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NITROGEN HEAT PIPE FOR CRYOCOOLER THERMAL SHUNT

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ABSTRACT

A nitrogen heat pipe was designed, built and tested for the purpose of providing a thermal shunt between the two stages of a Gifford-McMahan (GM) cryocooler during cooldown. The nitrogen heat pipe has an operating temperature range between 63 and 123 K. While the heat pipe is in this temperature range during the system cooldown, it acts as a thermal shunt between the first and second stage of the cryocooler. The heat pipe increases the heat transfer to the first stage of the cryocooler, thereby reducing the cooldown time of the system. When the heat pipe temperature drops below the triple point, the nitrogen working fluid freezes, effectively stopping the heat pipe operation. A small heat leak between cryocooler stages remains because of axial conduction along the heat pipe wall. As long as the heat pipe remains below 63 K, the heat pipe remains inactive. Heat pipe performance limits were measured and the optimum fluid charge was determined.

INTRODUCTION

Cooldown times for components attached to the second stage of two-stage GM cryocoolers are significantly influenced by the thermal capacitance of the components and by the parasitic heat loads on the cryocooler. For systems containing superconducting magnets with large thermal capacitance, the cooldown times can be considerable. These components are attached thermally and mechanically to the second stage of the cryocooler because, ultimately, this is where the lowest temperatures are attained. However, because the cryocooler's second stage has a significantly lower refrigeration capacity than its first stage, cooldown times can be reduced by temporarily shifting part of the heat load to the cryocooler's first stage where capacity is greater. A gravity-assisted, nitrogen heat pipe is well suited for this task.

The heat pipe is installed between the two stages of the cryocooler with the evaporator attached to the second stage and the condenser attached to the first stage. When the heat pipe is between about 123 and 63 K during cooldown, it acts as a thermal shunt connecting the first and second stages together. The heat removed from the load, which is mechanically attached to the second stage, is then shared by the first and second stages of the cryocooler. As the second stage temperature continues to decrease below 63 K, the triple point of nitrogen, the heat pipe becomes inoperable as the working fluid freezes, and the two cryocooler stages become thermally disconnected. There remains a parasitic heat leak along the heat pipe wall which can be made small by proper selection of the heat pipe material and geometry. At temperatures above the critical point (123 K), the heat pipe is also inoperable because of the absence of a two phase interface within the heat pipe. The freezing and thawing process is reversible and does not degrade the operation of the heat pipe.

DESIGN

The heat pipe is gravity-assisted, whereby the condenser is assumed to always be above the evaporator, and should be sized to handle the maximum first stage capacity of the cryocooler. In this way the heat pipe cannot be cooled below the triple point by the cryocooler during cooldown. Only when the second stage approaches the nitrogen triple point will the heat pipe begin to freeze and "shutoff". The simplest and most economical configuration for this heat pipe is a smooth, thin-wall, stainless steel tube joining copper evaporator and condenser sections. The stainless steel tube minimizes conduction heat leak between the cryocooler stages when the heat pipe is in the off mode and provides adequate strength to contain the heat pipe internal pressure at 300 K. The latter requirement results from the pressure increase between the maximum operating temperature of 123 K and the 300 K ambient temperature. The copper condenser and evaporator sections minimize thermal gradients at the heat transfer surfaces.

For a smooth-wall, gravity-assisted heat pipe, the capacity is primarily determined by the heat pipe cross-sectional area[1]. Figure 1 shows the calculated heat pipe capacity for different tube sizes as a function of the heat pipe temperature at the evaporator exit. Under most operating conditions the heat pipe vapor temperature is nearly isothermal and specifying a particular location for the heat pipe temperature is unnecessary. The 9.5 mm (3/8") OD x 0.7 mm (0.028") wall tube was selected to insure adequate capacity for the heat pipe. The maximum capacity of the cryocooler first stage in the temperature range 63 to 123 K was estimated at 80 W.

