Heat Load Sharing in a Loop Heat Pipe with Multiple Evaporators and Multiple Condensers

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Abstract

This paper describes the heat load sharing function among multiple parallel evaporators in a loop heat pipe (LHP). Each evaporator can be attached to an instrument. In the normal mode of operation, each evaporator will cool the corresponding instrument by absorbing the waste heat. When an instrument is turned off, the attached evaporator can keep it warm by receiving heat from other evaporators serving the operating instruments. This is referred to as heat load sharing. The fact that the wicks in the powered evaporators will develop capillary pressure to force the vapor that is being generated to flow to cold locations where the pressure is lower leads to the conclusion that heat load sharing is an inherent function of an LHP with multiple evaporators. This paper presents a theoretical basis of the LHP heat load sharing operation, and experimental results from ground tests of an LHP with two evaporator and two condensers. Factors that affect the amount of heat being shared are also discussed.

Introduction

Loop Heat Pipes (LHPs) are versatile two-phase heat transport devices that have gained increasing acceptance for spacecraft thermal control [1, 2]. They have been used on several commercial communications satellites and NASA's spacecraft such as ICESAT, AURA, SWIFT and GOES [3, 4]. All LHPs currently servicing the orbiting spacecraft have a single evaporator and a single condenser.

An LHP with multiple evaporators has several additional advantages. First, it can transport a significantly larger heat load than an LHP with a single evaporator. Second, it can be used to isothermalize a payload having a large thermal footprint. Third, it can serve as a thermal bus for multiple instruments that require similar operating temperatures. Fourth, the loop is self-regulating and requires no flow control devices because each evaporator will draw as much liquid as needed, according to the applied heat load, so that the exiting fluid will have vapor quality of one.

An inherent operating feature in an LHP with multiple evaporators is heat sharing among the evaporators. When some instruments are turned off, the attached evaporators will work in the condenser mode and draw heat from other evaporators serving the operating instruments, thus warming the non-operating instruments. This will save supplemental heater power required to maintain the instrument within the allowable range. The amount of heat that is shared among the evaporators is governed by mass, momentum, and energy conservation laws for the entire system, and is affected by the total system heat load, temperatures of the condenser sinks, and the thermal environment surrounding the non-operating instruments.

In the following sections, the theoretical background for the LHP heat load sharing operation will be discussed first. The performance characteristics will be illustrated by using a pressure drop diagram in the loop. This will be followed by a discussion of the experimental results in ground testing of an LHP with two evaporators and two condensers. Factors that affect the amount of heat being shared are also discussed.

Theoretical Background

The following discussion applies to the general case of an LHP consisting of N_E evaporators and N_C condensers. However, for simplicity, the schematic of an LHP having only two evaporators and two condensers is shown in Figure 1. A flow regulator consisting of a capillary wick is usually installed at the exit of each condenser to prevent vapor from flowing out of the condenser [5-7]. When a heat load $Q_{IN}^{(i)}$ is applied to the i[th] evaporator, part of the heat, $Q_L^{(i)}$, is transmitted to the CC (the so-called heat leak), and the remaining heat, $Q_E^{(i)}$, is used to vaporize liquid to generate a mass flow rate of $m_e^{(i)}$. The vapor flow from each evaporator then merges to form a total mass rate of m_t that flows to the condenser section. Thus,

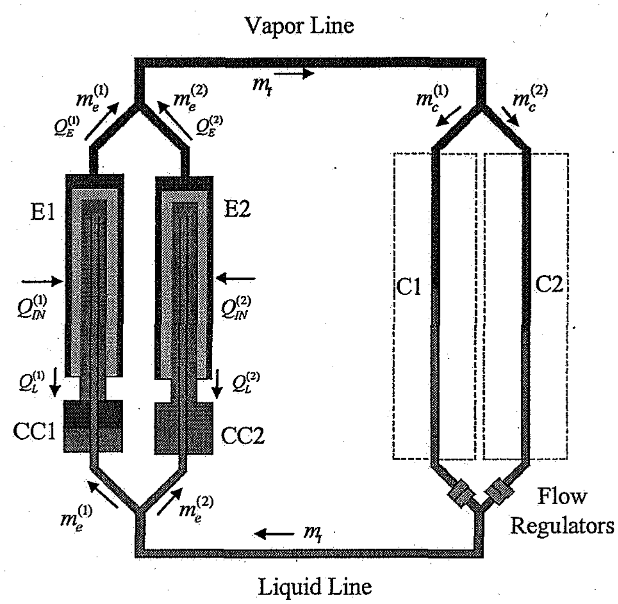
$$Q_{IN}^{(i)} = Q_E^{(i)} + Q_L^{(i)} \qquad i=1 \ to \ N_E \qquad (1)$$

$$Q_E^{(i)} = m_e^{(i)}\lambda \qquad i=1 \ to \ N_E \qquad (2)$$

$$m_t = \sum_{i=1}^{N_e} m_e^{(i)} \qquad (3)$$

where λ is the latent heat of vaporization of the working fluid. Note that the evaporators are passive and self-regulating in that each evaporator, based on the applied heat load $Q_{IN}^{(i)}$, will draw a liquid flow of $m_e^{(i)}$ so that equations (1) and (2) are satisfied and the vapor will exit the evaporator with a quality of unity.

It has also been experimentally verified that, under most circumstances, only one of the compensation chambers (CCs) in the LHP will contain two-phase fluid and control the loop operating temperature; all other CCs will be hard filled with liquid. Thus, the LHP operating temperature depends on the heat load distribution among the evaporators, the condenser sink temperatures and the ambient temperature. When evaporators are sharing heat loads, the operating temperature also varies with the thermal conditions surrounding the evaporators working in the condenser mode. Hence the operating temperature becomes more complex. On the other hand, the LHP operating temperature can always be controlled by actively controlling one or more CCs at the desired set point temperature regardless of the mode of operation. Even when all CCs are controlled at the same set point temperature, under most circumstances still only one of the CCs will contain two-phase fluid and control the loop operating temperature. All other CCs will be liquid filled. Figure illustrates that CC1 contains two-phase fluid and CC2 is liquid-filled.



