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Heat Pipes for Enhanced Cooldown of Cryogenic Systems

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ABSTRACT

In many important cryogenic applications the use of liquid cryogens for system cooling are either not feasible or are unsuitable. In such cases a cryogenic refrigeration system or multi stage cryocooler must be employed to provide the necessary cooling. To shorten cooldown time for such a system, especially if the thermal mass is large, a thermal shunt directly connecting the first stage of the cryocooler to the load during cooldown is desirable. This thermal shunt allows effective utilization of the greater cooling power available from the first stage of the cryocooler early in the cooldown. Upon reaching operating temperature, the thermal shunt must exhibit a high resistance to thermally isolate the first stage of the cryocooler from the load. Heat pipes are well suited to achieve these objectives. The Advanced Lightweight Influence Sweep System (ALISS), under development by the U. S. Navy for shallow water magnetic mine countermeasures, employs a large, conductively cooled, superconducting magnet that must be cooled from 300 to 4.2 K. Cryogenic heat pipes acting as cryocooler thermal shunts are used to shorten the cooldown time. Ethane, nitrogen and oxygen were evaluated as possible working fluids. A thermal model of the ALISS was developed to evaluate the cooldown performance of various heat pipe combinations. In conjunction with heat pipe performance tests, this model was used to select a suitable design for the heat pipe thermal shunts.

INTRODUCTION

Cooldown times for components or systems attached to multi stage Gifford-McMahon (GM) cryocoolers are significantly influenced by the thermal capacitance of the components, the thermal interface conductances and by parasitic heat loads on the system. For systems containing superconducting magnets with large thermal capacitance, the cooldown times can be considerable.

These components are attached thermally 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 compared with its first stage, cooldown times can be reduced by temporarily shifting part of the heat load during cooldown to the cryocooler's first stage where capacity is greater. Gravity-assisted, cryogenic heat pipes acting as thermal shunts are well suited for this task.

In the U. S. Navy's Advanced Lightweight Influence Sweep System[1] (ALISS), a superconducting magnet[2] will be used to sweep shallow water magnetic mines. The magnet is cooled by a pair of GM rare-earth cryocoolers. Figure 1 shows how the heat pipes connect the first stage of the cryocoolers directly to the load with the evaporators attached to the load and the condensers attached to the cooler's first stage. During cooldown the heat pipes act as thermal shunts connecting the load and the first stage of the cryocoolers. The heat removed from the load, which is mechanically attached to the second stage, is then shared thermally by the first and second stages of the cryocoolers. As the load temperature continues to decrease and falls below the triple point of the heat pipe working fluid, the heat pipe becomes inoperable as the working fluid freezes, thermally disconnecting the load from the first stage of the cryocoolers. By selecting different working fluids, the operating temperature range of the heat pipes can be tailored to the cooldown requirements of the application. There remains a parasitic heat leak along the heat pipe wall, when the working fluid is frozen, which can be made small by proper selection of the heat pipe material and geometry[3]. Because the optimum liquid fraction within the heat pipe is small, the freezing and thawing of the working fluid will not damage the heat pipe. At temperatures above the critical point of the heat pipe working fluid, the heat pipe is also inoperable because of the absence of a two phase interface within the heat pipe.

SYSTEM PERFORMANCE

Ethane and either nitrogen or oxygen are being considered as heat pipe working fluids to shorten the system cooldown time between 300 and 4.2 K. Gravity-assisted heat pipes containing these working fluids operate in different temperature ranges as shown in Fig. 2. By combining an appropriate number of ethane and either nitrogen or oxygen heat pipes connecting the first and second stages of the cryocoolers, the relatively high cooling capacity of the first stage can be applied to the large second stage heat load that arises as a result of the cooldown process. Each type of heat pipe is subjected to a different operating environment during system cooldown. The ethane heat pipes turn on almost immediately upon cryocooler startup and operate nearly isothermally from startup at 295 K down to about 150 K where the vapor pressure of the ethane becomes too low to provide significant heat transport. The working fluid eventually freezes at 89.9 K preventing any further circulation and resulting in shutdown of the heat pipe as desired.

By contrast, the nitrogen or oxygen heat pipes turn on when their condenser temperature reaches approximately 126 and 155 K respectively. With their higher critical temperature compared with nitrogen, the oxygen heat pipes startup sooner overlapping with the operation of the ethane heat pipes. This overlap results in smaller heat pipe axial temperature gradients at startup. As a result of a lower critical temperature, the nitrogen heat pipe operation does not overlap with the ethane. Therefore, the nitrogen heat pipes operate initially at higher axial temperature gradients. Eventually, as the load cools, the evaporator and condenser approach the same temperature and nearly isothermal operation follows. Upon further cooldown of the load, the working fluid in the heat pipes freezes stopping circulation. For nitrogen this occurs at 63.1 K and for oxygen at a lower temperature of 54.3 K. Heat transfer is then limited to conduction along the heat pipe walls.

Regardless of the temperature gradient imposed along the heat pipe length, in all cases the internal operation of the heat pipes is the same. The large temperature differences only change the effective length of the heat pipe. In the non isothermal mode the condensate does not reach all the way to the evaporator before it is vaporized. The point at which the liquid film is depleted defines the effective length of the heat pipe, which for strongly non isothermal operation is less than the total length. Heat from the load flows by conduction along the wall of the heat pipe to the point above the evaporator where the liquid film disappears. This conduction length varies according to the amount of superheat present in the evaporator. Large superheats require large conduction lengths and result in shorter effective heat pipe lengths. Eventually, as the load cools the conduction length shrinks to zero and the heat pipe becomes nearly isothermal over its entire length. The condensation, circulation and evaporation inside the heat pipe is similar in all cases, only the point at which evaporation occurs differs and therefore, changes the effective heat pipe length. Non isothermal operation of the heat pipe should not be viewed as abnormal or detrimental as thc heat pipe accommodates for this condition.

