An Experimental Study of the Operating Temperature in a Loop Heat Pipe with Two Evaporators and Two Condensers

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Abstract

This paper presents a comprehensive experimental study of the loop operating temperature in a loop heat pipe (LHP) which has two parallel evaporators and two parallel condensers. In a single evaporator LHP, it is well known that the loop operating temperature is a function of the heat load, the sink temperature and the ambient temperature. The objective of the present study emphasizes on the stability of the loop operating temperature and parameters that affects the loop operation. Tests results show that the loop operating temperature is a function of the total system heat load, sink temperature, ambient temperature, and heat load distribution between the two evaporators. Under most conditions, only one compensation chamber (CC) contains two-phase fluid and controls the loop operating temperature, and the other CC is completely filled with liquid. Moreover, as the test condition changes, control of the loop operating temperature often shifted from one CC to another. In spite of complex interactions between various components, the test loop has demonstrated very robust operation even during fast transients.

Introduction

Most existing loop heat pipes (LHPs) have a single evaporator and a single condenser. The evaporator is made with an integral compensation chamber (CC) whose saturation temperature determines the loop operating temperature. Because the compensation chamber is physically near the evaporator and is located in the path of the fluid circulation, its temperature is affected by the evaporator heat load, condenser sink temperature, and ambient temperature. In order for the loop to work properly, the overall pressure drop in the loop must not exceed the capillary pumping capability of the wick. In addition, the temperature difference between the evaporator and the CC must match the corresponding pressure drop across the primary wick as required by thermodynamics.

The LHP operation becomes more complex when multiple evaporators and multiple condensers are present. Each CC is still subjected to heat leak from the evaporator and liquid subcooling from the condenser. As will be analyzed in the following section, in most cases only one CC can contain two-phase fluid and controls the loop operating temperature. All other CC's will be filled with liquid. This brings challenges on sizing of CC's and imposes a constraint on the number of evaporators that can be integrated into a single loop. Moreover, control of the loop operating temperature can switch from one CC to another as the operating condition changes. Thus, the ability of the loop to adapt to fast transients such as rapid changes in the heat load and/or the sink temperature must be experimentally verified.

The feasibility of an LHP with multiple evaporators has been demonstrated [1-4]. An extensive test program has also been conducted to characterize the operating characteristics of an LHP with two evaporators and two condensers [5,6]. This paper will describe in detail the loop operating temperature under various test conditions. Tests performed included an array of even and uneven heat loads to the evaporators, high and low heat loads, even and uneven sink temperatures, a rapid change in the heat load distribution, and a rapid change in sink temperatures. Detailed descriptions of the phenomena observed will be given, and explanations on the physical processes involved will be offered. The implications on the operation of an LHP with more than two evaporators will also be addressed.

Test Article and Test Set-up

As shown schematically in Figure 1, the test loop, built by the Dynatherm Corporation, consists of two parallel evaporators, two parallel condensers, a common vapor transport line and a common liquid return line. Each evaporator has its own integral CC. Both evaporators are made of aluminum tubing with 15.8 mm (0.63 inch) O.D. by 76.2 mm (3 inches) length. One evaporator has a titanium wick with a pore radius of about 3 microns, while the other has a nickel wick with a pore radius of about 0.5 micron. Each CC is made of stainless steel tubing and has an O.D. of 14.8 mm (0.57 inch) and a length of 81.8 mm (3.22 inches). Both the vapor line and liquid line are made of 2.2mm O.D. (3/32 inch) stainless steel tubing, and have a length of 1168mm (46 inches). The vapor and liquid lines branch out to feed into the two evaporators and two condensers. Each condenser is made of 2.2mm O.D.(3/32 inch) stainless steel tubing and is 762mm (30 inches) long. A flow regulator made of capillary wicks is installed at the downstream of each condenser. The flow regulators prevent vapor from penetrating the wick before both condensers are fully utilized, and hence serve to balance the flows between the two condensers. Two 50.8 mm by 50.8 mm (2 inches by 2 inches) aluminum plates are installed on the vapor line. One is attached with coolant lines while the other is attached with an electrical heater. The two aluminum plates are used in the test to illustrate that in a capillary system a heat load can be added to the vapor line after some energy has been dissipated to a nearby radiator. The loop is charged with 15.5 grams of anhydrous ammonia.