' Figure 1 Schematic of an LHP with Multiple Evaporators and Condensers

The total mass rate of m_t will be distributed among the N_C condensers. The ith condenser will receive a mass rate of $m_c^{(i)}$ with an associated latent heat of $m_c^{(i)}\lambda$, and the vapor will be completely condensed over a length of $L_{c,2\phi}^{(i)}$. Thus,

$$m_t = \sum_{i=1}^{N_C} m_c^{(i)} \tag{4}$$

$$m_c^{(i)}\lambda = \pi D_c^{(i)} L_{c,2\phi}^{(i)} h_{c,2\phi}(T_{sat} - T_{c,wall}^{(i)}) \qquad i = 1 \text{ to } Nc \tag{5}$$

where $D_c^{(i)}$ is the diameter of the ith condenser and $L_{c,2\phi}^{(i)}$ is the length required to dissipate the latent heat, $h_{c,2\phi}$ is the condensation heat transfer coefficient, T_{sat} is the loop saturation temperature, and $T_{c,wall}^{(i)}$ is the condenser wall temperature. The liquid will then be subcooled over the remaining length of the condenser. If a condenser is completely utilized for vapor condensation, the vapor will be stopped by the flow regulator and the excess vapor will be diverted to other condensers, resulting in a flow re-distribution among all condensers. The liquid flow exiting each condenser then merges into a total liquid flow with a mass rate of m_t. The pressure drop in the ith condenser is the sum of the pressure drops in the two-phase region (over the length $L_{c,2\phi}^{(i)}$) and the liquid phase region, and the hydraulic pressure head due to gravity. Thus,

$$\Delta P_c^{(i)} = \Delta P_{c,2\phi}^{(i)} + \Delta P_{c,1\phi}^{(i)} + \rho_l g \Delta h_L^{(i)} = f(m_c^{(i)},\ L_{c,2\phi}^{(i)}) \qquad i = 1 \ to \ N_C \qquad (6a)$$

$$\Delta P_c^{(k)} = \Delta P_{c,2\phi}^{(k)} + \Delta P_{FR}^{(k)} \qquad \text{Condenser k is fully utilized} \qquad (6b)$$

where $\Delta P_c^{(i)}$ is the pressure drop over the i[th] condenser, ρ_l is the liquid density, $\Delta h_L^{(i)}$ is the end-to-end elevation of the i[th] condenser, $\Delta P_{FR}^{(k)}$ is the capillary pressure exerted by the flow regulator in the k[th] condenser. However, there can be only one pressure drop over the condenser section. Hence,

$$\Delta P_c^{(i)} \equiv \Delta P_c \qquad i = 1 \ to \ N_C \qquad (7)$$

where ΔP_c is the pressure drop across the entire condenser section.

There are a total of 2N +1 unknowns ($m_c^{(i)}$, $L_{c,2\phi}^{(i)}$ and ΔP_c), and 2N +1 equations in (4), (5), and (6). In essence, the conservation laws of mass, momentum and energy yield a set of mass flow rates through the condensers, a set of lengths over which vapor is condensed, and a pressure drop across the condenser section. A flow schematic and associated pressure drop diagram in the condenser section are shown in Figures 2(a) and 2(b). In Figure 2(a), all condensers contain two-phase and subcooled regions, where in Figure 2(b), one of the condensers is fully utilized and a pressure difference is sustained by the wick inside the flow regulator.

The fluid will exit each condenser with a temperature of $T_{c,out}^{(i)}$, which is a function of the flow rate through that condenser, the length of the two-phase region, and the condenser wall temperature. The exiting flows then merge and mix to yield a single temperature $T_{c,out}$. Thus,

$$T_{c,out}^{(i)} = f(m_c^{(i)},\ L_{c,2\phi}^{(i)},\ T_{c,wall}^{(i)}) \qquad i = 1 \ to \ N_C \qquad (8)$$

$$T_{c,out} = \frac{1}{m_t} \sum_{i=1}^{N_c} m_c^{(i)} T_{c,out}^{(i)} \qquad (9)$$

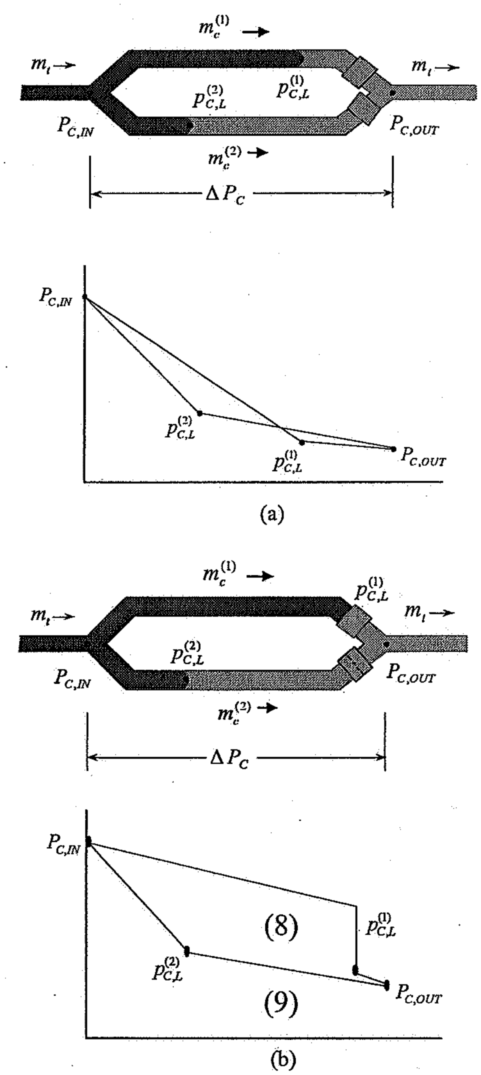


Figure 2 Flow distribution and Pressure Drops in Condenser Section

The liquid exchanges heat with the surroundings as it flows along the liquid return line. When it reaches the evaporator section, its temperature reaches $T_{LL,IN}$, which is a function of the total mass flow rate m_t, diameter and length of the liquid transport line, temperature of the surroundings, and mode of heat transfer. The mass flow then splits among all evaporators, and each individual liquid flow further exchanges heat with its surroundings along the liquid inlet line. At the inlet of the i[th] evaporator, the liquid flow has a temperature of $T^{(i)}_{LL,IN}$. The governing equation can be written as follows:

$$T_{LL,IN} = f\left(T_{c,out}, m_t, L_{LL}, D_{LL}, T_{amb}\right) \qquad (10)$$

$$T^{(i)}_{LL,IN} = f\left(T_{LL,IN}, m_c^{(i)}, L^{(i)}_{LL}, D^{(i)}_{LL}, T^{(i)}_{amb}\right) \qquad i=1 \ to \ N_E \qquad (11)$$

$$Q^{(i)}_{SC} = m_e^{(i)} C_P \left(T_{sat} - T^{(i)}_{LL,IN}\right) \qquad i=1 \ to \ N_E \qquad (12)$$

where C_p is the specific heat of the liquid. The temperature of the i[th] CC is determined by the energy balance between the liquid subcooling, $Q^{(i)}_{sc}$, the heat leak, $Q^{(i)}_{L}$, and the heat exchange between the CC and the surrounding, $Q^{(i)}_{RA}$. If there is no active measure to control the CC temperatures, the CC that reaches the highest temperature will contains two-phase fluid and control the loop operating temperature. All other CCs will be liquid filled. If all CCs are controlled at the same set point temperature, the CC that has the lowest absolute pressure will control the loop operating temperature, and all other CCs will be liquid filled under most circumstances. The energy balance can be described as follows:

$$Q^{(k)}_{L} - Q^{(k)}_{sc} - Q^{(k)}_{RA} = 0 \qquad T^{cc}_{sat} \ determined \ by \ k^{th} \ CC \qquad (13)$$

$$Q^{(i)}_{L} - Q^{(i)}_{sc} - Q^{(i)}_{RA} \leq 0 \qquad i=1 \ \ to \ N_E, \quad i \neq k \qquad (14)$$

Figure 3 shows the heat flows and temperatures on the liquid side of the LHP under the normal operation.

As each fluid flow completes its path, each evaporator is subjected to a total pressure drop which is the sum of pressure drops in the evaporator, vapor transport line, condenser section, liquid transport line, and individual wick. The total pressure drop must not exceed the maximum pressure drop that the capillary wick is able to sustain. Thus,

$$\Delta P^{(i)}_{cap} = \frac{2\sigma \cos\theta}{r_p^{(i)}} \qquad i = 1 \ \ to \ N_E \qquad (15)$$

$$\Delta P^{(i)}_{tot} \leq \Delta P^{(i)}_{cap} \qquad i = 1 \ \ to \ \ N_E \qquad (16)$$

where $r_p^{(i)}$ is the pore radius of the i[th] wick, $\Delta P^{(i)}_{cap}$ is the maximum pressure drop that the i[th] wick can sustain, $\sigma$is the surface tension force, and θ is the contact angle. Equations (1) to (16) describe the operation of a CPL with multiple evaporators and multiple condensers. A pressure drop diagram of the loop under normal operation where each evaporator receives an applied heat load is shown in Figure 3.

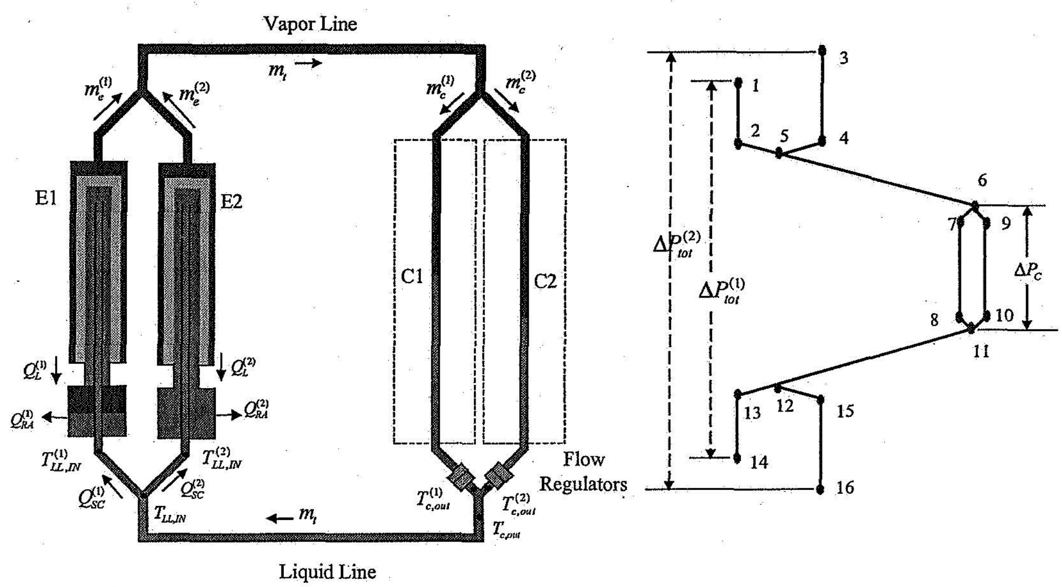


Figure 3 Pressure Drop Diagram of LHP Under Normal Operation

In the heat load sharing mode, the evaporator attached to a non-operating instrument works as a condenser, and its wick works as a flow regulator. The operating principles of heat load sharing can be explained by referring to Figure 4, where E2 receives an applied heat load and E1 is sharing the heat load. The vapor generated in E2 will flow to locations where the pressures and temperatures are low, i.e. the condensers and E1. In essence, a single pressure drop of $\Delta P_{EV} = P_5 - P_{12}$ exists among all "condensers", which share the total heat load according to the conservation laws of mass, momentum, and energy, as previously illustrated in Figure 2. Thus, heat load sharing is an inherent function of a CPL with multiple parallel evaporators. Note that at least one of the instruments must be operational and supplying heat to the system during the heat load sharing operation. Also note that equations (13) and (14) still apply in the heat load sharing mode of operation.