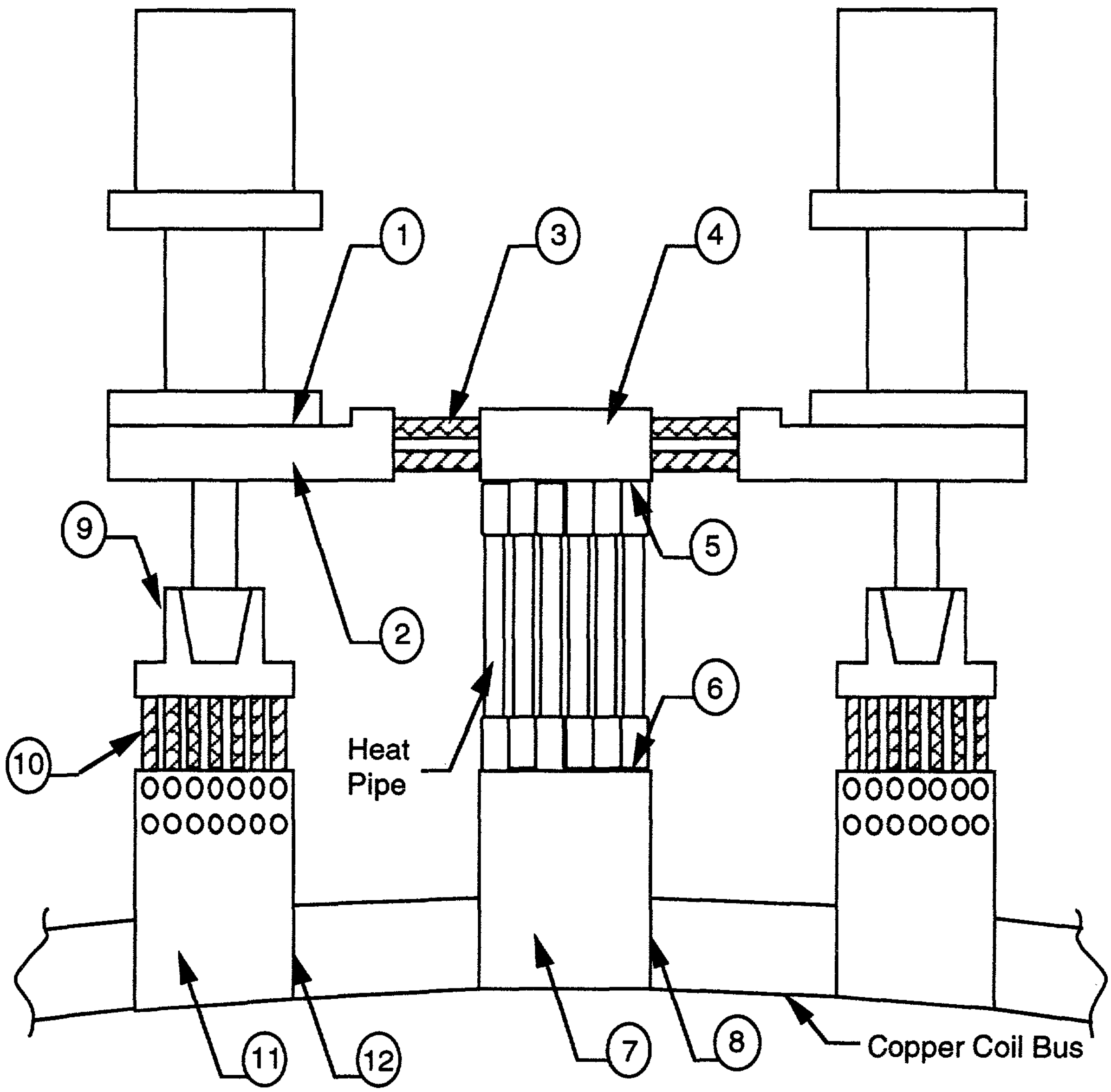

Figure 1. **Schematic of ALISS thermal configuration showing shunt heat pipes connecting the first stages of the cryocoolers to the load with thermal paths identified as 1) 1st stage contact conductance; 2) 1st stage cryocooler bus bar; 3) 1st stage flexible braids; 4) 1st stage collector bus; 5) heat pipe condenser contact conductance; 6) heat pipe evaporator contact conductance; 7) heat pipe cold bus; 8) heat pipe cold bus to coil contact conductance; 9) 2nd stage interface conductance; 10) 2nd stage flexible braids; 11) cryocooler cold bus; and 12) cryocooler cold bus to coil contact conductance.**

A thermal model for the ALISS system was developed to analyze the various cooldown scenarios and evaluate the optimum heat pipe configuration and selection of working fluids. The model is a lumped capacitance, finite-difference type using the SINDA[4] computer code. Temperature dependent heat capacities and interface conductances were used. Cryocooler performance was also temperature dependent and based on available test data.