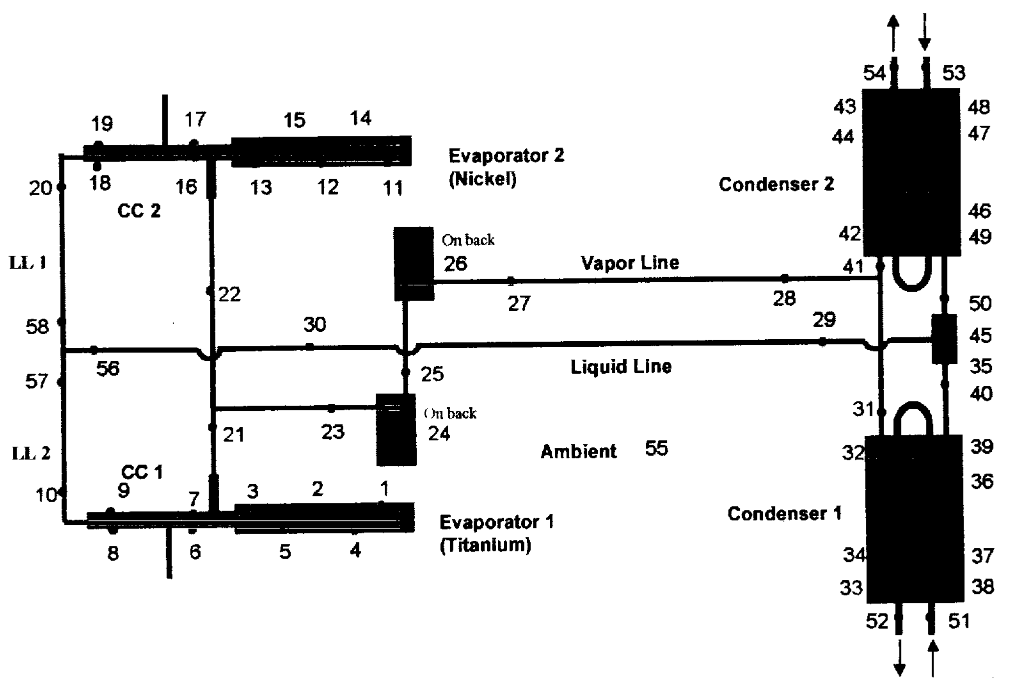

Figure 1. Schematic of an LHP with Two Evaporators and Two Condensers

Electrical heaters are attached to each evaporator and each compensation chamber, and are separately controlled. The two condensers are attached to two cold plates; each cooled by a separate chiller. Sixty thermocouples are used to monitor the loop temperatures. Notice that many thermocouples are installed on the liquid line between the two compensation chambers in order to monitor the interactions between the two elements during fast transients. A data acquisition system consisting of a datalogger, a personal computer, a CRT monitor, and Labview software is used to monitor and store data. The data is updated on the monitor and stored in the computer every second.

For ease of description, the following abbreviations will be used in subsequent discussions (see Figure 1): E1= evaporator 1, E2 = evaporator 2, CC1 = compensation chamber 1, CC2 = compensation chamber 2, C1= condenser 1, C2 = condenser 2, LL1=liquid line from TC56 to CC1, LL2 = liquid line from TC56 to CC2. In addition, the number in parenthesis next to the label for each curve in the following figures refers to the thermocouple number shown in Figure 1.

Theoretical Background

In an LHP with a single evaporator and a single condenser, the CC saturation temperature, which governs the loop operating temperature, is determined by an energy balance between the heat leak from the evaporator to the CC and the amount of subcooling of the returning fluid. Any factor that affects the energy balance will affect the CC. Conversely, any factor that does not affect the energy balance will be isolated from and become "invisible" to the CC. The liquid subcooling is a function of the heat load, sink temperature and ambient temperature. The heat leak is a function of the heat load and the vapor void fraction inside the evaporator core. The latter is especially important at low heat loads. For an LHP with a fixed heat load and a fixed sink temperature, the evaporator core may have different vapor void fractions at different times. A higher void fraction will lead to a higher heat leak, which in turn result in a higher CC temperature. Thus, an LHP can operate at different temperatures under seeming identical conditions (i.e. the same heat load and sink temperature). This is the essence of the temperature hysteresis [7-11].

In a two-phase system, the saturation temperature is directly related to the saturation pressure. Because both the CC and the evaporator outer grooves contain two-phase fluid, a relationship exists between the temperature difference and the pressure difference of theses two elements, as can be expressed by the Clausius-Clapeyron equation:

$$P_e - P_{cc} = \lambda\,(T_e - T_{cc}) / (T_{cc}\,\Delta v) \qquad (1)$$

where Δv is the difference in the vapor and liquid specific volumes, λ is the latent heat of vaporization of the working fluid, P_e and P_{cc} are the absolute pressures at the outer grooves of the evaporator wick and the CC, respectively, and T_e and T_{cc} are the saturation temperatures at the evaporator and the CC, respectively.

In an LHP with multiple evaporators and multiple condensers, the operating temperature becomes much more complex. A detailed discussion on this subject can be found in Reference 7. For the LHP shown in Figure 1, each CC is subjected to the same energy balance requirement based on the heat leak and liquid subcooling. The heat leak for each CC is a function the heat load to its evaporator and the total heat load. The liquid temperature at the liquid line right before the fluid branches out to individual CC's (TC56 in Figure 1) is a function of the total heat load, condenser temperature, and ambient temperature. The temperature of the liquid entering each CC, however, is also a function of the heat load to its evaporator. Because so many variables can affect the CC temperature, most likely the two CC's will reach different equilibrium temperatures. Since there can be only one operating temperature in the loop, the CC that has the higher equilibrium temperature will govern the loop operating temperature. This operating temperature is then imposed upon the other CC, resulting in more liquid subcooling than required to compensate for the heat leak. Consequently, that CC will be completely filled with liquid.

The thermodynamic constraint expressed in Equation (1) must be satisfied between *any* two components that contain two-phase fluid. If both CC's shown in Figure 1 have two-phase fluid, the following conditions must be satisfied:

$$P_{1,e} - P_{1,cc} = \lambda\,(T_{1,e} - T_{1,cc}) / (T_{1,cc}\,\Delta v) \qquad (2)$$

$$P_{2,e} - P_{2,cc} = \lambda\,(T_{2,e} - T_{2,cc}) / (T_{2,cc}\,\Delta v) \qquad (3)$$

$$P_{2,cc} - P_{1,cc} = \lambda\,(T_{2,cc} - T_{1,cc}) / (T_{1,cc}\,\Delta v) \qquad (4)$$

$$P_{2,e} - P_{1,e} = \lambda\,(T_{2,e} - T_{1,e}) / (T_{1,e}\,\Delta v) \qquad (5)$$

where the first subscript refers to the individual CC and its evaporator, and the second subscript refers to the component. Equation (5) will automatically be satisfied because the pressure difference between the two evaporators can be "absorbed" by the capillary wicks (by changing radius of the curvature of the meniscus). Whichever evaporator has the higher temperature will impose that temperature onto the other evaporator. Thus, the two evaporators will be at nearly the same temperature. On the other hand, Equations (2) to (4) can not be satisfied simultaneously under most conditions because the thermal and hydrodynamic conditions surrounding the two CC's may not yield the exact temperatures and pressures to satisfy the thermodynamic requirements. Most likely, one of the CC's will be completely filled with liquid due to too much liquid subcooling. Whichever CC is liquid-filled does not need to satisfy the Clausius-Clapeyron equation. As the operating condition changes, the thermal and hydrodynamic conditions surrounding each CC also change, creating a whole new set of thermodynamic states in the CC's. Fluid movement will occur if the prevailing condition so demands.

