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TRANSIENT CHARACTERISTICS OF A GROOVED WATER HEAT PIPE WITH VARIABLE HEAT LOAD

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ABSTRACT

The transient characteristics of a grooved water heat pipe have been studied by using variable heat load. First, the effects of the property variations of the working fluid with temperature were investigated by operating the water heat pipe at several different temperatures. The experimental results show that, even for the same heat input profile and heat pipe configuration, the heat pipe transports more heat at higher temperature within the tested temperature range. Adequate liquid return to the evaporator due to decreasing viscosity of the working fluid permits continuous vaporization of water without dry-out. Second, rewetting of the evaporator was studied after the evaporator had experienced dry-out. To rewet the evaporator, the elevation of the condenser end was the most effective way. Without elevating the condenser end, rewetting is not straight-forward even with power turned off unless the heat pipe is kept at isothermal condition for sufficiently long time.

NOMENCLATURE

- D : diameter of heat pipe, m
- H : latent heat of the vaporization, J/Kg
- L : total length of heat pipe, m
- N : liquid transport factor, kW/cm^2
- μ : viscosity, $N\text{-}s/m^2$
- ρ : density, Kg/m^3
- σ : surface tension, N/m

INTRODUCTION

The heat pipe is an effective device to transport a large amount of energy with a small temperature drop by using phase change of the working fluid in a closed system. Due to this extremely high thermal conductance at relatively light weight, the heat pipe has received much attention for the space applications to meet the demand for an effective thermal management device. However, the heat pipe has several operational restrictions such as the capillary limit, sonic limit, entrainment limit, and boiling limit. When heat pipes are operated within these limits, the inherent advantages of using heat pipes can be obtained.

For non-metallic heat pipes, the capillary limit is a major restriction so performance of the heat pipe depends largely on the capillary pumping force. When the heat input at the evaporator section is greater than allowed by the capillary limit, the evaporator section may experience dry-out which, if not relieved, will lead to burnout. So, defining the dry-out conditions and finding effective methods to rewet the evaporator section are very important to avoid any hazardous conditions during heat pipe operation. To achieve this goal, the experimental test was conducted by using the water heat pipe which is safer than the liquid metal heat pipe and is a prime candidate for manned applications, with operating temperatures up to 500 K.

Water heat pipes have been studied by several authors. First, Neal (1967) used a water heat pipe with stainless steel shell and screen wick of 2 foot length for a long term test of reliability and for a startup test. Also, Deverall et al. (1970) investigated to achieve a successful startup from the frozen state by using a water heat pipe with stainless steel shell and a screen wick. A conventional water heat pipe and a concentric annular water heat pipe with a grooved wick were tested to examine the increase of heat transport capacity per unit length and the effect of tilt angle by Faghri and Thomas (1988). All parts of the pipe were made of copper. The temperature difference of 12 C between the evaporator end cap and the adiabatic section was used as reference to define dry-out at the evaporator. Jang et al.

(1990) tested the Copper-water heat pipe with axial grooves to study transient behavior.

The viscosity change of the working fluid significantly affects the capillary force and the capillary limit is major restriction for the non-metallic heat pipe. The effect of property variations versus temperature was studied. Then, dry-out and rewetting was experimentally investigated to define conditions at dry-out and provide effective way to rewet the evaporator.

EXPERIMENTAL SETUP

To investigate transient characteristics of the heat pipe at low temperature, a water heat pipe with axial grooved wick was used as shown in Figure 1. The total length of this heat pipe is 1 m, and the lengths of the evaporator and condenser sections are 28.6 cm and 20 cm, respectively. The rest of the length is the adiabatic section. The outer diameter of the heat pipe is 5 cm and the inner diameter is 4.66 cm. The heat pipe shell is made of copper and the wick structure consists of 120 axial grooves cut on the inner surface of the shell. The depth and width of groove are .5 mm. Water is the working fluid. All processes of fabrication were described in a paper by Faghri and Thomas (1988). Figure 1 shows schematic diagram of the grooved water heat pipe and Table 1 shows summary of heat pipe design parameters.

