A study of thermal performance change of cryogenic heat pipes by wick structures for wide range of working fluid filling ratio

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Abstract. In this research the performances of nitrogen cryogenic heat pipes with different wick structures were compared under the condition of a wide range of heat load and several filling ratios. The heat pipes tested are commercially available ones, which were originally designed for room temperature applications using water as a working fluid. For the present research, working fluid, water, was replaced by liquid nitrogen. Heat pipe samples with three types of wicks, namely, axial grooves (G), sintered metal (S), and the combination of them (SG), were tested for comparing the thermal performance of the heat pipes with different wick structures, which is characterized by the effective thermal resistance $R_{th}$. The experimental data of the maximum heat transport capability $Q_{max}$ was compared with theoretical predictions on the basis of the capillary limit for each wick structure. Under high filling ratio condition, wide range of heat load was supplied to investigate the variation of the thermal resistance including such operations as dry-out state and the state where liquid puddle was formed in the pipe. The result indicates that the thermal performances of heat pipes with three kinds of wick structures are ordered as follows in descending order; SG, S and G wick.

1. Introduction

Conventional cooling methods have been used for the thermal management in electronic devices. However, failures in electronic circuits frequently occurred by excessive temperature rise due to low limitation of cooling capability. In order to solve this kind of problem, heat pipes have been widely used for high heat flux operation as heat pipes are highly conductive heat transfer devices using the latent heat of working fluid for efficient heat transfer over a very small temperature drop. A heat pipe is a tubular device with an inner surface lined with a porous wick, which can rapidly transfer heat away from a source to a sink with higher efficiency than through solid conductor. The wick is saturated with working fluid and the remaining volume of the tube is occupied with working fluid vapor [1].

There are many researches on heat pipes that use water as a working fluid for room temperature applications [2-5]. Sakulchangsatjatai et al. [2] studied, especially, a miniature heat pipe, 200.0 mm long, 6.0 mm in diameter with sintered copper powder as a wick structure. Their heat pipes had three
sections, 15, 100, 70 mm in lengths of the evaporator, the adiabatic and the condenser sections, respectively. By the application of the heat pipe, even 40 W of heat could be applied while maintaining the temperature lower than 60 °C. Yong Li et al. [3] fabricated and tested the heat pipes with four types of wicks, namely, axial groove (G), sintered metal (S), sintered-groove composite (SG) and groove with half sintered length (SGH). They found that the heat pipe with SG wick demonstrated the best thermal performance among them.

Cryogenic heat pipes have attracted considerable attention in the area of spacecraft cooling and high-temperature superconducting magnets cooling due to its high reliability, small size, low weight penalty, no need of extra electric power, zero maintenance and ability to perform satisfactorily even under zero gravity environment [6-9]. Muniappan et al. studied start-up characteristics [8] and the thermal performance of an axially grooved cryogenic heat pipe [9] with the condenser connected to the liquid nitrogen bath externally. The experiment was successfully conducted, and it resulted in 2.9 times improvement over simply utilizing metal conductor for cooling by using a copper rod at 100 K.

Based on a capillary wick design study [10], axial groove wick was formed by the extrusion of a groove-shaped plug with the same inner radius. Since the width of the grooves are generally larger than other wicks, the capillary pumping pressure is quite small. On the other hand, the permeability and the effective thermal conductivity are rather high. Sintered metal wicks are manufactured by packing tiny metal particles between the inner heat pipe wall and a mandrel in powder form. The capillary pressure developed by the sintered metal wick (and therefore capillary limit) which can be more readily predicted is generally large, but the permeability is rather low. Sintered-groove composite wicks employs the benefits of having small pores for generating high capillary pumping pressure by sintered wicks and having large pores for increasing the permeability of the liquid return path by groove wicks. The thermal behavior of heat pipes is commonly investigated by those with working fluid full in wick structure (100%) and increasing of the heat input at the evaporator until the evaporator temperature starts to increase rapidly. Then the experiment is terminated.

It has been already demonstrated in our previous study that the commercially available H$_2$O heat pipes can be used in the cryogenic temperature range by changing the working fluid to nitrogen. Thus, we may use the heat pipes as standard cryogenic heat pipes by changing the working fluid to cryogenic liquid. In the present study, the heat pipe was tested for several liquid filling ratios and wide range of heat load to investigate the thermal behavior even including the dry-out state. In order to make comparison with theoretical prediction for the thermal performance, the data was accumulated by measuring the thermal performance under various experimental conditions (wick structure, working fluid, filling ratio). Three types of wick structures, that is G, S, SG, were tested to investigate the performance change under liquid nitrogen temperature. In addition, the heat pipe with SG wick was tested further for the case of argon as a working fluid to compare the performance of different working fluid. The performance test under excess (over 100%) liquid fill condition was also conducted to consider the unusual thermal resistance state by regarding the liquid distribution inside heat pipe.