Test Results

Various types of tests were conducted to verify the robustness of the LHP. Tests performed include start-up, power cycle, sink temperature cycle, high power, low power, and active control of the compensation chamber temperatures. Each type of test was performed with various heat loads to the two evaporators and various sink temperatures for the two condensers. A summary of the overall test results was presented in Reference 5. The LHP demonstrated its ability to adapt to various test conditions, including some fast transients. This paper will focus on how the loop operating temperature changed in response to changes in the heat load distribution and the sink temperatures. The interaction between LHP components during transient will also be given.

In general, test results confirmed that the loop operating temperature is a function of the total heat load, the heat load distribution between the two evaporators, the sink temperatures of the two condensers, and the ambient temperature. Only one CC contained two-phase fluid in most cases and that CC controlled the loop operating temperature; the other CC was flooded with liquid. Control of the loop operating temperature could switch from one CC to the other as the test condition changed, especially when the heat load distribution changed. Moreover, the effect of the heat load distribution on the shift of temperature control was strongly dependent upon the condenser sink temperatures. As expected, whichever CC was at a higher temperature controlled the loop operating temperature. In addition, control of the loop operating temperature was a strong function of the initial state, leading to many temperature hystereses. In particular, the CC that was liquid-filled had a tendency to remain liquid-filled unless the test condition changed significantly enough to cause vapor generation in that CC.

Despite all the complex interactions between the two CC's, there are still some general rules that can be followed although exceptions do exist. The most important factor in determining the loop operating temperature is the combination of the sink temperature and the heat load distribution between the two evaporators. When the condenser sink temperatures are much lower than the ambient temperature (more precisely, when the liquid exiting the condensers is at a temperature lower than ambient), the liquid returning from the condensers will acquire parasitic heat gains along the liquid return line. Thus, the CC whose evaporator receives a higher heat load will be flooded with liquid due to high subcooling, and the other CC whose evaporator receives a lower heat load will control the loop operating temperature. When the condenser sink temperatures are higher than the ambient temperature, the liquid returning from the condensers would dissipate heat along the liquid return line. Thus, the CC with a higher heat load to its evaporator will control the loop operating temperature due to a higher heat leak and a warmer returning liquid. When both evaporators had the same heat load, either CC could control the loop operating temperature, depending on the initial condition which may favor one CC over the other.

There are four thermocouples installed on each of the compensation chambers as shown in Figure 1. In all tests, it was seen that the CC that controlled the loop operating temperature always showed a uniform temperature among all four thermocouples, while the CC that was flooded with liquid always showed divergent temperatures. One could determine which CC was in control by simply looking at the temperature distribution in both CC's. In the following figures, only one thermocouple from each CC will be depicted. The one that had a higher temperature always controlled the loop operating temperature.

Power Cycle at Low Sink Temperature

Figure 2 shows the loop operating temperature in a power cycle test where both condenser sinks were set at 273K and the heat load to E1/E2 varied as follows: 100W/0W, 75W/25W, 50W/50W, 25W/75W, and 0W/100W. Because the sink temperatures were lower than the ambient temperature (293K), one would expect that the loop operating temperature to be governed by CC2 at 100W/0W and 75W/25W, and by CC1 at 25W/75W and 0W/100W. Test results showed that CC2 was in control at 100W/0W, but control switched to CC1 in the remainder of the test. The shift of control from CC1 to CC2 at 25W/75W represents an exception to the rules described above. Note that E1 and E2 inlet temperatures decreased with an increasing heat load and increased with a decreasing heat load to each individual evaporator. On the other hand, the temperature difference between evaporator and the loop saturation temperature increased with an increasing heat load to each evaporator due to the heat transfer requirement.

Figure 2.

- 9/21/00, (8:30 to 13:00, sink 273K/273K), CC2 in control for 100W/0W, CC1 in control 75W/25W, 50W/50W, 25W/75W, 0W/100W.

When one of the evaporators has no heat load, that evaporator works as a condenser, and its CC will control the loop operating temperature because the CC receives heat from the vapor line but has no subcooled liquid coming in. In Figure 2, CC2 worked as a condenser at the heat load of 100W/0W, and controlled the loop operating temperature. In the same manner, CC1 worked as a condenser at 0W/100W and controlled the loop operating temperature. Figure 3 presents another example when only one of the evaporators received a heat load. In this test, both sinks were set at 273K and the heat load varied as follows: 25W/0W, 50W/0W, 0W/25W, and 0W/50W. As expected, CC2 controlled the loop operating temperature at 25W/0W and 50W/0W. Control switched to CC1 as heat load changed to 0W/25W and 0W/50W. Many more power cycle tests at low sink temperatures were tested. All test results verify that the CC whose evaporator had no heat load controlled the loop operating temperature. The only exceptions are when one evaporator received no heat load while the other evaporator received a very low heat load (5W). This will be elaborated on next.

Figure 3. 9/28/00, 8:30 to 16:30, 273K/273K, 25/0, 50/0, 0/25, 0/50.

Figure 4 shows the loop operating temperatures with a power profile of 0W/5W, 0W/100W, and 0W/5W. The chiller 1 was set at 273K while chiller 2 was idle (no coolant flow to C2). The loop operating temperature was controlled by CC2 at 0W/5W, and by CC1 at 0W/100W. At 0W/100W, CC2 had enough subcooled liquid to collapse vapor bubbles, while CC1 had no subcooled liquid yet still received heat from E2 via the vapor line (heat load sharing). This is why the CC without an external heat load to its evaporator would control the loop operating temperature when the other evaporator received moderate or high power (25W and above), as also illustrated in Figure 3. At 0W/5W, the mass flow rate was extremely low and the returning liquid was heated to the ambient temperature through parasitic heat gains. Thus, the heat leak required that CC2 operate at a temperature much higher than ambient. On the other hand, CC1 received very little heat from the vapor line because there was very little heat to be shared. The net result is that CC2 was at a higher equilibrium temperature and thus controlled the loop operating temperature. Notice that there were strong interactions between the two CC's at 0W/5W. It appeared that both CC1 and CC2 (or at least both E1 and E2 cores) had vapor bubbles. With a low heat load and low pressure difference between the two CC's, fluid and loop temperatures could oscillate. The fact that TC7 and TC17 oscillated simultaneously is a strong indication that both elements contained vapor. A periodic spike of the E1 inlet temperature (TC10) indicates a possible vapor expansion and shrinkage inside E1 core and CC1. As CC1 and CC2 temperature decreased, a larger "condenser" area is needed inside E1 and CC1 in order to dissipate the heat. This led to a sudden vapor expansion. Since the amount of heat to be dissipated was small, the vapor condensed quickly and the bubble shrank again. The cycled repeated itself. A similar test was conducted under the same sink condition with a power profile of 5W/0W, 100W/0W, and 5W/0W, which was a mirror image of that shown in Figure 4. In this case, the loop operating temperature was controlled by CC1 at 5W/0W, and by CC2 at 100W/0W. Similar temperature oscillations were also observed. The low power itself did not cause the temperature oscillation; having two-phase fluid in both CC's appeared to be the cause. As will be described in a later section, the loop could operate very stably at 5W/0W and 5W/5W at different sink temperatures, presumably because only one of the CC's contained two-phase fluid. Low power operation was tested very extensively under this test program, and details will be presented in a future paper.

