# Variable Conductance Heat Pipe for a Lunar Variable Thermal Link

Christopher J. Peters[1], John R. Hartenstine[2],
Calin Tarau[3], and William G. Anderson[4]
*Advanced Cooling Technologies, Inc., Lancaster, PA, 17601, U.S.A.*

**The Anchor Node Mission for the International Lunar Network (ILN) has a Warm Electronics Box (WEB) and a battery, both of which must be maintained in a fairly narrow temperature range using a variable thermal conductance link. During the lunar day, heat must be transferred from the WEB to a radiator as efficiently as possible. During the night, heat transfer from the WEB must be minimized to keep the electronics and batteries warm with minimal power, even with a very low (100 K) heat sink. Three different variable thermal links were identified that could perform this function: 1. A mini-loop heat pipe (LHP), 2. A mini-LHP with a thermal control valve, or 3. A Variable Conductance Heat Pipe (VCHP) with a hybrid wick. The mini-LHP has the highest Technology Readiness Level (TRL), but requires electrical power to shut-down during the 14-day lunar night, with a significant penalty in battery mass. The VCHP incorporates three novel features in order to achieve the design targets of the ILN program. The first is a hybrid wick, which allows the VCHP to operate with an adverse tilt in the evaporator. The second is locating the reservoir near the evaporator, rather than near the condenser, to prevent the reservoir temperature from dropping during the lunar night. Third, a bimetallic adiabatic section is used to minimize heat losses due to conduction when the VCHP is shut down. Testing included 1. Freeze/thaw, 2. Simulated lunar performance, with an adverse evaporator elevation, 3. Performance with a 2.54 mm (0.1 inch) adverse elevation, both for normal operation, and to demonstrate diode behavior when the condenser was heated. All of the tests were successful; however, the power with the heat pipe level was slightly lower than expected, probably due to problems with the hybrid wick interface.**

## I. Lunar Landers and Rovers

The lunar environment presents a number of challenges to the design and operation of thermal management systems. The heat rejection sink can be 330 K during daytime and can drop down to 50 K at night or in dark craters (Swanson and Butler, 2006). The Apollo landings were timed for lunar morning, so the environment was relatively benign. In contrast, future missions will need to operate over the entire temperature range. Typical lunar surface temperatures are shown in Figure 1. Instruments and equipment, such as batteries, will need to be maintained within -20ºC to 40ºC throughout the large diurnal temperature swings (Birur and Tsuyuki, 2009). In addition, depending upon the mission, the thermal system will be required to work both on the lunar surface after deployment and during the transit time from the earth to the moon.

Due to the wide temperature swings, future lunar landers and rovers will require a variable conductance thermal link that can reject heat during the day and passively shut off during the lunar night without requiring any electrical power. During the long lunar day, the thermal management system must remove waste heat from the electronics and batteries to prevent overheating. During the long lunar night, the variable thermal link must passively limit the amount of heat removed from the electronics and radiated to space since little to no power is available for temperature regulation. There are three basic elements to the thermal control system for a lander or rover:

[1] Research and Development Engineer, Aerospace Products, Chris.Peters@1-act.com
[2] Manager, Aerospace Products, AIAA Member
[3] Lead Engineer, Aerospace Products
[4] Chief Engineer, AIAA Member

1. A method to isothermalize the electronics and battery during the lunar night, and to remove heat to a variable conductance thermal link during the day.
2. A variable thermal link between the Warm Electronics Box (WEB) and the radiator. The WEB enclosure also contains the batteries.
3. A radiator to reject heat.

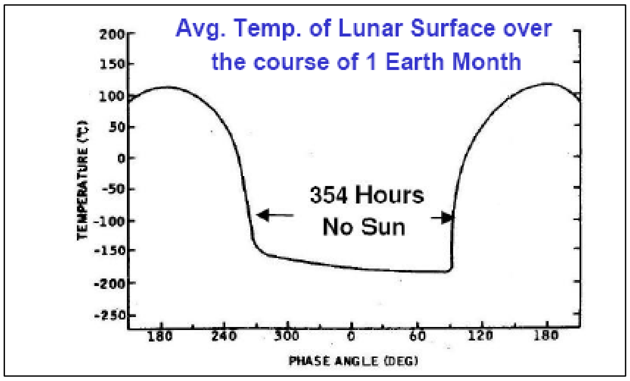



**Figure 1. Typical Lunar Surface Temperatures (Swanson and Butler, 2006).**

For solar powered systems, the variable link design is complicated by the heavy mass penalties associated with providing continuous power throughout the 14-day-long lunar night: It is estimated that 5 kg of batteries, solar cells, etc., are required to supply 1 W of electricity. Therefore, designs that operate without electrical power are highly desirable.

A number of technologies have been evaluated for a variable thermal link, or “heat switch”, including mechanical thermal switches, Variable Conductance Heat Pipes (VCHPs) and Loop Heat Pipes (LHPs). This work is discussed in Anderson et al. (2010). The thermal switch was dropped due to poor thermal conductance, less than one tenth that of a VCHP or LHP, while the pumped loop system was dropped as a result of moving parts, a higher mass, and higher power requirements. Both the VCHP and the LHP have similar “on” and “off” thermal conductances, both have flown in space, and both have similar masses. This paper discusses the design and testing of a VCHP; a companion paper discusses a LHP with a thermal control valve (Hartenstine et al. 2011).

**Table 1. Anchor Node Design Targets.**

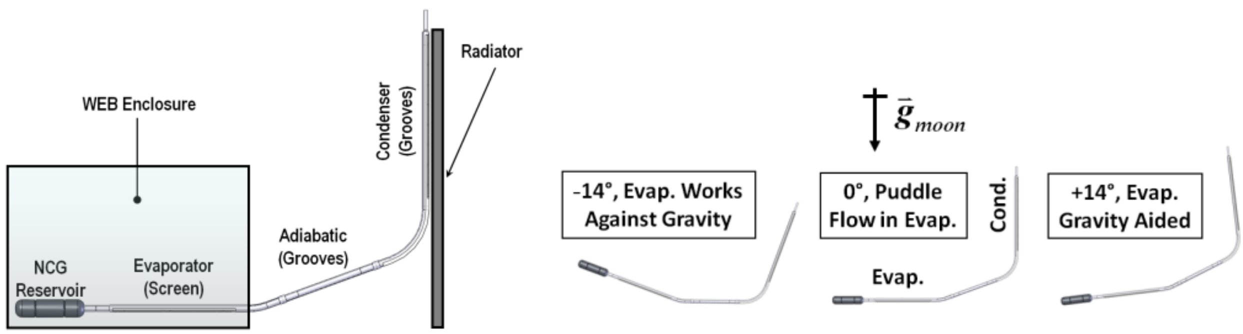
| | |
|---|---|
| Minimum Electronics Temperature | -10°C (263 K) |
| Maximum Electronics Temperature | 50 °C (323 K) |
| Maximum Radiator Load (Moon) | 73 W at lunar noon<br>(30 % margin: 94.9 W) |
| Power During Transit (Space) | 90 W during cruise<br>(30 % margin: 117 W) |
| Mission Length | ~ 6 years |
| Warm Electronics Box Geometry | 24” x 41” x 14” (height) |
| Distance from WEB to Radiator | 14” to side mounted vertical radiator |
| Minimum Radiator Sink Temperature (Moon) | 96 K |
| Maximum Radiator Sink Temperature (Moon) | 269 K |
| Maximum Tilt | ±14°<br>(8° slope, 6° obstruction) |

