Pressure Controlled Heat Pipe for Precise Temperature Control

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Abstract. This paper discusses the design and test of a pressure controlled heat pipe (PCHP) for spacecraft thermal management. The PCHP combines a conventional grooved aluminum-ammonia heat pipe with a variable-volume non-condensable gas reservoir to create a heat pipe whose conductance can be precisely controlled. Testing showed that a prototype PCHP was capable of maintaining a stable evaporator temperature within 0.1K despite wide swings in heat load and heat sink temperature. A similarly-sized variable-conductance heat pipe (VCHP) yielded temperature swings of over 3.5 K for the same variation of heat load and sink temperature. Using a non-optimized control system, the PCHP was capable of maintaining evaporator temperature within 0.05K over time. The PCHP had a much faster transient response than other devices such as heated-reservoir VCHPs, as well as providing a means for changing the set point temperature after assembly. The PCHP is a significant advance over other means of temperature control, even in its current non-optimized state.

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INTRODUCTION

Satellites are increasingly requiring better temperature control. Optical systems, lasers, and detectors frequently require temperature control to within ±1K or less as well as rejection of a significant amount of power. Some new missions even desire temperature control and thermal gradient control to the milli-Kelvin level. Heat pipes have long been used on spacecraft to move heat and route it to a radiator where it can be rejected (Wyatt, 1963). The major problem with conventional heat pipes is their fixed thermal conductance. Any change in the system, such as a change in heat load or an orbit-induced change in radiator sink temperature, will result in a change in the operating temperature of the device being cooled.

Some early means of temperature control include louvers, cold biasing, and variable conductance heat pipes. Louvers are thermostatically-driven surfaces which change the effective area of a radiator. While they do consume no extra power and can moderate temperature swings, they are not precision devices that can hold temperature within a few degrees K. Actuation differentials range from 10 °C and 18 °C (Gilmore, 2002). For example, the Pegasus satellite utilized louvers for thermal control of batteries and electronics and could maintain their temperature within 10 K of the desired set point. (Linton, 1966). Cold-biasing involves providing extra cooling capacity to assure that the device always runs below its operating temperature, then adding extra heat as needed to warm the device to its required operating temperature. This can provide temperature control within 0.1 K or better, (Aaron et al., 2002) but at the expense of additional radiator area, additional power consumption for the heaters, and additional complexity in the form of redundant thermostats and heaters to prevent overcooling due to a failure of the control system (Gasbarre et al., 2007). A Variable Conductance Heat Pipe (VCHP) is a conventional heat pipe with an added reservoir that contains a controlled amount of non-condensable gas. The gas volume changes in response to temperature and varies the conductance of the heat rejection region. The degree of control varies with the ratio of condenser to reservoir volume, however VCHPs having a reasonable reservoir volume can typically regulate temperature within ±2 K with no additional power consumption. (Marcus, 1971). The Orbiting Astronomical

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Observatory satellite was the first known use of a VCHP on orbit. (Mock, Marcus, and Adelman, 1974). That VCHP maintained an electronics box within ±2K while power dissipation changed from 15 to 35 Watts. Beinert (1971) demonstrated a VCHP with a heated reservoir and electronic feedback. The steady-state input temperature varied less than 2 K while the heat load varied from 5 Watts to 35 Watts. The transient response time was about 20 minutes.

Pressure Controlled Heat Pipes (PCHP) are an improvement on a VCHP. They provide temperature control by actively varying either the amount of control gas or the volume of the gas reservoir. The PCHP provides the high thermal conductivity of a heat pipe with the tight temperature control of cold biasing. PCHPs have fast transient response, consume little power, and are only slightly more complex than many other means of temperature control. This paper describes the construction and test of a developmental PCHP. The goal was to demonstrate a PCHP suitable for thermal management of a typical small satellite instrument of moderate but varying power, varying sink temperature, and with very tight temperature control requirements.

PRESSURE CONTROLLED HEAT PIPE TECHNOLOGY

Conventional heat pipes transport heat by two-phase flow of a working fluid. Shown in Figure 1 a heat pipe is a vacuum tight device consisting of a working fluid and a wick structure. The heat input vaporizes the liquid working fluid inside the wick in the evaporator section. The vapor, carrying the latent heat of vaporization, flows towards the cooler condenser section. In the condenser, the vapor condenses and gives up its latent heat. The condensed liquid returns to the evaporator through the wick structure by capillary action. The phase change processes and two-phase flow circulation continue as long as the temperature gradients between the evaporator and condenser are maintained.