Figure 4. 9/19/00, 9:15 to 17:15. 273K/Idle, 0W/5W, 0W/100W, 0W/5W.

Because the energy balance surrounding the CC involves the heat leak and liquid subcooling, adding a small heat load to an initially unheated evaporator could have a significant effect on the loop operating temperature. Figure 5 shows the loop temperature for the test where the sink temperatures were set at 273K/273K, and the heat load varied as follows: 100W/0W, 100W/5W, 5W/100W, 0W/100W. At 100W/0W, CC2/E2 worked as a condenser, and CC2 controlled the loop operating temperature. As 5W was added the E2, E2 began to pump fluid and E2 inlet temperature dropped from 298K to 295K. At such a low heat load, the CC2 temperature had to increase by more than 5K in order for the returning subcooled liquid to compensate for the heat leak. CC2 still controlled the loop operating temperature. When the heat load changed to 5W/100W, control switched to CC1 and CC2 was filled with liquid due to high subcooling of the returning liquid. When 5W was removed from E1, CC1/E1 worked as a condenser and the CC1 temperature decreased by more than 5K.

Figure 5

- 9/21/00, 13:30 to 18:00, 100W/0W, 100W/5W, 5W/100W, 0W/100W, sink at 273K/273K

Power Cycle at High Sink Temperature

Figure 6 shows the loop temperatures with the same power profile as shown in Figure 2 but with both sink temperatures set at 293K. It is seen that the loop operating temperature was controlled by CC1 for heat loads of 100W/0W and 75W/25W, and by CC2 for heat loads of 50W/50W, 25W/75W, and 0W/100W. All these were expected because the liquid leaving the condensers was at a temperature higher than ambient, and a higher heat load means a higher heat leak from the evaporator to the CC and a higher temperature of the returning liquid. This is true regardless of whether one or two evaporators received the heat load, or whether the heat load was small. This operating characteristic is further illustrated in Figure 7 where both sinks were set at 300K and the heat load varied as follows: 5W/5W, 50W/5W, 5W/50W, 5W/5W, 5W/0W, 0W/5W. At such a high sink temperature, the condenser extended all the way to TC56 (and TC10 and TC20). The temperature oscillation was caused by the vapor front moving back and froth around the "condenser" exit. This type of temperature oscillation was discussed in detail in Reference 12.

Figure 6.

- 10/20/00, 00 (7:30 to 12:30, sink 293K/293K), CC1 in control for 100W/0W, 75W/25W, CC2 in control for 50W/50W, 25W/75W, 0W/100W.

Figure 7.

- 11/17/00, 10:00 to 17:00, Sink at 300K/300K, power 5/5, 50/5, 5/50, 5/5/, 5/0, 0/5.
    - Consider use TC7, 17, 39, 49, 56, 30, see binder.

Figure 8 shows the loop temperature for a test with the same power profile as in Figure 5 except the sink temperatures were set at 293K. Unlike the case of low sink temperatures, increasing the heat load to the other evaporator from zero to a small amount had very little effect on the loop operating temperature since the CC whose evaporator had a heat load already controlled the loop operating temperature.

Figure 8.

- 10/20/00, 12:00 to 17:00, 100W/0W, 100W/5W, 5W/100W, 0W/100W, sink at 293K/293K. Plots for loop temperatures and CC1/CC2 temperature available.

Sink Temperature Cycle

The sink temperature cycle test was conducted by changing the set point of one or both chillers while keeping the heat load constant. As the sink temperature cycled up and down, the loop operating temperature also changed. Whether control would switch from one CC to the other was dependent upon whether the change of the sink condition could effect a change in the fluid status in the CC's. When both evaporators had the same heat load, changing the sink temperature usually did not cause a switch of control. When the evaporators had uneven heat loads, changing the sink temperature could effect a switch of control, especially when the sink was cycled above and below the ambient temperature.

Figure 9 shows the loop temperatures when the C1/C2 sinks were cycled between idle/293K and 273K/273K with a constant E1/E2 heat load of 100W/5W. As expected, the loop operating temperature was governed by CC1 at idle/293K and by CC2 at 273K/273K. A similar test was conducted with the same sink temperature cycle and a constant heat load of 5W/100W. Again, control of the loop operating temperature switched as the sink temperature changed.

Figure 9.

- 11/6/00, 12:30 to 17:20, 100W/5W, sink cycle: idle/293K, 273K/273K, idle/293K, 273K/273K.

Figure 10 show the results of another sink temperature cycle test where the E1/E2 heat load was kept constant at 25W/75W and the two chillers cycled independently between 293K and 253K. Since the C1/C2 temperature cycle included 253K/263K and 293K/293K, one would expect that control of the loop operating temperature to switch at the sink temperature extremes. However, Figure 11 shows that CC1 was in control throughout the test. Prior to this sink temperature cycle test, the loop operating temperature was controlled by CC1 in another test. This test showed that the CC that was flooded with liquid had a tendency to remain flooded. This will be discussed next.

Figure 10. 11/20/00, 18:00 to 23:30, 25W/75W, sink cycle 253K/253K to 293K/293K.

