above analysis is supported by ground testing of a water LHP having two evaporators with integral compensation chambers [30]. The two evaporators were tested with an elevation difference of 360 mm. It was shown that one of the compensation chambers was always hard filled with liquid. The elevation difference between the two evaporators in this test was modest. As the elevation increases, it will be even more difficult to maintain both compensation chambers at saturation states.

Even though flooding of some compensation chambers will not prevent the loop from its normal operation, it complicates the determination of fluid inventory and imposes limitation on the number of evaporators that can be used in an LHP. The loop must contain a minimum fluid inventory to accommodate the situation when all compensation chambers except one are flooded. On the other hand, the fluid inventory must not exceed the maximum amount expressed by equation (10) in order to avoid bursting at the maximum non-operating temperature. Bounded by these two extreme conditions, there exists a maximum number of evaporators that can be incorporated into an LHP. Increasing the size of the compensation chambers does not solve the problem because it is not known a priori which compensation chambers will be flooded; thus all compensation chambers must be sized the same. In fact, increasing the compensation chamber volume will actually reduce the number of evaporators allowed because of the condition imposed by equation (10).

The applicability of an LHP with multiple evaporators requires further studies. All the concerns described above have to be investigated and demonstrated. In practical applications, the fast transient of flow redistribution caused by rapid power or sink temperature changes is likely to be damped out because of the large thermal masses of the instruments attached to the evaporators. It is the author's opinion that the fluid inventory requirement imposes the real limitation on the number of evaporators that can be used in an LHP. Systems with more than three evaporators may be difficult to implement with the current LHP technology. One exception is when all evaporators are plumbed in parallel and are under the same or similar power profiles as is the case where all evaporators are built into a common cold plate.

<u>Redundant Loop Heat Pipes</u>

Figure 18 shows the schematic of a system with two-LHP's for redundancy. Heat pipes can be employed to connect the evaporators for better heat distribution. Redundant LHP's may be utilized to transfer the heat load which a single LHP is not able to, to enhance the system reliability, to isothermalize a heat source with large thermal footprints, or to serve as an alternative to a single LHP with multiple evaporators. Since each individual LHP has its own operating characteristics, there are some issues to be addressed for a system with redundant LHP's.

During the system start-up, all LHP's may not start at the same time because of different superheat requirements. The one that starts first may carry more and more heat load as its temperature continues to decrease. The other LHP's may not start at all until the active LHP increases its operating temperature due to high heat loads. In addition, since each evaporator will operate at its own equilibrium temperature and there is only one common heat source temperature, each evaporator will carry different heat loads. A corollary is that the heat source temperature depends more on the heat load distribution than the total heat load. Any disturbance that causes the heat load to redistribute will cause the heat source temperature to change. If any of the evaporators experiences a temperature hysteresis, the heat source temperature will change completely even though the total heat load and the sink condition remain the same.

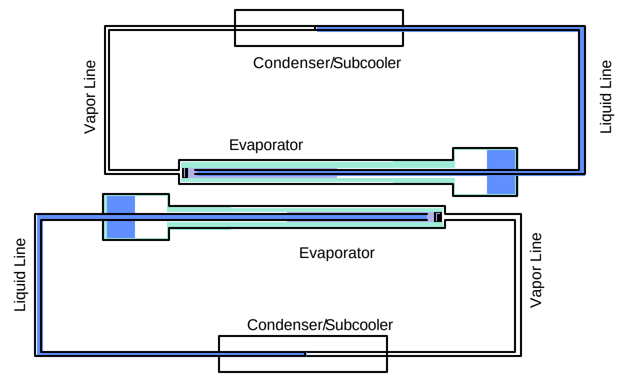


Figure 18. Schematic of a Redundant LHP System

If multiple LHP's are used to transport the total heat which is beyond the capability of a single LHP, enough margin must be given to the heat transport capability. All LHP's will not get an equal share of the total heat load. It is possible that some LHP's may not be operational until the others take very high heat loads. Even though this may not present any problems so far as the system operation is concerned, its impact on other thermal requirements such as temperature gradients on the heat source must be considered in the design. In this aspect, a single LHP with multiple evaporators can provide better isothermaization than redundant LHP's.

