JOURNAL OF THERMOPHYSICS AND HEAT TRANSFER
Vol. 7, No. 1, Jan.–March 1993

Freeze/Thaw Characteristics of a Copper/Water Heat Pipe: Effects of Noncondensible Gas Charge

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The freeze/thaw characteristics of a copper/water heat pipe of rectangular cross section were investigated experimentally to determine the effect of variations in the amount of noncondensible gases (NCG) present. The transient internal temperature profiles in the liquid channel are presented along with contours of the frozen fluid configuration obtained through visual observation. Several interesting phenomena were observed, including total blockage of the vapor channel by a solid plug, evaporator dryout during restart, and freezing blowby. In addition, the restart characteristics are shown to be strongly dependent upon the shutdown procedure prior to freezing, indicating that accurate prediction of the startup or restart characteristics requires a complete knowledge of the thermal history. Finally, the experimental results indicate that the freeze/thaw characteristics of room temperature heat pipes may be significantly different from those occurring in higher temperature, liquid metal heat pipes due to differences in the vapor pressures in the fluid's frozen state.

Introduction

IN general, the fundamental principles involved in the steady-state and transient operation of heat pipes are relatively well understood.[1,2] However, several phenomenon commonly found in heat-pipe operations have not been investigated fully, including the transient processes of freezing and restart from the frozen state. Experimental data exist primarily from investigations conducted on liquid metal heat pipes, as these designs must at one point always start from the frozen state, and largely ignore the effect of the initial frozen working fluid configuration on restart. Although heat pipes that utilize working fluids such as water and ammonia would not freeze normally during standard operation or manufacturing, several situations arise where the freeze/thaw behavior is important.[3,4] For example, high-capacity, external artery heat pipes with ammonia as a working fluid have been considered for use as part of the Space Station Freedom heat rejection system. Many problems, such as those associated with zero-g priming, liquid/vapor passage blockage, and noncondensible gas formation either have been resolved or are currently under investigation. One potential problem, however, that of restart from a frozen state, has received little attention and only a limited amount of analytical and experimental data are presently available.[3] In order to restart the heat pipes from a frozen state, the freeze/thaw characteristics must be well understood. Also, an investigation of several deprimings in arterial, variable conductance heat pipes (VCHPs) flown aboard the communications technology satellite (CTS), indicated that the freezing and subsequent thawing of a portion of the condenser due to variations in the satellite viewing attitude, precipitated the formation of gas bubbles in the liquid arteries and resulted in depriming.[4] Avoiding conditions in which the condenser could freeze was seen as the optimal solution to the problem. Although practical for the CTS, attitude adjustments on other platforms such as Space Station could be very expensive and, for long-term missions, may not be feasible.

Three types of freezout that occur in heat pipes have been proposed and described by Antoniuk and Edwards[4]: suction freezout, freezing blowby, and diffusion freezout. Suction freezout is the freezing of the fluid and corresponding depletion of the available liquid inventory in the active region of the heat pipe. Freezing blowby corresponds to a phenomenon occurring when a complete solid blockage of the vapor and liquid passages is thawed. During melting of the blockage, a high-pressure region (evaporator) exists on one side of the blockage and a low-pressure region (condenser) exists on the other. When breakthrough of the blockage occurs, liquid is rapidly driven from the higher pressure evaporator into the lower pressure condenser region and may result in rapid dryout of the evaporator wicking structure. Diffusion freezout (also discussed by Edwards and Marcus[5]) is the freezing of the working fluid that diffuses from the vapor located in the condenser region into the portion of the condenser blocked by noncondensible gases.

The ability to restart a frozen heat pipe has been discussed by Chi[1] and Ivanovskii et al.[2] as being a direct function of the condenser heat removal rate, heat-pipe geometry, working fluid, and condenser-to-evaporator length ratio. Both sources also noted that the addition of noncondensible gases (NCG) would aid in the startup due to a progression of the melt front along the length of the heat pipe. It was hypothesized that the amount of noncondensible gas required for frontal heating and successful startup corresponded to the amount necessary to achieve a noncondensible gas pressure in excess of the working fluid saturation pressure at the melting temperature.[2] Neither source, however, discusses the process of the working fluid freezing, and both assume an essentially uniformly distributed working fluid in the discussions of frozen startup.