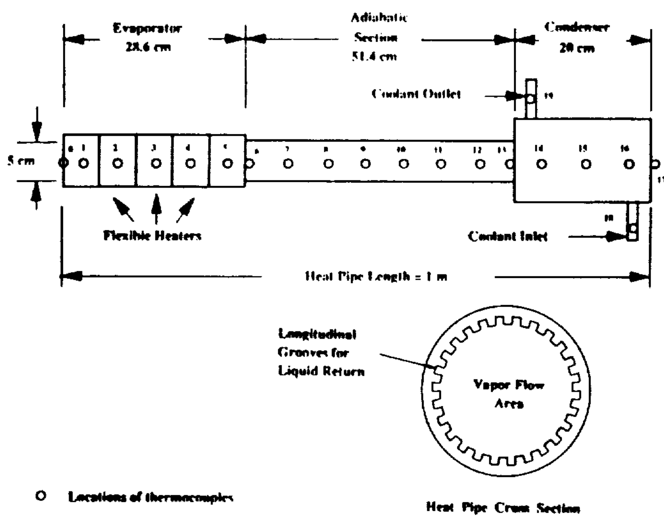

Figure 1. Schematic of a water heat pipe.

Table 1. Summary of physical dimension of the heat pipe.

Heat Pipe Shell		Grooves	
Material	Copper	Number	120
Total length	100.0 cm	Width	0.5 mm
Evaporator	28.6 cm	Depth	0.5 mm
Condenser	20.0 cm		
Adiabatic	51.4 cm		
Outer diameter	5.0 cm		
Inner diameter	4.66 cm		
Pipe thickness	0.17 cm		

If the working heat pipe configuration is to be maintained, it is difficult to instrument the inside of the heat pipe to monitor vapor and liquid flow. This is because performance of the heat pipe depends mainly on the internal flow, and instrumentation at the inside of the heat pipe would lead to internal flow disturbances which would affect operation of the heat pipe. So, the temperatures at the surface of the evaporator, adiabatic, and condenser sections, and the evaporator and condenser end caps were measured. Also, temperatures measured at the evaporator and condenser surfaces may be influenced by heaters on the evaporator section and cooling coil on the condenser section. Therefore, temperatures at the adiabatic section may represent heat pipe temperatures. The wick structure with the working fluid is located inside heat pipe shell. However, there is no wick structure attached on the evaporator and condenser end caps. One side of these thin caps is insulated and another side is in contact with the vapor space in the heat pipe. So, thermocouples mounted at the center of insulated surfaces of the evaporator and condenser end caps measure temperatures closest to the vapor temperature in the heat pipe.

To measure the surface temperature, a total of 16 copper-constantan thermocouples were installed on the outer surface of heat pipe shell as shown in Figure 1. Two thermocouples were mounted at the center of the evaporator and condenser end caps. Also, another two thermocouples were used to measure coolant inlet and outlet temperatures. A Fluke 2280A data logger was used to record temperature. Five flexible heaters with a rectangular shape were tightly wrapped on the surface of the evaporator and connected to two Variacs. The length and width of each heater are 15.95 cm and 5.72 cm, respectively. Also, each heater can provide 250 watts with 115 volts. An electrical switch was installed between a heater and a Variac output so each heater can be operated independently. Also, voltmeters and ammeters were installed to get exact electrical power output from the Variac. The cooling coil was tightly wrapped around the condenser section to provide controlled condenser cooling. The cooling coil made of .25 inch outer diameter soft copper tube was held in place by stainless steel clamps at each end.

Since it is desired to have the liquid condensed at the condenser section return to the evaporator by using solely the capillary force, the effect of a gravity field is significant on the heat pipe performance. So, to minimize this effect, the water heat pipe was placed on the optical bench to level accurately between the evaporator and condenser. Then, insulation material made from ceramic fibers was wrapped on the entire heat pipe in 1.5 inch thickness. The thermal conductivity of this insulation material is .03 W/m-C. A schematic diagram of experimental test setup is shown in Figure 2. Cooling water was pumped to the cooling coil from the coolant bath which keeps constant water temperature. Flow meter was installed at the inlet of the cooling coil to measure cooling water flow rates. Table 2 shows all heat pipe test cases.