<u>Analytical Modeling</u>

Development of the single-evaporator LHP analytical model began a few years ago with the Russian company TAIS Ltd. This steady state model uses a numerical solution of the energy and pressure balances and is written in PASCAL language. Bienert and Wolf presented the first LHP analytical model developed in the United States [13], based on concurrent pressure and energy balances at each element for steady state. The energy balances were solved using a combined lumped parameter and nodal approach: the evaporator and compensation chamber were treated as lumped parameters, the remainder of the loop was treated as a nodal network. Several institutions have since developed their own

LHP models using similar approaches [15, 32]. Most of these models were written in a spreadsheet format.

Developments of mathematical models for LHP transient behaviors are being undertaken at several institutions. Most models utilize the thermal analyzer SINDA/FLUINT for interactions between the LHP and the surroundings [33]. No analytical model of multiple-evaporator LHP's, steady state or transient, has been reported. However, Cullimore and Ring Technologies has made available a free SinapsPlus™ "pre-built" model with graphical spreadsheet-like access to an underlying SINDA/FLUINT thermal/hydraulic model of a generic LHP [10].

Accurate predictions of the LHP performance require accurate tracking of all heat leaks and boundary conditions, especially of the compensation chamber and the liquid line. In addition, the pressure drops must be accurately modeled since they affect the heat leak from the evaporator to the compensation chamber. Like other two-phase systems, accurate mathematical modeling of LHP's is difficult due to the complex physical processes involved.

Flight Experiments

Several flight experiments have been conducted in order to characterize the LHP performance in micro-gravity and to verify the technology readiness for spacecraft applications.

A Russian LHP has been flown onboard the Granat spacecraft since December 1989, and the experiment is still on-going [34]. This LHP contains a nickel wick and uses propylene as the working fluid. Tests conducted include start-up, power cycle, and steady state operation. The heat load ranges from 5W to 38W and the sink temperature varies between –100 ^{o}C and + 66 ^{o}C.

An American made LHP was flown twice on the Space Shuttle STS-83 and STS-94 in 1997. The primary wick was made of titanium and ammonia was used as the working fluid. The flight experiment demonstarted heat loads between 22W and 292W. Successful tests were verified for start-up, power cycle, low power, high power and temperature control. Flight test also demonstrated the robustness of LHP operation for being vapor tolerant. Good correlation between one-G and micro-G was obtained.

Another LHP made in American was flown on the Space Shuttle STS-87 in 1997 [35]. One major goal of this project was to demonstrate the aility of duplicating the Russian technogy in the United States. The LHP utilizes a nickel wick and ammonia was selected as the working fluid. The flight test demonstrated excellent LHp performance over 213 hours of operating time. Test performed included start-up, power cycle, low power, high power, and temperature control. The heat load ranged between 12.5W and 400W while th the operating temperature varied between –27 ^{o}C and +66^{o}C. Flight data correlated well with ground test results.

SUMMARY

The operating principles and performance characteristics of LHP's were described. The design of the evaporator and the compensation chamber is mainly responsible for all the temperature behaviors of the LHP. The key to understanding LHP operation is to know the thermal and hydraulic interactions between the compensation chamber and other elements; the effects of various parameters on the operating temperature can then be explained.

The physical proximity of the compensation chamber to the evaporator and the use of a secondary wick simplify the LHP start-up and make the evaporator very vapor-tolerant. Both contribute to the robustness of LHP operation. On the other hand, because the compensation chamber is plumbed in-line with the flow circulation, the operating temperature is affected by the loop operating parameters and ambient conditions. The existence of two saturation states across the evaporator wick imposes additional thermodynamic constraints on the loop operation because the loop operating temperature is now directly linked to the loop *pressure drops*. Such effects become more pronounced at low powers and high adverse elevations. The use of metal wicks with fine pores increases the capillary pumping head. However, metal wicks increase the heat leak from the evaporator to the compensation chamber, and hence reduce the overall thermal conductance. The phenomenon of temperature hysteresis results most likely from the existence of different liquid/vapor structures inside the evaporator core under different conditions. The liquid/vapor structure in the evaporator core also affects many other aspects of the LHP operation.

LHP's with a single evaporator have gained increasing acceptance in spacecraft applications because of their high pumping capability and robust operation. Little has been studied regarding dual LHP's or LHP's with multiple evaporators. Advances in these areas will greatly enhance the LHP applications. Development of an analytical model which predicts the LHP transient behaviors is also urgently needed.

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