The heat pipe tube wall thickness was determined by the heat pipe internal pressure and the tube wall stress[2]. Figure 2 shows the allowable internal pressure for each tube OD considered as a function of wall thickness. An allowable tensile stress in the tube wall of 68.9 MPa (10,000 psi) was assumed with a factor of safety of 2.0. Also shown for reference is the heat pipe internal pressure at 300 K, which is the maximum pressure anticipated. From the figure the minimum wall thickness that does not exceed one half of the allowable stress for a 9.5 mm (3/8") OD tube is 0.7 mm (0.028"). This minimum value was selected to minimize axial conduction along the tube. The calculated axial conduction as a function of tube size at steady state operating temperatures (4.2 to 80 K) is shown in figure 3. For the 9.5 mm (3/8") OD x 0.7 mm (0.028") wall tube selected, the axial heat leak is approximately 50 mW.

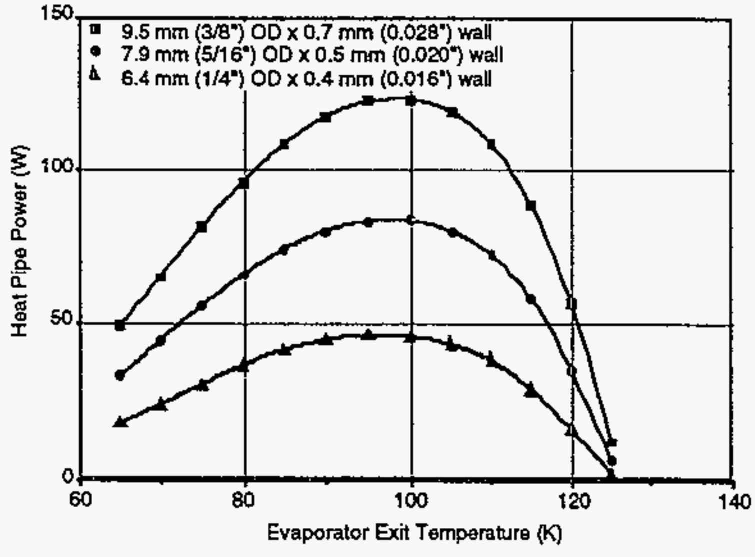

Figure 1. Calculated heat pipe performance as a function of operating temperature and tube size.

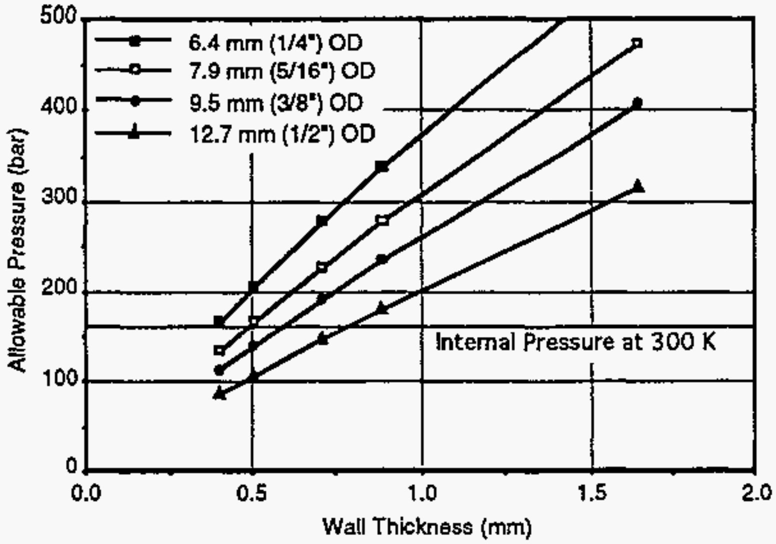

Figure 2. Allowable pressure for heat pipe tube wall as a function of OD and wall thickness. N_2 temperature is 80 K, fill fraction is 0.2, tube length is 127 mm (5.0"), and allowable stress is 68.9 MPa (10,000 psi) (FS = 2.0).