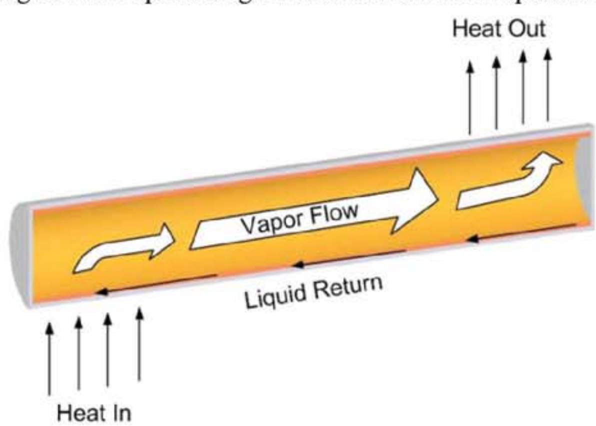

FIGURE 1. Cross Section Of Typical Heat Pipe Showing Vapor And Liquid Flow And Heat Input And Rejection Areas.

A simple Variable Conductance Heat Pipe, or VCHP, is shown in Figure 2. It is similar to a conventional heat pipe but has a reservoir and controlled amount of non-condensable gas (NCG). When the heat pipe is operating, the gas 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 (Fig 2a, 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 (Fig 2a, bottom). This reduces the amount of available condenser, decreases the heat pipe conductivity, and drives the evaporator temperature up. The typical reservoir volume is about ten times larger than the swept volume of the condenser, which results in about a ±2K control band. Precision temperature control is possible if very large reservoir volumes are used. For temperature control to the milliKelvin level the reservoir would have to be about 2000 times larger than the swept volume of the condenser. This would be mass and volume prohibitive for a spacecraft.

Pressure controlled heat pipes (PCHPs) are similar to VCHPs. The vapor/non-condensable gas (NCG) interface position in the condenser moves to vary the conductance of the heat pipe. The control mechanism in a PCHP, however, is active. This provides much closer control of conductance and can allow changing of the set point temperature after assembly of the heat pipe. Two different control mechanisms are possible: varying the number of moles of gas in the system or varying the volume of the reservoir. In practice, control is obtained by actively injecting or removing NCG or by varying the volume of the reservoir by contracting or expanding a bellows. PCHPs had been built in the past for precise control of annular heat pipes operating in the region of 1300K. Using a gas injection mode of control, they were able to stabilize the operating temperature against changes of input power or heat loss to within 0.001K for several hours (Bienert, 1991).

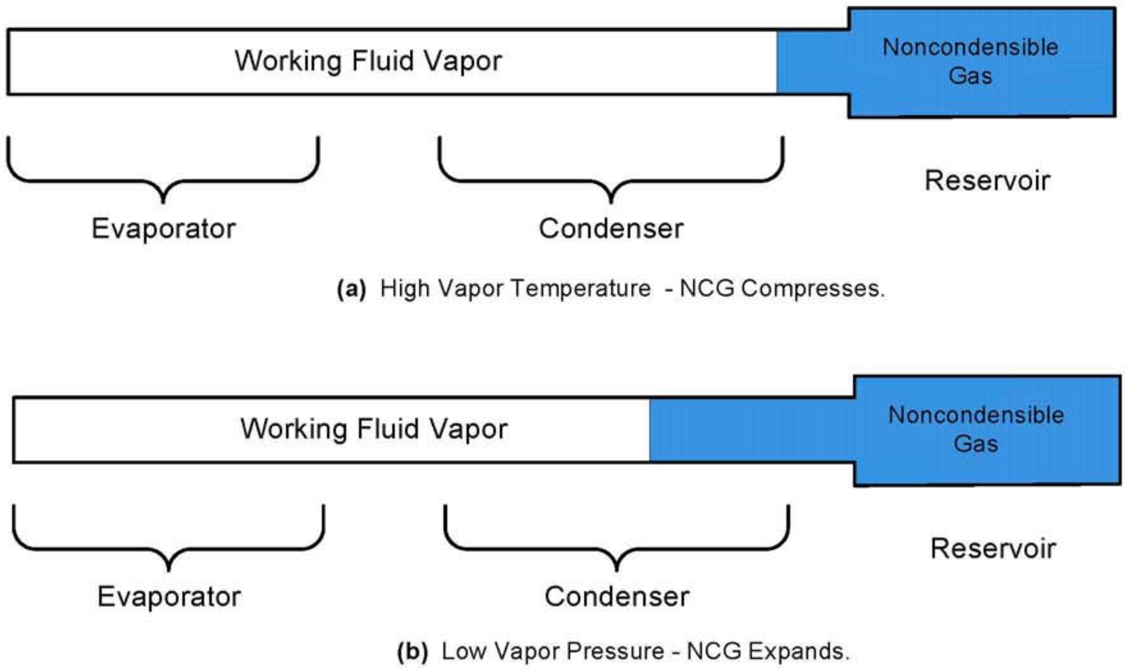

FIGURE 2: Conventional VCHP Operation.