2. Theoretical prediction of maximum heat transport capability

In order to theoretically estimate the maximum heat transport capability, the capillary limit is usually selected as a measure, because it gives the lowest value compared with other limitations, namely, the boiling, the entrainment, the sonic and the viscous limits [1]. The equation of the maximum heat transport capability is;

$$Q_{\text{max}} = \frac{P_c}{(F_l + F_v)L_{\text{eff}}}$$  \hspace{1cm} (1)

Here, the maximum effective capillary pressure is $P_c = \frac{2\mu}{r}$, the frictional coefficient for the liquid flow is $F_l = \frac{\mu_l}{k \cdot A_v \cdot \rho_l}$, the frictional coefficient for the vapor flow is $F_v = \frac{32 \mu_v}{d_v \cdot A_v \cdot \rho_v}$, and the effective length of heat pipe; $L_{\text{eff}} = L_e + L_a + L_c$. 

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where $\sigma$ is surface tension, $r_c$ is effective capillary radius, $\mu_l$ and $\mu_v$ are liquid and vapor kinematic viscosities, $\rho_l$ and $\rho_v$ are liquid and vapor densities, $\lambda$ is latent heat of vaporization, $K$ is permeability, $A_w$ and $A_v$ are cross-sectional area of wick and vapor core of the heat pipe, and $d_v$ is vapor core diameter.

3. Experimental Set up
The experimental set up was constructed on the basis of the one described in our previous paper [11] with some modifications, especially for the Argon-heat pipe test. The experimental apparatus for the cryogenic heat pipe is shown in figure 1. It consists of the pressure unit, the cooling unit and the data acquisition unit.

![Figure 1. Experimental apparatus](image)

**Figure 1.** Experimental apparatus

![Figure 2. Heat pipe and locations of temperature sensors (in mm.)](image)

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![Figure 3. The condenser section of (a) N$_2$-heat pipe, (b) Ar-heat pipe](image)

**Figure 3.** The condenser section of (a) N$_2$-heat pipe, (b) Ar-heat pipe

The temperature sensors with ± 0.25 K of an accuracy were mounted as shown in figure 2. Six temperature sensors (T$_1$ to T$_6$) are Lake Shore DT-670 Silicon Diodes. The calibrated temperature sensor (T$_1$) was installed at the middle of the evaporator section. The second to the fifth temperature sensors (T$_2$, T$_3$, T$_4$, T$_5$) were all installed in the adiabatic section. The sixth temperature sensor (T$_6$) was installed at the middle of the condenser section. The condenser sections are shown in figure 3(a) for the Nitrogen-heat pipe and in figure 3(b) for the Argon-heat pipe. For the Nitrogen-heat pipe, indium foil was used on contact surface area between the condenser and the liquid nitrogen bath and between the condenser and the heat pipe to improve the heat transfer performance.

For the Argon-heat pipe experiment, the additional calibrated temperature sensor (T$_7$) and the film heater were mounted on the condenser section in order to control the temperature of this section at 87.3 K, which is higher than the normal liquid nitrogen temperature. For the sake of this, a sheet of stainless-
steel thin plate was inserted between the condenser block at 87.3 K and the bottom plate of the nitrogen bath at 78 K as a thermal barrier.

The three-types of wick structures and its dimensions are shown in figure 4 and table 1, respectively.

| Parameter                        | Value (designed) |
|----------------------------------|------------------|
| Tube wall material               | Copper           |
| Working Fluid                    | Nitrogen or Argon|
| Outer Diameter (OD)              | 6.0 mm           |
| Heat pipe length                 | 200 mm           |
| Evap., Adia., Cond. lengths      | 15, 120, 65 mm   |
| Thickness of wall (Axial groove wick) | 0.26 mm |
| Groove depth                     | 0.20 mm          |
| Groove width                     | 0.27 mm          |
| Number of grooves                | 50               |
| Thickness of wall (Sintered metal wick) | 0.20 mm |
| Thickness of sintered wick       | 0.80 mm          |
| Sphere radius of copper powder, \( r_s \) | 100 ± 50 µm |
| Permeability (Grooved, Sintered) | 3.0 x 10^{-9}, 6.67 x 10^{-11} m² |
| Porosity of wick, \( \varepsilon \) (Grooved, Sintered) | 0.82, 0.57 |

(a) Groove (b) Sintered metal (c) Sintered metal-Groove

Figure 4. Wicks structures.