**A. Anchor Node**

The design targets were taken from the International Lunar Network (ILN) Anchor Node mission. The Anchor Nodes are small landers that include a seismometer, a laser reflector, and a probe for measuring heat flow from the moon’s interior. To accurately locate moonquakes, several seismometers on separate Anchor Nodes must operate simultaneously. Each Anchor Node lander has a WEB and a battery, both of which must be maintained in a fairly narrow temperature range. A variable thermal link between the WEB and radiator is required. During the day, the thermal link must transfer heat from the WEB electronics to the radiator as efficiently as possible, to minimize the radiator size. On the other hand, the thermal link must be as ineffective as possible during the lunar night. This will keep the electronics and battery warm with minimal power, even with the very low temperature (100 K) heat sink. At this time, heat must be shared between the electronics and battery, to keep the battery warm. Design targets for the program are shown in Table 1.

## II. VCHP Design

An important design target consideration is that the VCHP must work both in space and in a gravity-aided mode on the lunar surface. While on the moon, the VCHP needs to operate against a maximum slope. For the Anchor Node, it is ±14° (due both to the general tilt of the terrain, as well as any local rock or depression). The maximum slope for rovers can be as high as ±45°. The basic VCHP layout is shown in Figure 2 (a). The VCHP evaporator sits on the WEB enclosure baseplate. The adiabatic section is tilted so that it is always gravity aided on the moon.




(a) (b)

**Figure 2. a. Schematic of the VCHP layout from the WEB to the radiator. b. Depending on the terrain, the evaporator can have a ±14° tilt relative to horizontal. In some cases, the evaporator wick must work against gravity on the lunar surface, which is not possible with a conventional grooved wick.**

Due to the ±14° slope, the conventional spacecraft all-grooved wick is not suitable in the evaporator section. Typically, grooved heat pipes are tested on earth with an adverse tilt (evaporator above condenser) of 0.1 inch. As shown in Figure 2 (b), the evaporator can have an adverse tilt of several inches. To accommodate this high adverse elevation, a hybrid wick was developed. All of the condenser and most of the adiabatic section employ axial grooves for liquid return. A small portion of the adiabatic section and all of the evaporator use wrapped screen mesh as the wicking structure. The screen mesh wick, which has higher capillary pumping pressure than grooves, enables the evaporator to operate against the tilt caused by the uneven lunar surface.

A schematic of the VCHP is shown in Figure 3. The VCHP incorporates three novel features in order to achieve the design targets of the ILN program:

1. Hybrid-Wick, discussed above.
2. Reservoir near Evaporator, to prevent the reservoir temperature from dropping during the lunar night.
3. Bimetallic Adiabatic Section, to minimize heat losses during the lunar night.

A stainless steel heat pipe section of 12.7 cm (5 inches), which replaces the original aluminum portion in the adiabatic section, acts as a thermal dam and minimizes axial heat leak to the cold radiator during shutdown.

The heat pipe uses anhydrous ammonia as the working fluid, since it is the best working fluid when the heat pipe is operational. The Non-Condensable Gas (NCG) has two purposes: 1. Provides a variable thermal link that turns off as the evaporator temperature drops, and 2. Suppresses the freezing of ammonia in the condenser during the lunar night. The freezing point of ammonia is 195 K, while the radiator could conceivably cool down to 96 K. In a CCHP, ammonia would tend to freeze in the condenser. However, in a VCHP, the gas in the adiabatic and condenser sections blocks the flow of ammonia from the evaporator to the condenser. Ammonia can only slowly diffuse through the gas. The gas also aids in starting up the heat pipe after sunrise (Ellis and Anderson, 2009).

VCHPs typically use argon as the NCG. Due to the low temperatures, neon was selected for this VCHP instead. The reason is that the critical temperature of Argon is 151 K, so the argon would not be a perfect gas during the lunar night, and could actually condense. Neon, with a critical temperature of 44 K, will behave like a perfect gas.

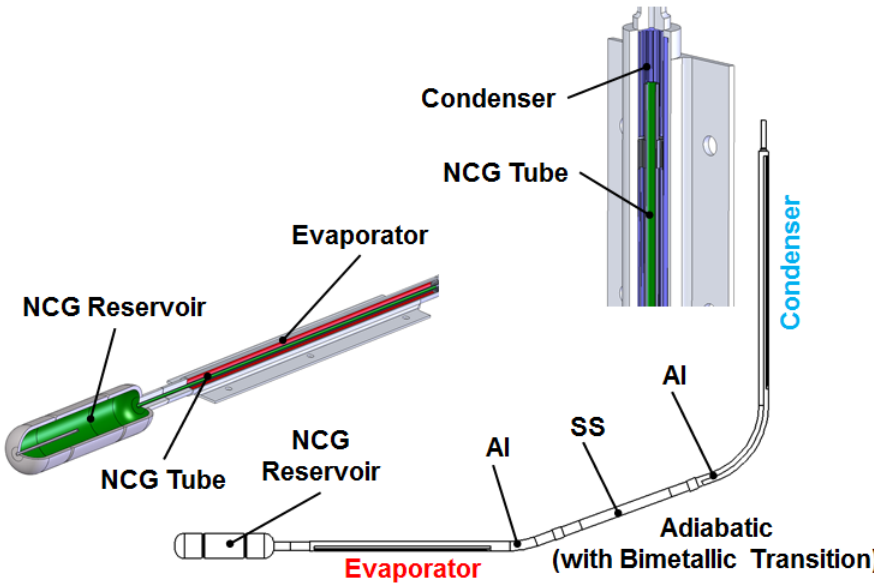



**Figure 3. Schematic of the VCHP with a hybrid wick, to allow operation at different tilts. Placing the reservoir near the evaporator keeps the reservoir warm, minimizing the required reservoir size. Also, part of the adiabatic section is stainless steel, which minimizes heat leaks when the VCHP is turned off.**

### B. Reservoir Location

Placing the NCG reservoir near the evaporator, as opposed to the traditional location near the condenser, keeps the gas reservoir warm and minimizes the reservoir size. A conventional VCHP, shown in Figure 4, has the reservoir located next to the condenser. The VCHP reservoir temperature is controlled by tying it thermally to another portion of the spacecraft to cold bias the VCHP, then adding heaters on the reservoir to control power.