Use of the shunt heat pipes results in significant reduction in cooldown time for the ALISS system as shown in Fig. 3. The 961 kg load temperature is shown as a function of time for the case without any heat pipes and for the case with two ethane and four nitrogen heat pipes corresponding to the ALISS configuration. With heat pipes present the cooldown time is reduced to nearly half of the case without heat pipes. Figure 4 shows a heat load map for the two cooldown cases. Included in the figure is the predicted heat load on the first stage of the cryocoolers both with and without heat pipes present. These curves illustrate the effect of shunting a portion of the cooldown load to the first stages of the cryocoolers resulting in faster cooldown for the system. Also shown in Fig. 4 is the heat load carried by the ethane and nitrogen heat pipes. The ethane heat pipes operate for the

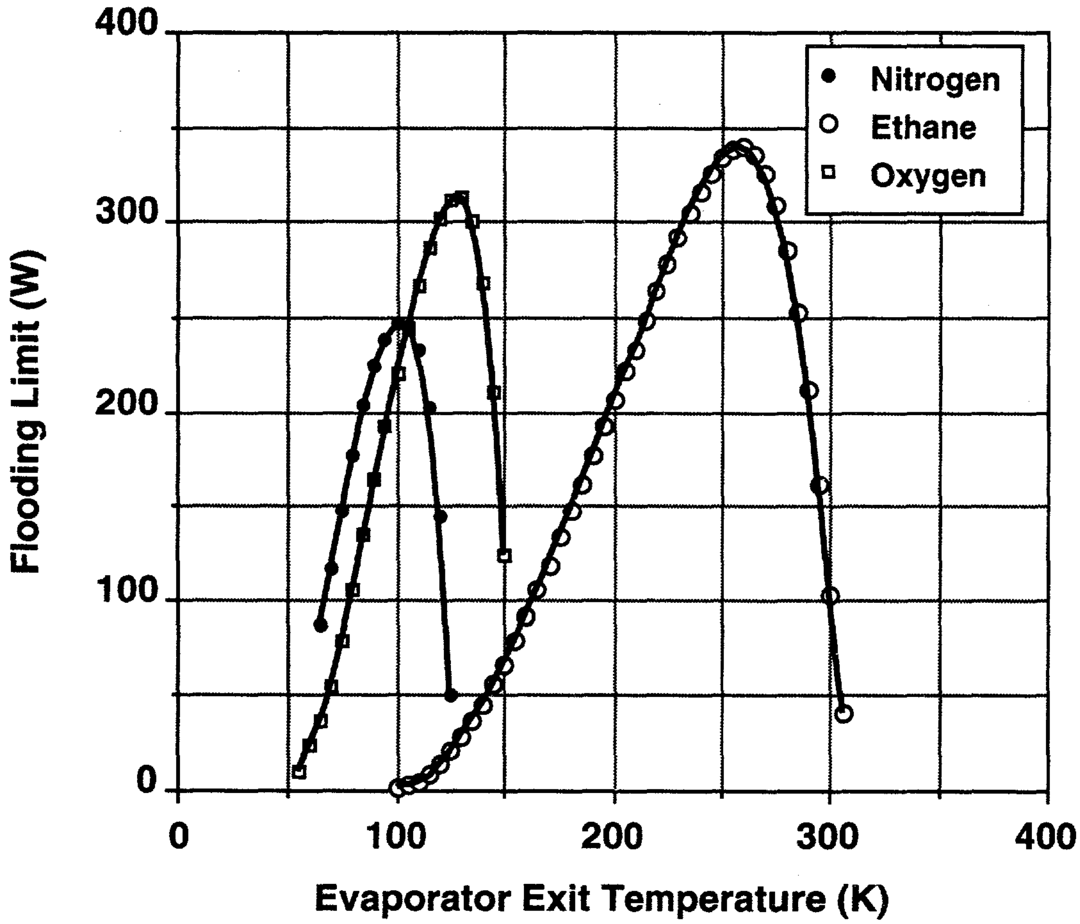

Figure 2. Predicted heat pipe performance due to flooding limit for 13-mm-OD heat pipe with ethane, nitrogen and oxygen working fluids.

first 65 h of the cooldown during which time the nitrogen heat pipes remain supercritical and do not operate. After a brief period of approximately 4 h when none of the heat pipes are operational because the ethane is frozen and the nitrogen is still supercritical, the nitrogen heat pipes begin operation in the non isothermal mode described earlier. The effect of the 4 h interruption in heat pipe duty can be seen in Fig. 4 as a drop in the first stage heat load. Finally, at 105 h the nitrogen also freezes and all heat pipes cease operation. Except for some heat conduction along the heat pipe wall, this terminates the shunting of heat between the load and the upper stages of the cryocoolers.

From this point, only an additional 10 h are required for the magnet to reach 4 K. Control of the heat pipes is passive in the sense that no external action is required to initiate or terminate their operation. The heat pipes respond automatically to the imposed thermal conditions.

HEAT PIPE DESIGN

The heat pipes are gravity-assisted and therefore require the evaporators to be below the condensers in a gravity field when operating. All of the heat pipes are identical and are approximately 300-mm-long with an outside diameter of 13 mm. The evaporators and condensers are copper while the adiabatic section is fabricated from low thermal conductivity, thin-walled tubing. The evaporators and condensers are sized to accommodate the radial heat load and the geometry of the thermal interfaces. A schematic of the heat pipes is shown in Fig. 5.

