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A study of the thermal behavior of a nitrogen heat pipe for a wide range of heat loads at several filling ratios

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Abstract. The primary purpose of this research was to investigate the thermal behavior of cryogenic heat pipes with a wide range of heat loads, from normal heat pipe operation to film boiling states. The heat pipes tested are commercially available from Fujikura Electronics (Thailand) Ltd., and they were originally designed and optimized for room temperature applications using water as a working fluid. We replaced the water with nitrogen as a working fluid, and we operated it as a cryogenic heat pipe at a liquid nitrogen temperature. The size of the tubular copper heat pipe was 6 mm in the outer diameter and 200 mm long, with a wick structure composed of fine axial grooves and sintered fine metal particles. The performance of the heat pipes with different liquid filling ratios was examined. The effective thermal resistance and the axial temperature distribution were measured for a wide range of heat loads at several filling ratios. The thermal behavior in the film boiling and the local dry states for large heat input were also examined for the potential application to a heat pipe heat switch.

1. Introduction

A heat pipe is a tubular device with an inner surface lined with a porous wick; it can rapidly transfer heat away from a source to a sink with higher efficiency than a solid conductor. The wick is saturated with a working fluid and the remaining volume of the tube has the working fluid vapor. The length of a heat pipe is divided into three parts: the evaporator section, the adiabatic section, and the condenser section. Heat applied externally to the evaporator section is conducted across the pipe wall and the wick structure, where it vaporizes the working fluid. The pressure difference between the evaporator and the condenser sections drives vapor from the evaporator section to the condenser section where it condenses and releases the latent heat of vaporization to a heat sink. The capillary pressure pumps the condensed liquid back to the evaporator for re-evaporation. This process continues if the flow passage for the working fluid is not blocked and sufficient capillary pressure is maintained [1].

Muniappan et al. [2] studied a cryogenic grooved heat pipe 180 mm long, 12.72 mm in diameter, and 1.34 × 0.84 × 0.6 (depth) mm of the trapezoidal axial grooves wick using nitrogen as a working fluid. The maximum heat transport capability was 3 W. In the present study, the wick structure of the heat pipe was a composite wick structure of grooved and sintered fine powder copper with smaller effective capillary radius. This can improve the capillary pumping pressure [3], increasing the maximum heat transport capability, even though the heat pipe has a smaller diameter.

This study used commercially available heat pipes. Repurposing commercially available heat pipes to cryogenic heat pipes avoided the use of specially designed and fabricated heat pipes for cryogenic temperatures. In addition, the cost for a new design is higher than a repurposed design for commercially available heat pipes, which are readily available and have uniform performance quality due to mass production. The objectives of this study were to investigate the performance of a commercially available heat pipe as a cryogenic heat pipe under liquid nitrogen temperatures and to test the design of a cryogenic heat switch that uses the dry-out behavior of heat pipes. When the heat switch was switched on, it worked like a heat pipe with good heat transfer; when it was switched off and it was dry-out state, it had a large thermal resistance.

2. Experimental Apparatus

The experimental apparatus for the cryogenic heat pipe is shown in Figure 1. It consisted of a cooling unit, a pressure unit for evacuation and working fluid filling for the heat pipe, and a data acquisition unit to control the heat input and to acquire the output from the temperature sensors, the pressure sensor (P1 for heat pipe and P2 for cryostat), and the heater's power supply.

The cryostat was stainless steel. The cylindrical liquid nitrogen bath, which was welded to the top of the cryostat flange via a neck tube, was made of stainless steel, and it was located inside the cryostat. The copper bottom of the cylindrical nitrogen bath was brazed with the stainless-steel side surface of the bath to use the copper's high thermal conductivity to maintain the condenser section at liquid nitrogen temperature. The stainless-steel capillary tube was used as an evacuating and filling line to the heat pipe. The outer surfaces of the liquid nitrogen bath, the stainless-steel capillary tube, and the heat pipe were wrapped with 30 layers of multilayer insulation (MLI). The top cryostat flange had two 22-pin connectors linking the lead wires of the heater and the temperature sensors.