TEST HARDWARE AND HARDWARE DESIGN

The demonstration test hardware was designed to accommodate a hypothetical but typical small satellite electronics box with a variable heat load, tight operating temperature tolerance, and wide heat sink temperature swing. The requirements are summarized in Table 1 below. The heat pipe envelope was based on a standard grooved aluminum extrusion with an integral flange. The flange was machined away as required to form the discrete evaporator and condenser sections. Ammonia was selected as the working fluid because it was compatible with the aluminum envelope and had good performance over the required temperature range.

TABLE 1. Heat Pipe Design Requirements and Performance Goals.

Parameter	Value
Power (W)	50-150
Set point temperature (K)	300
Evaporator Size (cm)	3.8 x 15.2
Condenser Size (cm)	3.8 x 30.5
Overall Length (cm)	100
Sink Temperature (K)	223 - 283
Temperature Stability (K)	0.001

The demonstration PCHP used the variable reservoir volume mode of control, where the mass of the NCG remains constant and the volume of the reservoir is varied to adjust the fraction of the NCG that resides in the heat pipe condenser. The required reservoir volume was computed using Equation 1. The volume of the NCG is a function of temperature and operating pressure. For a given heat load and the known condenser thermal resistance per unit length, some portion of that NCG volume will reside in the condenser. The remainder must be contained by the reservoir. Using equation 1, the necessary reservoir volume and NCG mass were found by iteratively increasing the amount of NCG from a low value and testing for zero or larger reservoir volume at all four bounds of operating temperature and heat load requirements.

Equation 1 makes two simplifying assumptions. It assumes that there was no diffusion between the NCG and the working fluid (the so-called flat-front model), and it ignored axial thermal conductivity along the walls of the heat pipe envelope. Both effects tend to increase the amount of heat leakage between the active and the inactive portions of the condenser. As a result, this first-order model slightly under-estimates the motion required of the gas front when large changes in conductivity are required. It is good for scoping calculations such as rough heat pipe sizing, and is accurate when small displacements were involved or for calculating the sensitivity of heat pipe conductivity to changes in gas front position.

$$V_{res} = \frac{nRT_g}{P_v} + L_c A_{HP} \left(\frac{QR_c}{T_v - T_c} - 1 \right). \tag{1}$$

The intent was to use two different diameters of flexible bellows as reservoirs, with the large bellows used for coarse changes of set point temperature and the small bellows used for fine temperature regulation. Time constraints mandated the use of O-ring sealed pistons that ran in precision-honed cylinders. The diameter of the small piston was found from the known thermal resistance per unit length of the evaporator and condenser and the required temperature resolution. The temperature drop of the heat pipe is

$$\Delta T = Q\left(R_e \frac{1}{L_e} + R_c \frac{1}{L_c} \right). \tag{2}$$

That equation can be differentiated on condenser length to find the change in condenser length required to produce a given change in temperature drop

$$\frac{d}{dL_c} \Delta T = QR_c \frac{1}{L_c^2}. \tag{3}$$

Based on a heat load of 50 Watts and a measured condenser resistance of 0.238 C/watt-in for the selected extrusion, the resulting sensitivity of the gas front position to temperature was 0.33 C/inch or 0.003” per milliKelvin. That resolution is easily achievable by standard components such as a stepper motor with a lead screw drive. For example, a stepper motor with a 1.8° step size (200 steps per revolution) and a 40 TPI lead screw (0.025” per revolution) are readily available commercial components. The resolution of that system would be 0.025”/200 steps or 0.000125”/ step, or 30 times better than the required resolution.

TABLE 2. Design Parameters for Piston-Cylinder Assemblies.