**Figure 4. Standard VCHPs have the reservoir located next to the condenser.**

In contrast, no electrical heating is available to control the VCHP temperature in a lunar lander or rover. If the VCHP reservoir was located at the top of the condenser in Figure 2 (a), then the VCHP reservoir would operate near the sink temperature of 96 K during the lunar night. Anderson, Ellis, and Walker (2009) developed the equations to size this type of radiator, and showed that with a wide variation in sink temperature, there is a minimum $\Delta T_{VCHP}$, even with an infinite reservoir.

Placing the NCG reservoir near the condenser would necessitate a very large reservoir that can only provide coarse temperature control ($\Delta T_{VCHP} \geq 30$ °C). $\Delta T_{VCHP}$ is defined as the difference between the operating and shutdown temperatures of the evaporator ($\Delta T_{VCHP} = T_{On} - T_{Off}$). A very precisely controlled VCHP would have a very small, but finite $\Delta T_{VCHP}$. Such temperature control is only possible with a reservoir near the evaporator. Figure 5 illustrates these principles.

In Figure 5, items colored red correspond to a warm reservoir placed near the evaporator and items colored in blue correspond to a cold reservoir located near the condenser. The two dashed vertical lines represent the asymptotes for either the warm or cold reservoir. In both reservoir types, the mass of the system trends exponentially towards infinity as the degree of control increases and $\Delta T_{VCHP}$ decreases. Observe that an infinitely large cold reservoir provides $\Delta T_{VCHP} \approx 30$ °C and an infinitely large warm reservoir provides $\Delta T_{VCHP} = 0$ °C. For this reason, the reservoir is located in the WEB, next to the evaporator, where it can be kept warm by the heat pipes used to transfer heat in the WEB. A small NCG tube passes through the entire length of the heat pipe to pneumatically connect the NCG reservoir to the condenser (see Figure 3). This NCG tube allows the NCG reservoir to be located near the evaporator rather than the condenser.

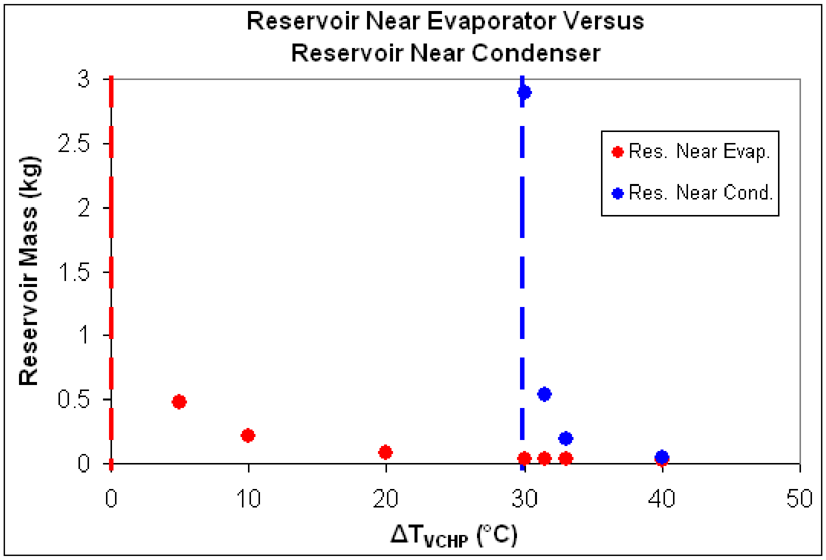



**Figure 5. Comparison between NCG Reservoir Locations.**

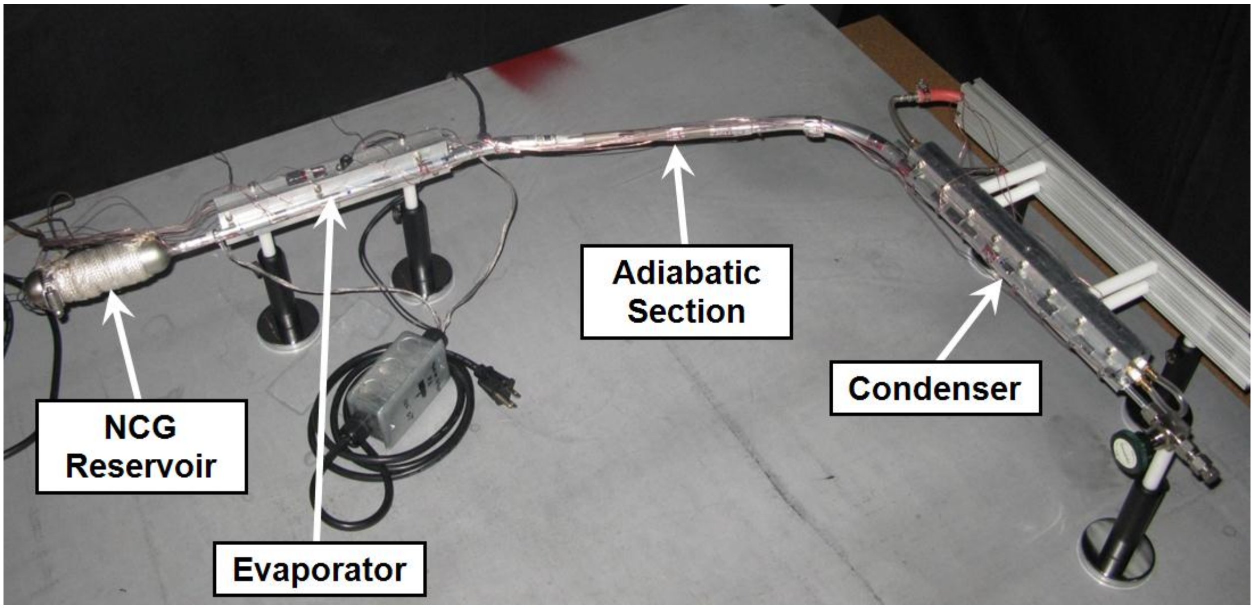



**Figure 6. Hybrid wick VCHP with reservoir adjacent to the evaporator.**

## III. VCHP Fabrication and Test Setup

The VCHP had the following specifications:

