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Hot Reservoir Stainless-Methanol Variable Conductance Heat Pipes for Constant Evaporator Temperature in Varying Ambient Conditions

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Abstract

Precision electronics require tight temperature control to operate at maximum performance. This becomes challenging with large fluctuations in ambient conditions. For this high altitude (20Km) application, ambient temperatures vary from -85°C to -45°C and the allowable temperature range for the cooled device is from 15°C to 30°C. Traditional approaches to cool a device with these requirements include active means of throttling coolant flow, whether it's from a fan or pumped single phase system. These active systems require additional power and weight, which results in decreased reliability. Variable conductance heat pipes (VCHPs) represent an attractive solution since they can be passive devices. VCHPs are similar to conventional heat pipes but have a reservoir and a controlled amount of non-condensable gas (NCG) inside the reservoir. The NCG expands and contracts depending on operating conditions in the evaporator to occupy condenser surface area which will cause a change in thermal resistance from the evaporator to the condenser. This work showed in two distinct steps how a hot reservoir VCHP was used to provide a passive thermal solution. The first step demonstrated theoretically the feasibility of a full scale VCHP capable to reject the required power of 90W maintaining the device temperature within the allowable range while the heat sink temperature swept the earlier mentioned range. For this VCHP, a finned condenser was designed to reject the heat into the ambient. During the second step a stainless steel/methanol lab hot reservoir VCHP prototype was developed to validate the full scale predictions obtained in the first step. Since the lab prototype used a chiller block attached to the condenser tube to replace the fin structure and the ambient air, the full scale boundary conditions were recalculated to accommodate the lab conditions. The new sink temperature range became -20°C to 15°C while the new target device temperature became 26 to 37°C. The paper also discusses some fabrication and charging details. Overall, the lab VCHP prototype testing validated the full scale modeling predictions through two experimental data points.

Keywords: Variable conductance heat pipe, Heat pipe, Thermal control, Passive thermal solution

1. INTRODUCTION

While continuously increasing in complexity, the payloads of high altitude atmospheric vehicles need thermal management systems to reject their waste heat and to maintain a stable temperature as the air (ultimate heat sink) temperature swings between -85°C and -45°C. It is shown in [1] that constant conductance heat pipes have been utilized on balloon payloads to effectively move the waste heat over significant distances and dissipate the heat into the ambient [1]. The drawback of this thermal solution is that the conductance cannot effectively be reduced under cold operating or cold survival environment conditions, so an active heater requiring significant energy is required to keep the payload warm. As discussed in this paper, a variable conductance heat pipe (VCHP) can be used to increase the thermal resistance as the sink temperature drops, keeping the instrument section warm.

Typical spacecraft VCHPs have the non-condensable gas (NCG) reservoir located at the end of the condenser where its temperature is maintained with electrical heaters. This solution is active.

A VCHP theoretical and experimental study is presented in [1] where the thermal control performance of two VCHP configurations, a cold reservoir VCHP (standard) and a hot reservoir VCHP are compared. For the cold reservoir VCHP configuration, the reservoir is thermally and physically connected to the condenser following its temperature. For the hot reservoir VCHP configuration, the reservoir is thermally and physically connected to the evaporator also following its temperature. Although both solutions are passive, the hot reservoir shows a significantly tighter temperature control, as the ultimate heat sink (ambient air) experiences wide temperature swings.

The proposed solution in this work is similar to the hot reservoir VCHP configuration presented in [1, 2] where the reservoir follows the evaporator temperature, eliminating the need for electrical energy to regulate the heat flow and, consequently, the payload temperature. Again, this thermal management solution is passive.

2. BACKGROUND – VARIABLE CONDUCTANCE HEAT PIPES

A simple VCHP is shown below in Figure 1. It is similar to a conventional heat pipe but has a reservoir and controlled amount of non-condensable gas (NCG) inside the reservoir. When the heat pipe is operating, the NCG is swept toward the condenser end of the heat pipe by the flow of the working fluid vapor. The NCG then blocks the working fluid from reaching a portion of the condenser. The VCHP works by varying the amount of condenser available to the working fluid. As the evaporator temperature increases, the vapor temperature (and pressure) rises, the NCG compresses (Figure 1 top) and more condenser is exposed to the working fluid. This increases the conductivity of the heat pipe and drives the temperature of the evaporator down. Conversely, if the evaporator cools, the vapor pressure drops and the NCG expands (Figure 1 bottom). This reduces the amount of available condenser, decreases the heat pipe conductivity, and maintains the evaporator temperature up [2].