Parameter	**Large Piston**	**Small Piston**
Bore (cm)	3.175	1.422
Piston Area (cm^2)	7.92	1.588
Stroke (cm)	3.175	6.350
Volume Displacement (cm^3)	25.1	10.1

A block diagram of the control system is shown in Figure 3. Major components include a Eurotherm model 2404 process controller, a digitizer, and an Applied Motion STAC-6 stepper/servo drive controller. A Minco S667PD 100-ohm thin-film platinum RTD was mounted to the adiabatic section of the heat pipe approximately 5 cm from the end of the evaporator section. The temperature that it sensed was compared to the set point temperature by the Eurotherm controller and converted to an analog output voltage proportional to the magnitude of the error. That signal was digitized and converted to a command to the servo drive which then moved the piston to a position proportional to the error signal.

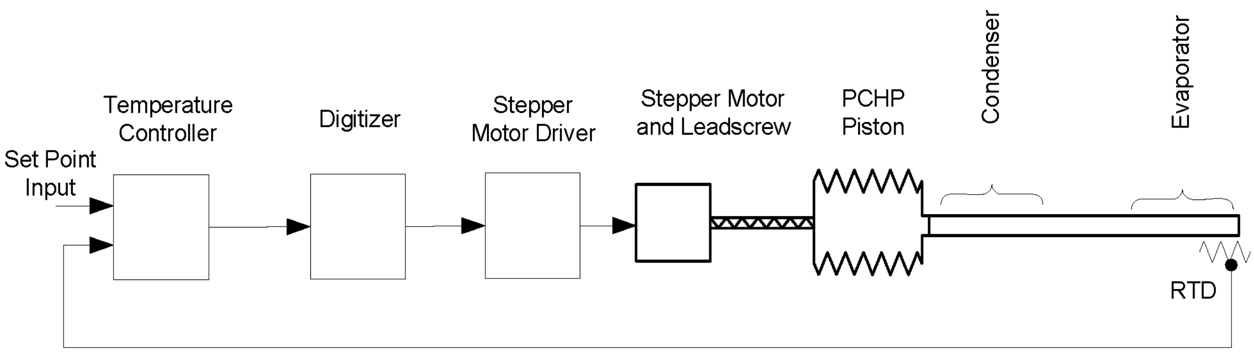

FIGURE 3. Block Diagram of the Control System.

Digitizing the signal from the process controller resulted in some uncertainty in the heat pipe operating temperature. The digitizer had a total range of 4096 counts for an input of ±5 volts. The actual output of the controller was limited to between 0.8 and 5.0 volts, which was 42% of the total range or 1720 counts. The piston, therefore, could assume any of 1720 discrete positions. Given a measured temperature difference of between 3°C and 5°C for a full piston stroke depending on the operating condition, the temperature resolution of the system was 5°C/1720 counts or about 3 milliKelvins per step. The measured uncertainty of the digitizer was ±3 counts, which resulted in a temperature uncertainty of ±9 milliKelvins. Since the stepper had a resolution of 625,000 counts per full piston stroke, the digitizer was the limiting factor in system resolution. This could easily be improved by adding an amplifier to better scale the analog signal or by purchasing a digitizer with more resolution.

The heat pipe evaporator and condenser were mounted to aluminum blocks for heating and cooling. The heater block contained a pair of cartridge heaters whose input power was regulated by a variac and monitored by a Wattmeter. The condenser block was cooled by liquid nitrogen (LN). A temperature controller and solenoid valve connected to the LN supply maintained the condenser block at a known and constant temperature. Heat pipe temperatures were measured by an array of thermocouples mounted to the top of the extrusion, out of the heat flow path. The array was read by a computer-based data acquisition system at least once every 10 seconds. An overall view of the test hardware just prior to adding insulation is shown in Figure 4. All major heat pipe and control system components are visible, as well as the array of thermocouples mounted to the top of the condenser with strips of Kapton tape.

There are two sources of uncertainty in the temperature measurements. The data acquisition system used to take most temperature measurements has an uncertainty of ±0.23 °C. The wire used to form the thermocouples has an uncertainty of ±0.5 °C. The accuracy of the measurement system, including both the thermocouple and the data acquisition system, is the RMS sum of the two uncertainties, or ±0.24 °C. That value includes both random and systemic errors. The random error is small. Analysis of the collected data showed that for a constant temperature the standard deviation is typically 0.08 °C for ten or more measurements.