The amount of heat that is shared by a condensing evaporator is a function of the system heat load and the sink temperatures of all "condensers". The evaporator sharing the heat will resume its normal operation automatically when the attached instrument is turned on again. When the condensing evaporator is fully utilized, the liquid exiting the evaporator will be close to the saturation temperature. This may affect the operation of evaporators located downstream of this condensing evaporator due to reduced subcooling of the returning liquid.

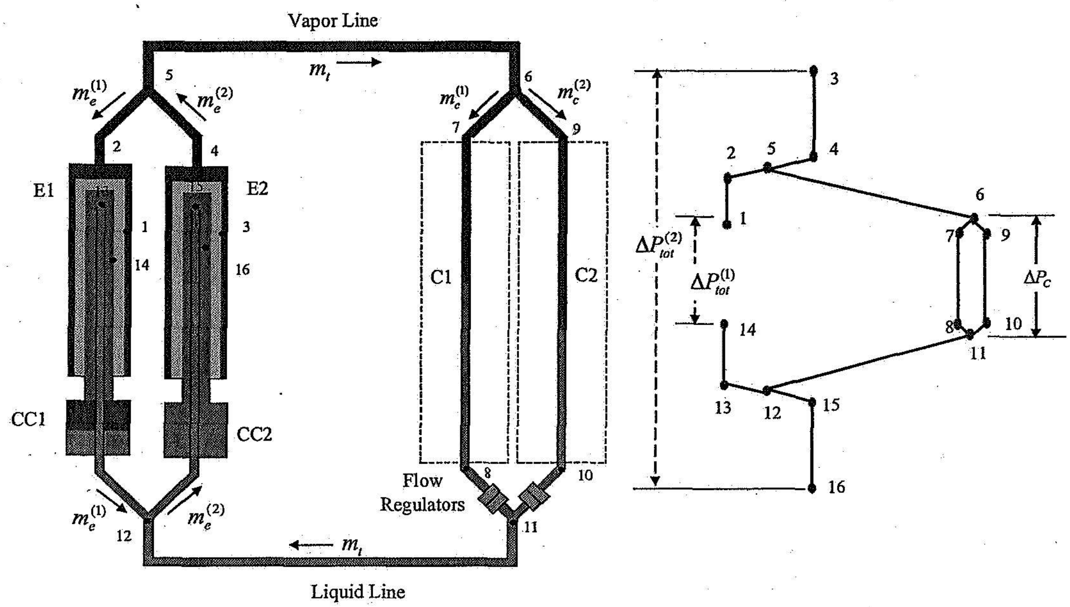


Figure 4 Pressure Drop Diagram in an LHP Under Heat Load Sharing Mode

Figures 4 shows that some heat will always flow to the condenser section regardless of how much heat is applied to the evaporators, i.e. the condensing evaporator cannot share 100 percent of the applied heat load. To increase the heat that can be shared by the condensing evaporator, a back pressure regulator (BPR) consisting of a capillary wick can be installed on the vapor line [5-8]. Figure 5 shows that the condensing evaporator will share 100 percent of the applied heat load when the pressure drop $\Delta P_{EV} = P_5 - P_{12}$ is smaller than the capillary pressure of the wick in the BPR. Vapor will flow to the condensers only if ΔP_{EV} is greater than the capillary pressure across the BPR.

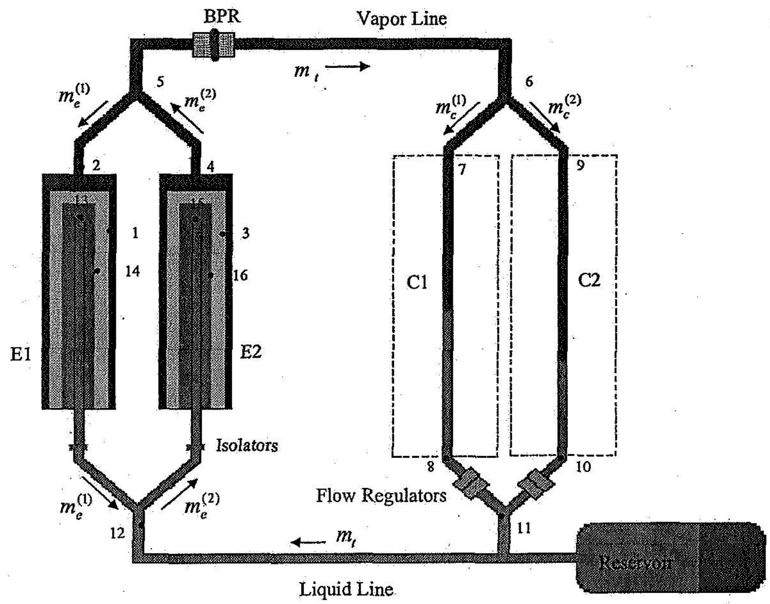


Figure 5 A CPL with a Back Pressure Regulator in the Vapor Line

The reservoir in the CPL can maintain a constant saturation temperature for the loop operation. However, temperatures of the instruments will deviate from the saturation temperature due to the heat transfer requirement. When an instrument is turned on, the instrument will be at a higher temperature than the saturation temperature, i.e. $\Delta T_1 = Q_E/G_{evap}$, where Q_E is the heat dissipation of the instrument and G_{evap} is the overall thermal conductance associated with the evaporation process. Conversely, when the instrument is turned off, the instrument temperature will be lower than the saturation temperature, i.e. $\Delta T_2 = Q_{shared}/G_{cond}$, where Q_{shared} is the heat being shared by the instrument and G_{cond} is the overall thermal conductance associated with vapor condensation process. Thus, a temperature difference of $\Delta T = \Delta T_1 + \Delta T_2$ can be expected between the "on" and "off" states of the instrument as illustrated in Figure 6.