- 30.5 cm (12 inch) Condenser / ≈ 48.3 cm (19 inch) Adiabatic Section – Grooved aluminum extrusion (6063-T6 Al)
    - Bimetallic Transition – 1.25 inch 6061-T6 Al × 5 inch 304 SS × 1.25 inch 6061-T6 Al
- 22.9 cm (9 inch) Evaporator – Nickel 200 50×50 screen mesh
- NCG Tube (304 SS) – 0.32 cm (0.125 inch) outer diameter
- NCG Reservoir (304 SS) – 73.7 $cm^3$ (4.5 $inch^3$) internal volume
- Working Fluid (Ammonia) – 20.8 grams
- Non-Condensable Gas (Neon) – ≈0.65 grams

The completed hybrid wick VCHP is shown in Figure 6, while the thermocouple locations are shown in Figure 7. Heat was supplied to the evaporator with electric cartridge heaters embedded in an aluminum block. Heat was removed from the condenser with a cold plate, cooled with liquid nitrogen to a fixed temperature.

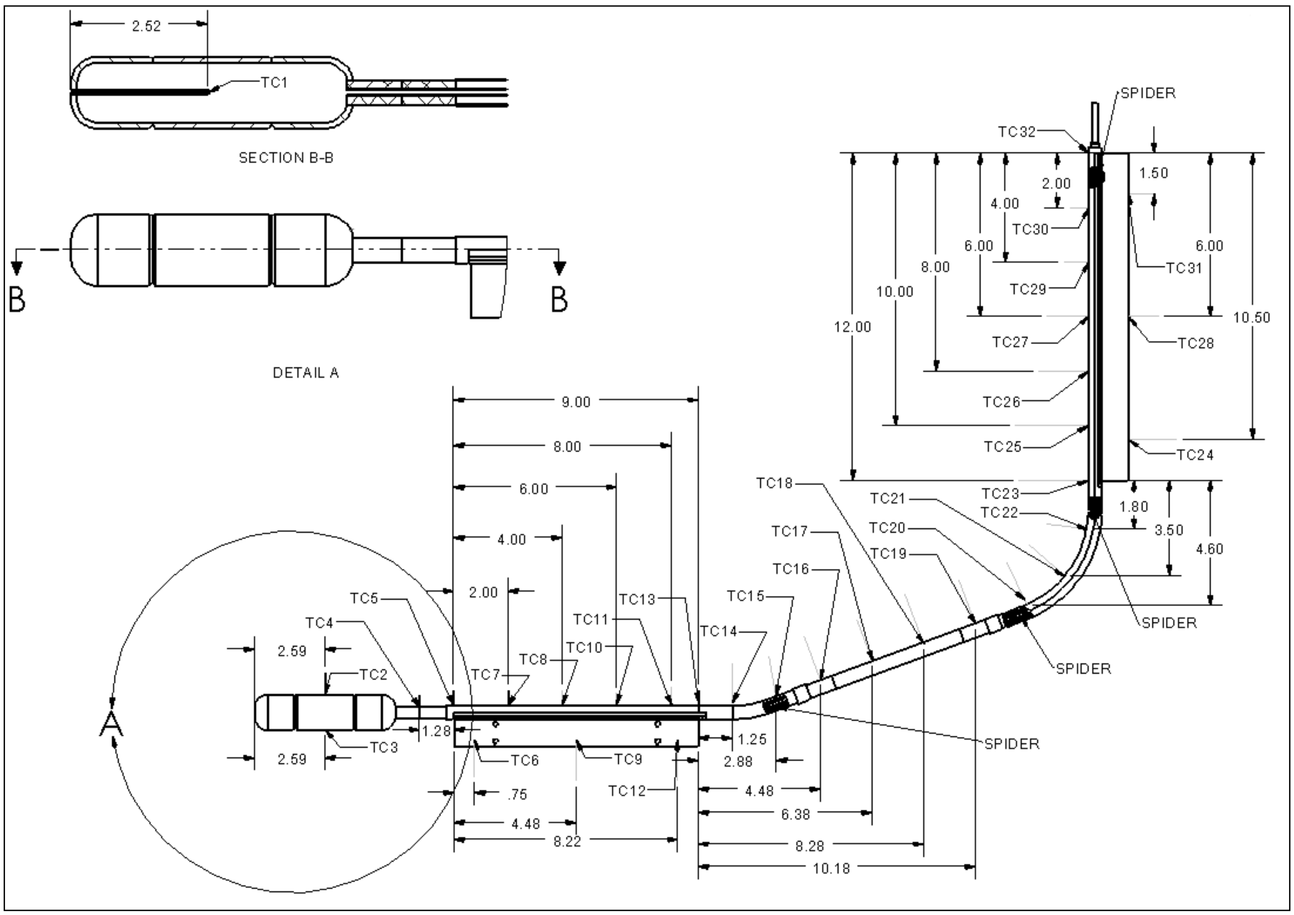



**Figure 7. VCHP Thermocouple Placement.**

## IV. VCHP Testing

### C. Test Objectives

There were four main test objectives:

1. Freeze/Thaw (Lunar): Demonstrate the ability of the VCHP to act as a variable thermal link on the moon, and minimize the heat transfer as the condenser temperature dropped below the freezing point of

ammonia. In addition, verify that the VCHP can operate for short periods with a cold condenser (which can occur in some lander and rover scenarios).

2. Maximum Power (Lunar): Measure the power that the VCHP can transfer on the moon when the evaporator tilt is adverse, level, or favorable.
3. Maximum Power (Space): Measure the power at different adverse elevations, and estimate the maximum power that the VCHP could carry in space.
4. Diode (Space): Demonstrate that the VCHP will act as a diode in space, preventing heat from solar insolation from heating the WEB.

### D. Lunar Freeze/Thaw Results

For the simulated lunar tests, the VCHP was mounted in a test fixture on an optical table that set the condenser vertical and allowed the evaporator to be configured for gravity aided (+2.3°), gravity neutral (0°) and gravity adverse (-2.3°) inclinations (2.3° on earth is equivalent to a 14° inclination on the moon). Test conditions for the lunar freeze/thaw tests are shown in Table 2.