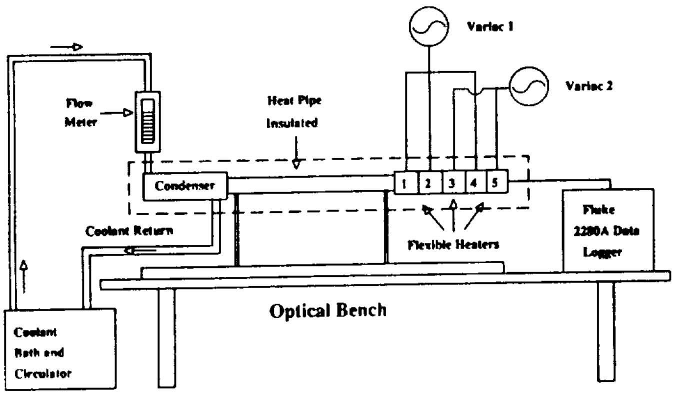

Fig. 2 Schematic diagram of the water heat pipe experimental setup.

Table 2. Summary of heat pipe test.

Case No.	Initial Temp. (K) of the Heat Pipe	Heat Input (W)	Time Variation of Heat Input
1	297	50	Constant
2	313	50	Constant
3	333	50	Constant
4	350	50	Constant
5	313	150	Stepped (see Fig. 6)
6	333	200	Stepped (see Fig. 6)
7	350	200	Stepped (see Fig. 6)
8	313	110	Stepped (see Fig. 8)
9	313	150	Stepped (see Fig. 9)

RESULTS AND DISCUSSIONS

Effects of Property Variations

The effects of the property variations of the working fluid with temperature were investigated by operating the water heat pipe at several different temperatures. The initial heat pipe temperatures of 297 K, 313 K, 333 K, and 350 K were used with 50 watts uniform heat input at the evaporator section. Except for the 297 K initial temperature, to operate the heat pipe at the elevated temperatures, the heat pipe was slowly heated until the temperatures at the surface and end caps reached the desired initial temperatures. During the heating period, instead of turning the heater on, the coolant inlet temperature was increased from ambient temperature to the desired initial temperature to add heat to the condenser section. This way can avoid partial dry-out at the evaporator section while the heat pipe reaches the desired initial temperature. Since the heat pipe was insulated and the coolant inlet temperature was fixed at the initial temperature, the heat pipe finally reached the steady state at the desired initial temperature with uniform temperature distribution. Once this condition was obtained and then 50 watts heat input was applied on the evaporator while the coolant inlet temperature was kept constant. The coolant inlet temperature was the same as the initial temperature.

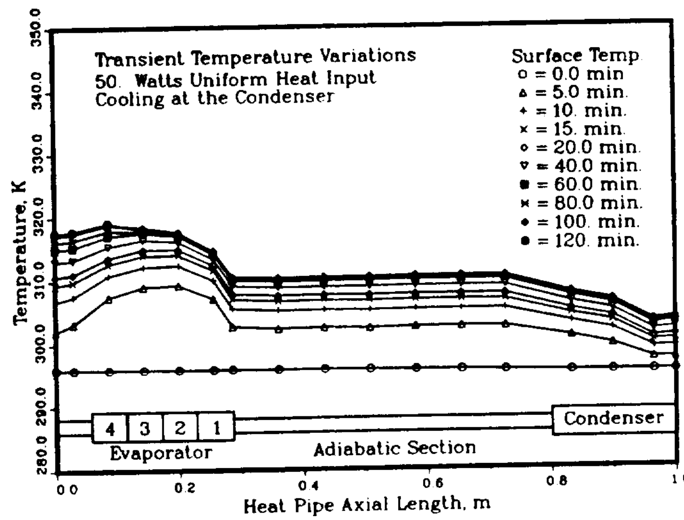

Fig. 3 Transient surface temperature distributions for case 1.