4. Experimental Procedure

To ensure isolation of the heat pipe from the surrounding environment, a leak test of the cryostat and the tube connection parts was performed by using a helium leak detector. The cryostat was continuously evacuated down to 1.0 x 10^{-2} Pa that was measured by \( P_2 \). In order to evaluate the heat leak from the room temperature environment, liquid nitrogen was transferred into the liquid nitrogen bath and heat input was added to the evaporator section while the heat pipe was continuously evacuated. The temperature difference between the evaporator and the condenser during the experiment was measured. It was found there was approximately 0.1 W of heat input as the parasitic heat input from the room temperature environment mostly by radiation, which was included in the total heat input to the heat pipe during the experiment.

The heat pipe and all the connecting tubes were evacuated and flushed five times with gas phrase (N₂ or Ar) of working fluid at room temperature to remove any non-condensable foreign gas from the heat pipe. Gas phrase of working fluid from the high-pressure gas cylinder was charged into the gas reservoir (\( V_1 \) and \( V_4 \) were closed) at \( P_i \) pressure; then \( V_3 \) was closed. Liquid nitrogen was transferred into the liquid nitrogen bath. The liquid nitrogen cooled the heat pipe to approximately 78 K and the pressure of the cryostat was down to 5.0 x 10^{-4} Pa. After \( V_1 \) was opened, the pressure of gas reservoir was changed to \( P_f \). Before starting the experiment, \( V_2 \) 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:

\[
f_r = \frac{(\rho_f - \rho_l) V_r}{\frac{\rho_i}{R_i} A_{vL} T} - \rho_i A_{vL},
\]

where \( V_r \) and \( T \) are the volume and temperature of the gas reservoir (0.3 L), \( R_i \) is an individual gas constant, \( \varepsilon \) is the porosity of the wick, \( L \) is the total heat pipe length.

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 \( Q \):

\[
R_{th} = \frac{T_i - T_o}{Q},
\]

where \( Q \) is the heat input (W), and \( T_i \) and \( T_o \) are the wall temperatures (K) of the evaporator and the condenser, respectively.
5. Results and discussion

5.1. Temperature and pressure variations of the heat pipe

The variations of the temperatures $T_i$ to $T_e$, measured at several locations, and the pressure $P$ inside the heat pipe as a function of $Q$ are shown in figure 5 for two working fluids. The saturated vapor temperature $T_{sat}P$ converted from the pressure $P$ is also presented there to compare with the experimental temperature data. The experiment was started after cooling down from room temperature to LN$_2$ temperature. As seen in figure 5(a), the heat input $Q$ was step by step increased, and after 10.0 W of the heat input, $T_f$ started to increase with the lapse of time and the pressure started fluctuating due to the onset of film boiling. Furthermore, $T_i$ rapidly increased for $Q$ larger than 13.5 W, which indicates that dry-out occurred in the evaporator section. It should be noted for the argon heat pipe, shown in figure 5(b), that 7.0 W of heat input was the upper limit of the heat input to the condenser section in the normal heat pipe operation, as the condenser section $T_c$ could no more be maintained at 87.3 K. Once $Q$ was greater than 7.0 W, all the temperatures started to increase, and the thermal performance was highly degraded. Thus, the heat transport capability could not be obtained for the Ar heat pipe in this series of experiment. It is, in fact, suggested the thermal barrier in the condenser section should be replaced by thinner one or one with lower thermal resistance for further experiments.

![Figure 5](image.png)

**Figure 5.** Temperature and pressure variations of the heat pipe at 100% fill with change of heat input $Q$. (a) N$_2$-heat pipe, (b) Ar-heat pipe.
5.2. Effect of filling ratio

The correspondence between the model illustration for the liquid distribution in the heat pipe and the experimental result of the thermal resistance is shown in figure 6 in the case of SG wick.

Figure 6. Schematic illustration of working fluid distribution in the N₂-hear pipe with SG wick.

For low filling ratio, dry-out occurred even at low heat input $Q$ because liquid amount was insufficient in the evaporator section; cases ① and ②. Under proper filling (100%) condition, the minimum value of the thermal resistance was achieved during the normal heat pipe operation; ④. Film boiling started when the heat load was applied between 12.0-13.5 W; ⑤. Local dry-out occurred in the evaporator section after 13.5 W of heat input that caused the temperature in the evaporator section to rapidly rise, and thus the thermal resistance also started increasing; ⑥. For excess filling, $R_\text{th}$ is larger than that in case of normal heat pipe operation, because in the condenser portion heat is transferred by thermal conduction through existing excess liquid column to the condenser; ⑦.