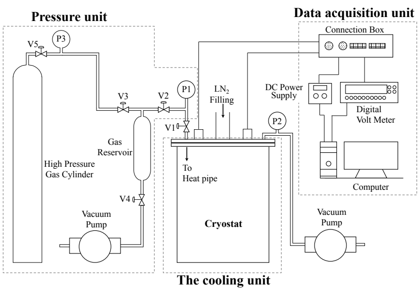


Figure 1. Experimental apparatus

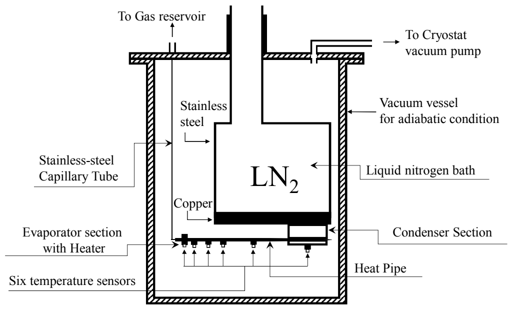


Figure 2. Cross-section of the cryostat

The heat pipe was attached to the bottom of the liquid nitrogen bath with copper blocks, as shown in Figure 2. The lengths of the copper blocks for the heater and the condenser, which were connected to the liquid nitrogen bath, were regarded as the lengths of the evaporator and the condenser sections, respectively. To avoid thermal shrinking, only the copper block for the condenser section was fixed. The axial temperature distribution along the external surface of the heat pipe was measured by six temperature sensors (T_1 to T_6) with ±0.25 K accuracy (Lake Shore-DT-670 Silicon Diodes). The temperature sensors were mounted at various locations, as shown in Figure 3. The wick structure of the heat pipe and its dimensions are shown in Figure 4. A cartridge-type heater with maximum input voltage of 25 W and 120 V, respectively, was installed at the evaporator section to provide heat to the heat pipe.

3. Experimental Procedure

A leak test of the experimental apparatus was performed using a helium leak detector to ensure isolation of the heat pipe from the surrounding environment. Heat input was added to the evaporator section while the heat pipe was evacuated. We observed the temperature change between the evaporator and the condenser during the experiment. There was 0.4 W of parasitic heat input by conduction through the containing tube wall, stainless-steel capillary tube, and instrument wires, and by radiation, which were included in the total heat input to the heat pipe.

Table 1. Specifications of the heat pipe

Parameter	Value (designed)
Container wall material	copper
Working fluid	nitrogen
Outer diameter (mm)	6.0
Inner diameter (mm)	5.4
Heat pipe length (mm)	200
Evap., Adia., Cond. Lengths (mm)	15, 120, 65
Wick	grooved - sintered copper powder
Wick structure thickness (mm)	0.7
Number of grooves	50
Sphere radius of the sintered copper power (μm)	50
Permeability (m^2)	2.40 x 10^{-11}
Porosity of wick	0.57

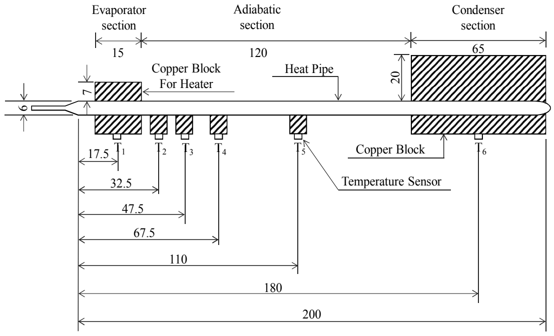


Figure 3. Schematic diagram of the heat pipe with temperature sensor locations. (All dimensions are in mm.)

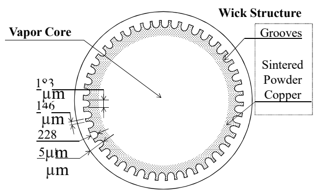


Figure 4. Cross-section of the heat pipe

The heat pipe and all the connecting tubes were evacuated and flushed three times with nitrogen at room temperature to remove any non-condensable foreign gas from the heat pipe. The cryostat was continuously evacuated during the experiment, which was started 24 hours after the start of the evacuation process. Nitrogen gas from the high-pressure gas cylinder was charged into the gas reservoir (V1 and V4 were closed) at P_i pressure; then V3 was closed. The pressure P_f was also measured after V1 was opened and liquid nitrogen was transferred into the liquid nitrogen bath. The liquid nitrogen cooled the heat pipe to approximately 78 K within 2-hours; then V2 was closed. The measurement data of $P_i - P_f$ was used to calculate the filling ratio f_r of the working fluid, which is the amount of condensed liquid nitrogen saturated in the wick structure divided by the void volume of the wick structure, as defined by:

$$f_r = \frac{\frac{(P_i - P_f)V_r}{R_V T} - \rho_v A_v L}{\varepsilon \rho_l A_w L}, \tag{1}$$

where V_r is the volume of the gas reservoir (0.3 L), R_v is the gas constant of nitrogen, ε is the porosity of the wick, T is the temperature, L is the total heat pipe length, ρ_v and ρ_l are the vapor and liquid densities of nitrogen, respectively, and A_v and A_w are the cross-sectional areas of the vapor core and wick structure, respectively.