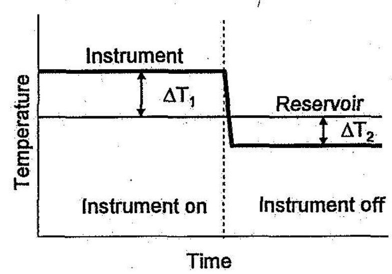


Figure 6 Temperature Drop between Operating and Non-Operating Instruments

Description of the Test Article and Test Set-up

The test article, shown in Figures 6 and 7, consists of two evaporators, two condensers, a common vapor transport line and a common liquid return line. Each evaporator has an integral CC. Both evaporators are made of aluminum tubing with an O.D. of 15mm and a length of 76.2mm. One evaporator has a titanium wick with a pore radius of about 3 μm, while the other has a nickel wick with a pore radius of about 0.5 μm. The use of two different wicks and pore sizes was intentional. Each CC is made of stainless steel tube of

14.8mm O.D. x 81.8mm L. The vapor line and liquid line, each 1168mm long, are made of stainless steel tube with an O.D. of 3.3mm and 2.2mm, respectively. Each condenser is made of stainless steel tube of 2.2mm O.D. x 762mm L. A flow regulator consisting of a capillary wick is installed at the downstream of the condensers. The loop is charged with 15.5 grams of ammonia. The design parameters are summarized in Table 1.

Each condenser was attached to a cold plate, and each cold plate was cooled by a separate chiller. A 500-gram aluminum mass was attached to each evaporator to simulate the instrument mass. Two cartridge heaters were attached to each thermal mass to provide heat loads between 1W and 200W per evaporator. To demonstrate heat load sharing, each thermal mass had two channels to accommodate a coolant flow. Since only one thermal mass needed to be cooled at a time, a third chiller was used and the coolant flow was directed to the designated thermal mass using control valves.

A TEC was installed on each CC through an aluminum saddle. The hot side of the TEC was connected to the evaporator through a copper strap. Each TEC was controlled by a bi-polar power supply. Changing the polarity on the power supply changed the TEC operation between heating and cooling modes.

Ninety type T thermocouples were used to monitor the temperatures. A data acquisition system consisting of a personal computer, a CRT monitor, and two data loggers was used to display and store test data every two seconds.

Table 1. Summary of LHP Design Parameters

Component	Description
Evaporator (2)	Aluminum, 13mm O.D. x 76.2mm L each
Primary wick	Nickel, 0.6 μm pore radius, 60% porosity, 1.4x 10-14 m2 permeability
Primary wick	Titanium wick, 3 μm pore radius, 60% porosity, 1.0x 10-14 m2 permeability
CC (2)	Stainless steel, 18mm O.D. x 61mm L, 18cc each
Vapor line	Stainless steel, 2.38mm O.D. x 1200mm L
Liquid line	Stainless steel, 1.59mm O.D. x 1200mm L
Condenser (2)	Stainless steel, 2.38mm O.D. x 760mm L each
Flow regulator	Polyethylene wick, 40 μm pores
Working fluid	Anhydrous ammonia, 15.5 grams

