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MEASUREMENTS OF THE TRANSIENT BEHAVIOR OF A CAPILLARY STRUCTURE UNDER HEAVY THERMAL LOADING

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Abstract—The transient operating characteristics of a slab type capillary structure in a refrigerant 11 heat pipe have been studied under conditions which cause partial or complete drying and rewetting to occur. Detailed changes with time in internal liquid and vapor temperature distributions, internal pressure, and heat transport capability are presented. It was found that the device will, under some conditions, approach steady operation with a portion of the capillary structure dried and that rewetting can be quickly accomplished. However, when the entire structure is dried, rewetting is much more difficult to accomplish.

NOMENCLATURE

$A_i, B_i, C_i, A_k, B_k, C_k$	coefficients [K^{-1}]
c	specific heat [$J\ kg^{-1}\ K^{-1}$]
D_i, D_k	coefficients [dimensionless]
d	wire diameter [m]
h_{cj}	film coefficient in cooling jacket [$W\ m^{-2}\ K^{-1}$]
K	thermal conductivity [$W\ m^{-2}\ K^{-1}$]
L	axial length [m]
l	wetted length [m]
M	mass [kg]
P_v	internal vapor pressure [bar]
Q_c	heat to coolant jacket [W]
Q_e	heat to pipe [W]
R_o	outer radius of heat pipe [m]
r	radius [m]
T	temperature [K]
t	time [min]
T_{cj}	mean fluid temperature in cooling jacket [K]
$T_{i,k}$	temperature at node (i, k) [K]
$\dot{V}$	volume flow rate [$m^3\ s^{-1}$]
V	velocity [$m\ min^{-1}$]
z	axial position [cm].

Greek symbols

β	gap between layers of capillary structure [m]
μ	viscosity [$N\ min\ m^{-2}$]
ρ	density [$kg\ m^{-3}$]
σ	surface tension [$N\ m^{-1}$]
ϕ	inverse permeability [m^{-2}].

Subscripts

a	adiabatic section
cap	capillary structure
c, cond	condenser
cj	cooling jacket
cr	critical
e, evap	evaporator
eff	effective
f	coolant
i	radial position
in	entering coolant
k	axial position
l	liquid
p	pore
s	solid
v	vapor
w	liquid saturated wick.

Superscripts

n	instant of time
$n+\frac{1}{2}$	time after half step
$n+1$	time after one step
$'$	during second time step
$\bar{}$	mean value.

INTRODUCTION

THE TRANSIENT internal operating characteristics of heat pipes during heavy thermal loading have been under study for a number of years by several investigators. However, there still exists a need for detailed internal measurements which can be used as a basis for developing mathematical models which can be used for predicting transient behavior. The data presented herein should prove to be quite useful in this respect.

Transient heat pipe operation may be classified as one of four types. In the first category heat transport is modest as compared to capillary limitations. Under these conditions the heat pipe smoothly and easily accommodates external changes on either heated or cooled ends. Internal operation is characterized by a fully wetted capillary structure and nearly uniform temperatures and pressures with respect to position. Internal temperatures and pressures do of course change with respect to time. With the second type of transient operation a portion of the capillary structure in the heated zone dries and rather large internal temperature variations may occur at an instant of time and while pressures vary with time they probably are essentially uniform with respect to position. Under these conditions thermal conductance of the heat pipe is reduced due to the inactive portion of the evaporator but the heat pipe smoothly accommodates external changes which occur on heated or cooled ends and will approach steady operation after changes are made. It is quite possible for a heat pipe to operate steadily with a portion of the heated capillary structure dried. The third type of operation is characterized by a complete failure of the device. Generally the failure is brought about by excessive heat input and/or high temperatures on the cooled end. While the heat pipe may for a time appear to approach steady operation after a sudden increase in heat input to excessive levels, eventually failure will occur. During failure very large temperature gradients develop in both liquid and vapor regions and pressure rises rapidly to very large values. Eventually the heated and cooled ends become thermally uncoupled, except for axial conduction, and evaporator temperature rises while condenser temperature falls. Another type of transient operation of heat pipes consists of startup after failure or from a thermodynamic supercritical state. It is not uncommon, particularly with cryogenic working fluids, for the working fluid to be in the supercritical regime. In order for normal operation to be established under these circumstances, the working fluid must be brought into the thermodynamic mixture region and a rewetting of the capillary structure must occur.

Chang [1] theoretically studied the four types of transient heat pipe operation. He modeled the heat pipe by breaking it into several regions such as evaporator shell, evaporator working fluid–capillary structure combination, vapor, etc. He then wrote differential governing equations for each region which included energy, momentum, and continuity balances and solved these using alternating implicit–explicit techniques. In those cases where rewetting occurred, the model accounted for the effects of liquid dynamics in the capillary structure. Chang [1] and Colwell [2] made extensive internal measurements during heat pipe transients. Generally good agreement was found between measurements and predictions.

Priester [3] theoretically studied the transient performance of a nitrogen heat pipe using both analog and digital computers. Under normal operation with small temperature changes with time the simple analog model worked very well. It was necessary to use a digital approach when large temperature changes occurred with time and also in those situations where drying and rewetting required that the model include liquid dynamics in the capillary structure. Priester showed that care must be taken during startup after capillary drying in order to avoid excessive temperatures and pressures in the evaporator zone.