Because of limited surface area in the heat pipe condenser, the thickness of the liquid condensate film on the inside wall of the heat pipe could be unnecessarily large. This film in the condenser has a thermal resistance and creates an unwanted temperature drop in the heat pipe. Increasing the internal diameter of the condenser reduces the condensate film thickness because the surface area is increased. This leads to a reduction in thermal resistance and a smaller temperature drop across the liquid film. The temperature difference across the film was calculated as a function of condenser length at an operating temperature of 80 K and is shown in figure 4. Results are shown for the nominal heat load of 60 W and, because the heat pipe is capable of higher performance, for a heat load of 100 W. The active condenser length is approximately 32 mm, which at the nominal heat load gives a film temperature drop of 9 K.

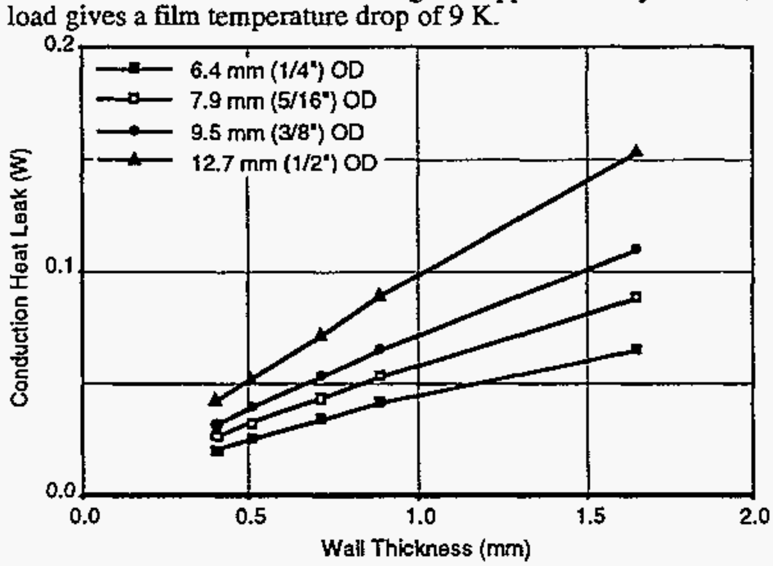

Figure 3. Calculated conduction heat leak along the heat pipe wall with boundary temperatures of 4.2 and 80 K.

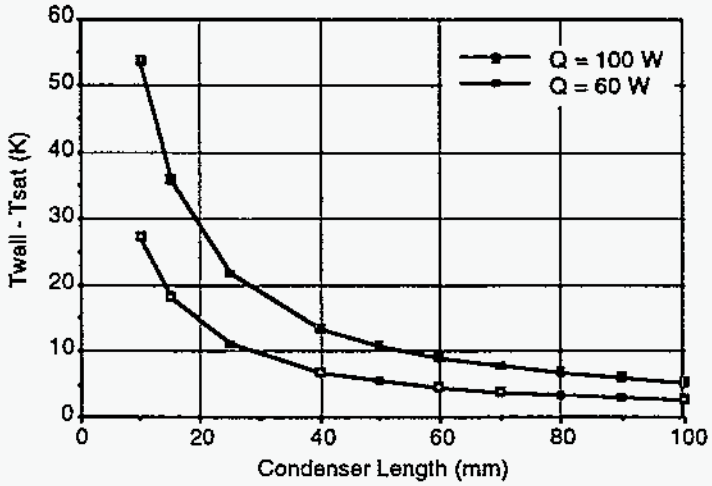

Figure 4. Calculated temperature drop across the liquid film in the heat pipe condenser.

A schematic of the heat pipe is shown in figure 5. The copper evaporator was externally threaded to mate with an internally threaded boss on the cryocooler second stage. Thermal contact was enhanced by mechanically loading the threaded joint with a locknut. The condenser extended above the first stage of the cryocooler. The smooth external surface of the condenser was inserted into a socket that was fitted with flexible louvers and was mounted on the first stage of the cryocooler. The louvers had an interference fit and maintained contact with the heat pipe condenser surface. This joint accommodated relative movement between the heat pipe and the cryocooler.