Figure 3 shows transient surface temperature distributions for an initial temperature 297 K. Up to 10 minutes, temperature at the evaporator end cap was close to that at the adiabatic section, and then the former rose faster than the latter. Temperature at the evaporator end cap may be considered to be close to that of vapor space at the evaporator end. So, a higher temperature at the evaporator end cap than that at the adiabatic section implies that the vapor near the evaporator end cap may be super-heated. Figure 4 shows temperature distributions along the axial direction at steady state conditions for four different operating temperatures. Note that the

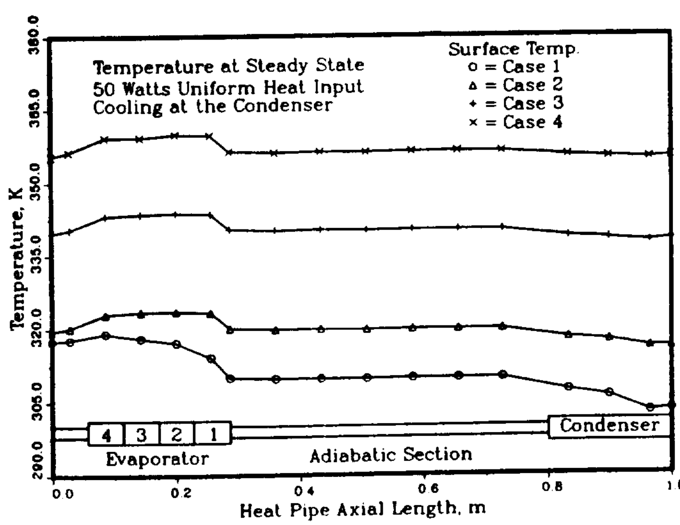

Fig. 4 Temperature distributions at steady state for various operation conditions (cases 1, 2, 3, and 4).

temperature distribution becomes more uniform as the operating temperature increased. The temperature distribution for the initial temperature of 350 K shows that the heat pipe was operating at near isothermal condition. Temperatures under heaters were slightly higher than those at the adiabatic section due to influencing of the heaters. Temperature distributions at the adiabatic section for all four cases are very uniform.

To understand the effects of the property variations, the liquid transport factor of the working fluid was calculated for different temperatures based on following equation:

$$N = \frac{\rho \sigma H}{\mu} \qquad (1)$$

This liquid transport factor describes the capillary pumping ability of the working fluid. For the water heat pipe, unlike liquid metal heat pipes, the capillary pumping limit may be the first limit encountered. So, the highest performance of the water heat pipe may be obtained at operation temperature which gives the greatest value of liquid transport factors. Figure 5 shows thermo-physical properties (Incropera and DeWitt, 1981) and liquid transport factor variations, calculated by using Equation (1), versus temperatures for water. Up to temperature about 460 K, the liquid transport factor is increased mainly due to decrease in the viscosity and then decreased because the surface tension, density, and heat of vaporization decrease more rapidly than the viscosity. This implies that for given heat input the heat pipe can approach more isothermal condition until liquid transport factor reaches the maximum.

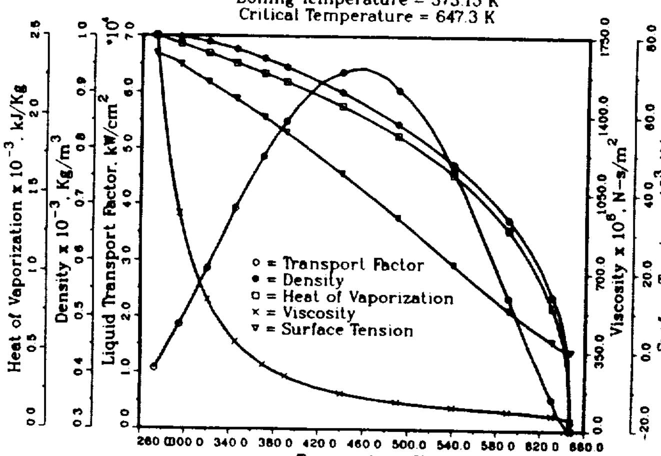

Fig. 5 Thermophysical properties and liquid transport factor for water.