**Table 2. Testing Procedure for Lunar Freeze/Thaw.**

| Inclination (°) | Evaporator Minimum Temperature (°C) | Condenser Temperature (°C) |
|---|---|---|
| -2.3 | -10 | -60 |
| -2.3 | -10 | -177 |

Figure 8 plots VCHP temperatures as a function of time during the lunar freeze/thaw test. TC1 corresponds to the gas temperature in the NCG reservoir. TC10 measures the vapor temperature of the evaporator. TC23, TC26, TC27 and TC30 detect the vapor temperature of four locations within condenser, with TC23 at the entrance of the condenser and TC30 close to the tip of the condenser. The power curve shows the electrical power input into the heater block of the evaporator. For more information on the locations of these TCs, consult Figure 7.

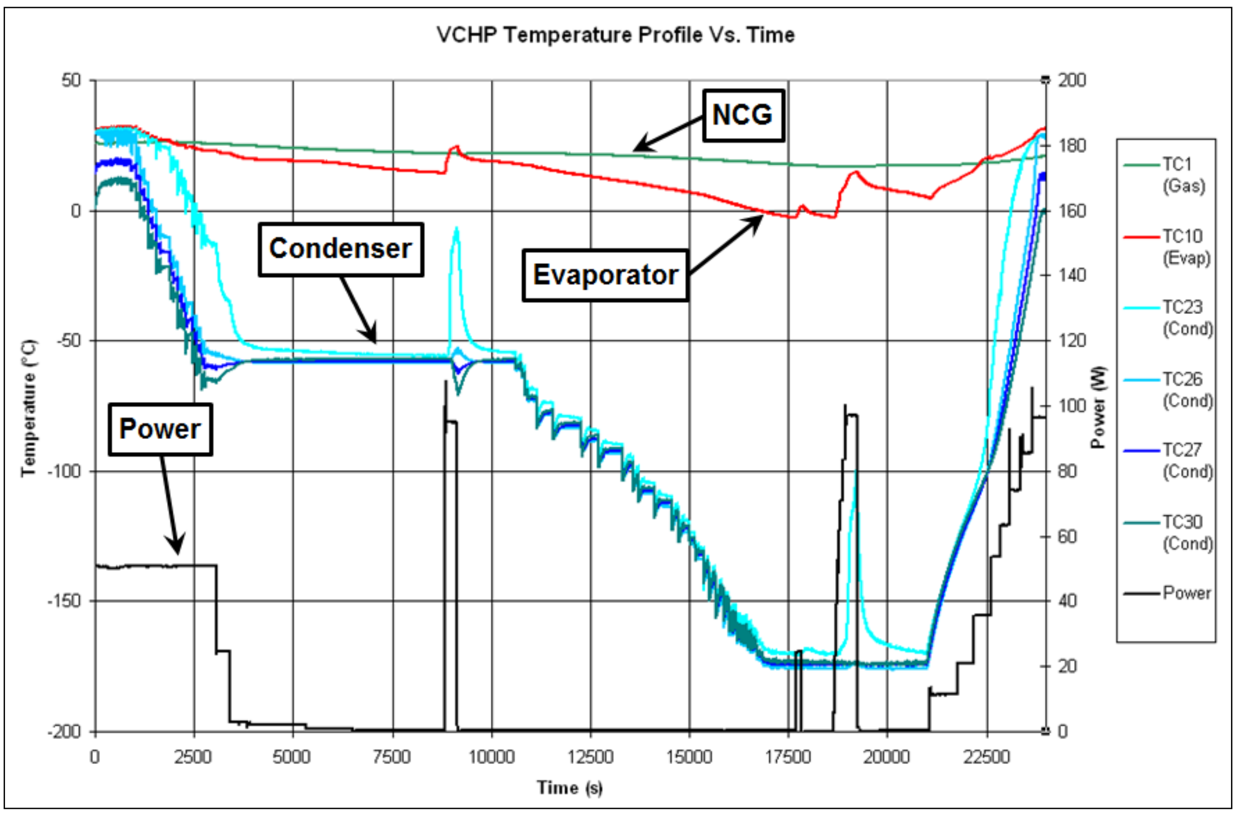



**Figure 8. VCHP Temperature Profile as a Function of Time (-2.3° Orientation).**

Initially, the pipe is operating at a nominal 25 °C and 50 W. At about 6000 seconds the pipe temperature and power input are reduced to -60 °C and 0.2 W, respectively. The purpose of 0.2 W of heat input was to maintain the evaporator above -10 °C. At around 9000 seconds, power is temporarily increased to the full 95 W, to simulate a brief period of activity during the lunar night. After this power increase, the pipe was returned to the -60 °C shutdown state. Next, the sink temperature is further reduced to -177 °C (96 K, ammonia freezes at 195 K). The pipe reaches a steady-state shutdown at -177 °C and 0.1 W. At approximately 17,500 seconds, the power is briefly increased to 25 W and the transient response of the frozen pipe was observed. With no indication of problems, the pipe is returned to -177 °C shutdown. Power is then increased to a full 95 W for a short duration. After the full power increase, the pipe is returned to a state of shutoff until around 21000 seconds when the power is gradually increased and heat pipe startup begins. Finally, the VCHP is brought to nominal steady-state operation at 95 W and 25 °C.