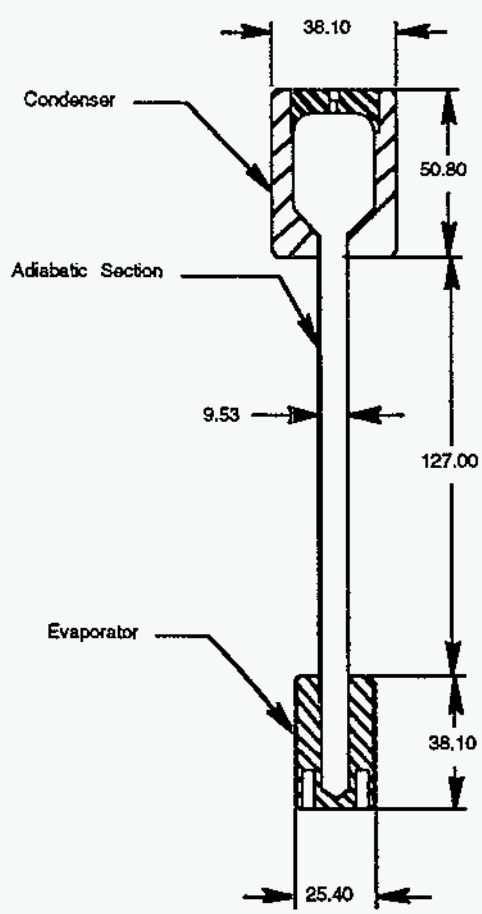

Figure 5. Shunt heat pipe configuration (dimensions in mm).

FILL CHARGE

The heat pipe was evacuated and charged with nitrogen gas at 300 K. Figure 6 shows the relationship between charge pressure and liquid fraction in the heat pipe at 75 K. The heat pipe performance is sensitive to the amount of liquid in the evaporator. If there is insufficient liquid the evaporator will not be isothermal. In a smooth wall, gravity assisted heat pipe the evaporator is cooled from above by the descending liquid film and from below by splashing from the liquid pool[3]. Initially, the heat pipe was charged to 41 atm (600 psia) giving a liquid ratio at 75 K of 0.05. This corresponded to a liquid height at no load of 25 mm. Normally, for a constant diameter heat pipe, this would be a sufficient quantity of liquid for satisfactory operation, however, because of the abnormally large volume of the condenser the heat pipe underperformed at this fill charge (4.1 MPa or 600 psia). The fill charge was then increased to compensate for the holdup of liquid in the oversized condenser. Improved performance resulted.

TESTING

The heat pipe was tested in a vacuum insulated cryostat. The evaporator was electrically heated with a circumferentially wrapped Ni-Cr alloy wire. The heat pipe condenser temperature was controlled by clamping the heat pipe condenser to a variable temperature nitrogen bath. A vacuum pump and throttle valve were used to control the pressure of the nitrogen bath and provided means to vary the bath temperature between 63 and 80 K. Because temperatures higher than 80 K were not required, pressurization of the bath was not necessary. With the variable temperature bath acting as heat sink and the electric heater controlling the input power to the heat pipe, the heat pipe operating temperature could be varied independent of the heat pipe power. Using this technique, a performance map of maximum power versus operating temperature of the heat pipe was generated.

Silicon diode thermometers were used to measure heat pipe wall temperatures and were attached to the heat pipe at the evaporator end, on the adiabatic section near the evaporator exit and on the condenser 20 mm from the condenser inlet. All instrumentation and heater leads were thermally anchored to the nitrogen bath and the thermometer leads were thermally tempered at the measurement points as well. Phosphor-bronze wire was used for the thermometer leads to further minimize heat leak and reduce measurement errors.

The heat pipe performance limit is defined as the maximum power that the heat pipe can transmit under nearly isothermal conditions. Exceeding this power limit results in an interruption of the circulation of the working fluid and subsequent dryout of the evaporator region. With an evaporator no longer supplied with liquid, the heat pipe cannot continue to transfer heat by internal fluid circulation and the axial temperature gradient increases as axial conduction becomes the primary heat transfer mode. For the smooth wall gravity assisted heat pipe, the entrainment limit is reached when flooding or entrainment of the liquid layer occurs[4].