Tendency for Flooded CC to Remain Flooded

The general rules presented earlier encompassed most of the test results. However, there were exceptions. Most notably, the CC that was liquid-filled had the tendency to remain liquid-filled. This is not too surprising considering that a liquid-filled CC need some superheat to generate vapor bubbles. Thus, the outside force, be it a change in the heat load distribution or a change in the sink temperature, must be sufficient to produce such an effect in order for the loop temperature control to switch from one CC to the other. Table 1 lists some of the tests which demonstrated such a characteristic. In general, changing the sink temperature was less effective in producing such a switch, as shown in Figure 10. As long as both evaporators had the same heat load, control never switched regardless of the change in the total heat load or the sink temperature. This is because the current test set-up is more or less symmetrical for the two evaporators and CC's, and the liquid returning to each CC was at the same temperature. Thus, the first five tests in Table 2 are in line with expectation. However, in the next test (11/2/00), CC1 continued to control as the heat load changed from 5W/5W to 100W/5W at 273K/273K, as shown in Figure 11. Notice that during the transient from 5W/5W to 100W/5W, CC1 temperature dropped sharply and was below the CC2 temperature. CC1 still controlled the loop operating temperature because the vapor line temperature TC23 still followed TC7. Apparently, CC2 never had enough superheat to generate vapor bubbles and thus remained flooded. As TC7 approached a steady state, its temperature increased above TC17. The last test was also an exception in that CC2 continued to control at 25W/5W and 25W/0W when the sinks were set 293K/293K.

- Figure 11.
    - 11/2/01, 10:30 to 17:30.

Table 1. Operating Temperature Tests

E1/E2 Power (W)	Chiller 1/Chiller 2	Date	Results
50/50	Varied between 253K and 293K	11/8/00	CC1 controlled throughout CC2 remained flooded throughout
50/50	Varied between 253K and 293K	11/20/00	CC2 controlled throughout CC1 remained flooded throughout
5/0, 5/5, 50/50, 5/5	293K/293K	10/30/00	CC1 controlled the loop throughout CC2 remained flooded throughout
0/5, 5/5, 50/50, 5/5, 5/100	293K/293K	10/31/00	CC2 controlled the loop throughout CC1 remained flooded throughout
5/0, 5/5, 50/50, 5/5, 100/5	273K/273K	11/2/00	CC1 controlled the loop throughout CC2 remained flooded throughout
0/5, 5/5, 50/50, 5/5, 5/100	273K/273K	11/1/00	CC2 controlled the loop until 5/100, then control switched to CC1
25W/75W	Varied between 253K and 293K	11/20/00	CC1 controlled the loop throughout CC2 remained flooded throughout
0/25, 5/25, 25/5, 25/0, 25/5, 5/25, 0/25	293K/293K	10/26/00	CC2 controlled the loop throughout CC1 remained flooded throughout

Capillary Limit

The titanium wick used in E1 is three times weaker than the nickel wick in E2. Thus, E1 will reach capillary limit first regardless of the heat load distribution. When the capillary limit is reached, vapor will penetrate the titanium wick and the CC1 temperature will increase. Because of the continuous vapor penetration, the CC1 temperature will rise and begin to control the loop operating temperature regardless of which CC is in control prior to the capillary limit. Figure 12 shows the loop temperature in a capillary limit test where the heat load was applied to E1 only. Since E2 received no heat load, E2/CC2 worked as a condenser and CC2 controlled the loop operating temperature for heat loads between 50W/0W and 120W/0W. The capillary limit was reached at 130W/0W as indicated by a rapid increase in the CC1 temperature exceeding the CC2 temperature. Vapor penetration also pushed cold liquid from TC10 to TC20, causing the TC20 temperature to drop temporarily. The temperature difference between E1 and CC1 also increased after the capillary limit was exceeded due to an increased thermal resistance. Nevertheless, the loop continued to function at a higher temperature because the secondary wick still drew liquid from the CC. The loop also approached another steady temperature as the heat load increased to 140W/0W. The loop completely recovered as the heat load was reduced to 100W/0W, and the heat load could be increased to 115W/0W without exceeding the capillary limit.

Figure 12
9/12/00, sink at 263K/258K, high power: 50/0, 75/0, 100/0, 120/0, 130/0, 140/0, 150/0, 100/0, 110/0, 115/0.

Summary and Concluding Remarks

An extensive test program has been implemented to investigate the operating temperature of an LHP with two evaporators and two condensers. Test results confirmed that, under most conditions, only one CC contained two-phase fluid and controlled the loop operating temperature, the other CC was completely liquid filled. The loop operating temperature was affected by the total heat load, heat load distribution between the two evaporators, sink temperatures of the two condensers, and ambient temperature. As the operating condition changed, control of the loop operating temperature could switch from one CC to the other. In addition, the effect of the heat load distribution on the shift of temperature control was strongly dependent upon the condenser sink temperature. The loop operating temperature was controlled by the CC whose evaporator received the smaller heat load if the sink temperature was lower than ambient, and by the CC whose evaporator received a higher heat load if the sink temperature was higher than ambient.

Control of the loop operating temperature was also a strong function of the initial states, leading to many temperature hystereses. In particular, the CC that was liquid-filled had a tendency to remain liquid-filled unless the test condition changed significantly enough to cause vapor generation in that CC. It is expected that, in a multiple evaporator LHP, all CC's except one will be liquid filled. Therefore, there may be an upper limit on the number of evaporators that can be integrated into a single loop. A detailed explanation was given in Reference 7.

Heat load sharing between the evaporators was also demonstrated. However, since the evaporator/ CC that worked as a condenser had to dissipate heat to the ambient while at the same time controlling the loop operating temperature, the loop could not operate at a temperature lower than the ambient temperature unless both CC's had two-phase fluid simultaneously. Heat load sharing at a low total system power was one of the cases that both CC's seemed to have two-phase fluid. Unfortunately, this condition also led to temperature and fluid oscillations.

