TRANSIENT ANALYSIS OF A TWO-PHASE HYDROGEN HEAT SWITCH

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TRANSIENT ANALYSIS OF A TWO-PHASE HYDROGEN HEAT SWITCH*

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ABSTRACT

A transient, thermal analysis of a two-phase hydrogen heat switch is presented. The heat switch has application to a zero-g, no-moving-part magnetic refrigerator that operates between 7 K and 25 K and provides a one-way thermal path for heat rejection from the refrigerator to the 20 K heat sink. Incorporation of the heat switch is dependent on operating frequency. However, the thermal diode operating characteristic and the high axial thermal conductivity of the heat switch provide advantages over other methods.

INTRODUCTION

Magnetic refrigeration shows promise as a space-based cryocooler technology for two important reasons: first, its potentially high operating efficiency enables reduced input power requirements, and second, its low inventory of moving parts, combined with the slow motion of those parts, provides increased reliability. It would be of significant advantage to eliminate all moving parts in such a device. To date, low-temperature applications1-3 have been investigated because of the well-understood properties of gadolinium gallium garnet (GGG), as a low-temperature working material, and the need for superconducting magnets to generate sufficient field strength. The application for a magnetic refrigerator (MR) has focused on sensor cooling at temperatures near 7 K, with heat rejection to a 20 K heat sink. Thus, an MR could be a bottom stage in a refrigeration system spanning 7 to 300 K, or a single stage, rejecting heat to a liquid hydrogen heat sink at 20 to 25 K.

In an MR the working fluid must experience a cyclic magnetic field. There are several ways to accomplish this, including moving the magnetic working material relative to the field, moving the field relative to the working material, or cycling the field, thereby eliminating relative motion. The latter method requires charging and discharging the magnets, but no relative motion of the magnetic material is necessary and, hence, forms

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the basis for a no-moving-part magnetic cryocooler. Several important issues relative to the no-moving-part design must be addressed. One issue that arises in a Carnot cycle design is that of periodic heat transfer to and from the magnetic working material. Thermal switches, operating as one-way or diode heat pipes, have been proposed as a means of transferring heat from the magnetic working material to the heat sink at 25 K. The transient response of these cryogenic thermal switches and their impact on the MR performance is the subject of this investigation.

ANALYSIS

THERMAL SWITCH DESIGN

A no-moving-part magnetic refrigerator operating on a Carnot cycle between 7 K and 20 K is shown schematically in Fig 1. The magnet is cyclicly charged to produce the required field change and the magnetic material is stationary relative to the magnet. Thermal switches are required to transfer heat from the load at 7 K to the magnetic material and from the magnetic material at 25 K to the heat sink. The heat load is cyclic, with period determined by the magnet-charging frequency. Hydrogen-filled, diode heat pipes (hydrogen heat switches) are being considered for connection to the heat sink at 20 K.

Fig. 1. Schematic of Carnot cycle no-moving-part magnetic refrigerator showing location of thermal switches.
An active two-phase thermal switch, based on heat pipe operating principles, has several advantages for this application. The high axial thermal conductivity resulting from two-phase operation permits separation of the heat source from the heat sink without imposing large temperature gradients in the system. This provides much needed design flexibility and enables the heat sink to be located away from the cyclic magnetic field. The two-phase system also provides isothermal heat addition and heat removal zones that can accommodate nonuniform heat generation in the magnetic material. The relatively large axial heat flux transmitted by the thermal switch minimizes the void fraction of the magnetic working material. More importantly, operation of the device as a thermal diode or one-way thermal conductor provides the required thermal switching needed for the MR operation. Finally, the two-phase system has a low inventory of working fluid that minimizes thermal addenda in the magnetic cooler.

The evaporator of each thermal switch is embedded in the matrix of magnetic working material. A low-resistance thermal joint is achieved by using a low-melting-point solder, such as gallium on the external surface of the evaporator. The cold end or condenser of the thermal switch is an integral part of the heat sink heat exchanger. The geometry of the thermal switch, shown in Fig. 2, may contain a right-angle bend between the evaporator and condenser to provide a compact envelope.