Another test was conducted with stepped heat input to investigate the effects of the property variations with temperature. At uniform initial temperature of 350 K (case 7), the stepped heat inputs of 50, 100, 150, and 200 watts at 20 minute intervals were applied on the evaporator. As shown in Figure 6, up to 150 watts the heat pipe was near isothermal condition and then with 200 watts heat input

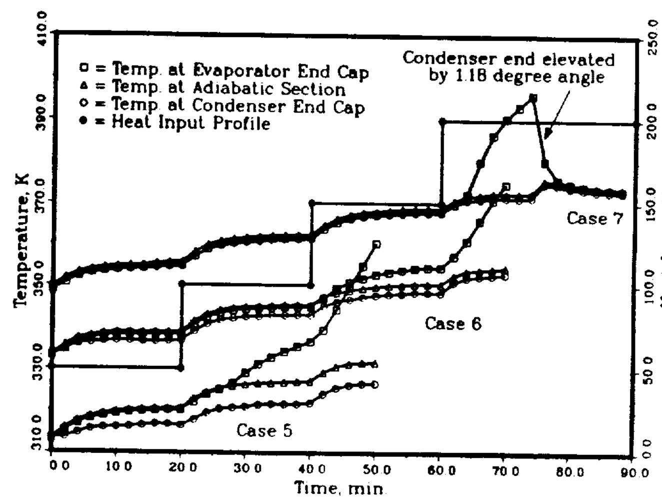

Fig. 6 Transient temperature variations at the evaporator and condenser end caps and the adiabatic section with stepped heat input (cases 5, 6, and 7).

temperature at the evaporator end cap rapidly increased while temperatures at the condenser end cap and adiabatic section approached steady state condition. So, the condenser end was elevated by 1.18 degree angle while heat input at the evaporator was continued. Then, temperature at the evaporator end cap rapidly decreased and temperatures at the adiabatic section and condenser end cap increased. The water heat pipe again reached an isothermal condition. This means that the part of the evaporator section had been dry-out. When the condenser end was elevated, adequate liquid returned from the condenser to the evaporator section to rewet the evaporator. Consequently more vaporization occurred at the evaporator section so more heat was transported from the evaporator to the condenser section.

For initial temperature of 313 K (case 5), the stepped heat inputs of 50, 100, and 150 watts at 20 minute intervals were added at the evaporator. However, temperature at the evaporator end cap started to deviate from that at the adiabatic section with 100 watts heat input. Then, with 150 watts heat input that temperature rapidly increased. So heat input was terminated. Also, a greater temperature gradient between the evaporator end and the condenser end was observed.

For an initial temperature of 333 K (case 6), the same heat input profile as that for 350 K was used. With 150 watts heat input some temperature gradient between the evaporator end cap and adiabatic section was observed. When the heat input was increased to 200 watts, the temperature at the evaporator end cap rapidly increased while temperatures at the condenser end cap and adiabatic section slowly increased. Again this implies that the vapor in vapor space at near evaporator end cap is super-heated. So, power input was turned off. As shown in Figure 6, the water heat pipe tested at the highest temperature was clearly working at the most isothermal condition and also transported more heat. Figure 7 shows temperature distributions along the entire heat pipe for the initial temperature of 350 K.

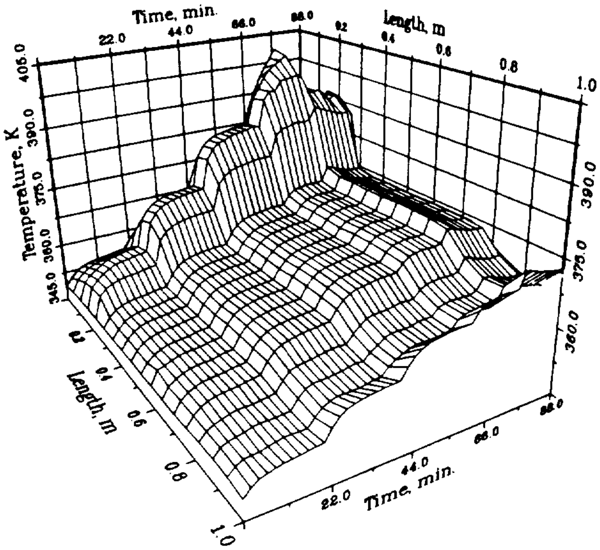

Fig. 7 Transient surface temperature distributions for the initial temperature of 350 K (case 7).