Experimental investigations of frozen startup have been conducted by several investigators, including Merrigan et al.[6] for a molybdenum/lithium heat pipe with varying radiation heat rejection rates during freezing of the working fluid. In this investigation, uneven rates were found to hinder startup due to partial dryout of the evaporator during the freezing process. Also, successful restart was found to be directly dependent on the heat rejection rate during startup. Startup of sodium and potassium heat pipes with instantaneous power

Presented as Paper 91-1402 at the AIAA 26th Thermophysics Conference, Honolulu, HI, June 24–26, 1991; received June 28, 1991; revision received Dec. 6, 1991; accepted for publication Dec. 10, 1991.

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additions have been investigated by Tolubinsky et al.[7] while Bystrov et al.[8] conducted experiments on vacuum and gas-loaded sodium heat pipes. A significant decrease in the startup time was noted by Ivanovskii et al.[2] for increasing amounts of gas loading in sodium heat pipes. Sockol[9] presented an analytical model for an argon-lithium system and predicted that startup is not initiated until the vapor pressure of the working fluid is on the same order of magnitude as the pressure of the initial gas charge. Analytical and experimental investigations for gas-loaded liquid metal heat pipes have been presented by Bystrov and Goncharov[10] for an argon/sodium system and by Ponnappan et al.[11] for a gas-loaded, sodium heat pipe. Addition of NCG was found to enhance the restart capabilities for each of the two respective systems.

Deverall et. al.[12] conducted an experimental investigation of frozen startup with a water heat pipe. Evaporator dryout was observed for an initially frozen heat pipe at −65°C and an evaporator heat addition of 10 W. Restart and isothermal operation was achieved only after the fluid in the condenser had melted and rewet the evaporator. A temperature gradient of approximately 100°C from the evaporator to the condenser was required to thaw the heat pipe completely. Abramenko et al.[13] performed an experimental investigation using an aluminum/ammonia heat pipe with radiation heat rejection, thus resulting in conditions similar to those found in spacecraft thermal control systems. Startup from the frozen state was obtained only after the radiative sink temperature was increased, thus reducing the heat rejection rate at the condenser.

Two recent numerical investigations have been presented that examine the startup of a frozen heat pipe.[14,15] The numerical model of Bowman[14] was compared with experimental data obtained for a copper/water heat pipe and a molybdenum/lithium heat pipe. A more extensive finite element numerical model has been presented by Jang et al.[15] in which both the free molecular flow and continuum flow regimes were examined. This model was used to simulate the startup of a frozen stainless steel/sodium heat pipe, but was not compared with any experimental data.

Overall, published information that describes the freeze/thaw characteristics of room temperature heat pipes are limited and additional investigations are required. Although the analytical/numerical models successfully predict the startup processes occurring in frozen heat pipes, the basic assumption is of a uniformly distributed frozen working fluid in the heat-pipe wick or liquid channel, thus ignoring the freezing process and the thermal history of the heat pipe. To understand the importance of the thermal history, two parameters must be investigated. The first of these concerns the final frozen configuration of the working fluid within the heat pipe and the second concerns the corresponding restart characteristics once the working fluid has frozen. Therefore, an experimental investigation of the freeze/thaw characteristics of a copper/water heat pipe was conducted to obtain freezing/restart data for a heat pipe utilizing a room temperature working fluid. Also, the effects of varying levels of NCG were specifically investigated herein, as all previous work revolving around frozen startup in the presence of NCG have shown a strong dependence on the level of the NCG charge.

Experimental Apparatus

The experimental apparatus used in this investigation consisted of a 2.0-m copper heat pipe with a 25.4-mm-wide by 11.0-mm-deep internal rectangular cross section, as shown in Fig. 1. A small step was machined in the internal channel to accommodate a 40-mesh copper wire screen placed between the vapor and liquid channels along with four additional layers of copper mesh beneath the top screen separating the liquid and vapor channels. A transparent lexan cover with a 1.58-mm o-ring was used to seal the heat pipe. The lexan cover provided the ability to visually observe the operation of the heat pipe during the freeze/thaw process. Prior to installation of the lexan cover, the heat pipe was rinsed several times with acetone to clean the wick and the internal surfaces of the heat pipe. After cleaning, the cover was installed and the heat pipe was evacuated and charged with 170 cc of distilled deionized water.