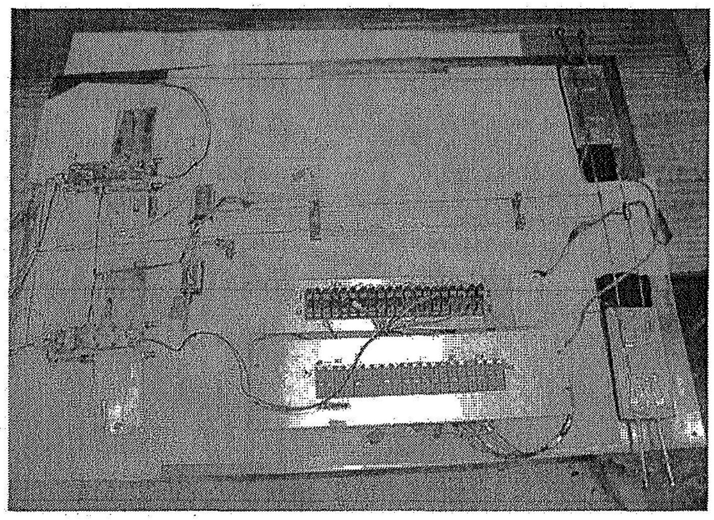


Figure 6 Picture of the Test Article

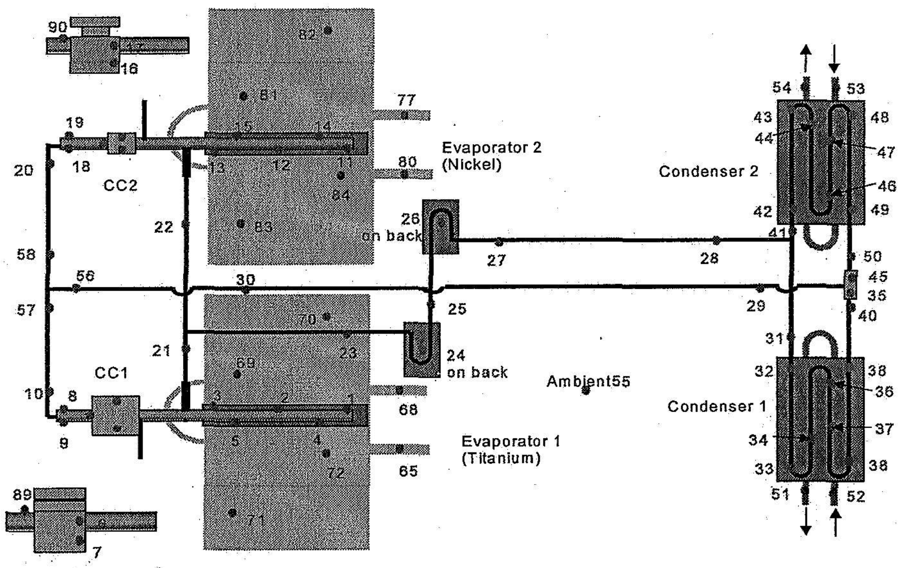


Figure 7 Schematic of the Test Article and Thermocouple Locations

Experimental Results

A comprehensive test program was carried out to investigate the effect of various parameters on the heat load sharing operation. Tests were conducted with and without active control of the CC temperature. The following conditions were varied: a) the heat load applied to the operating evaporator; b) the temperature of the evaporator sink; c) temperatures of the condenser sinks; and d) the CC set point temperature. For a given applied heat load to the operating evaporator, the amount dissipated by the two condensers and the heat sharing evaporator is governed by the conservation laws of mass, momentum, and energy conservation. Changing any one of the above parameters will cause the fluid flow to redistribute among the three "condensers", and the heat shared by the condensing evaporator.

When the CC temperatures are not actively controlled, the operating temperature will change with the applied heat load, the evaporator sink temperature, and the condenser sink temperatures. Under this condition, the loop operating temperature cannot be lower than the ambient temperature. When a tight operating temperature is needed, any one or all of the CCs can be controlled at the set point temperature. As long as the set point temperature is higher than the loop natural equilibrium temperature under the given operating condition, the loop operating temperature will following the CC set point temperature.

<u>Varying Applied Heat Load</u>

Figure 8 illustrates that the loop operating temperature was controlled by setting CC2 at 303K while E2 was sharing heat from E1. E1 power varied between 100W and 20W. The temperature of coolant flowing to E2 sink was kept at 293K and the flow rate was constant at 0.4 ml/s. The temperatures of the two sink were maintained at 283K. Heat shared by E2 decreased with a decreasing E1 power. E2 was fully utilized at 100W and its temperature was close to the CC2 set point temperature. E2 became partially filled with liquid at all other powers, and it temperature became more subcooled with a decreasing E1 power. As E1 power was lowered to 20W, the heat shared by E2 was even less and the E2 temperature was close to its sink temperature.

Figure 9 shows the heat sharing operation when CC1 was controlled at 303K and E1 power varied between 125W and 25W with increments of 25W. The temperature of coolant flowing to E2 sink was kept at 288K and the flow rate was constant at 0.4 ml/s. The C1 and C2 sinks were kept at 273K and 293K, respectively. E2 was fully utilized at 125W and 100W and its temperature was close to the loop saturation temperature. As E1 power decreased to 75W, 50W and 25W, heat shared by E2 continued to decrease and E2 was partially filled with liquid. E2 became increasingly subcooled with a decreasing E1 power.