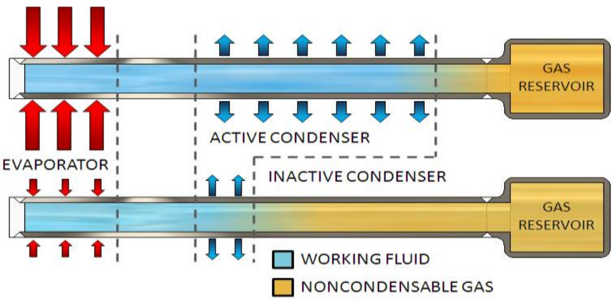

Figure 1. The working of a variable conductance heat pipe (VCHP) is illustrated. At high heat load the temperature dependent saturation pressure of the working fluid is high and compresses the non-condensable gas (NCG) into the reservoir. At lower heat input the working fluid temperature and pressure is lower, and the non-condensable gas expands into the condenser.

For the VCHP shown in Figure 1, the degree of control depends primarily on two factors: the slope of the working fluid vapor pressure curve and the ratio of reservoir and condenser volumes. Working fluids having steeper vapor pressure curves at the particular operating temperature result in tighter temperature control. Small changes in temperature result in large changes in pressure and subsequently large changes in the NCG volume/condenser length. Similarly, large reservoir volumes improve control because a given pressure change results in a larger change in the position of the gas/vapor interface in the condenser.

Schematics of the two VCHP configurations, hot and cold reservoir, are shown in Figure 2. Configuration 1 (hot reservoir VCHP) has the reservoir attached to the evaporator and the reservoir temperature will mainly follow the evaporator (payload) temperature. Configuration 2 (cold reservoir VCHP) has the reservoir attached to the condenser and the reservoir temperature will follow the condenser (or sink) temperature.

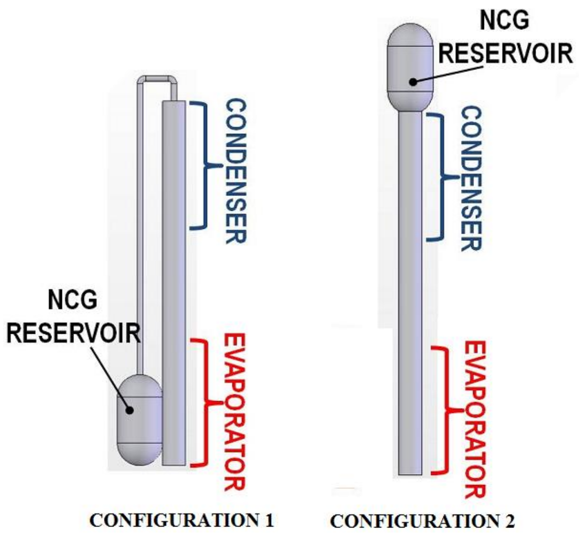

Figure 2. Potential VCHP configurations: Configuration 1 – reservoir attached to the evaporator (hot reservoir); Configuration 2 - reservoir attached to the condenser (cold reservoir).

Configuration 1 was selected as passive thermal control solution for further development that is discussed in this paper.

3. FULL SCALE VCHP MODELING AND DESIGN

Two main parameters are initially set during the process of VCHP modeling and design. First parameter is finding/setting the thermal resistance between the ultimate heat sink and vapor in the condenser such a way that the maximum allowable vapor temperature is reached when the device rejects full power into a heat sink at its highest temperature. This parameter involves the condenser size and its external features. Since, for the discussed application, heat is removed from the condenser by forced convection, these features are fins. Since the drag induced on the vehicle must be minimized, the intrusiveness of the condenser into the environment must be also minimized. The result will be a short condenser. Then second parameter that needs to be set is the NCG reservoir size. This task will be done based on both the condenser size and the fact that when minimum power is rejected into a heat sink at its lowest temperature, the vapor temperature will still be above the minimum allowable value. If survival is one of the operation modes, then minimizing the survival power by placing the NCG front into the adiabatic region of the VCHP, as far as possible from the condenser, is another consideration in setting the reservoir size.