TEST METHOD AND TEST RESULTS

Testing began with measurement of the heat pipe transient response and sensitivity of the temperature drop to the position of the small piston. The small piston was withdrawn to its outermost position and the heat pipe was allowed to reach thermal equilibrium. The piston was then quickly advanced to its innermost position and the heat

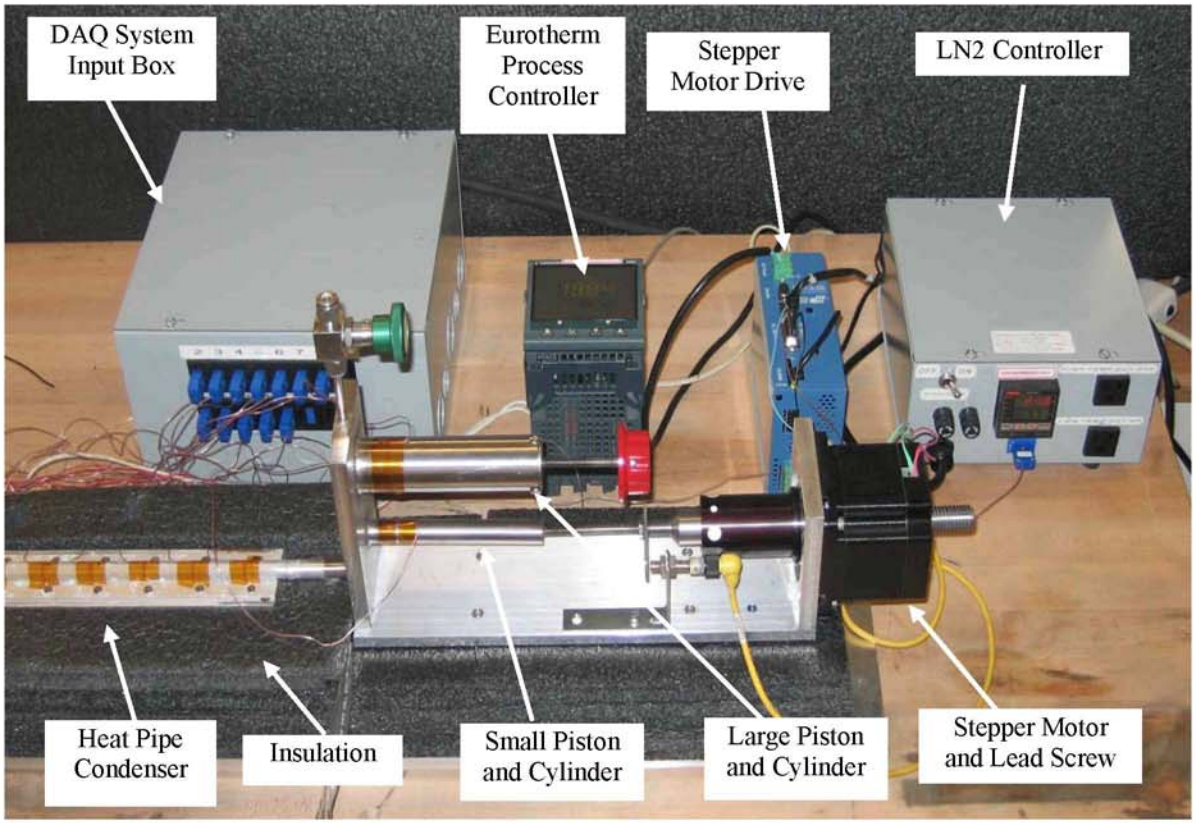

FIGURE 4. Photograph of Condenser End of Heat Pipe Showing All Major Components.

pipe was again allowed to reach equilibrium. The resulting evaporator temperature change was 3°C, which met the design predictions. The measured evaporator time constant was about 12 seconds. While the evaporator and its heat source reached equilibrium quickly, the remainder of the heat pipe took about 10 minutes to reach a new steady state temperature distribution.

The next test measured the ability of the PCHP to maintain a constant evaporator temperature over changes of input power. The sink or condenser block temperature was held constant at -50 °C while the input power was varied from 50 to 250 Watts in increments of 50 Watts. These power limits corresponded to the limit of travel of the small piston. The heat pipe was allowed to reach steady state at each power increment. The test was repeated with the control system inactive (open loop mode) to show the performance of a representative VCHP. The results are shown in Figure 5. While in open loop mode the temperature varied 3.4 degrees. In closed loop mode the temperature varied less than 0.1 °C.