5.3. Overall thermal resistance

The overall thermal resistance of the heat pipe is plotted as a function of $Q$ in figures 7 and 8, where the calculation result of $R_\text{th}$ of a copper tube and a copper rod at LN₂ temperature with the same diameter and length as the heat pipe is added. At the beginning of experiment when the heat input is very small, $R_\text{th}$ decreases as the heat input increases. This confirms previous findings in many studies of the thermal behavior of heat pipe, which can be explained that at low heat input/temperature, the surface tension on the solid-liquid interface is high and thus somewhat inhibits evaporation, and this effect will be subsequently reduced with the increase of the heat input. Moreover, the film thickness of liquid layer is high at low heat input. This creates high thermal resistance, but the film thickness reduces with the increase of the heat input [12]. The value of $R_\text{th}$ decreases to the minimum value during the normal heat pipe operation, and increases again due to film boiling or even dry-out occurrence. The experimental results of $R_\text{th}$ of the heat pipe with three type of wick structures under similar filling ratio condition were compared in figure 7. The lowest $R_\text{th}$ was equally 0.25 K/W for three types of wicks during the normal heat pipe operation. On the other hand, $Q_{\text{max}}$ is significantly different, 3.0 W, 11.0 W, and 12.5 W for G, S, and SG wicks, respectively. In figure 8, the experimental results of $R_\text{th}$ of Ar and N₂ heat pipes with SG wick and similar filling ratio were compared. It is found that both the value $R_\text{th}$ are almost equivalent, but $Q_{\text{max}}$ value of N₂ is slightly larger than that of the Ar heat pipe.
Figure 7. Thermal resistance variation of N$_2$ heat pipe with different wick structure

Figure 8. Comparison of the thermal resistance between N$_2$ and Ar heat pipes with SG wick

5.4. Maximum heat transport capability

The maximum heat transport capability $Q_{\text{max}}$ was experimentally determined as the upper limit of the heat input above which the evaporator temperature started to rise in a large scale. In the experiment, the heat input was stepwise increased until the local dry-out state was detected, then $Q_{\text{max}}$ is determined.

Figure 9. The maximum heat transport capability as a function of filling ratio.

In figure 9, the experimental result and the theoretical prediction of $Q_{\text{max}}$ were plotted as a function of filling ratio for different wick structures and working fluid. It is found that the experimental results of G wick and S wick with 100 $\mu$m of average sphere radius agree with theoretical calculation data [1]. In order to theoretically estimate $Q_{\text{max}}$ of the sintered-groove composite wick (SG), it is assumed that the maximum effective capillary pressure $P_e$ exclusively results from sintered metal wick and the frictional
coefficient for the liquid flow $F_l$ is from axial groove wick, respectively. It is found in figure 9 that the assumption for the capillary limit in the case of SG wick leads to considerable success in the theoretical prediction of $Q_{\text{max}}$ for the $N_2$ heat pipe with SG wick. The difference in $Q_{\text{max}}$ between $N_2$ and Ar heat pipes with the identical SG wick may be explained by the difference of fluid property such as the surface tension, the liquid and vapor densities, the liquid and vapor viscosities and the latent heat of vaporization. However, the experimental value of $Q_{\text{max}}$ for the Ar heat pipe is considerably smaller than the theoretical value for the capillary limit. In fact, this was only a preliminary attempt for the case of Ar-heat pipe, and the experimental examination has not been sufficiently done due to the difficulty in controlling the heat input to maintain the condenser temperature at liquid Ar temperature, 87.3 K. We would like to leave the verification to future experiments.

6. Conclusion
The results from this study suggest as follows: in order to normally operate the heat pipe, liquid should be filled at least 50%. The minimum value of $R_{th}$ is 0.25 K/W for three types of wicks during the normal heat pipe operation. The performance rank by the wick structures on the basis of $Q_{\text{max}}$, is as follows in descending order; SG, S and G. The result of thermal performance test with the same wick structure (SG) indicates that both $N_2$ and Ar heat pipes have nearly identical $R_{th}$ but the results are significantly different in $Q_{\text{max}}$. In case of the $N_2$-heat pipe with S or SG wick structure at sufficient liquid fill, $Q_{\text{max}}$ can be extend to 12.0 W. It is interesting to note in the case of considerably high liquid fill, $R_{th}$ is constant without dry-out even at high heat input although $R_{th}$ is higher than that in case of normal heat pipe operation for 100 % fill because of the appearance of excess liquid column in the condenser section of heat pipe. The experimental results for homogeneous wick structure, G and S wicks, are in complete agreement with the theoretical calculation data. The satisfactory agreement of the experimental results with the theoretical prediction for the case of composite wick structure, SG wick, seems to confirm our hypothesis for the wick function.

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