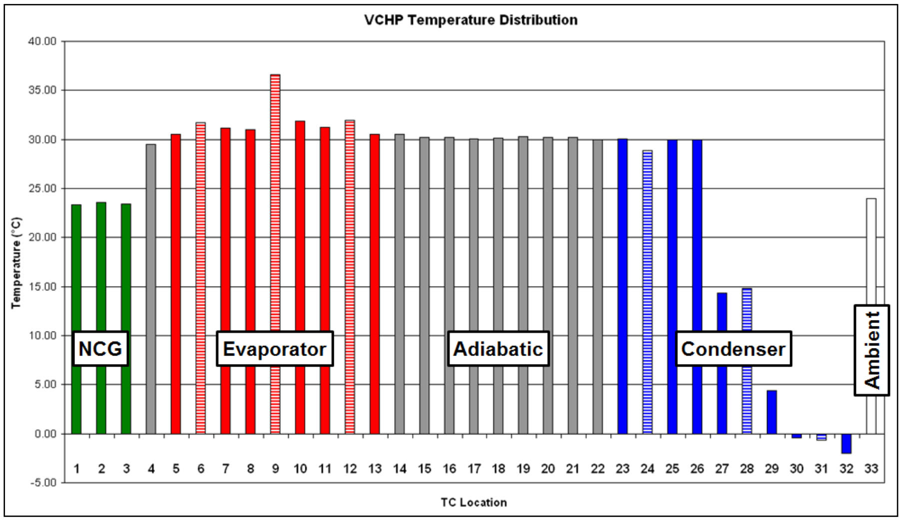



**Figure 9. VCHP Operation (25 °C, 95 W, -2.3° Orientation).**

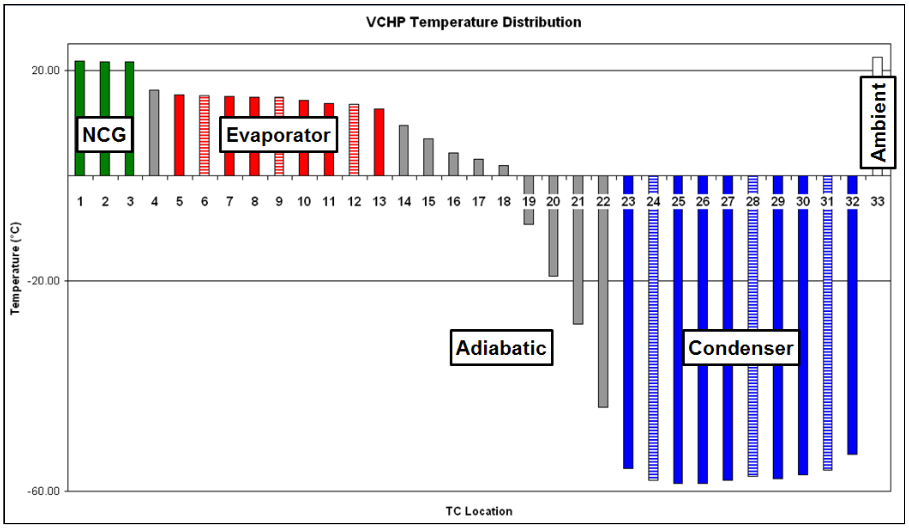



**Figure 10. VCHP Cold Shutoff (-60 °C, 0.2 W, -2.3° Orientation).**

During the freeze/thaw tests, TC10 on the evaporator experienced relatively small temperature fluctuations compared to the rest of the system. The condenser followed the temperature of the sink, especially when the heat pipe was shut off; however, the evaporator maintained its temperature, staying well above the -10 °C lower limit. The evaporator required only minimal input power (0.1 W to 0.2 W). There was a large temperature difference between the evaporator and condenser throughout most of the test. This indicates that the VCHP is working correctly and successfully shutting off. This temperature difference becomes small during the start and end of the test when the heat pipe is operating.

Figure 9, Figure 10 and Figure 11 show the instantaneous temperature profile within the heat pipe during operation, -60 °C shutoff and -177 °C shutoff, respectively. In these figures, green denotes the non-condensable gas temperature, red represents evaporator temperatures, gray stands for the adiabatic section temperatures (TC6 is the temperature of the bimetallic end cap), blue corresponds to condenser temperatures and white signifies the ambient temperature. Solid color bars are vapor temperatures whereas striated color bars symbolize the interfacial temperatures in the condenser and evaporator heat transfer paths, measured by plunger TCs.

When the heat pipe is operating, the temperatures are nearly uniform as seen in Figure 9. The gas/vapor front is in the condenser at approximately TC27. Note that the evaporator and gas reservoir temperatures are very similar. In the ILN Anchor Node spacecraft, both the evaporator and gas reservoir would be in the WEB enclosure together and would be thermally coupled. TC9, a plunger-style thermocouple consistently measured higher temperatures than the rest of the evaporator. It is speculated that there might have been some separation between the screen wick and the evaporator wall at this location, resulting in additional thermal resistance and a local rise in temperature.

Figure 10 shows the VCHP shut down with the heat sink at a temperature of -60 °C. The gas/vapor front is rather diffuse, with no sharp temperature transition from the warm operating region to the cold shutdown region. The center of the gas/vapor front is near the middle of the adiabatic section. At -60 °C, the ammonia working fluid is still above its freezing point. The VCHP does a good job of shutting off, evident from the large temperature gradient across the adiabatic section and the uniformity of temperatures in the evaporator. Note that the evaporator temperatures are well above the -10 °C lower limit, with a maintenance power input of 0.2 W.

Figure 11 illustrates the temperature profile of the VCHP when it has shut down and the sink temperature is at -177 °C. Ammonia freezes at -78 °C and any working fluid condensate left in the condenser will have frozen. The diffuse gas/vapor front has its center in the adiabatic section, closer to the evaporator than the condenser. The temperature of the evaporator is above the -10 °C lower bound with a maintenance power of 0.1 W.

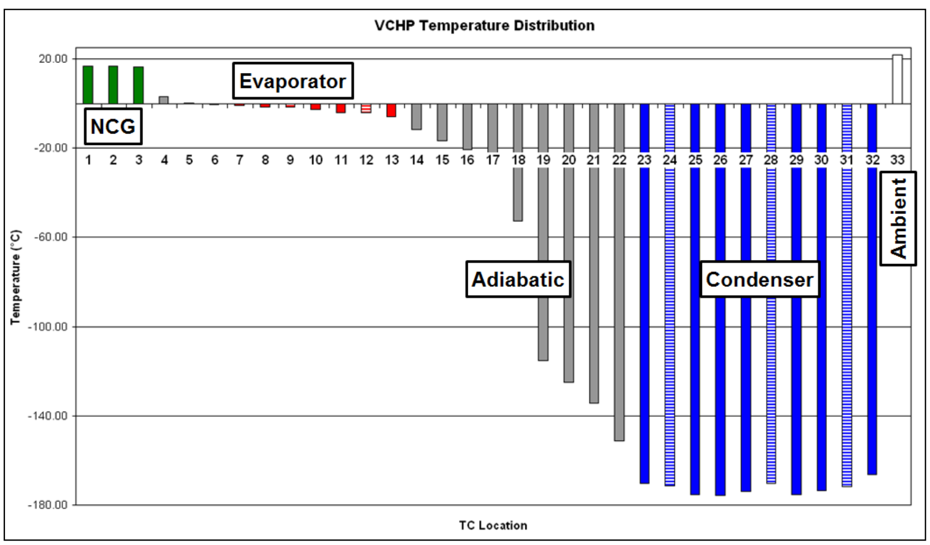



**Figure 11. VCHP Very Cold Shutoff (-177 °C, 0.1 W, -2.3° Orientation).**

**Table 3. VCHP Overall Conductances for Lunar Freeze/Thaw.**

| Testing Condition | Overall Conductance (W/°C) | Power (W) |
|---|---|---|
| 25 °C Operation | 4.7 | 95 |
| -60 °C Shutdown | 0.0031 | 0.2 |
| -177 °C Shutdown | 0.00057 | 0.1 |
| -2.3° Inclination; 9 Inch Evaporator; 12 Inch Condenser | | |