One of the reasons to have the two evaporators made of different wicks was to investigate the effect of the wick thermal conductivity on the heat leak and the loop operating temperature. Because the nickel wick has a much higher thermal conductivity, one may expect that E2, which uses the nickel wick, will be at a higher temperature than E1 when an even heat load is applied to both evaporators. Test results show that the void fraction inside the evaporator core has a much higher impact on the loop operating temperature than the wick thermal conductivity. The initial condition of the evaporator cores directly influenced the operation that followed. The implication is that the void fraction must be predicted accurately in order for an analytical model to yield accurate loop temperature predictions.

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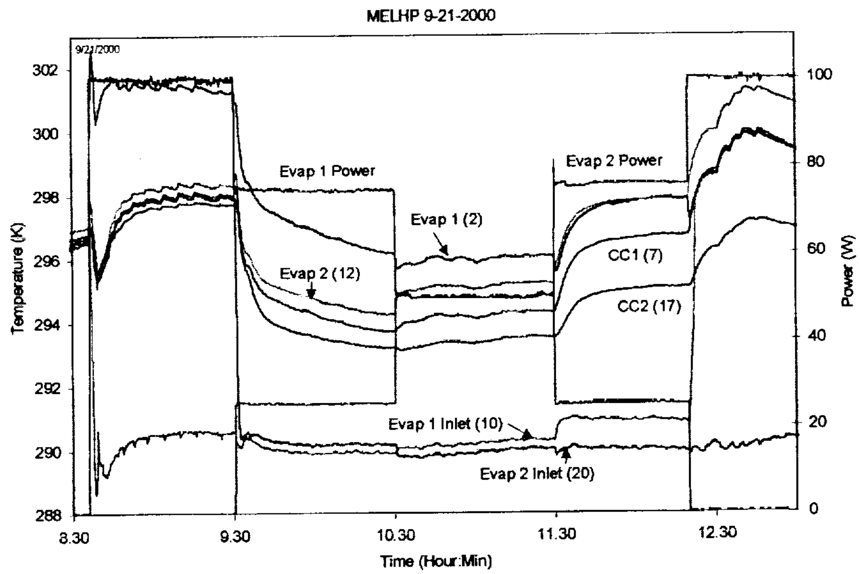


Figure 2. Loop Temperatures in Power Cycle Test at 273K/273K

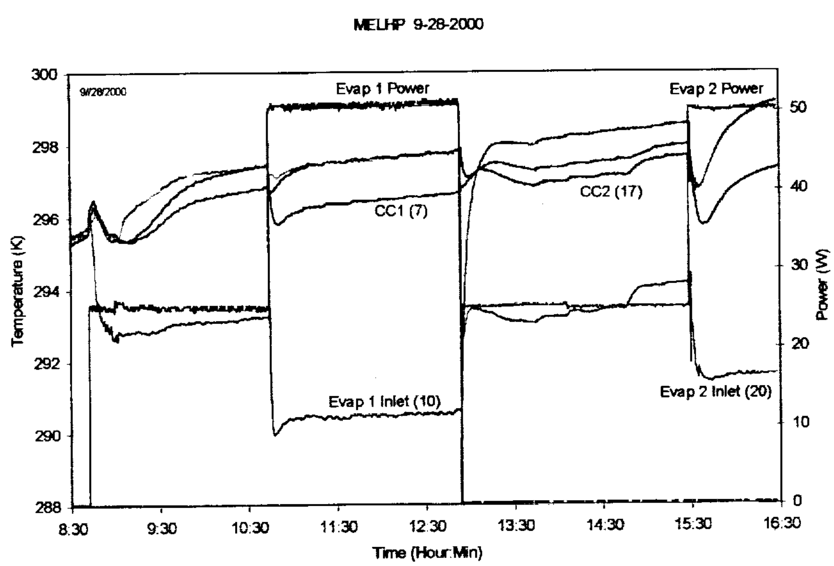
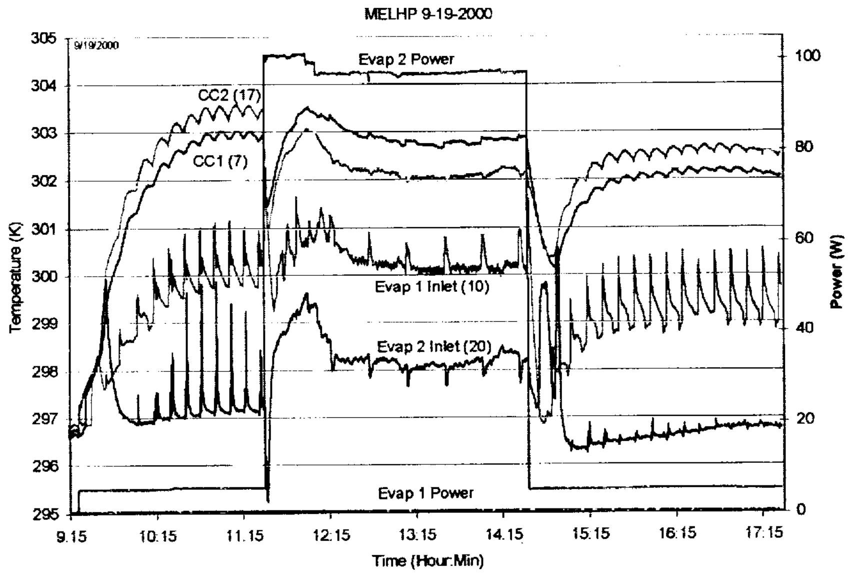