3.1 Finned Condenser Design

The ultimate heat sink for this application is free-stream air. To aid heat rejection, the condenser end of the VCHP is fitted with aluminum fins as seen in Figure 3.

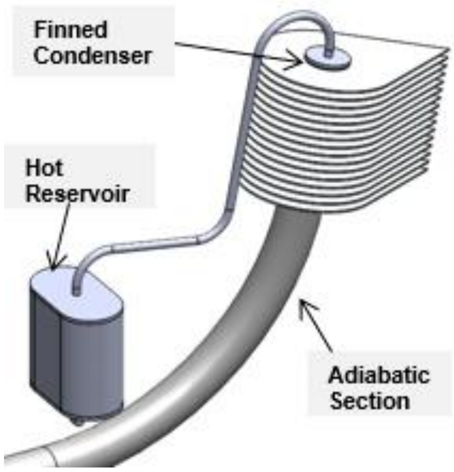

Figure 3. Aerodynamically shaped fins are attached to the condenser

The fin pitch and thickness are optimized to balance fin efficiency and overall weight. This calculation is performed using a parallel resistance network of flow through the heat sink and the bypass air flow. Figure 4. shows the results from this calculation. For this application, the fin stack is made of .81mm (0.032”) thick fins with a 3.34mm (0.131”) pitch. With this design the resultant ΔT is 59°C which is the temperature difference between the ultimate heat sink, the air, and the outer surface of the condenser tube.

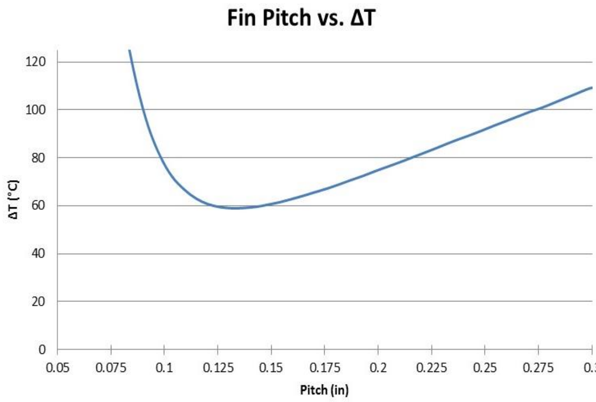

Figure 4. Optimum fin pitch for an annular fin in a free air stream using a .81mm thick fin.

3.2 Thermal Control Modeling Predictions

As mentioned above, once thermal resistance between vapor and ultimate heat sink is set, the reservoir size is the next parameter to be set in defining the thermal control capability of the device.

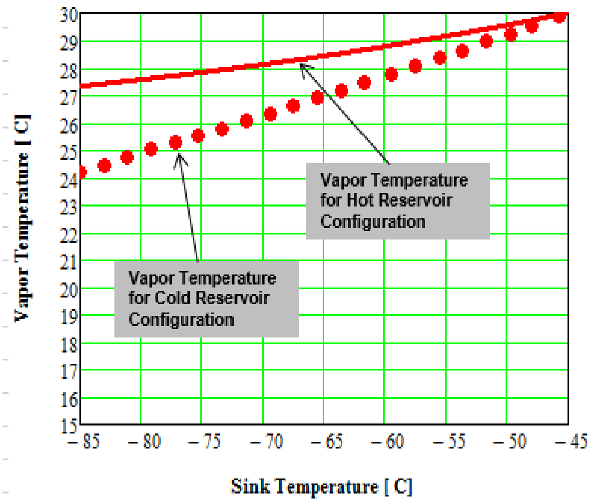

Figure 5. Full scale VCHP evaporator temperature variation as the sink temperature sweeps the entire design range.

A VCHP model based on flat front theory [3] was developed and used to analyze the full scale VCHP solution (hot reservoir) selected for the presented work. The cold reservoir configuration was analyzed as well for reference. The most relevant results are presented above in Figure 5 where VCHP thermal control performance is shown as a function of sink temperature for a 37 cm^3 reservoir volume and a $10cm^3$ condenser volume.