The heat pipe performance, as shown in Fig. 2, is expected to be limited by flooding of the condenser by the counter flowing vapor. At high vapor velocities the liquid return flow along the heat pipe wall is impeded by the liquid-vapor interaction. At the onset of flooding the liquid is pushed into the condenser by the upward flowing vapor, interrupting the flow of liquid to the evaporator. As a result, partial dryout of the evaporator occurs and the heat pipe capacity is reduced. At temperatures near the freezing point of the working fluid, the vapor pressure is low

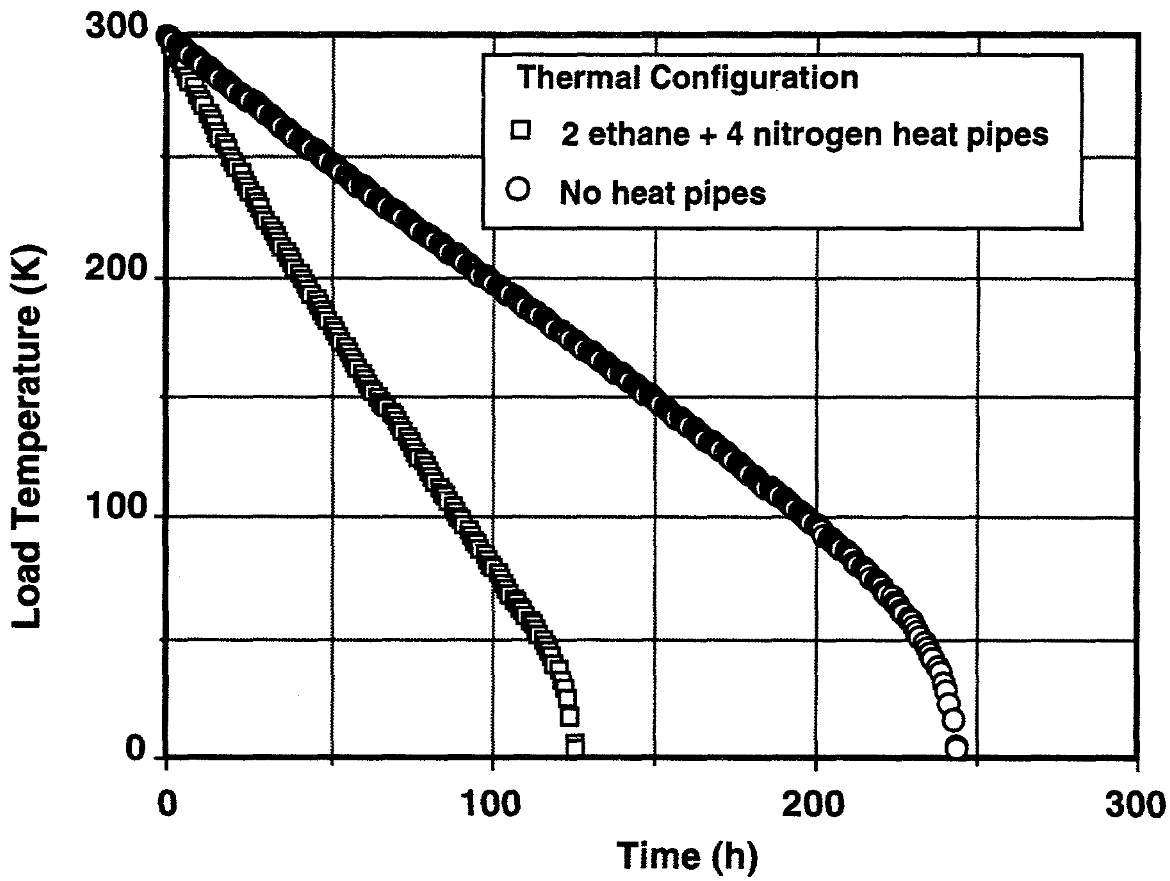

Figure 3. Cooldown performance of ALISS system with and without 2 ethane and 4 nitrogen 13-mm-OD heat pipes.