The experiment was conducted for filling ratios 40 %, 80 %, 90 %, 100 %, 110 %, and 120 % to investigate the thermal behavior of the heat pipe for different f_r. The heat input was increased to 5.5 W by 0.1 W increments. The pressure inside the heat pipe and the wall temperature of the heat pipe were recorded after a steady state was reached for each heat input step. Thermal resistance of the heat pipe, R_{th}, indicated the heat transfer performance of the heat pipe. It was calculated as the temperature difference between the evaporator section and the condenser section divided by the heat input Q:

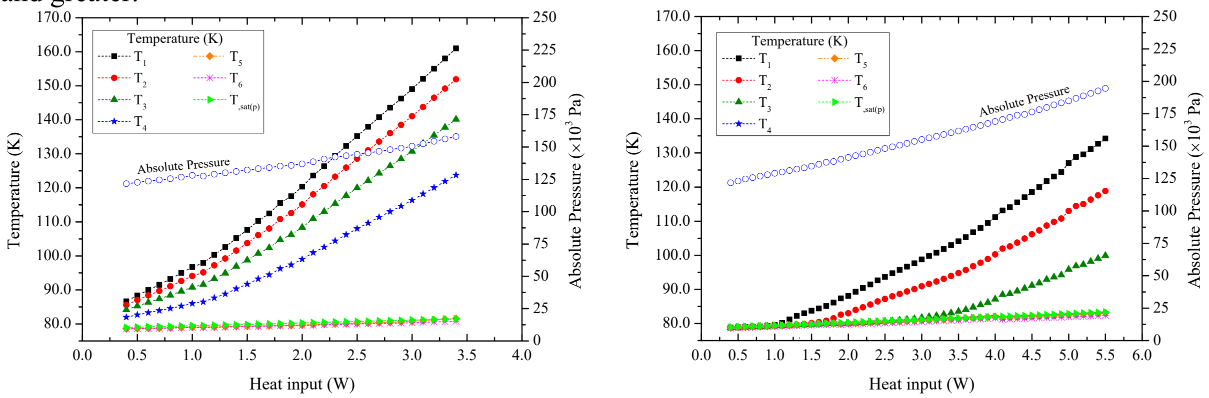
$$R_{th} = \frac{T_e - T_c}{Q}, \tag{2}$$

where Q is the heat input (W), and T_e and T_c are the wall temperatures (K) of the evaporator and the condenser, respectively.

4. Results and discussion

4.1. Temperature variation of the heat pipe

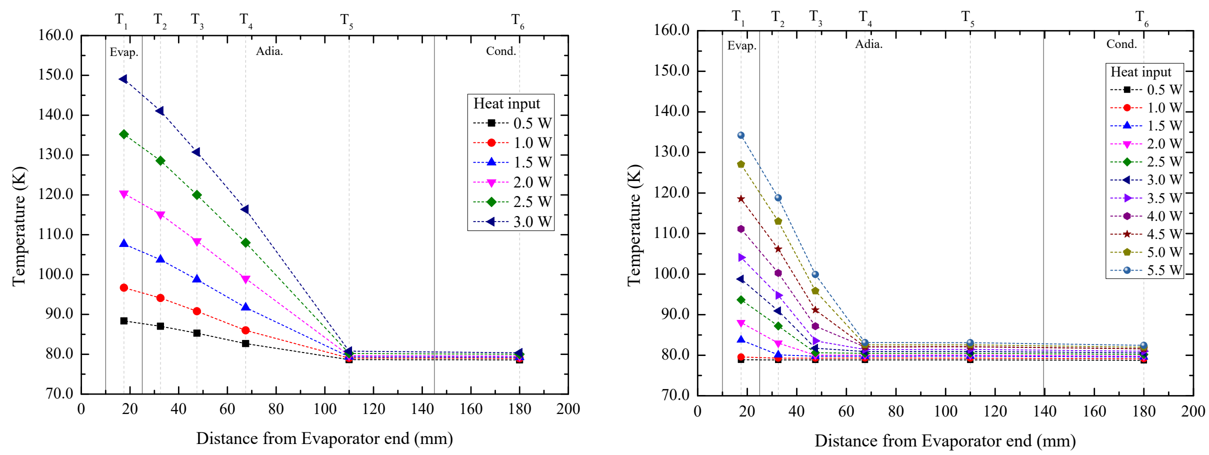
Temperature variations of the heat pipe, measured at several locations, are shown in Figure 5 as a function of Q for two filling ratios. The large temperature increase for the 40 % filling ratio (Figure 5(a)) was primarily due to an insufficient volume of the working fluid. This result indicated that the wick structure was saturated with condensate liquid nitrogen only in the condenser section side of the heat pipe. However, all six temperatures were equal to the saturated temperature of nitrogen for the 80 % filling ratio (Figure 5(b)) when Q < 1.0 W, which indicated that the heat pipe was operating normally. The evaporator temperature gradually increased for Q greater than 1.0 W. The adiabatic section temperatures (T_2 and T_3) gradually increased when Q was greater than 2.0 and 3.0 W, respectively. However, the other two adiabatic section temperatures (T_4 and T_5) remained equal to the saturated temperature of nitrogen. The pressure inside the heat pipe gradually increased from 140 kPa to 200 kPa with Q increased along the vapor pressure curve. The evaporator temperature started to increase above 2.5 W for the 90 % filling ratio (figure not shown). All six temperatures along the heat pipe were approximately equal to the saturated temperature for Q < 5.5 W for filling ratios of 100 % and greater.