As seen in Figure 5 the sink temperature sweeps the entire range between -85 and -45ºC specified by the design requirements. As expected, Configuration 1 (hot reservoir) provides a better (tighter) temperature control than Configuration 2 (cold reservoir). The vapor temperature change while heat sink experiences all the temperatures within the prescribed range is 2.7°C for the hot reservoir configuration and 6°C for the cold reservoir configuration. In conclusion, despite the choice for the hot reservoir configuration, the cold reservoir configuration also shows suitability for the application.

The final design parameters of the full scale hot reservoir VCHP are shown in Table 1.

Table 1. Full scale VCHP dimensions

Envelope Material	304 Stainless Steel
Working Fluid	Methanol
Non-Condensable Gas	Argon
Heat Pipe Diameter	16mm
Heat Pipe Length	0.630m
Evaporator Length	0.445m
Condenser Length	57mm
Condenser Volume	$10cm^3$
Reservoir Volume	$37cm^3$
Connecting Tube Diameter	3.2mm
Connecting Tube Length	0.4m
Tube wall thickness	0.5mm
Number of Screen Wraps	4
Total Mass	190g (no fins)

4. PROTOTYPING AND TESTING

To demonstrate the validity of the design, prototype units were designed and fabricated. Stainless steel and methanol were selected as the envelope and working fluid material pairing. The condenser end was vertically oriented with gravity assistance to return the fluid back to the evaporator. The evaporator was horizontal with a single wrap of mesh screen for promoting vaporization around the full circumference of the pipe. The pipe dimensions are listed below in Table 2. For testing, a liquid nitrogen chiller block replaced both the air cooled heat sink and the aluminum fins to obtain better control of testing at various condenser temperatures. Because of these changes on the heat rejection side, the boundary conditions applied to the full scale design were replaced by new values to accommodate the new thermal resistance between the vapor and the ultimate heat sink that in the lab was liquid nitrogen. As lab boundary conditions, the heat sink temperature changed between -25 and 15ºC while the evaporator temperature was allowed to change between 26 and 37ºC.

Table 2. Lab prototype VCHP dimensions

Envelope Material	304 Stainless Steel
Working Fluid	Methanol
Non-Condensable Gas	Argon
Heat Pipe/Reservoir Diameter	17.5mm
Heat Pipe Length	0.627m
Evaporator Length	0.445m
Condenser Length	50.8mm
Reservoir Length	177.8mm
Connecting Tube Diameter	3.2mm
Connecting Tube Length	0.355m

The lab prototype modeling was carried based on the specific boundary conditions that are presented above. The results are presented later in the paper, in Figure 9, where they are compared to the experimental results.

4.1 Fabrication

The heat pipe envelope was constructed from thin-wall 304 stainless steel tubing with the condenser length bent at a 101° angle from the evaporator centerline. A single wrap of mesh screen was attached along the inner diameter of the evaporator and end caps were welded at each end. End caps were welded onto the reservoir section with a fill tube and bellows valve for charging the pipe with methanol and argon. The connecting tube was bent and welded in place to join the reservoir and heat pipe envelope.

After dimensional inspection, the empty assembly underwent a vacuum bake to eliminate contaminates prior to fluid filling. Once a nominal mass of methanol was charging in the pipe, the heat pipe was tested in the processing fixture to vent any remaining non-condensable gas and adjust the fluid charge. The heat pipe was put under load to observe baseline performance at design power.

4.2 Testing Preparations

A fixture was developed for charging the assembly with methanol to make a functional heat pipe, for charging with argon for thermal control, and for testing the performance in cold and hot ambient conditions. This assembly is shown in

Figure **6**. The test and charging stand consisted of two aluminum heater blocks: one for the evaporator region to apply fixed power into the pipe, and another for the reservoir that could hold the envelope temperature to match the evaporator temperature to prevent pooling of methanol outside the heat pipe region. A flexible strip heater was wrapped around the connecting tube between the heat pipe envelope and reservoir for the same purpose. Each evaporator block used electrical cartridge heaters for power input.