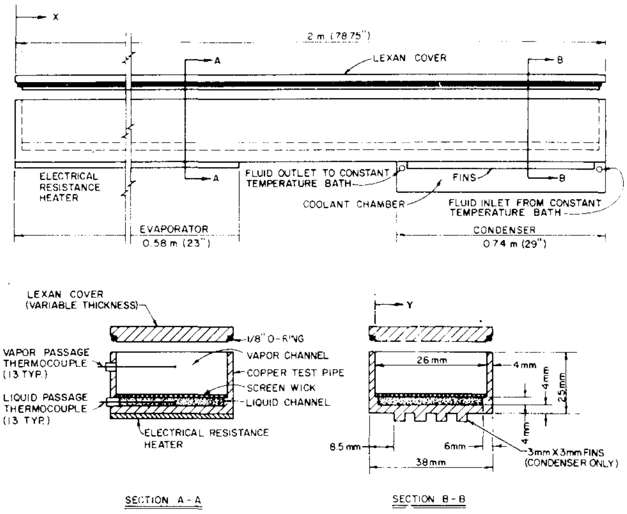

Fig. 1 Experimental heat pipe configuration.

Heat addition to the evaporator section was provided by a 0.58-m mica strip resistance heater. Power to the resistance heater was provided by a 120-V ac variable transformer and measured using a digital wattmeter. In this way, the input power could be measured and controlled to ±1.0 W. The condenser section was cooled by an ethylene glycol/water mixture and a constant temperature circulating bath, which pumped the coolant through a 0.74-m coolant chamber located along the base of the heat pipe condenser.

To monitor the internal temperature distributions within the heat pipe, 26 stainless steel sheathed copper-constantan thermocouples (AWG-30) were used, 13 located in the vapor channel and 13 in the liquid channel. Starting at 0.143 m from the leading edge of the evaporator in the liquid channel and 0.156 m in the vapor channel, the thermocouples were placed at 0.152-m intervals along the length of the heat pipe. Data acquisition was provided by a Dell 325D personal computer in conjunction with a Hewlett-Packard 75000 series B data acquisition system. The thermocouples along with the data acquisition system provided internal liquid and vapor temperatures accurate to within ±0.3°C.

The effects of convective heat transfer with the surroundings on the freezing and thawing processes were minimized by placing the heat pipe in a 0.152-m-diam, 3.0-m-long lexan vacuum chamber. A mechanical roughing pump was used to evacuate the chamber to a vacuum of less than 0.5 Torr. This vacuum was maintained and monitored throughout the tests and also helped reduce leakage of noncondensible gas into or out of the heat pipe during the tests.

To control the amount of noncondensible gas present during the tests, the vapor channel of the heat pipe was fitted with a three-way, high-vacuum valve at the end of the condenser section. To add the noncondensible gas charge, small amounts of room air were vented slowly into the heat pipe through this valve. To remove noncondensible gas, the condenser was first elevated 5 cm above the evaporator, then, with the coolant to the condenser off, 30 W of power was input to the evaporator. This heat input was maintained until the overall temperature of approximately 40°C was obtained, thus resulting in an increase in the internal heat pipe pressure. Once this temperature had been obtained and the noncondensible gas had collected at the condenser end, the valve to the vacuum pump was opened and the desired amount of noncondensible gas was evacuated. The heat pipe was then leveled and the temperatures allowed to stabilize. By using this method for NCG removal, loss of the working fluid was minimized by collecting the NCG at the condenser end of the heat pipe. Throughout testing, no degradation in performance was observed and no visual observation of working fluid depletion was found.

Experimental Procedure

To provide a common initial reference state, all of the tests were started from a uniform isothermal ambient (25°C) condition. The heat pipe was then brought to a steady-state condition with 160 W of power input and a condenser cooling-fluid temperature of 0°C prior to each freeze/thaw cycle. This condition was maintained for a minimum of 1 h to ensure that steady state had been obtained and to verify the amount of noncondensible gas present in the heat pipe. The initial noncondensible gas charge and the boundary between the noncondensible gas and the active condenser region could be identified visually by the condensation (in the form of a fine mist) on the inside of the lexan cover in the active condenser region. In the region containing noncondensible gas, no condensation occurred, thus making the gas/vapor interface very distinctive in nature. The coolant temperature provided to the condenser was then lowered to −10°C at a rate of −0.08°C/min and the heat pipe was again operated at steady-state for a minimum of 1 h to reconfirm the amount of noncondensible gas present.