The ability of the PCHP to maintain a constant evaporator temperature over varying sink temperatures was measured next. The input power was held constant at 50 Watts. The condenser block temperature was varied from -70 °C to -40 °C in increments of 10 °C. Those temperature limits corresponded to the freezing point of the working fluid and the limit of adjustability of the small piston. This test was repeated in open loop and closed loop mode. Figure 6 shows the results of this test. As shown in Figure 6, the evaporator temperature changed less than 0.1 °C while in closed loop mode but varied by 4.5 °C when in open loop mode.

A final test measured the time stability of the PCHP. Two changes were made to the test hardware. The control system had an extra integrator in the control loop, and the resolution of the data acquisition system resolution was increased to 0.01 K. The input power was held constant at 25 watts. The major disturbance in the system was a periodic fluctuation of about 0.5 °C in the condenser block temperature which was due to the 10 second duty cycle of the LN2 controller. In open loop mode, this fluctuation was reflected to the evaporator end of the heat pipe and resulted in an evaporator temperature fluctuation of 0.5 °C. Results from testing in closed-loop mode are shown in

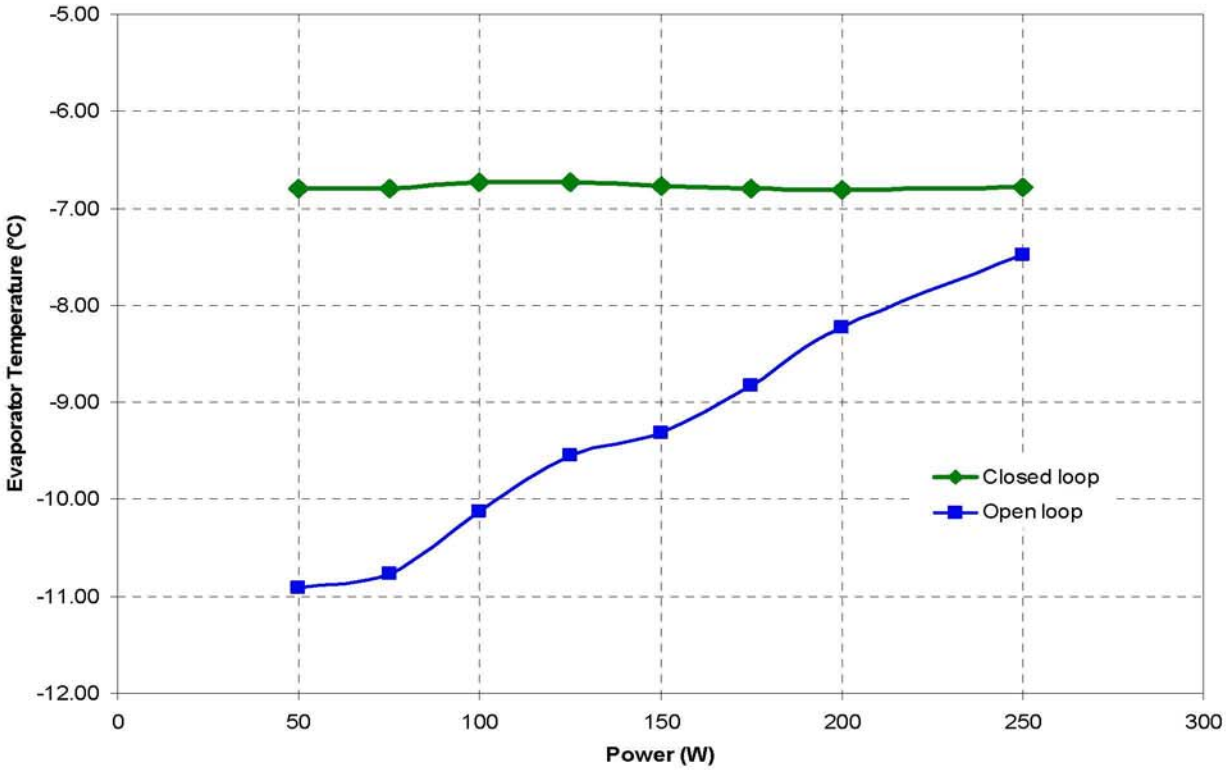

FIGURE 5. Input Power vs Evaporator Temperature Showing Good Temperature Stability in Closed-Loop Mode.