The simplest internal configuration for the thermal switch consists of a homogeneous, screened wick inserted into and located against the inner wall of the container. This configuration allows vapor flow along the centerline of the container tube, with return liquid flow in the annular homogeneous wick. This design requires the largest working fluid inventory and, therefore, provides the slowest transient response and largest thermal addenda. The objective of the study was to evaluate the effect of thermal switches on the refrigerator performance. This switch design was selected because it produces the

![Diagram of magnetic heat switches](image)
maximum performance impact on the MR operation. Figure 3 shows the dimensions and a cross section of the thermal switch. The wick material was assumed to be copper to give adequate radial thermal conductance, and the outer tube was assumed to be 300-series stainless steel to minimize the wall thickness and reduce eddy-current heating.

The thermal switch was sized to carry a 3-W heat load at an operating temperature of 20 to 25 K. The wick thickness was optimized at 20 K as shown in Fig. 4. Increasing the wick thickness reduces the liquid pressure drop while simultaneously increasing the vapor pressure drop. An optimum thickness of 0.025 cm (.010") was selected. With a fixed wick thickness the switch performance is shown as a function of operating temperature in Fig. 5. The results show that the switch performance is reduced by one-third between 20 and 25 K. This behavior is typical for cryogenic working fluids.

This thermal switch design, with a nominal capacity of 3 W and a diameter of 0.8 cm, was used as a basis for evaluating the transient thermal response and thermal addenda for the no-moving-part magnetic refrigerator.

TRANSIENT THERMAL RESPONSE

A finite difference thermal model of the hydrogen thermal switch was developed. A nodal diagram is shown in Fig. 6. Separate nodal elements are defined for the wall, the liquid-saturated wick, and the vapor. Because the operating temperature of the refrigerator is 7 to 25 K, the heat of fusion for the hydrogen working fluid is included. Boundary nodes are external to the evaporator, and condenser and contact resistance is included. Also included are temperature-dependent heat capacity for the wall and wick materials. The temperature change of the magnetic working material is simulated by a time-dependent boundary-node temperature profile. This temperature profile can be altered to simulate different operating conditions for the MR. Transient temperature profiles and energy flows are output from the model.

Fig. 3. Internal cross section of the thermal switch showing homogeneous wick.
Fig. 4. Optimum wick thickness for thermal switch at 20 K.

Fig. 5. Capacity of thermal switch as a function of operating temperature.
The magnetic refrigerator operating frequency was chosen as the independent variable. The nominal operating conditions for the refrigerator are given in Table 1.

Three MR designs were investigated, based on three operating frequencies of 0.1, 0.033, and 0.0167 Hz, giving periods of 10, 30, and 60 s, respectively. For a fixed cooling power, varying the operating frequency requires a variable mass of magnetic material. For a fixed length over diameter (L/D) ratio for the magnet configuration, changes in the magnetic material mass cause changes in the diameter and length of the magnetic material envelope. The magnetic envelope, in turn, changes the evaporator length of the thermal switch.

TABLE 1. Nominal magnetic refrigerator operating conditions.

<table>
<thead>
<tr>
<th></th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cooling Power (W)</td>
<td>1.0</td>
</tr>
<tr>
<td>Load Temperature (K)</td>
<td>7</td>
</tr>
<tr>
<td>Sink Temperature (K)</td>
<td>20</td>
</tr>
<tr>
<td>Working Material</td>
<td>GGG or DAG</td>
</tr>
<tr>
<td>Max Field Strength (T)</td>
<td>6</td>
</tr>
<tr>
<td>L/D for GGG</td>
<td>2</td>
</tr>
<tr>
<td>Thermal Switches</td>
<td>3</td>
</tr>
</tbody>
</table>
The results of the transient model are shown in Figs. 7-12 for the three frequencies investigated. Figure 7 shows the thermal switch wall temperature profile for a 0.1-Hz operating frequency. The evaporator boundary is cycled between 7 and 25 K, with the condenser boundary temperature held constant at 20 K. The boundary temperature for all three cases was the same, with only the operating frequency changed. In Fig. 7, note the 1.0-s delay required for heat pipe startup after the material is magnetized at 5 s. Figure 8 shows the power and energy profiles for the 0.1-Hz case. The power through the condenser wall is labeled "QCOND" while the power transmitted by the thermal switch is labeled "QMAX". In the absence of axial conduction in the switch wall, QMAX = QCOND. The total energy transferred to the evaporator is labeled "EVAP". This parameter is used to determine the thermal addenda (the energy absorbed by the thermal switch prior to startup.) After startup, EVAP is a measure of the total energy removed from the magnetic working material.

Figures 9 and 10 show temperature and power profiles for 0.033-Hz operation, while Figs. 11 and 12 show the results from operating at 0.017 Hz.