The power to the evaporator was then shut off instantaneously and the heat pipe was allowed to freeze. Throughout the freezing process, continuous temperature measurements and visual observations of the freeze front of the working fluid were conducted and recorded. The freeze/thaw cycles for each case were repeated several times to verify the repeatability of the obtained results. Once an equilibrium state had been reached, startup of the frozen heat pipe was attempted. This consisted of initially adding varying amounts of evaporator heat input while maintaining a constant coolant flow to the condenser. The heat input was discontinued upon evaporator dryout or increased incrementally if an equilibrium condition was obtained, until the initial power input of 160 W for the cycle had once again been reached. Upon completion and verification of each freeze/thaw cycle, the amount of noncondensible gas was changed and the process was repeated.

Results and Discussion

The experimental results obtained in this investigation are divided into two separate sections for clarity. One describing the freezing characteristics and the other describing the restart characteristics corresponding to the different initial working fluid distributions resulting from the freezing tests.

Freezing

The results of the freezing tests for noncondensible gas lengths of 0.07 m (9.5% blocked condenser at initial steady-state condition), 0.30 m (40.5%), 0.36 m (48.7%), and 0.40 m (54%) are presented. These four NCG lengths resulted in marked differences and characteristics in comparison to each of the other NCG lengths. The initial and final liquid channel temperature profiles for noncondensible gas lengths of 0.07 and 0.36 m are presented in Figs. 2 and 3, respectively. The temperature profiles for NCG lengths of 0.30 and 0.40 m were essentially equivalent to those presented in Fig. 3 for the 0.36 m length and therefore, are not introduced.

Referring to Fig. 2 for a 0.07-m NCG length, at an elapsed time of 55 min the working fluid was confirmed visually to be evenly distributed throughout the wick and completely frozen. Intuitively, this should represent a steady-state condition with the working fluid distribution remaining unchanged once frozen. This was assumed to occur at 55 min, however, as seen by the temperature profiles for an elapsed time of 1300 min, this did not transpire. The temperature profiles exhibited a frontal heating mode in the evaporator with time until a steady-state temperature with the environment (coupled by radiation) of 10°C at the beginning of the evaporator was reached. The solid in the evaporator wick was observed visually to be gradually depleted during this time period (55–1300 min) until the wick in the evaporator and adiabatic sections was completely dried. It is important to note that no melting was observed during this time period. One possible explanation for the continued migration of fluid to the condenser is the vapor pressure difference of the solid due to the temperature gradient across the heat pipe (3.28 Torr at −4°C and 2.33 Torr at −8°C) was great enough to transport the fluid gradually by sublimation.

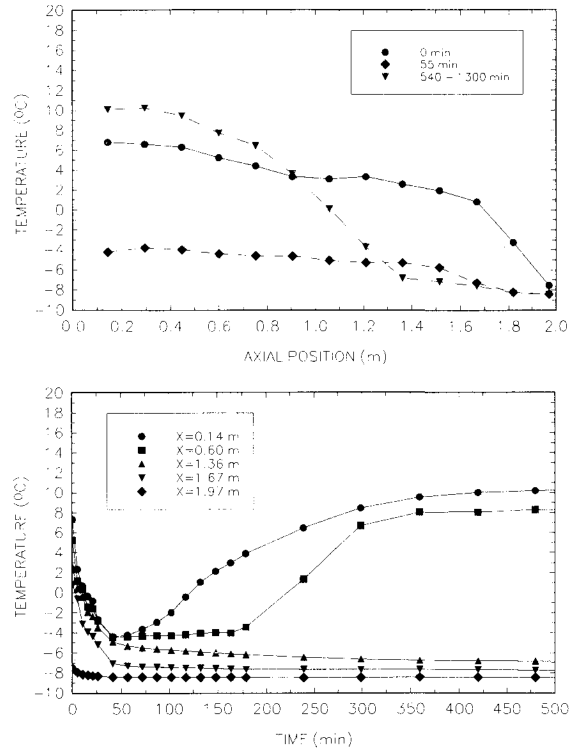

Fig. 2 Transient liquid channel temperatures for 0.07-m NCG length.