The heat pipe was charged with nitrogen gas to 134 bar (1950 psia) at 300 K. This fill charge corresponded to a liquid fraction of 0.19 at 75 K. Figure 7 shows the measured performance data for the heat pipe compared with the predicted performance using the LANL heat pipe model[6]. The operating temperature range extended to 123 K, near the critical temperature, as predicted by the model. Heat pipe performance limits below 95 K were not measured because of limitations of the test apparatus. The nitrogen bath was capable of reaching 63 K but the thermal resistance between the heat pipe condenser and the nitrogen bath was larger than expected. Also, the heat pipe capacity was higher than expected and at a heat load of 100 W the heat pipe temperature was 95 K. To measure performance limits below 95 K would require decreasing the thermal resistance to the heat sink.

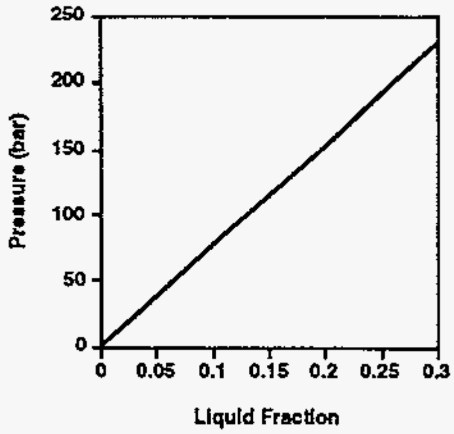

Figure 6. Relationship between liquid fraction in the heat pipe at 75 K and charge pressure at 300 K.

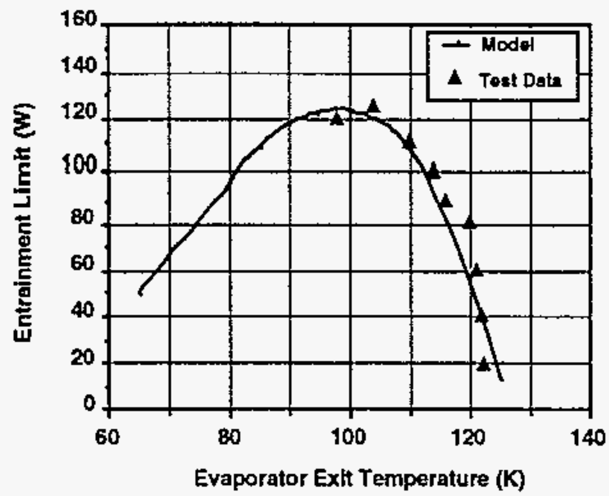

Figure 7. Heat pipe performance map at a liquid fraction of 0.19.

SYSTEM COOLDOWN TESTS

After performance testing was completed, the heat pipe was shipped to the Naval Surface Warfare Center (NSWC), Carderock Division for system cooldown tests. The heat pipe was installed between the first and second stages of a Balzers model UCH-130 cryocooler equipped with a neodymium second stage regenerator. The total weight of the second stage cold mass (copper disc and NbTi coil) was 4.1 kg.

The heat pipe was connected to copper plates on the first and second stages of the refrigerator. The external threads of the evaporator were mated with an internally threaded plate mounted to the refrigerator's second stage. To enhance thermal contact, the threaded joint was packed with Apiezon "N" grease and a nut was installed to increase the joint contact pressure. The condenser end was inserted into a socket on a copper plate mounted to the first stage of the cryocooler. The socket contained flexible louvers (Multi-Lam^{TM}) of silver plated beryllium-copper that maintained an interference fit but would allow some relative motion due to thermal contraction or during installation. In practice, the Multi-Lam connection was not as compliant as expected and was difficult to assemble. Significant temperature differences developed across this interface during cooldown of the magnet, which hindered performance.