Table 3 lists the overall conductance (evaporator to condenser) of the VCHP during operation, -60 °C shutdown and -177 °C shutdown. The goal of the ILN VCHP was to minimize conductance during shutdown, especially at very cold sink temperatures. As evidenced from Table 3, the heat pipe satisfied this goal, dropping in conductance three orders of magnitude for the -60 °C shutdown state and four orders of magnitude for the -177 °C shutdown states. These conductances neglect any heat in-leak from the environment.

The freeze/thaw tests were of relatively short duration, and do not demonstrate conclusively that the VCHP can start up after a 14-day-long shutdown period, corresponding to the lunar night. Previous tests with titanium/water VCHPs have demonstrated startup after they were shut down with the condenser frozen for a 14-day-long period (Ellis and Anderson, 2009). Further, longer duration freeze/thaw testing of the current VCHP is planned.

### E. Lunar Thermal Performance Results

During the lunar thermal performance tests, the heat pipe condenser was again nominally vertical. Table 4 outlines the procedure for the lunar performance tests. As in the freeze/thaw testing, TC9 (plunger-style) consistently measured a higher temperature than the rest of the thermocouples on the evaporator. In these tests, dryout was not reached. Instead, the test was halted whenever the temperature difference between TC9 and the rest of the heat pipe reached approximately 20 °C. Table 5 tabulates the maximum powers for the different evaporator inclinations (-2.3° adverse; 0° neutral; +2.3° aided). The VCHP could probably transport higher powers than those recorded in Table 5, but testing was stopped at these wattages because of the 20 °C ΔT between the hotspot and the rest of the pipe. Observe that these powers are over twice the value of the heat transport target for the ILN VCHP (94.9 W).

**Table 4. Testing Procedure for Lunar Thermal Performance.**

| Inclination (°) | Evaporator Temperature (°C) |
|---|---|
| -2.3 | 25 |
| 0 | 25 |
| +2.3 | 25 |

**Table 5. VCHP Powers for Lunar Thermal Performance (Target: 95 W).**

| Inclination (°) | Maximum Power Tested (W) |
|---|---|
| -2.3 | 220 |
| 0 | 212 |
| +2.3 | 220 |
| 25 °C Operating Temperature | |

### F. Space Thermal Performance Results

The orientation of the ILN VCHP was changed for the space tests. Instead of being in a vertical orientation, the VCHP was placed in a horizontal orientation, with a slight relative elevation between the evaporator and condenser. Figure 6 shows the horizontal orientation of the un-insulated heat pipe during all simulated space testing. The VCHP, along with the heating and cooling blocks and their respective stands are visible.

For the space thermal performance test, the elevation was adverse with respect to the capillary flow; the evaporator was 0.1 inch, 0.2 inch and 0.3 inch above the condenser and the pipe was operated at 25 °C (Table 6). Heat was inputted into the evaporator and rejected from the condenser.

**Table 6. Testing Procedure for Space Thermal Performance.**

| Adverse Elevation (in) | Evaporator Temperature (°C) |
|---|---|
| 0.1 | 25 |
| 0.2 | 25 |
| 0.3 | 25 |

Simulated space testing began with the pipe nearly horizontal and 0.1 inch of adverse elevation between the evaporator and condenser. Anomalous results were observed: the pipe only transported 30 W and had a very non-uniform temperature distribution (a correctly functioning pipe is practically isothermal). It was hypothesized that there could be a possible asymmetry within the heat pipe. The symmetry plane is the same plane which contains the 20° and 70° bends in the adiabatic section. The pipe was flipped 180° with respect to the gravity vector. Retesting showed a marked improvement in transport capability and isothermality of the heat pipe. This confirmed the presence of an internal asymmetry that impairs liquid return. All of the space testing results (thermal performance and thermal diode) are with the pipe in this flipped orientation. We believe that this anomaly is due to a problem at the grooved/screen wick interface. With data from three different elevations, one can perform a linear extrapolation to predict the transport capability of the heat pipe in zero gravity; see Table 7.

**Table 7. VCHP Dryout Powers for Space Thermal Performance (Target: 117 W).**

| Adverse Elevation (in) | Dryout Power (W) |
|---|---|
| 0.0 (Zero-Gravity) | 83* |
| 0.1 | 64 |
| 0.2 | 37 |
| 0.3 | 22 |
| 25 °C Operating Temperature<br>*Extrapolated Power | |

**G. Space Thermal Diode Results**

During transit, at some times the radiator will be hotter than the WEB due to solar insolation. During these times, it is desirable for the VCHP to act as a diode, preventing overheating of the WEB. A conventional VCHP would behave like a gas-loaded diode in this situation. Tests were conducted to verify that a VCHP with the reservoir near the evaporator would also act as a diode. During these tests, the ILN VCHP was kept in a horizontal orientation; however, the adverse elevation and heat input/output were reversed compared to the space thermal performance test. The goal of this test was to verify that the pipe inhibited heat transfer in the reverse direction; therefore, the test intentionally attempted to operate the heat pipe backwards. Heat was input into the condenser and rejected from the evaporator. Since the capillary flow would travel from the evaporator to the condenser, the adverse elevation was defined as the condenser being 0.1 inch, 0.2 inch and 0.3 inch above the evaporator. Table 8 shows the testing procedure for the space thermal diode.