Dry-out and Rewetting

As shown in Figure 6, the rapid increase of the temperature at the evaporator end cap means that part of the evaporator section is experiencing dry-out. Using this phenomenon, dry-out and rewetting of the water heat pipe was studied experimentally (case 8). First, a uniform heat input of 50 watts was added to the evaporator section until the heat pipe reached steady state condition. Then, heat input was increased by 10 watts at 5 minute intervals so the maximum heat input reached 110 watts. Figure 8 shows heat input profile, and temperatures at the evaporator and condenser end caps, and the adiabatic section

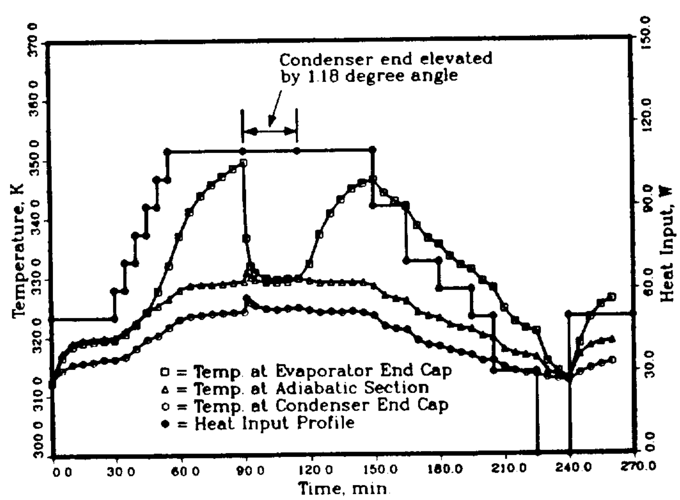

Fig. 8 Dry-out and rewetting of a water heat pipe with variable heat load and tilting (case 8).

versus time. During this period, temperature at the condenser end cap is lower than those at the evaporator end cap and adiabatic section. Up to heat input of 80 watts, temperature at the evaporator end cap is close to that at the adiabatic section. However, more heat input lead to rapid increase of temperature at the evaporator end cap while temperatures at other locations stayed steady. This indicates that the vaporization of the working fluid at the evaporator was not enough to cool the evaporator section. Then, the condenser section end was elevated to rewet the evaporator so temperature at the evaporator end cap rapidly decreased and temperatures at the adiabatic section and condenser end cap increased at the same time. This temperature change shows that the part of the evaporator section had been dryed out. The experimental results shown in Figure 8 indicate that about 20 K temperature difference between at the evaporator end cap and adiabatic section existed.

After the heat pipe reached the steady state with the elevation of the condenser end, the heat pipe was leveled again while the same heat input was being added. The temperature at the evaporator end cap rose rapidly as before, indicating that part of the evaporator section was undergoing dry-out again. From this one may conclude that the gravity field resulting from elevating the condenser end aided the capillary force to return the liquid from the condenser to the evaporator.

In an attempt to rewet the evaporator section, heat input was reduced in a stepwise manner to the initial heat input rate of 50 watts as shown in Figure 8, but the temperature at the evaporator end cap did not drop to its previous value. This indicates that even at a lower heat input rate part of the evaporator section was still in a dry-out condition. Next, power was completely turned off until the heat pipe reached the initial condition, and then 50 watts heat input was re-applied in the same manner as at the beginning of the test. However, the temperature distribution was quite different from that obtained previously. The temperature at the evaporator end cap increased faster than other temperatures. From this one may conclude that the evaporator section was not successfully rewetted.