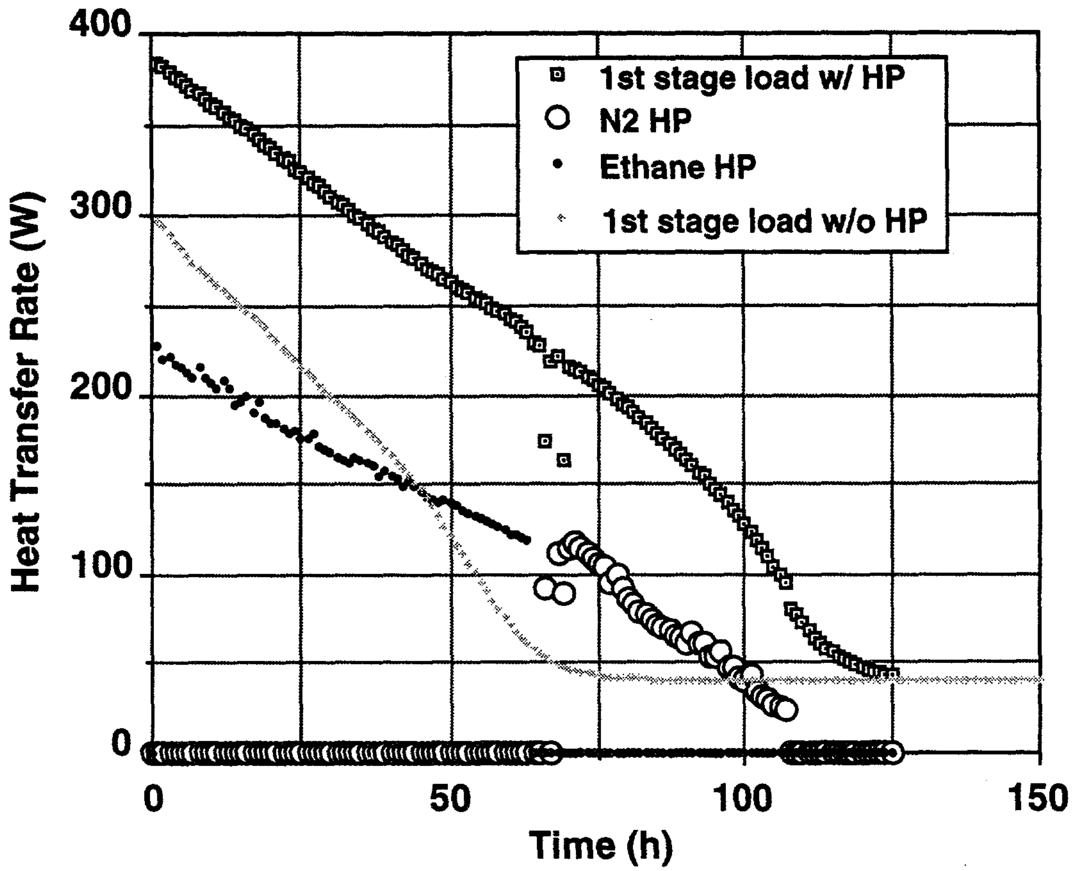

Figure 4. Load map of ALISS system cooldown with and without 2 ethane and 4 nitrogen 13-mm-OD heat pipes.

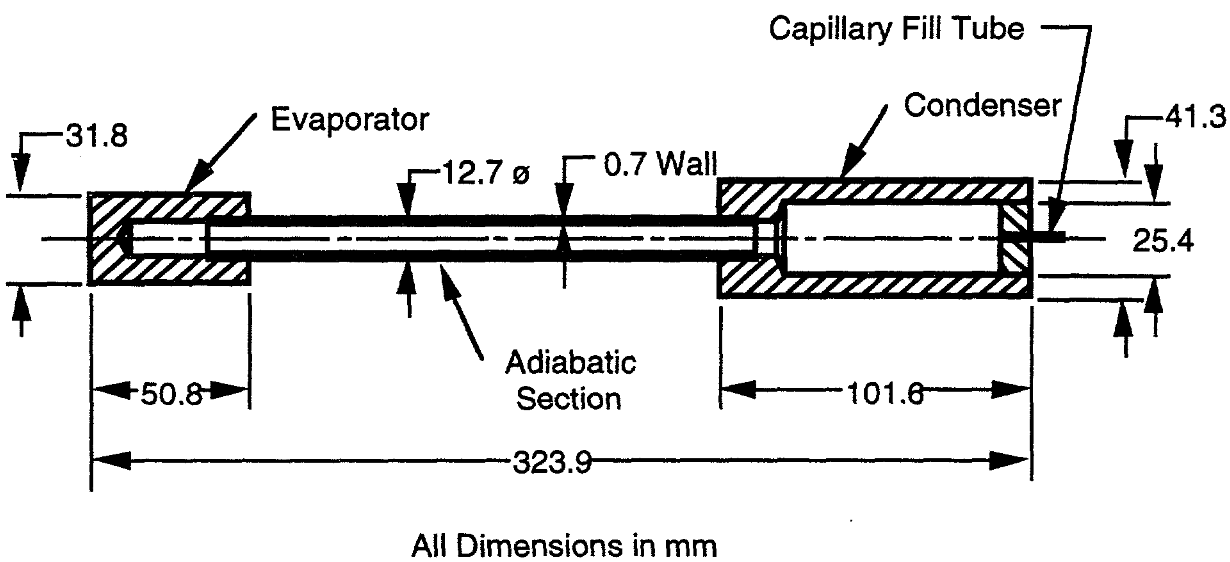

Figure 5. Preliminary design of shunt heat pipe showing evaporator and condenser geometry.

resulting in high vapor velocities and correspondingly low performance limits also shown in Fig. 2. The flooding limit is strongly temperature dependent requiring the heat pipes to be slightly oversized to carry the required heat load over a wide temperature range. The data in Fig. 2 were generated using the Los Alamos Heat Pipe Code HTPIPE[5].

HEAT PIPE TESTS

Heat pipe performance testing was conducted at the Carderock Division of the Naval Surface Warfare Center, Annapolis Detachment (CDNSWC/A). Based on the heat pipe design described previously, two scaled up heat pipes with a 16-mm-OD (0.625 in) adiabatic section were fabricated. A photograph of one of the heat pipes is shown in Fig. 6.