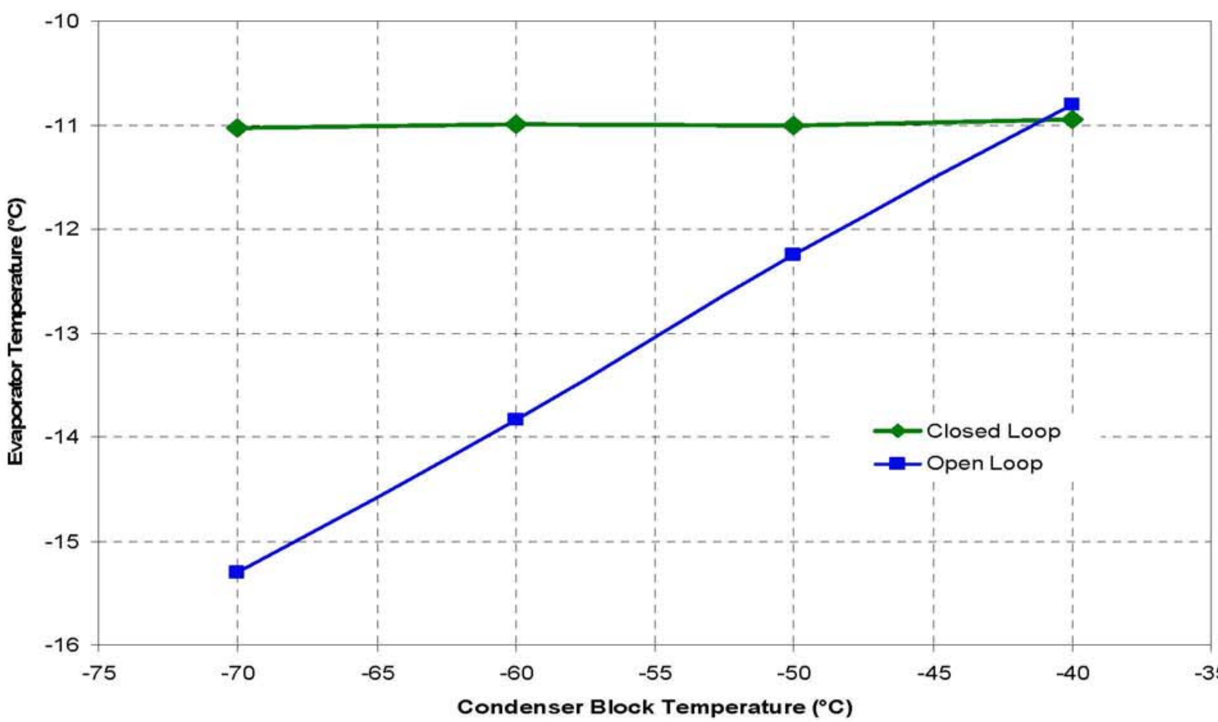

FIGURE 6. Condenser Block Temperature Vs Evaporator Temperature Showing Excellent Temperature Stability In Closed-Loop Mode.

Figure 7. The temperature varied less than 0.05 °C over the duration of this test. At least half of that variation was due to the control system compensating for manual changes of the position of the large piston made at 50 and 70 seconds. While the plot shows only a short amount of data, the test ran for over 10 minutes with similar tight temperature control.