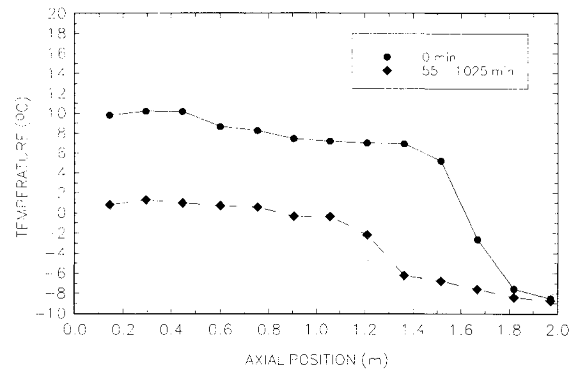

Fig. 3 Transient liquid channel temperatures for 0.30-m NCG length.

The liquid channel temperature distributions for a noncondensible gas length of 0.36 m, presented in Fig. 3, were significantly different from those obtained for a 0.07-m NCG length. The most noticeable differences between the temperature profiles for a low level of NCG (0.07 m) and those presented in Fig. 3 is that the fluid in the liquid channel was never frozen and the frontal heating along with the evaporator dryout was not present. The evaporator was confirmed visually to have remained primed and thawed.

Visually, for each test, layers of frozen working fluid were observed to form along the side walls of the heat pipe in the condenser region. These frozen fluid layers continued to increase in thickness with time and are presented by the contours in Figs. 4–6 for NCG lengths of 0.07, 0.30, and 0.36

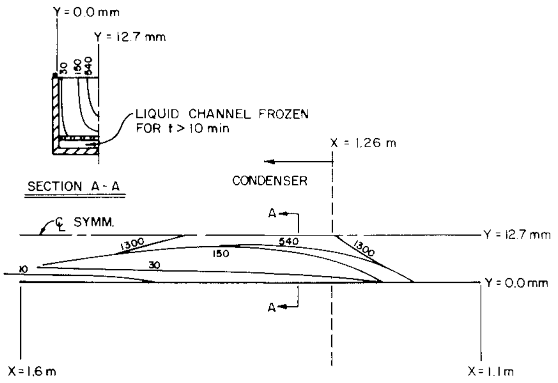

Fig. 4 Solid formation contours for 0.07-m NCG length.

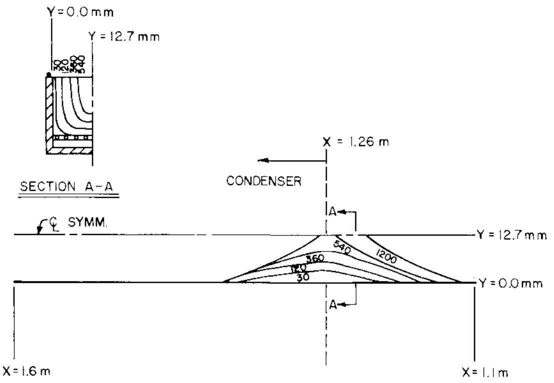

Fig. 5 Solid formation contours for 0.30-m NCG length.

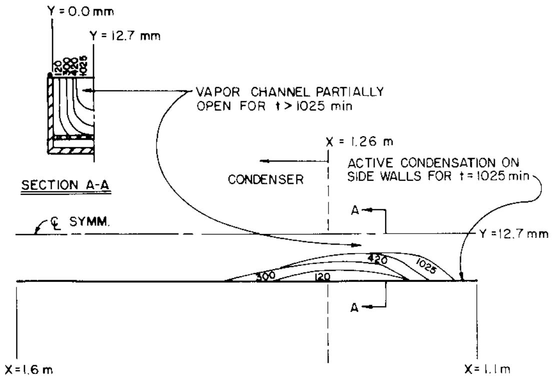

Fig. 6 Solid formation contours for 0.36-m NCG length.

m, respectively. With elapsed time, the solid formations resembled a converging-diverging nozzle and eventually closed for levels of NCG below 0.30 m, forming a complete blockage of the vapor channel. In comparing the solid formation contours of 0.30 and 0.36 m (Figs. 5 and 6), the blockage of the vapor channel for 0.36 m was not complete and an approximately 12-mm-wide by 5-mm-deep region was present along the length of the solid formation. For both levels of noncondensible gas, the heat pipe in the region from the evaporator to a position adjacent to the inlet of the blockage was observed to be active with very slow—but definitely present—dropwise condensation occurring along a 50-mm-long region in front of the blockage. Dryout of the evaporator did not occur because the liquid could return to the evaporator from the condensation zone adjacent to the blockage, thereby explaining the essentially isothermal region from the beginning of the evaporator to an axial position of approximately 1.05 m.