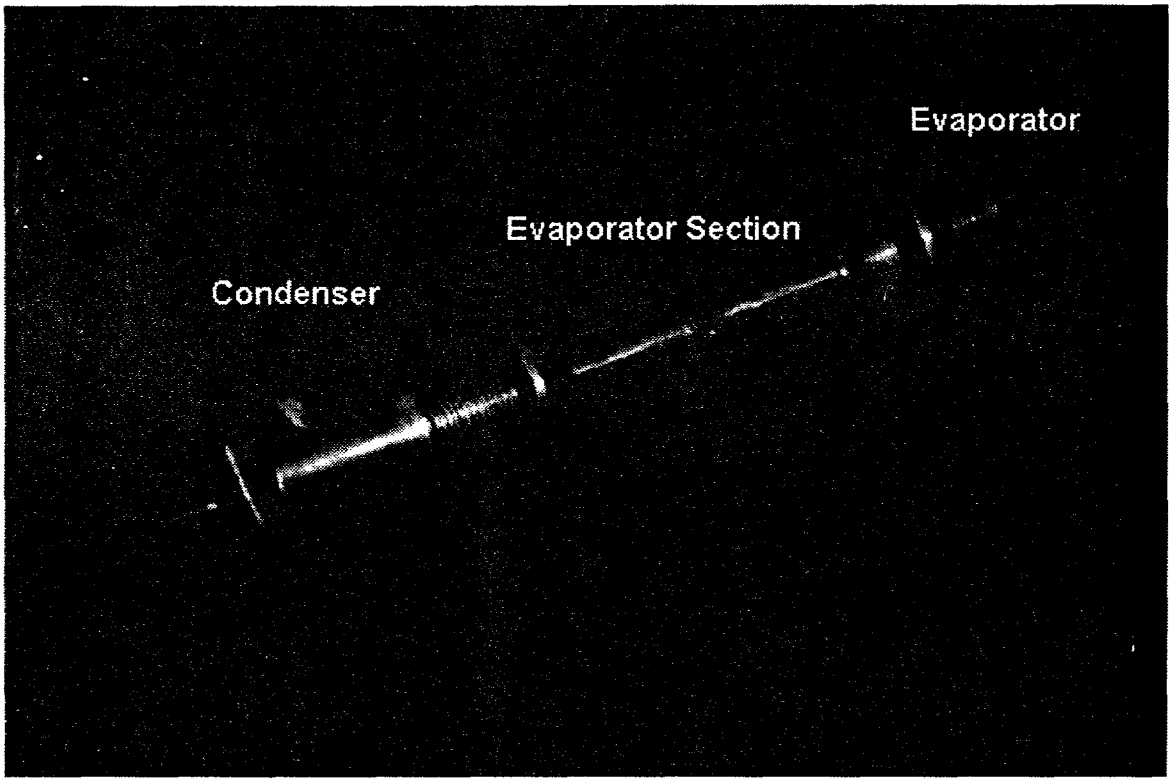

Figure 6. Heat pipe showing mounting provision for evaporator and condenser

The isothermal performance of the 16-mm-OD heat pipe can be accurately predicted using the Los Alamos computer code HTPIPE.[5] Figure 7 shows the heat pipe performance predicted by HTPIPE for the various working fluids together with test data obtained during isothermal heat pipe operation for ethane and nitrogen. These measurements were limited by large temperature gradients between the cooler and the heat pipe interfaces and by the cooler capacity. Because of these limitations, only heat pipe performance limits near the working fluid critical point could be measured. The measured data are consistent with the performance predictions for isothermal operation.

However, the non-isothermal (evaporator and condenser at different temperatures) performance is not predicted by HTPIPE and; therefore, additional tests were performed to determine these characteristics. The heat pipe condenser was thermally connected to the cold plate of a single stage GM cooler from Cryomech (AL 200). The heat pipe was installed in a vacuum vessel with a 1-stage GM cooler, five Lake Shore diode temperature sensors, and two cartridge heaters. Three diode temperature sensors were located on the heat pipe: one at the condenser; one at the evaporator; and one at the middle of the heat pipe (adiabatic section). The remaining two temperature sensors were located on the cold plate and the hot plate respectively of the test apparatus. The first heater, which was computer controlled, was placed near the heat pipe condenser to provide a nearly constant condenser temperature. The second heater was located on the evaporator plate to provide the thermal load for the heat pipe. Figure 8 shows the arrangement of the heat pipe, cooler, and instrumentation.

The heat pipe was filled with nitrogen to a pressure of 136 atm (2000 psig) and the non-isothermal performance measured. In simulating non-isothermal operating conditions the heat pipe temperature was increased until partial dryout of the evaporator occurred. The condenser temperature was then reduced and maintained at 105 K while the evaporator temperature was varied from 160 K to 255 K by varying the power supplied to the evaporator.