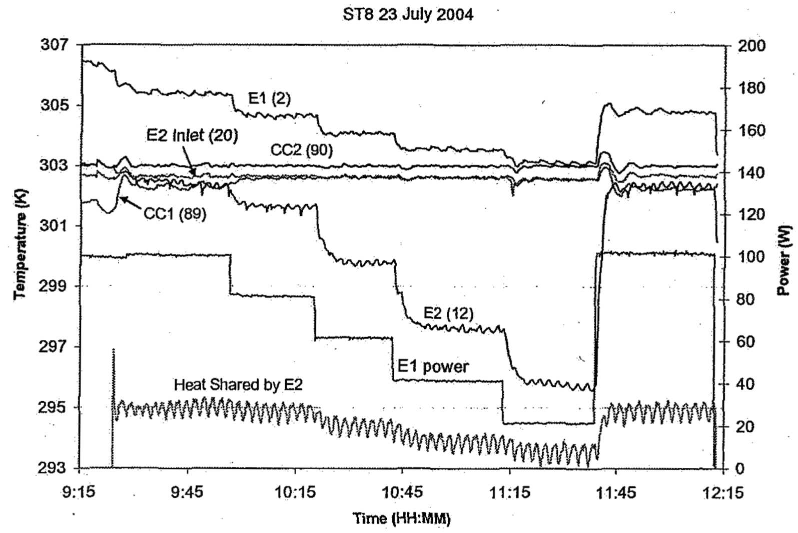


Figure 8 Heat Sharing with Varying Applied Heat Load

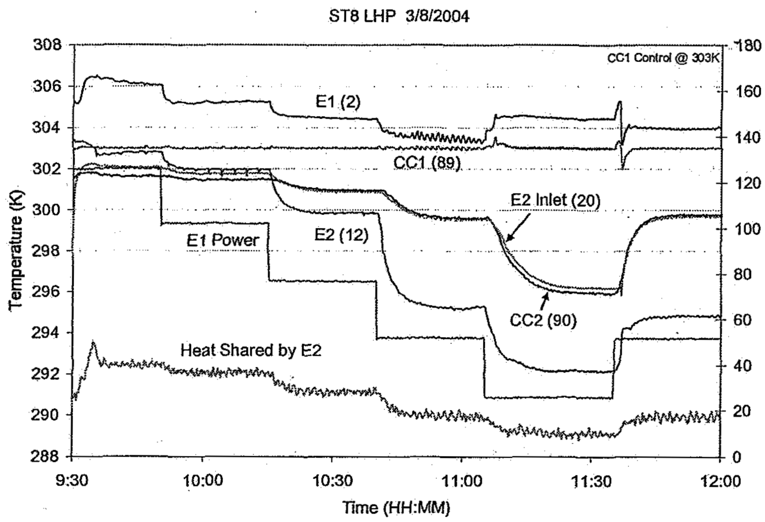


Figure 9 Heat Sharing with Varying Applied Heat Load

Figure 10 shows the heat sharing operation when the CC temperatures were not actively controlled. The heat load applied to E1 changed in steps while E2 shared heat from E1. The temperature and the mass flow rate of the coolant flowing to the E2 sink was kept constant at 293K and 0.4 ml/s, respectively, starting at 10:00. The two condenser chillers were kept at 283K. When the E1 heat load was between 100W and 60W, CC2 controlled the loop operating temperature. E2 was fully utilized for vapor condensation as evidenced by the fact that E2 pump and its inlet temperatures were following the CC2 temperature. E2 dissipated about 10W. As E1 heat load was reduced to 40W, CC1 began to control the loop operating temperature, and E2 was no longer fully utilized. Both the middle of the E2 pump (TC12) and E2 inlet were at subcooled temperatures. When the E1 power was further reduced to 20W, E2 dissipated about 5W and become more subcooled.

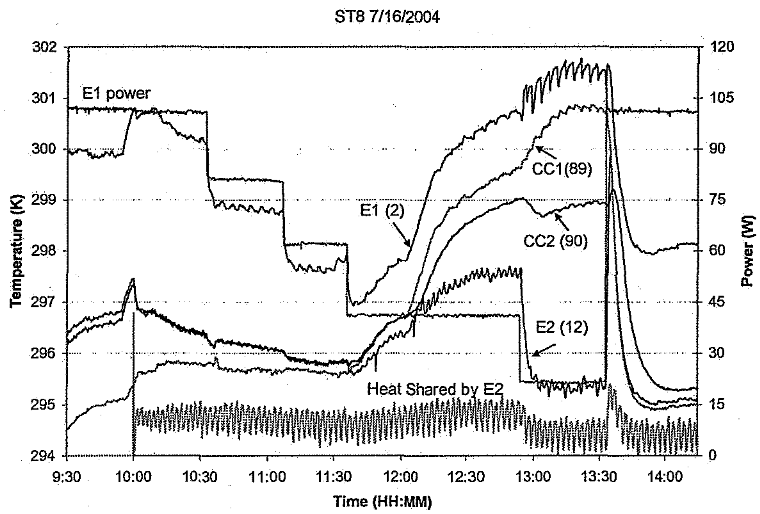


Figure 10 Heat Sharing with Varying Applied Heat Load (without CC temperatures Control)

Varying Evaporator Sink Temperature

Figure 11 shows the test results when a constant heat load of 100W was applied to E2, and CC1 was controlled at 303K. The temperature of the coolant flowing to E1 sink varied as follows: 293K/288K/283K/298K/303K. E1 was fully utilized at sink temperatures above 288K, and its temperature was close to 303K. E2 became partially filled with liquid at 283K sink, and its temperature was below 303K.

Figure 12 shows a test similar to that shown in Figure 11 except the CC temperatures were not actively controlled. A constant power of 100W was applied to E2. The E1 sink temperature varied as follows: 293K/288K/283K/298K/303K. CC1 controlled the loop operating temperature throughout the test. E1 was fully utilized for coolant temperatures between 283K and 298K. The heat shared by E1 increased with a decreasing coolant temperature. When the coolant temperature was raised to 303K, higher than the CC1 saturation temperature of 298K, heat was transmitted from the coolant to E1, and E1 changed its working mode from a condenser to an evaporator. A net heat load to the E1 evaporator was shown as a negative heat dissipation by E1.