For an NCG length of 0.40 m, no blockage of the vapor channel or significant amounts of frozen working fluid were observed in the condenser, although the temperature distributions were essentially equivalent to those for NCG lengths of 0.30 and 0.36 m. The working fluid in the liquid channel was frozen from an axial position of 1.13 m to the end of the condenser section, with the remainder of the fluid in the liquid state. Once again, an active condensation zone was present in a 50-mm region adjacent to the beginning of the frozen portion of the liquid channel.

From the above results, it is apparent that the central location of the blockage is a weak function of the NCG charge in the heat pipe. After the power to the evaporator is discontinued, the evaporator temperatures decrease and, thus, the pressure in the heat pipe decreases as well. This pressure reduction results in the expansion of the NCG charge and requires the gas/vapor front to move accordingly. This expansion of the NCG charge may explain the weak function of the blockage location on the level of noncondensible gas charge.

Similarly, a direct correlation between the NCG charge and the magnitude of the vapor channel solid blockage is apparent. Although a flat-front approximation was used in this investigation to determine the amount of noncondensible gas present, the gas/vapor front contains a diffusion region where the noncondensible gas and working-fluid vapor phase are intermixed.[5] In this diffusion region, the concentration of vapor decreases from a maximum of the vapor side of the region to an essentially zero concentration at the gas side. As in the previous discussion of the location of the blockage, when the overall pressure in the heat pipe decreases, the NCG charge expands accordingly. This expansion forces the diffusion region at the front of the NCG to move closer to the inlet of the condenser and the adiabatic region. The percentage of the diffusion region exposed to the subfreezing condenser temperatures is then reduced and hinders a larger portion of vapor in the diffusion region from freezing on the heat-pipe walls, thus decreasing the magnitude of the vapor channel solid formation. Eventually as the NCG level is great enough, the diffusion region is forced entirely into the adiabatic region and prevents any solid formation on the heat pipe walls.

To stop the migration of the fluid and formation of the vapor channel blockage as well as to obtain a uniformly distributed working fluid for room temperature fluids, it may be required to freeze the heat pipe to temperatures significantly below the triple point (see Fig. 7). Therefore, the vapor pressure of the solid is several orders of magnitude less than the vapor pressure near the triple state. This is precisely the condition in liquid metal heat pipes where the frozen state corresponds to ambient temperatures and significantly reduced vapor pressures. From Fig. 7, it is apparent that the vapor pressure for the liquid metal working fluids are on the same order of magnitude at twice the melting point temperature as the vapor pressure for room temperature fluids at or near the melting point temperature. It should also be noted that for heat pipes with large condenser to evaporator length ratios, formation of a solid plug may not occur due to an inadequate amount of working fluid to create a total blockage. However,

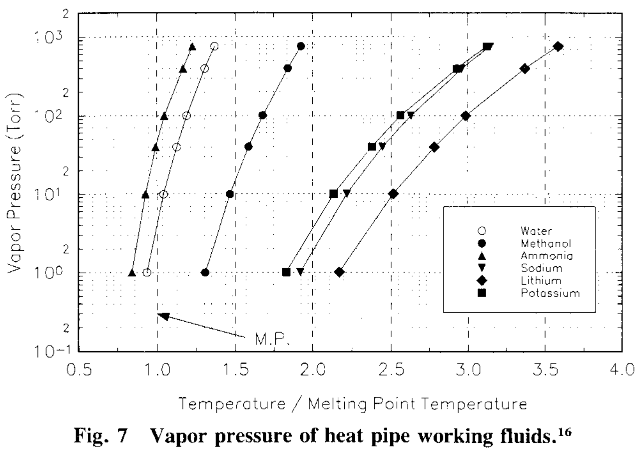

Fig. 7 Vapor pressure of heat pipe working fluids.[16]

the condition of evaporator dryout for low levels of NCG may still result.