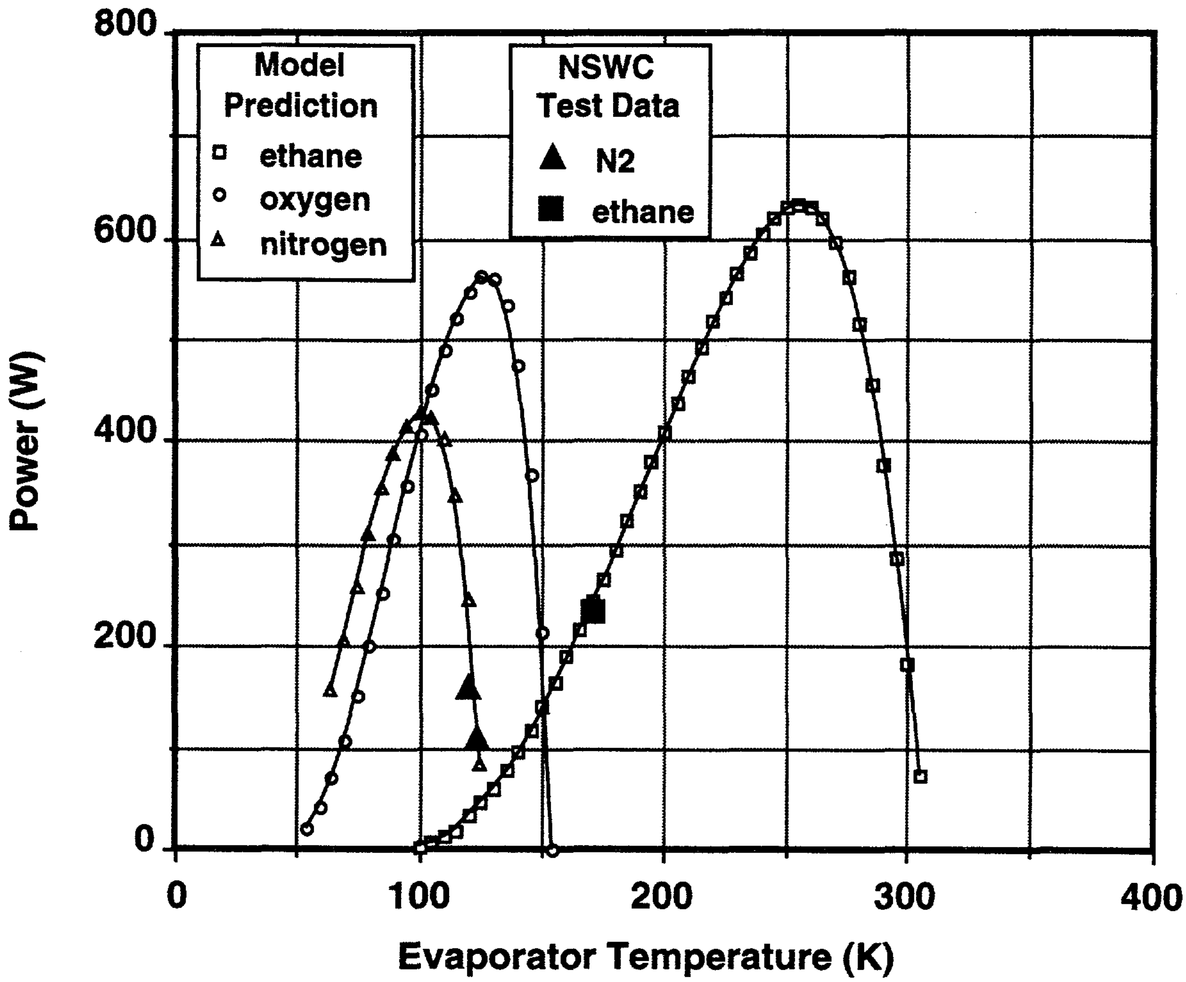

Figure 7. Comparison of predicted isothermal heat pipe performance with test data for 16-mm-OD heat pipe.

The results of the non isothermal tests are shown in Fig. 9. The non-isothermal capacity of the nitrogen heat pipe is a strong function of the evaporator temperature. The length of the evaporator dryout zone increases non linearly with increasing evaporator temperature. Consequently, the heat transfer rate which is proportional to the temperature difference along the heat pipe and inversely proportional to the evaporator dryout length, goes through a minimum as the evaporator temperature increases. Initially, the increase in the evaporator dryout length increases faster than the heat pipe temperature difference causing the heat transfer rate to decrease. Eventually, the temperature difference increases faster than the dryout length and the heat transfer rate increases with increasing evaporator temperature. The non isothermal performance of the oxygen heat pipes is expected to be similar to the nitrogen heat pipes and testing of the oxygen heat pipes is planned. The heat pipe evaporator response determined from the non isothermal tests will be incorporated into the system cooldown model.