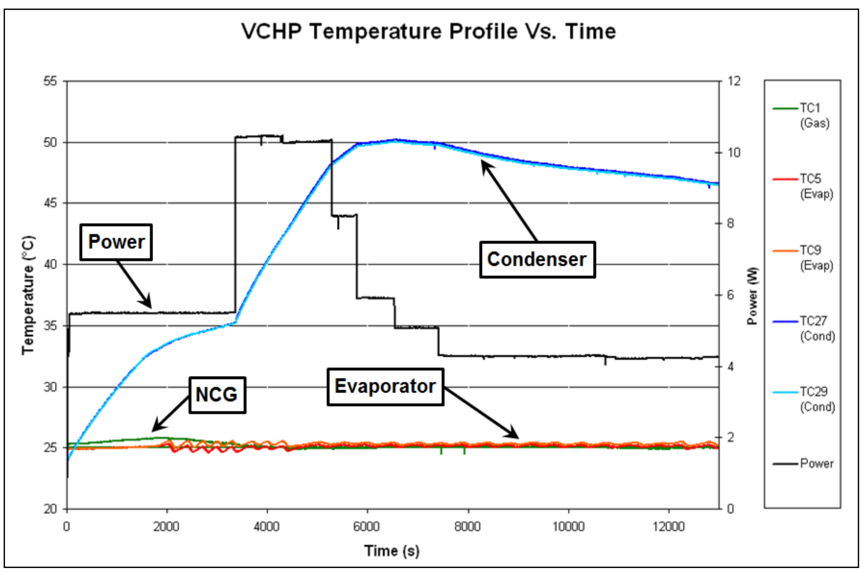



**Figure 12. VCHP Temperature Profile as a Function of Time (Evaporator at 25 °C, 0.1 Inch).**

Figure 12 illustrates the temperatures within the pipe as a function of time for the 0.1 inch adverse elevation trial. The plot shows that the input power was adjusted until a steady-state temperature difference of 20 °C was observed between the evaporator and condenser. Once this temperature difference was achieved, the input power was recorded as the reverse heat transfer rate. For this particular test, the reverse heat transfer rate was 4.3 W.

**Table 8. Testing Procedure for Space Thermal Diode.**

| Adverse Elevation (in) | Evaporator Temperature (°C) | Condenser Temperature (°C) |
|---|---|---|
| 0.1 | 25 | 45 |
| 0.2 | 25 | 45 |
| 0.3 | 25 | 45 |

Table 9 lists the results of the thermal diode experiment. All of the reverse powers are low (less than 4 % of the 117 W target for space) and the conductances are minimal (two orders of magnitude less than the values of the space thermal performance test). The conductances are negative because the condenser is hotter than the evaporator, which is the opposite of normal operation.

**Table 9. VCHP Results for Space Thermal Diode (Evaporator at 25 °C; Condenser at 45 °C).**

| Adverse Elevation (in) | Reverse Heat Transfer Rate (W) | Overall Conductance (W/°C) |
|---|---|---|
| 0.1 | 4.3 | -0.205 |
| 0.2 | 3.2 | -0.165 |
| 0.2 | 3.2 | -0.168 |
| 9 Inch Evaporator; 12 Inch Condenser | | |

As shown in Table 9, the ILN VCHP effectively shuts down during backwards operation and minimizes heat transfer in the reverse direction.

## V. Conclusion

A VCHP was developed to act as a variable thermal link for lunar landers and rovers, passively minimizing heat losses during the lunar night, without requiring electric power to shut off. Three major changes to a conventional spacecraft VCHP were incorporated to allow operation in different orientations:

1. Hybrid wick, to allow the evaporator to operate when tilted in an adverse orientation.
2. Reservoir near the evaporator, to minimize the reservoir size and mass.
3. Bimetallic adiabatic section, with a length of grooved stainless steel to minimize heat leaks during the lunar night.

Four different tests were run:

1. Lunar Freeze/Thaw.
2. Lunar Performance.
3. Simulated Space Performance.
4. Simulated Space Diode.

The simulated lunar performance testing demonstrated:

- VCHP shut off as the condenser temperature was lowered, reducing heat transfer.
- Freeze/thaw cycles without performance degradation.

- Can operate briefly at low temperatures – accommodated short-duration full-power bursts during -60 °C and -177 °C cold shutdown.
- Design can meet target power at adverse elevations.
- VCHP started up with frozen condenser.

The simulated 0-g testing demonstrated:

- Effective thermal diode operation.
- Performance shortfalls encountered in testing indicated potential hybrid wick design and fabrication issues.

The VCHP satisfied all of the ILN targets except for the 117 W zero-gravity power transport. A sintered wick insert is under investigation.

Further testing is planned at NASA Marshall. The low temperature tests were for relatively short durations, much less than the 14-day-long lunar night. Longer term tests are planned. In addition, tests will be made in a thermal vacuum chamber, to more closely simulate the actual thermal conditions.

## Acronyms

| | |
|---|---|
| CCHP | Constant Conductance Heat Pipe |
| ILN | International Lunar Network |
| LHP | Loop Heat Pipe |
| NCG | Non-Condensable Gas |
| VCHP | Variable Conductance Heat Pipe |
| WEB | Warm Electronics Box |

## Acknowledgments

This program was sponsored by NASA Marshall Space Flight Center under Purchase Order No. NAS802060. We would like to thank the technical monitor, Jeffery Farmer of NASA Marshall, for many helpful technical discussions. Tim Wagner was the technician at ACT. We would like to thank Kyle Van Riper for technical discussions about the VCHP. Any opinions, findings, and conclusions or recommendations expressed in this article are those of the authors and do not necessarily reflect the views of the National Aeronautics and Space Administration.

## References

1. W. G. Anderson, M. C. Ellis, and K. Walker, "Variable Conductance Heat Pipe Radiators for Lunar and Martian Environments," SPESIF 2009, Huntsville, AL, February 24 - 27, 2009.
2. W. G. Anderson, J. R. Hartenstine, K. L. Walker, and J. T. Farmer, "Variable Thermal Conductance Link for Lunar Landers and Rovers," 8th IECEC, Nashville, TN, July 25-28, 2010.
3. Birur, G., Tsuyuki, G., "JPL Advanced Thermal Control Roadmap – 2009", presented at the Spacecraft Thermal Control Workshop, March 10-12, 2009.
4. M. C. Ellis and W. G. Anderson, "Variable Conductance Heat Pipe Performance after Extended Periods of Freezing," SPESIF 2009, Huntsville, AL, February 24 - 27, 2009.
5. J. R. Hartenstine, K. L. Walker, and W. G. Anderson, "Loop Heat Pipe with Thermal Control Valve for Variable Thermal Conductance," 41st International Conference on Environmental Systems (ICES 2011), Portland, OR, July 17-21, 2011.
6. Swanson, T., and Butler, D., "NASA/Goddard Thermal Control Technology Roadmap-2006", 17th Aerospace Spacecraft Thermal Control Workshop, Los Angeles, CA, March 14-16, 2006.