The heat pipe startup time lengthens at lower operating frequency because of the lower starting temperature that results from the longer dwell time between magnetic cycles. The thermal addenda is also larger at the lower frequencies; however, the thermal addenda, as a fraction of the total energy transferred, decreases at lower frequencies, resulting in less degradation of refrigerator performance.

The transient response data for the thermal switch also shows that, upon demagnetization, very little energy flows in the reverse direction. This is an important operating characteristic of a two-phase heat switch that prevents heat leak from the heat sink back into the refrigerator working material and acts as a thermal diode. The results of the transient analysis indicate that no modification to the wick structure is required to achieve the diode behavior. The low thermal capacitance of the heat switch permits a rapid decrease in the working fluid temperature, which prevents fluid circulation when the magnet material demagnetizes. This unexpected result simplifies the design of the wick for the thermal switch.

The impact of the thermal addenda, introduced by the thermal switch on the refrigerator's performance, is discussed in the following section.

**IMPACT ON REFRIGERATOR PERFORMANCE**

The incorporation of the 25-K thermal switches into the refrigerator design affects the refrigerator's performance in two ways:

- Increased heat capacity or thermal addenda of the heat switch reduces the adiabatic temperature change of the magnetic working material, and

- The finite response time or startup delay of the thermal switch following magnetization reduces the net cooling power of the refrigerator.
Fig. 7. Thermal switch wall temperature profiles for operating frequency of 0.10 Hz.

Fig. 8. Thermal switch energy power flows for operating frequency of 0.10 Hz.
Fig. 9. Thermal switch wall temperature profiles for operating frequency of 0.033 Hz.

Fig. 10. Thermal switch energy power flows for operating frequency of 0.033 Hz.
Fig. 11. Thermal switch wall temperature profiles for operating frequency of 0.017 Hz.

Fig. 12. Thermal switch energy power flows for operating frequency of 0.017 Hz.
Both of these effects were evaluated as a function of operating frequency of the MR. Heat rejection to the thermal sink was held constant at 7.5 W, requiring three thermal switches of nominal 2.5-W capacity. In addition, the load temperature was held constant at 7 K. The decrease in cooling power and the heat sink temperature required were determined as a result of incorporating the thermal switches into the refrigerator. The results are given in Table 2. The thermal efficiency, relative to the Carnot efficiency, is defined as

$$
\eta = \frac{T_h}{T_c} - 1
$$

and for the baseline case, where

$$
Q_c = 1 \text{ W} ,
Q_h = 7.5 \text{ W} ,
T_c = 7 \text{ K} , \text{ and}
T_h = 27 \text{ K} ,
$$

then

$$
\eta = 0.44 .
$$

### Table 2. Effect of thermal switches on magnetic refrigerator performance.

<table>
<thead>
<tr>
<th>MR Operating Frequency (Hz)</th>
<th>0.1</th>
<th>0.033</th>
<th>0.017</th>
</tr>
</thead>
<tbody>
<tr>
<td>Period (s)</td>
<td>10</td>
<td>30</td>
<td>60</td>
</tr>
<tr>
<td>Mass GGG (g)</td>
<td>858</td>
<td>2572</td>
<td>5145</td>
</tr>
<tr>
<td>Evaporator Length (cm)</td>
<td>8.6</td>
<td>12.4</td>
<td>15.6</td>
</tr>
<tr>
<td>Diameter GGG (cm)</td>
<td>4.3</td>
<td>6.2</td>
<td>7.8</td>
</tr>
<tr>
<td>Switch Addenda (J)</td>
<td>25</td>
<td>51</td>
<td>81</td>
</tr>
<tr>
<td>Adiabatic DT Ratio (DTa'/DTa)</td>
<td>0.71</td>
<td>0.78</td>
<td>0.82</td>
</tr>
<tr>
<td>$T_h$ (K) (Nominal 20 K)</td>
<td>21.2</td>
<td>22.6</td>
<td>23.4</td>
</tr>
<tr>
<td>Thermal Length (cm)</td>
<td>0.53</td>
<td>0.95</td>
<td>1.29</td>
</tr>
<tr>
<td>Startup Delay (s)</td>
<td>1</td>
<td>3</td>
<td>6</td>
</tr>
<tr>
<td>Cooling Power (W) (Nominal = 1 W)</td>
<td>0.83</td>
<td>0.83</td>
<td>0.91</td>
</tr>
<tr>
<td>Efficiency</td>
<td>0.25</td>
<td>0.28</td>
<td>0.32</td>
</tr>
<tr>
<td>Efficiency Ratio (Nominal = 0.44)</td>
<td>0.58</td>
<td>0.63</td>
<td>0.74</td>
</tr>
</tbody>
</table>
The reduced efficiencies with the thermal switches present are shown in Table 2 and range from 0.254 to 0.324. Normalizing these values to the baseline efficiency gives a measure of the degradation caused by incorporating the thermal switches. The efficiency ratio as a function of operating frequency, is plotted in Fig. 13. At a high frequency, when the cycle time is short, the additional thermal addenda reduces refrigerator performance more than at a lower frequency at which the mass of magnetic material is greater. The effect is less pronounced at the higher frequency because the switch temperature change is reduced as a result of the low dwell time between magnetizations. This effect is shown as a change in slope at a frequency of 0.03 Hz.