Figure 3. Loop Temperatures in Power Cycle Test at 273K/273K



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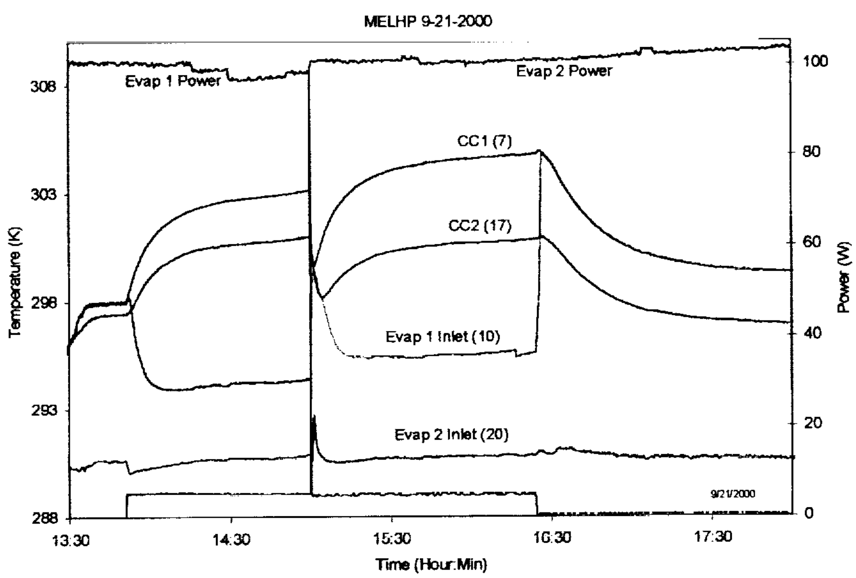