The goal of the first test was to determine the cooldown time from room temperature to 5 K for the system without the heat pipe installed. The cooldown time was approximately 3.9 h for this case as shown in figure 8. After the heat pipe was installed, the test was repeated resulting in a cooldown time of 3.5 h. During this test the heat pipe's performance was impaired because of overcooling of the condenser and freezing of the working fluid while the evaporator remained above the critical temperature. While some heat was transferred by the heat pipe during the cooldown, freezing of the working fluid in the condenser most likely reduced the fluid inventory reducing heat pipe performance. As a result, the heat pipe operated for only a short time. A third test was run during which supplemental heat was added to the heat pipe condenser for a few minutes during the cooldown to prevent overcooling of the condenser. This increased the effectiveness of the heat pipe and shortened the cooldown time to 2.5 h. Cooldown temperature profiles measured using the heat pipe with and without heat addition are also shown in figure 8. The inability of the heat pipe to increase the heat load on the first stage of the cryocooler at temperatures above about 125 K resulted in the observed overcooling of the heat pipe condenser. Addition of a second heat pipe with a higher operating temperature (ethane) would eliminate the overcooling problem and further reduce the cooldown time of the system.

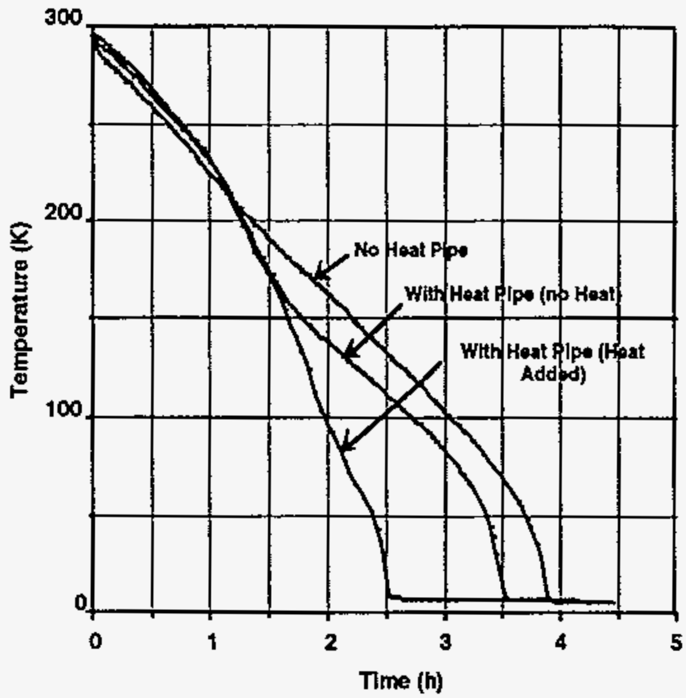

Figure 8. System cooldown performance with and without shunt heat pipe installed.

CONCLUSIONS

This heat pipe, with a liquid fraction of 0.19 at 75 K, achieved a 30 % higher transport power than predicted; however, the liquid fraction needed to achieve this performance was also larger than expected. This larger liquid inventory is required because the liquid tends to accumulate in the oversized condenser, which is required to reduce the film thickness and, therefore, the temperature drop. The heat pipe operating temperature should be between 63 and 123 K. Although not tested at the lower end of this range, the LANL heat pipe computer code was used to determine the heat pipe performance.

As expected, heat pipe performance was sensitive to the liquid inventory within the pipe. Filling the heat pipe at liquid fractions below 0.15 resulted in a reduced operating temperature range with the maximum operating temperature reduced substantially below the nitrogen critical temperature. The maximum operating temperature of the heat pipe at a liquid fraction of 0.05 was 106 K. Larger liquid fill fractions increased the maximum operating temperature.

Upon cooldown of the cryocooler to below the nitrogen triple point, 63 K, the heat pipe working fluid freezes and heat transfer along the heat pipe wall is by conduction only. This heat leak was calculated to be 50 mW with heat pipe end temperatures of 4.2 and 80 K.

System cooldown tests at the NSWC showed a 30 percent decrease in cooldown time when using the nitrogen heat pipe. Because of the narrow operating temperature range of the nitrogen heat pipe, some supplemental heating was required to maintain the heat pipe condenser above the nitrogen triple point during the initial cooldown phase when the heat pipe was not yet fully operational. This problem could be eliminated with the addition of a second heat pipe with a higher operating temperature (such as ethane). By increasing the heat load to the cryocooler at a higher temperature, subcooling of the nitrogen heat pipe would be eliminated, and the additional capacity of the second heat pipe would further reduce the cooldown time.

ACKNOWLEDGMENTS

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