(a) 40 % filling ratio (b) 80 % filling ratio

Figure 5. Variations of the wall temperature and absolute pressures

4.2. Temperature distribution along the length of the heat pipe



(a) 40 % filling ratio (b) 80 % filling ratio

Figure 6. Wall temperature distribution along the length of the heat pipe

The temperature rise throughout the adiabatic section was high with the 40 % filling ratio, even with low Q (Figure 6(a)). However, the temperature rises throughout the adiabatic section (T4 and T5) and the condenser section (T6) were relatively small for the 80 % (Figure 6(b)) and 90 % filling ratios when Q was less than 1.0 W and 2.5 W, respectively. Temperatures along the heat pipe and the saturated vapor temperature were approximately equal for filling ratios of 100 % and greater with Q < 5.5 W.

4.3. Overall thermal resistance

The overall thermal resistance of the heat pipe is plotted as a function of Q for several filling ratios (Figure 7). The calculation of R_{th} of a copper tube and a copper rod used the same diameter and length as the heat pipe. The experimental results of R_{th} of the heat pipe without working fluid filling was consistent with the thermal resistance of a copper tube. The experimental results of the 40 % filling ratio was slightly less than the 0 % filling ratio due to thermal conduction through the liquid nitrogen inside the heat pipe. For the 80 % filling ratio and greater, R_{th} decreased rapidly, which indicated that the heat pipe was operating normally.

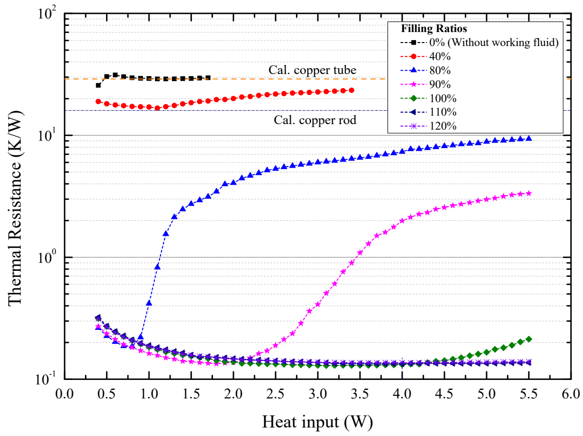


Figure 7. Overall thermal resistance of the heat pipe at different heat inputs for several filling ratios

5. Summary

Based on the experimental results, we concluded that it is possible for commercially available room temperature heat pipes (designed with grooves and sintered powder copper wick structure) to achieve relatively high effective thermal conductivity in the operating temperature range when the working fluid is replaced with nitrogen, though the performance was not optimized for the temperature range. For the heat pipes tested in this study, the 40 % filling ratio of the working fluid was insufficient for normal heat pipe operation, and the overall thermal resistance was approximately equal to the thermal resistance of heat pipes without working fluid. The overall thermal resistances of the heat pipe were approximately 200 and 100 times lower than a simple copper tube and a copper rod at 80 K, respectively, for heat inputs less than 0.8 W, 2.0 W, and 3.5 W (for 80 %, 90 %, and 100 % filling ratios, respectively) and heat inputs greater than 5.5 W (filling ratios of 110 % and 120 %). As the fill ratio increased, the onset of film boiling started at higher heat loads. Since the condenser section was in thermal contact with the liquid nitrogen bath (even for film boiling and local dry states for large Q values), the evaporator temperature did rise precipitously.

Acknowledgements

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References

[1] Chi S W 1976 *Heat pipe theory and practice* (United States: McGraw-Hill)
[2] Muniappan S K *et al* 2012 *Therm. Sci.* **16** 133
[3] Faghri A 2014 *Frontiers in Heat Pipes.* **5** 1