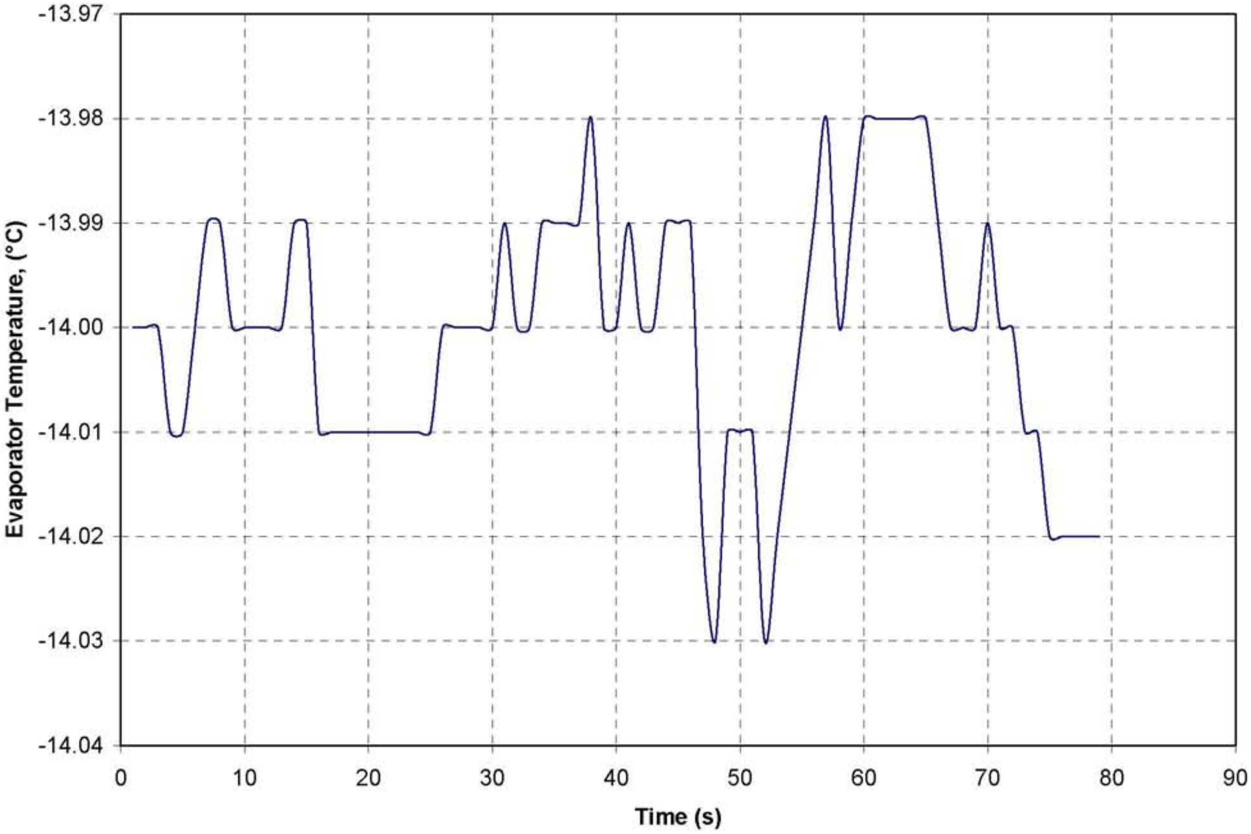

FIGURE 7. Time Stability of PCHP with 25 Watts Input and Modified Control System Showing Less Than 0.05K Variation of Evaporator Temperature Over Time.

The power consumption of the stepper motor and drive were measured. The drive itself drew about 6 Watts even with the stepper motor inactive. The motor itself can draw up to 32 Watts if fully loaded and operating continuously. Based on the duty cycle and mechanical loading it encountered, the motor drew an average power of 2 Watts. The combined power consumption of the stepper and drive system was therefore 8 Watts, which was slightly lower than the power consumption of the heated-reservoir VCHP described by Bienert and Brennan (1971).

CONCLUSIONS

A PCHP with variable reservoir volume control was successfully demonstrated. It met, or approached, all of its design goals, and it is a significant advance over other means of temperature control even in its current non-optimized state. It was able to maintain its evaporator temperature within 0.1 °C over changes in heat sink temperature from -70 °C to -40 °C and changes of input power from 50 Watts to 250 Watts. In contrast, operating as a conventional VCHP resulted in a 3.5 °C variance for the same change of input conditions. The PCHP was able to maintain the evaporator temperature within 50 milliKelvins over time, which was nearly at the limit of resolution of the current feedback control system. The transient response time of the PCHP evaporator was 30 seconds. The response time of a heated-reservoir VCHP having similar power consumption was 20 minutes. The PCHP provides the same tight temperature control as cold biasing, but without the mass penalty of larger radiators and the power consumption of extra trim heaters. Although its use was not discussed in this paper, the large piston allows the set point temperature of the PCHP to be adjusted after charging the heat pipe. This capability does not exist with a standard VCHP. The major limitation with the current PCHP technology is in the control system and its components. Based on the error analysis discussed previously and a frequency-domain analysis of the control system, the necessary reduction in steady-state error and system stability are readily attainable with existing technology and components.

NOMENCLATURE

A_{HP} = Cross Sectional Area of the Condenser Vapor Core (m^2)
L_c = Condenser Length (m)
L_e = Evaporator Length (m)
n = Number of Moles of NCG
P_v = Vapor Pressure (N/m^2)
Q = Power (W)
R = Ideal gas constant
R_c = Condenser Thermal Resistance (K/W-m)
R_e = Evaporator Thermal Resistance (K/W-m)
T_c = Condenser Temperature (K)
T_g = NCG Temperature (K)
T_c = Sink Temperature (K)
T_v = Vapor Temperature (K)
Vres = Reservoir Volume (m^3)
ΔT = Temperature Drop (K)

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