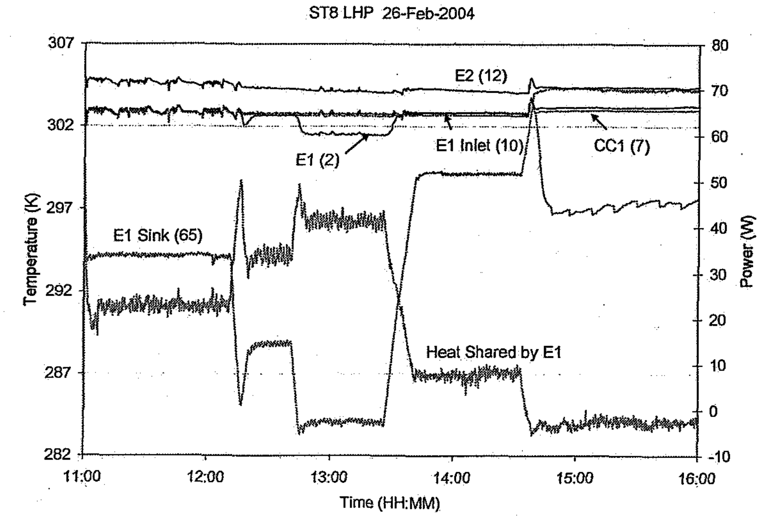


Figure 11 Heat Load sharing with Varying Evaporator Sink Temperatures

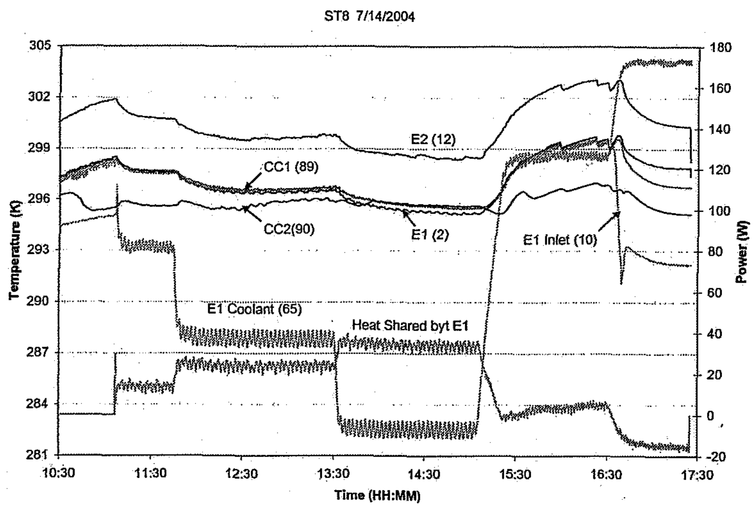


Figure 12 Heat Load Sharing with Varying Evaporator Sink

Varying CC Set Point Temperature

Figure 13 illustrates how the set point temperature may affect the amount of heat being shared. E1 power was constant at 100W. E2 sink temperature and flow rate were kept at 283K and 0.4 ml/s, respectively. CC1 set point varied as follows: 303K/298K/293K/288K/286K/283K/288K/292K/293K. Heat shared by E2 decreased as the CC1 saturation temperature decreased. Near the end of the test, the coolant flow to the E2 sink was shut off, and a heat load of 50W was applied to E2. E2 immediately changed its operation from an evaporator mode to a condenser mode.

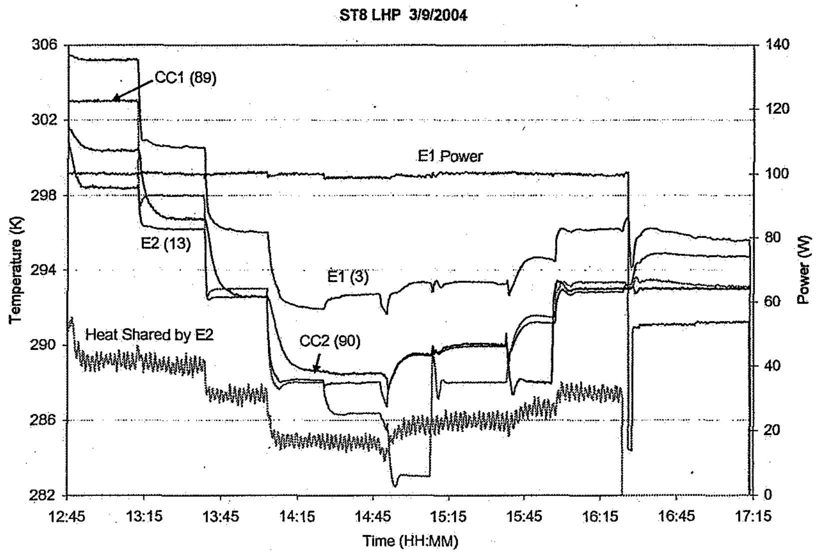


Figure 13 Effect of Spacecraft Maneuver on CPL Heat Load Sharing Operation

Varying Condenser Sink Temperature

Figure 14 shows the effect of C1 and C2 sink temperature on the heat load sharing operation. E2 had a constant heat load of 100W. The temperature and mass rate of coolant flowing to E1 sink were kept at 283K and 0.4 ml/s, respectively. Temperatures of, C1 sink/C2 sink varied as follows: 273K/273K, 293K/273K, 293K/283K, 298K/283K. The amount of heat shared by E1 was little affected by changes of the condenser sink temperature. This is because E1 was much closer to E2 than C1 or C2, and E1 sink was set at low enough temperature.

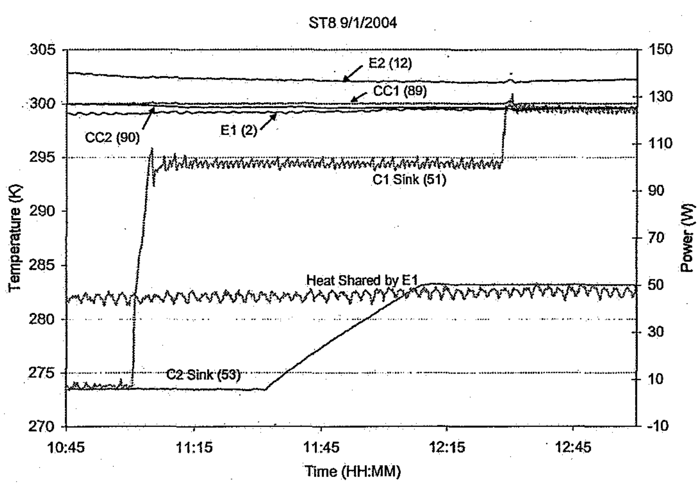


Figure 14 Heat Sharing with Varying Condenser Sink Temperatures

Summary and Concluding Remarks

Heat load sharing among evaporators is an inherent and useful feature provided by an LHP with multiple evaporators. It allows the evaporators attached to non-operating instruments to receive heat from other evaporators serving the operating instrument, thus eliminating or reducing the supplemental heater power that is otherwise required to maintain the non-operating instrument within the allowable temperature range. The amount of heat that can be shared by an evaporator is governed by the conservation laws of mass, momentum and energy in the entire loop, and is a function of the system heat load, temperatures of the condenser sinks, and temperatures of sinks surrounding the evaporators that share the heat load.

The heat load sharing operation was demonstrated in ground tests of an LHP with two evaporators and two condensers. The experimental results verified that the heat shared by the evaporator was mainly a function of the heat load applied to the other evaporator and the temperature of the sink surrounding the evaporator sharing the heat, and was much less affected by the condenser sink temperatures. Moreover, test results verified that whether the evaporator would work in the normal evaporator mode or the condenser mode was purely a function of the thermal environment surrounding that evaporator. The evaporator would automatically switch the mode of operation as its environmental condition changed. Thus, heat load sharing is an inherent and passive function of an LHP with multiple evaporators.

References

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