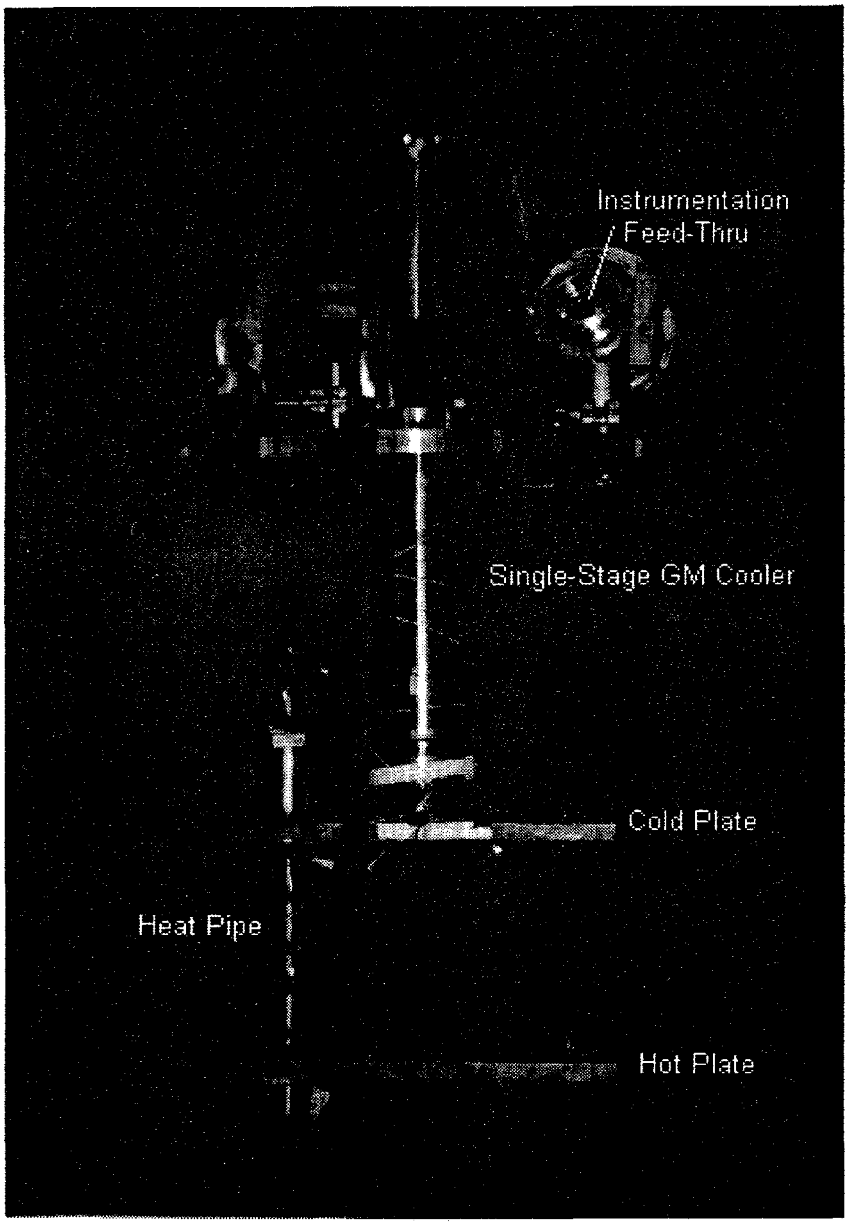

Figure 8. Heat pipe mounted on cryocooler first stage for testing.

CONCLUSIONS

This study investigated the use of ethane, oxygen, and nitrogen heat pipes to enhance the cooldown of a large superconducting magnet system. High capacity ethane, oxygen, and nitrogen cryogenic heat pipes were modeled, designed, fabricated, and tested. In addition, the non-isothermal performance of the nitrogen and oxygen heat pipes, where the evaporator temperature exceeded the working fluid critical point, were experimentally determined.

The impact of the various types of heat pipes on the cooldown time for a large superconducting magnet system was determined through the use of a system cooldown thermal model. The thermal model predicted that the use of heat pipes as thermal shunts between the first and second stages of the cryocooler can reduce the cooldown time by 50 percent.

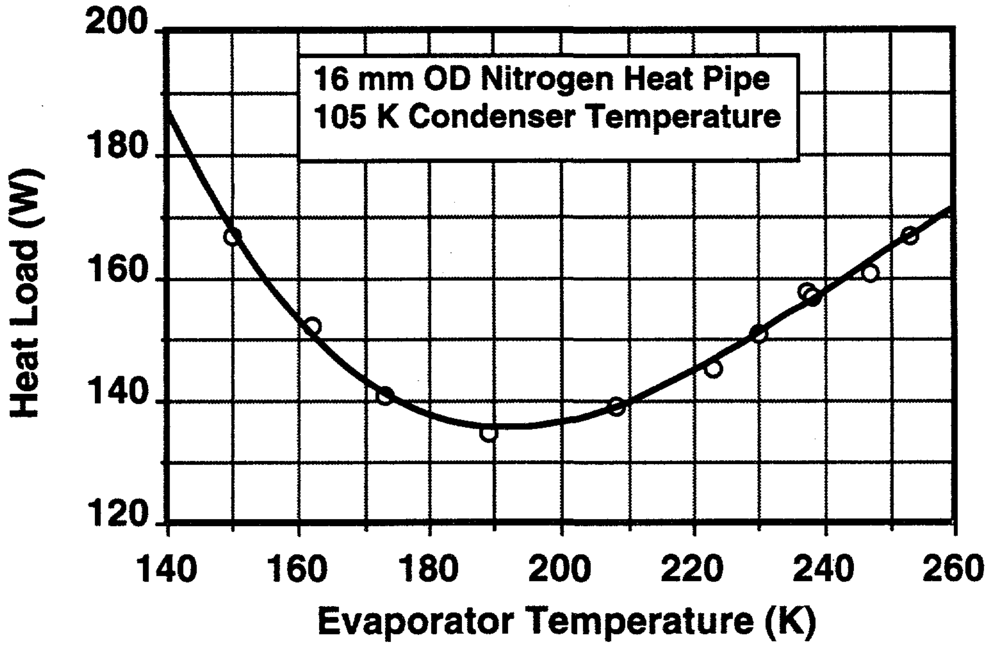

Figure 9. Non isothermal heat pipe performance based on constant condenser temperature.

Testing and evaluation of the three candidate heat pipes centered around the determination of an effective heat pipe combination. The use of oxygen heat pipes in conjunction with ethane heat pipes provided a continuous, nearly isothermal shunt for the cryocooled system from room temperature down to about 60 K. In contrast, it was found that nitrogen heat pipes in combination with ethane heat pipes did not perform isothermally in the temperature range of 150 K - 125 K.

Because of oxygen's wider operating temperature range (60 - 155 K) and its greater transport properties compared with nitrogen, oxygen heat pipes were selected to operate with the ethane heat pipes as a thermal shunt between the first and second stages of the cryocooled system.

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