The changing mass ratio also influences the maximum temperature reached by the magnetic material upon magnetization. The smaller mass ratio at higher frequency gives the larger adiabatic temperature change.

The homogeneous wick heat pipe design selected for this study has some reverse conduction or heat leak from the condenser to the evaporator. This heat leak together with the thermal capacitance of the conservatively large volume of working fluid in the heat pipe acts as a parasitic load on the MR. A measure of the effectiveness of the heat switch is the ratio of energy flow in the on mode to the energy flow in the off mode. For this switch design the ratio \( \approx 8.5 \). The use of an artery wick will both reduce the liquid inventory and improve the heat pipe performance. In addition, the use of a stainless steel, inverted wick design will decrease reverse conduction because of the lower thermal conductivity of the wick. These changes will significantly reduce parasitic losses to the MR.

**Fig. 13.** Reduction in refrigerator efficiency caused by thermal addenda of the thermal switches.
DISCUSSION

The following observations can be made regarding the operation of a no-moving-part magnetic refrigerator.

a. When operating in a nonregenerative or Carnot cycle a means of passive thermal switching is required. Passive, in this case, refers to a system with no moving parts or valves with which to accomplish the switching function. Because two physical changes occur in the system when it is magnetized, a change in field and a change in temperature, the switching function must be triggered by one of these mechanisms. The two-phase hydrogen thermal switch responds to the temperature change and its effect on the working fluid transport properties. A material with a suitable magnetic, field-dependent thermal conductivity could also be used as a thermal switch.

b. Any material placed in thermal contact with the magnetic material must have low thermal addenda so that the adiabatic temperature change of the refrigerant is not significantly reduced.

c. The transient response of the switch must be fast relative to the operating frequency of the MR. Otherwise, a reduction in the average cooling power occurs because of the time lapse required to establish the steady state temperature profile of the thermal switch.

d. A high turndown ratio (ratio of the on to off thermal conductance) for the switch is required to effectively isolate the magnetic material from the heat load or heat sink at the appropriate point in the cycle.

e. A regenerative cycle eliminates the need for thermal switching by relying on convective heat transport. However, this concept requires a no-moving-part pump for circulating the working fluid.

Results of this analytical study for evaluating the performance of two-phase thermal switches can be summarized as follows:

f. The operational temperature range of the switch is limited by the choice of working fluid. For hydrogen, the useful range is 15 to 30 K. In the gap between 5 and 15 K, no suitable working fluid is available. At temperatures above 30 K, neon is a suitable working fluid.

g. The mass of the two-phase thermal switch is small compared with that of a solid conductor of equal thermal conductance. Low mass enables rapid transient response and low thermal addenda.

h. The diameter of the thermal switch must remain small to contain the internal pressure of the working fluid at 300 K. Stainless steel is the best candidate because of high-strength and low-axial conduction; however, the radial conduction of the stainless steel is poor.
i. The high effective thermal conductivity of the switch permits flexibility in specifying the geometry of the magnetic material. Changing the length of the evaporator or heat addition zone has a minor impact on the switch capacity.

j. The switching function can be achieved without special modification to the wick structure. The rapid temperature change realized upon demagnetization is sufficient to prevent heat flow in the reverse direction, that is from the heat sink to the magnetic material.

Results of this study indicate that a two-phase hydrogen heat switch would be suitable for incorporation into a no-moving-part magnetic refrigerator operating between 7 and 25 K. The heat switch would provide a completely passive, one-way thermal path between the magnetic material and the heat sink. An estimate of the refrigerator performance degradation was made and compared with an ideal case for which no thermal path was incorporated. These data can be used to compare this thermal configuration with other concepts.

REFERENCES