Figure 5. Loop Temperatures in Power Cycle Test

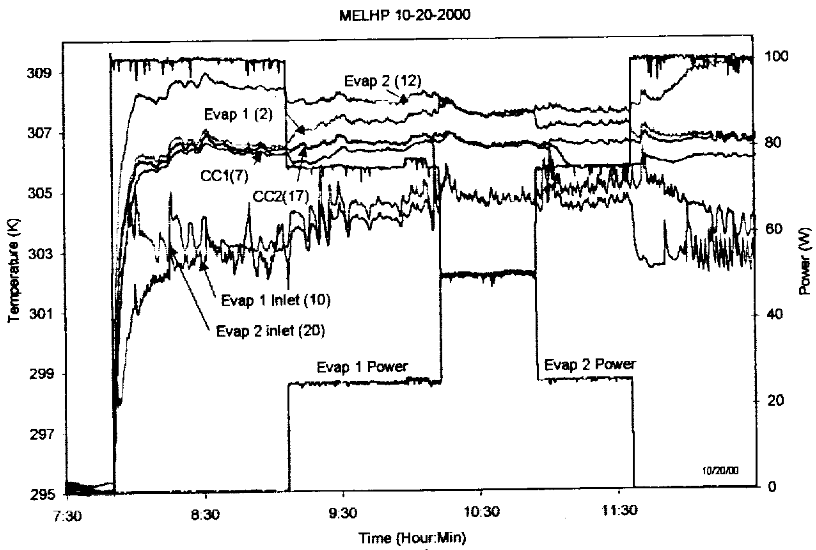

Figure 6. Loop Temperatures in Power Cycle Test at 293K/293K

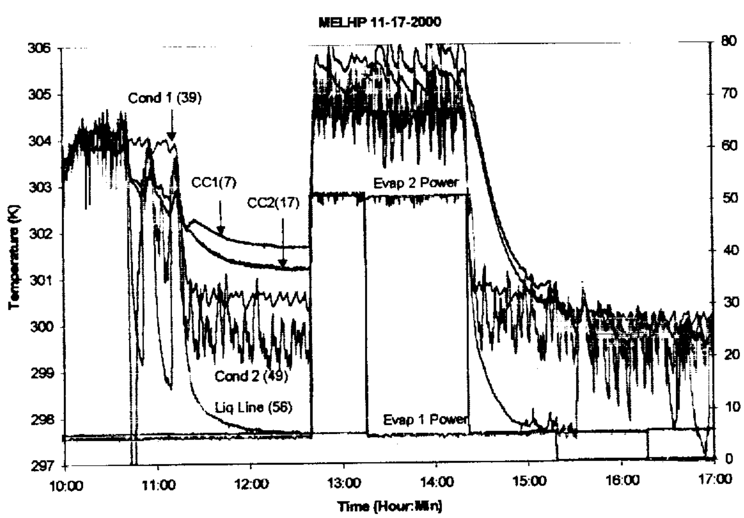

Figure 7. Loop Temperatures in Power Cycle Test at 300K/300K

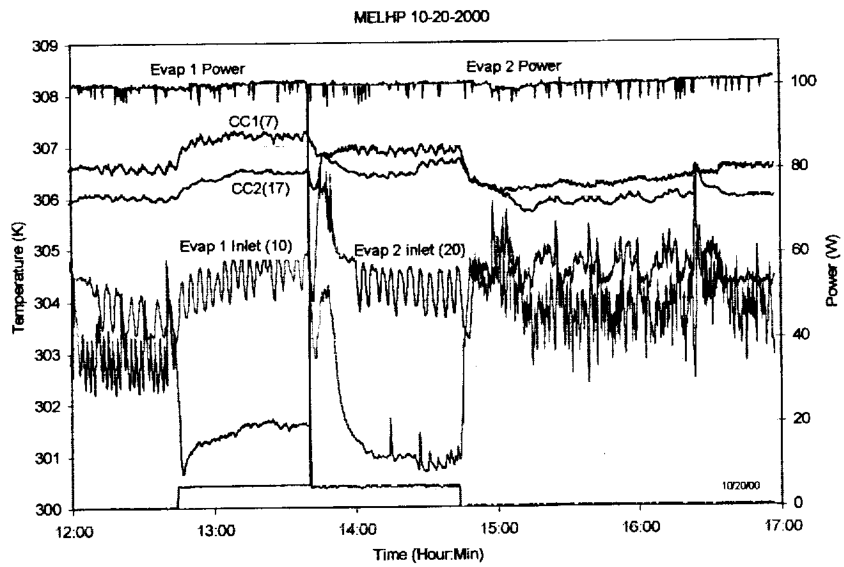

Figure 8. Temperatures in Power Cycle Test at 293K/293K

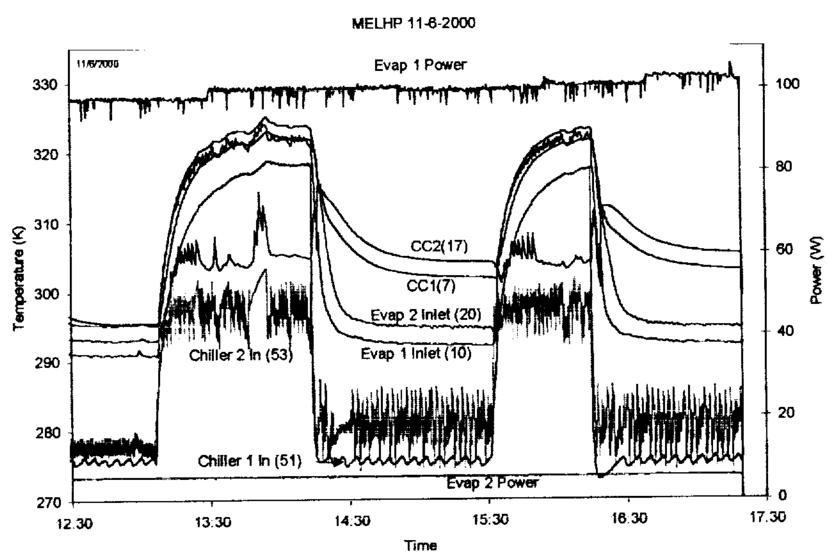

Figure 9. Loop Temperature in the Sink Temperature Cycle Test

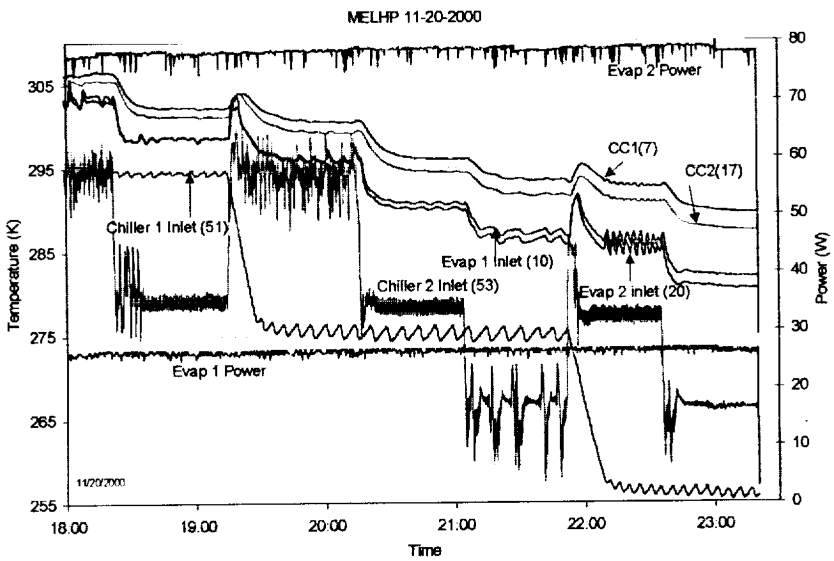

Figure 10. Loop Temperatures in Sink Temperature Cycle Test

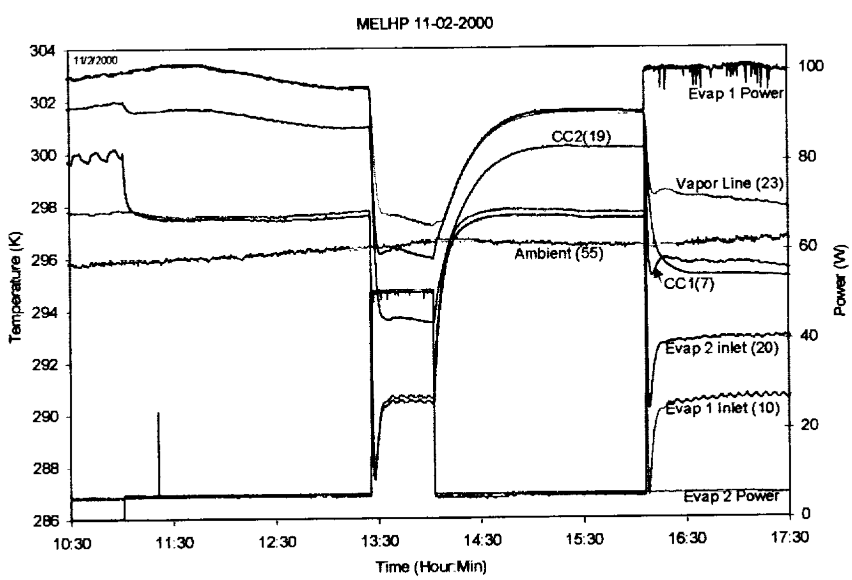

Figure 11. Loop Temperature in Power Cycle Test

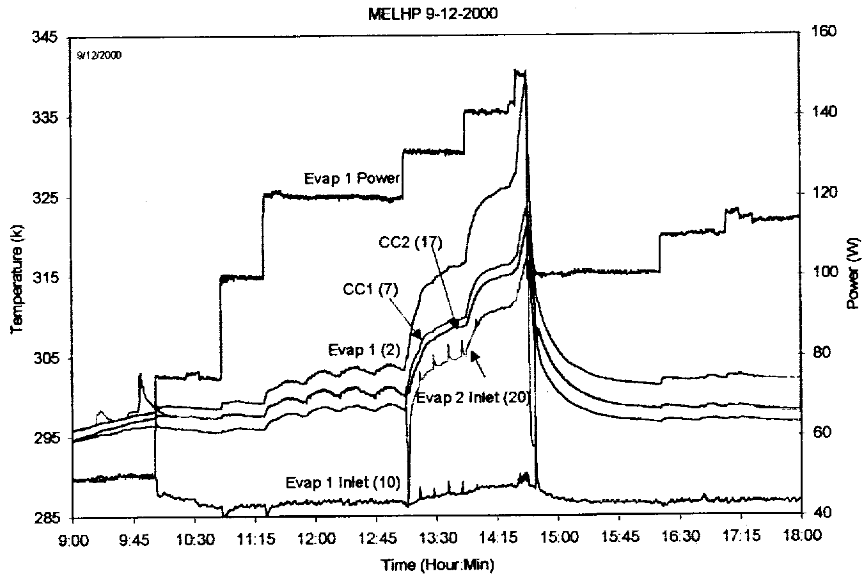

Figure 12. Loop Temperature in Capillary Limit Test