Restart

For each of the previous tests, restart from the frozen state for several levels of evaporator heat addition was attempted. Two restart scenarios were performed for the frozen configurations corresponding to a 0.07-m NCG length. The first corresponded to the reported case of a dry evaporator and a solid formation blocked vapor channel (see Figs. 2 and 4). Because the evaporator wick was dry and the vapor passage was completely blocked, successful startup by evaporator heat addition was not anticipated. This was indeed the result for restart attempts with power inputs ranging from 30–80 W, as restart would require melting of the fluid entirely by axial conduction through the heat-pipe walls.

In the second startup attempt with the same level of noncondensible gas, the blockage of the heat-pipe vapor channel was not permitted sufficient time to form and startup of the heat pipe was attempted after an elapsed time of 55 min. This scenario corresponds to the initial state of an evenly distributed frozen working fluid and an open vapor passage, examined in the previously discussed numerical models.[14,15] Initially, 30 W of evaporator heat input was added. The transient liquid temperature profiles for this case are presented in Fig. 8. As shown, dryout of the evaporator resulted because the working fluid along the liquid channel remained frozen and could not be pumped back to the evaporator. As heat was applied to the evaporator, the solid layers along the side walls in the condenser were observed visually to increase slightly in thickness due to the freezing of the working fluid from the evaporator. Similar results were obtained for evaporator heat inputs up to 80 W.

The startup for the frozen configurations with NCG lengths of 0.36 and 0.40 m were successful for a variety of initial power inputs and were very similar in nature. The temperature profiles for the restart of the frozen configuration of a 0.40-m NCG length are presented in Fig. 9 with an initial

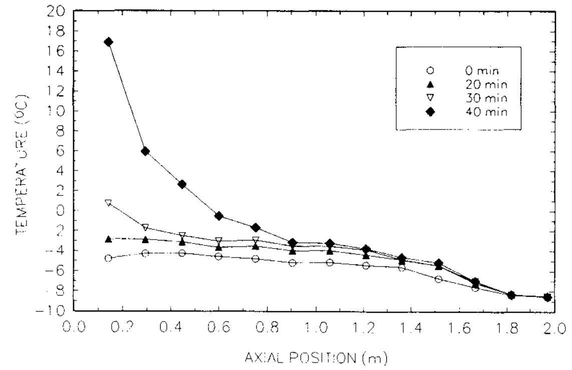

Fig. 8 Transient liquid channel temperatures for restart of 0.07-m NCG length (blocked vapor channel; P = 30 W).

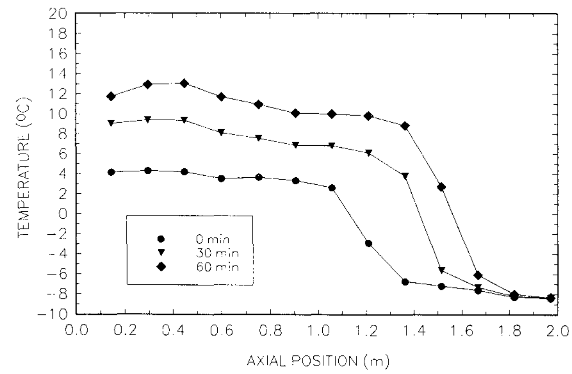

Fig. 9 Transient liquid channel temperatures for restart of 0.40-m NCG length (P = 80 W at 0 min; P = 160 W at 30 min).

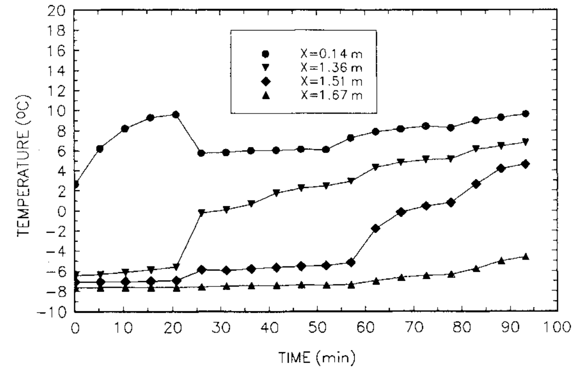

Fig. 10 Transient liquid channel temperatures for restart of 0.30-m NCG length (P = 30 W at 0 min; P = 80 W at 40 min; P = 160 W at 75 min).