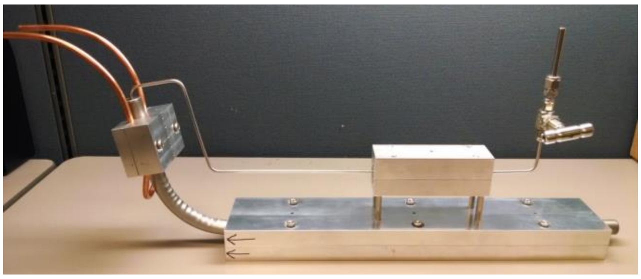

Figure 6. Hot reservoir VCHP in heater block and condenser block assembly

In the real-world application, an array of epoxied aluminum fins on the condenser length would be used for heat rejection to ambient. However due to the need to easily ramp between a range of sink temperatures to demonstrate variable conductance, the fins were replaced with a liquid nitrogen-cooled cold plate. Liquid nitrogen flowed through the copper tubing embedded in an aluminum block at a controlled rate to simulate the cold and hot ambient temperature. The surface temperature of the cold plate was set to match the ambient air temperature plus the temperature drop from the resistance of the fin stack at design power.

Type T thermocouples were fixed to the reservoir and evaporator regions. A series of thermocouples was incrementally placed on the condenser for tracking the gas front based on the read-out of adjacent points and identifying a steep temperature gradient, as shown in

Figure **7**. The valve was left on the fill tube for quickly adjusting the charge of methanol and argon during and between tests for optimal performance.

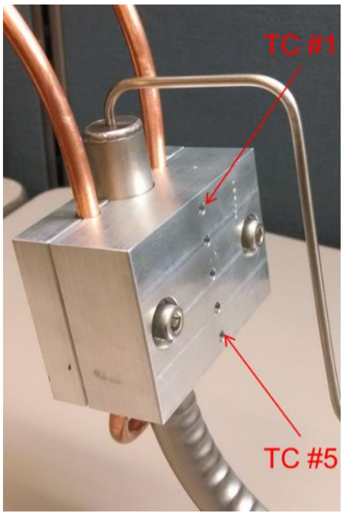

Figure 7. Condenser end thermocouple layout

4.3 VCHP Charging

After the prototype with a methanol-only charge was validated as a functioning heat pipe, ACT made preparations for inputting a discrete amount of argon for controlling the evaporator temperature within the desired range. A large capacity tank with a pressure gauge was placed in between an argon cylinder and the heat pipe fill tube and valve. A vacuum pump evacuated the tank and the plumbing that connected all components. After drawing a vacuum, the tank was filled with argon to the pressure that corresponded to methanol's saturation pressure at the desired operating temperature of 27°C. The heat pipe processing assembly was configured to design power at the minimum ambient condition of -25°C. The system was allowed to start-up before introducing the argon into the heat pipe vapor space.

4.4 VCHP Demonstration Test

After a brief period of operation at the minimum ambient condition, the fill tube valve was opened to allow argon to enter the heat pipe. The plot shown in Figure 8 details the time history of the initial argon charge and cold/hot ambient cycle. At time 2200s when the valve is opened, the condenser region thermocouples sharply decrease to meet the cold block temperature. This is evidence that the argon has swept into the condenser and pushed back the methanol vapor, which has weaker vapor pressure coming from a cold evaporator.

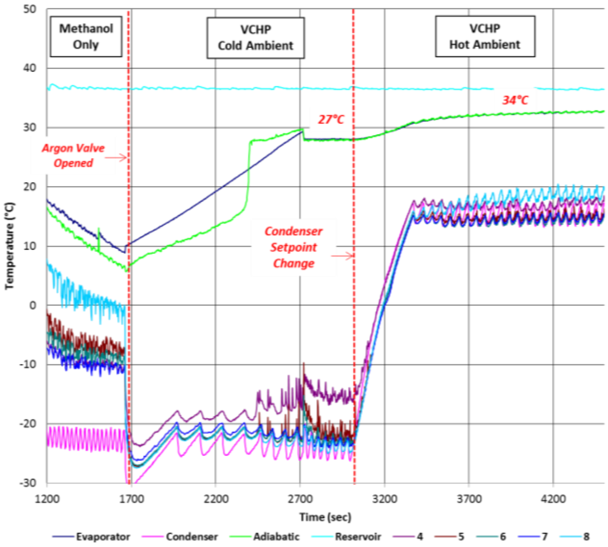

Figure 8. VCHP charging and demonstration from a functioning methanol heat pipe. Vapor temperature is investigated for two condenser set points.