Figure 9 shows another heat input profile and temperature variation versus time for a dry-out and attempted rewetting test (case 9) without elevating the condenser section. The initial temperature was 313 K. After the heat pipe reached a steady state condition with a uniform heat input rate of 50 watts, the heat rate was increased by 10 watt increments at 10 minute intervals to a maximum of 150 watts. When the heat input rate was 100 watts, the temperature at the evaporator end cap started to increase faster than did all other temperatures on the heat pipe. As the heat input rate was further increased, the temperature at the evaporator end cap continued to increase at a fast rate while temperatures at the adiabatic section and the condenser end cap increased more slowly. When the temperature difference between at the

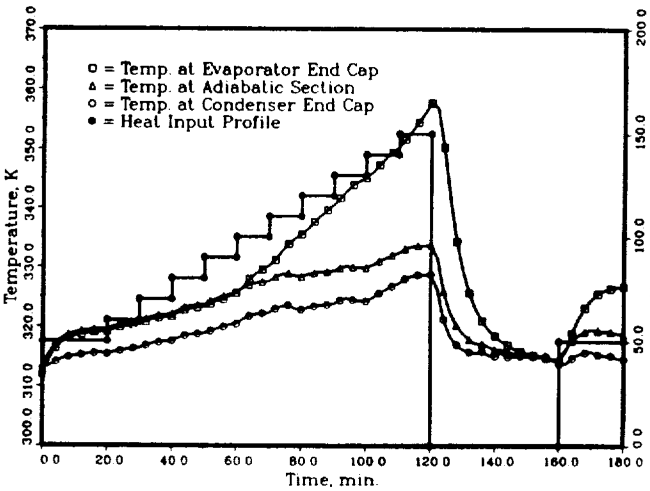

Fig. 9 Dry-out and rewetting of a water heat pipe with variable heat load (case 9).

evaporator end cap and the adiabatic section was 25 K, instead of reducing heat input step-wise as before, heat input was suddenly terminated for 40 minutes in an attempt to rewet the evaporator section without elevating the condenser end, as shown in Figure 9. Then, a uniform heat input of 50 watts was applied at the evaporator section. However, temperature at the evaporator section increased faster than did all other temperatures, indicating a persisting dry-out condition. Figure 10 is a 3-D plot showing

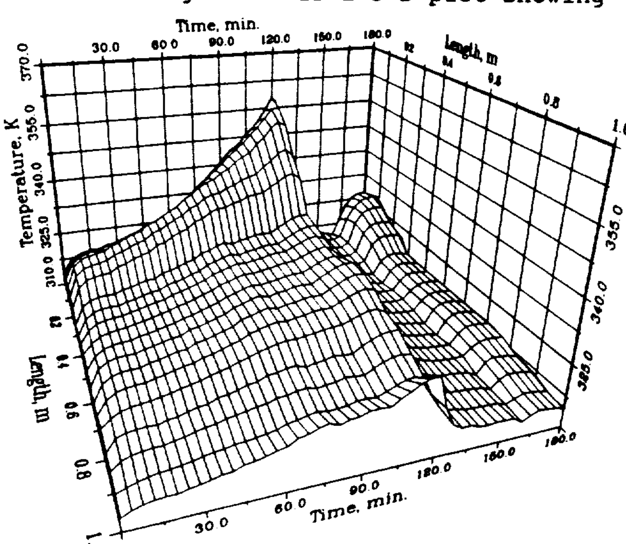

Figure 10. Transient surface temperature distribution for case 9.

temperature distribution along the heat pipe length versus time. Figures 8 shows that elevation of the condenser end is an effective way to rewet the evaporator, assuming of course that a gravity field exists. On the other hand, reducing the heat input at the evaporator did not results in successful rewetting of the evaporator as shown in Figures 8 and 9.

CONCLUSIONS

A grooved water heat pipe was tested to study the effects on heat pipe performance of the property variations of the working fluid with temperature. Also studies were dry-out and rewetting conditions of the evaporator section. The test results show that, even for the same heat input profile and heat pipe configuration, the heat pipe operated at a more isothermal condition and transported more heat at higher operating temperature within tested temperature range. Dry-out at the evaporator section was characterized by rapid temperature increase at the evaporator end cap, while temperatures at the adiabatic section and the condenser end cap stayed steady. When the evaporator had experienced dry-out, elevation of the condenser end was the most effective way to rewet the evaporator if a gravity field exists. Without elevating the condenser end, rewetting was not achieved even with power turned off for 40.0 minutes.

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