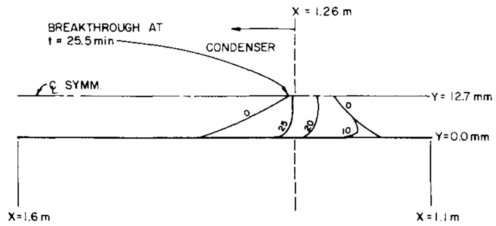

Fig. 11 Solid formation contours during restart of 0.30-m NCG length.

power input of 80 W followed by an increase to 160 W after 30 min. Because the wick in the evaporator and adiabatic regions were primed with liquid and the vapor channel was not completely blocked, restart was accomplished with the melt front of the condenser wick proceeding frontally. These temperature profiles are consistent and indicative of the frontal heating mode characteristic of startup in gas-loaded heat pipes.

The frozen configuration for a noncondensible gas length of 0.30 m yielded the most interesting results during startup with an initial power input to the evaporator of 30 W. The liquid channel temperatures as functions of time for several axial locations are given in Fig. 10 along with the solid formation contours in Fig. 11. The previously discussed condition of freezing blowby was exhibited by this startup test once breakthough of the solid blockage occurred (approximately 25.5 min). Upon breakthrough, the remaining frozen working fluid located on the side walls melted rapidly and the temperature of the evaporator liquid dropped due to the decrease in pressure. With step function power increases to 80 W at a time of 40 min and to 160 W at a time of 75 min, the heat pipe returned to a steady-state operating condition. Although this test demonstrated successful startup of a blocked vapor passage, in some cases freezing blowby may result in dryout of the evaporator wick and unsuccessful startup. This is particularly true if the liquid loss through blowby is large in comparison to the evaporator liquid inventory or in heat pipes with large condenser to evaporator length ratios.

Conclusions

The freeze/thaw characteristics of a copper/water heat pipe have been investigated experimentally to determine the effect of variations in the amount of noncondensible gas charge. During the freezing process, visual observations noted a complete solid blockage of the vapor passage near the inlet of the condenser for low levels of noncondensible gas along with a dry evaporator wick. Startup attempts under these conditions resulted in an inability to restart the heat pipe by evaporator heat addition alone. As the level of noncondensible gas increased, the position of the blockage moved toward the adiabatic section and the magnitude of the solid formation decreased. For further increases in noncondensible gas, this continued until a transition region was obtained where the evaporator was still primed with liquid and the blockage eventually was not complete. Startup under these conditions was obtainable; however, freezing blowby occurred for thin blockage levels. A condition of no significant solid formation was reached for high levels of noncondensible gas due to expansion of the noncondensible gas beyond the inlet of the condenser. Startup for this state was obtainable and exhibited a frontal startup mode characteristic of gas-loaded and liquid metal heat pipes.

Comparing the results of the present investigation and those of previous investigations available in the literature, it is apparent that significant differences exist in the freeze/thaw characteristics of room temperature and liquid metal heat pipes. These primarily result from the extreme differences in the vapor pressures near the melting point of the fluid. For room temperature heat pipes, the frozen configuration normally will not correspond to temperatures far below the triple point of the working fluid, as is the general case for liquid metal heat pipes. As a result in room temperature heat pipes, the vapor pressure of the working fluid may be great enough for migration of the fluid to colder regions of the heat pipe to occur. Also, the vapor pressure of room temperature fluids upon melting is generally much greater than the vapor pressure of liquid metal working fluids upon melting. Thus, dryout of the evaporator may occur due to the immediate vaporization of the fluid once the solid has melted. In liquid metal heat pipes, this low vapor pressure upon melting permits sufficient time for large portions of the frozen fluid to be melted by conduction prior to large-scale vaporization of the working fluid.

Acknowledgment

The authors acknowledge the support of the NASA Office of Commercial Programs (Code C) through the Texas A&M University Center for Space Power, the McDonnell Douglas Space Systems Division, the Grumman Aerospace Corporation, and the Lockheed Missiles and Space Company, Inc.

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