Over the next several minutes, the thermal mass of the evaporator and heater block rose in temperature from -5°C to before reaching an abrupt plateau at 27°C, which was dictated by the pressure of the argon tank. At this stage, the valve leading to the argon tank is closed and the system continues to operate at its new steady state value.

Slightly past 5000s, the condenser setpoint is manually increased to 15°C to simulate the hot ambient condition. Once again, the system undergoes a brief transient to a new steady state value. However, rather than increasing by the same 40°C margin as the condenser, the evaporator increases only 7 degrees before settling at 34°C.

The noticeable difference between the cold and hot ambient tests shown in Figure 8 is the change in temperature difference across the evaporator and condenser. During the cold ambient test, the delta is 52°C compared to 19°C in the hot ambient test with no change in power input. This is another demonstration of the variable conductance characteristic of the heat pipe. When the condenser set point is increased, the hot methanol vapor coming from the evaporator is able to sweep the argon back into the reservoir to open up more condenser surface area to dump heat, thus lowering its resistance relative to the cold ambient demonstration.

4.5 Testing Results Comparison with the Modeling Predictions

A comparison between the experimental data and the corresponding modeling predictions performed for the testing prototype geometry and boundary conditions is shown below in Figure 9. As seen only two experimental data points were obtained that correspond to steady states of the testing VCHP for two sink set points, -24°C and 15°C respectively. For the first sink temperature (-24°C) the vapor temperature was 27°C while for the second sink temperature (15°C) the vapor temperature was 34°C. Theoretical vapor temperature predictions that correspond to the same sink temperatures were 30°C for the cold sink data point and 33.6°C for the hot sink data point as shown in Figure 9.

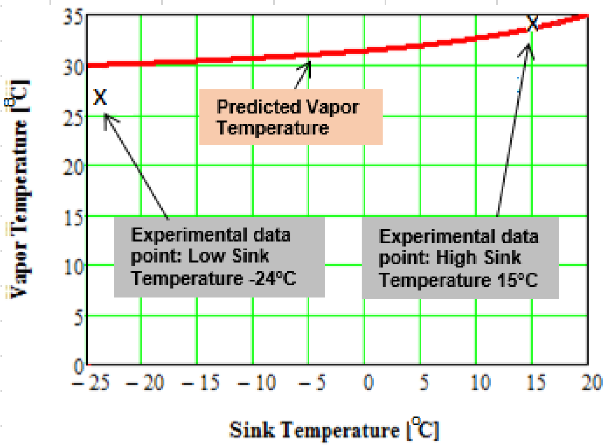

Figure 9. Comparison between the modeling predictions performed for the testing prototype geometry and boundary conditions and actual testing results

5. CONCLUSIONS

A hot reservoir VCHP was selected for an aerospace application where the heat sink is high altitude ambient air. A full scale design was performed that demonstrated full feasibility of the chosen thermal management solution. To validate the model used to design the full scale device, a lab hot reservoir VCHP prototype was designed, fabricated, and tested. During a single continuous test, ACT demonstrated a working methanol heat pipe, initial charging of argon to achieve the VCHP thermal control function, and a narrow window of evaporator temperatures after cycling through a much wider range of heat sink temperatures. The lab prototype testing results validated the VCHP model used to design the full scale hot reservoir VCHP.

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REFERENCES

[1] Tarau, C., Anderson, W. G.,"Variable Conductance Thermal Management System for Balloon Payload, 20th AIAA Lighter-Than-Air Systems Technology Conference, Daytona Beach, FL, March 2013

[2] Anderson, W.G. "Variable Conductance Heat Pipes for Variable Thermal Links," 16th International Heat Pipe Conference, Lyon, France, May 20-24, 2012

[3] Marcus. B.D., "Theory and Design of Variable Conductance Heat Pipes," NASA Report NASA- CR-2018, April, 1972. Available from the NASA Technical Reports Server, http://ntrs.nasa.gov/.