Williams [4] and Williams and Colwell [5] theoretically and experimentally studied the effects of partial evaporator drying and Reynolds number in the condenser vapor region on steady state performance of water and methanol heat pipes. They found that both effects significantly affect overall heat pipe conductance and internal vapor temperature. Groll and Zimmermann [6] developed a simple mathematical model to predict dynamic behavior of heat pipes. They used a lumped parameter model which worked well for small power variations around the nominal operating condition. Their discussion was mainly concerned with low temperature heat pipes in the temperature range between 200 and 500 K.

Luikov [7] has developed systems of three-dimensional (3-D) differential equations for heat and mass transfer in capillary porous materials. For various special cases such as steady state, one dimension, and simple moving boundary, he shows how these coupled Fourier and Fick equations can be simplified. Experimental data is also presented for the non-steady drying of a moist ceramic plate, drying of moist peat in a metallic spherical flask, and drying of a moist clay sphere.

Plesset and Prosperetti [8] theoretically considered the flow of vapor in liquid enclosures under both steady and unsteady conditions where different parts of the liquid surface have different temperatures. They predict that vapor pressure should be essentially uniform except where very high vapor velocities exist. In the case of two plane parallel liquid walls at different temperatures they predict that vapor temperature should remain at the value for the high temperature surface throughout the vapor region up to a small distance from the cool surface where the temperature falls exponentially to the value of the cool surface. Vasilév and Konev [9] discuss transient behavior of cryogenic and low temperature heat pipes. They point out that it is very important to consider the effects of property variations with temperature. Hall [10] indicates that unusual thermodynamic and heat transfer characteristics may occur near the thermodynamic critical point. Thermal conductivity and specific heat may become very large. There is a lack of agreement among researchers as to the effects on convection of approaching the critical point.

The purpose of the project described herein was to obtain some detailed transient measurements of pressure and temperature inside an operating heat pipe under a variety of operating conditions including some conditions where failure of the device occurred. In addition, mathematical models which describe normal and some adverse operating conditions were developed. Under normal operating conditions it was found that transient conduction equations with variable properties applied to the shell and combination fluid-capillary structure predicted performance accurately. Under adverse conditions, mathematical models must include internal fluid dynamics and agreement between theory and experiment is not as good.

EXPERIMENTS

The test stand, which is shown schematically in Fig. 1, incorporates equipment for vacuum pumping, working

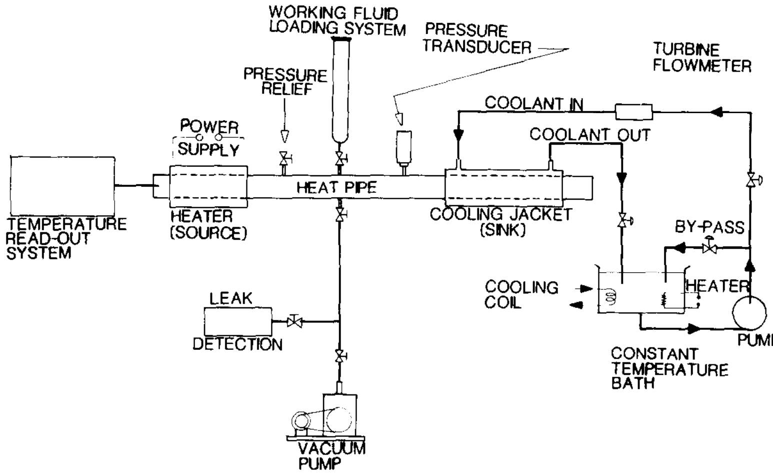

FIG. 1. Schematic of the experimental setup.

fluid loading, heating, removing thermal energy from the test section, and measuring and recording temperatures, pressures and flow rates. The test section, which is shown in Fig. 2, consists of a cylindrical 316 stainless steel heat pipe of 80 cm active length, 1.91 cm O.D. and 1.57 cm I.D. The capillary structure, also of 316 stainless steel, consists of two circumferential layers of 100 mesh screen placed along the inside wall of the pipe in the evaporator and condenser sections and a central slab which runs the entire length of the pipe and contracts the circumferential sections. The slab consists of four layers of 40 mesh screen covered on both sides with two layers (four layers total) of 100 mesh screen. Fifty millilitres of refrigerant 11, which has a critical temperature of 471.2 K, was used as the working fluid in all tests reported herein. The test section is supplied thermal energy from an electric clamshell heater with resistance elements cast in a ceramic material and cooled by circulating a silicone fluid through a jacket around the condenser end of the pipe. The silicone fluid passes through a constant temperature bath which either heats or cools the fluid in order to maintain the desired bath exit temperature. The heated section is 15.24 cm in length.

Thirteen very small stainless steel sheathed thermocouples are embedded in the central slab of the capillary structure inside the heat pipe as shown in Fig. 2. In addition, three thermocouples, designated T18, T19 and T20 in Fig. 2, are placed in the vapor region in the interior of the heat pipe. One thermocouple is placed in each of the heated, adiabatic, and cooled sections. A high pressure, high temperature pressure transducer is connected to the vapor region in the adiabatic section.

The equipment was designed and fabricated so that a wide variety of transient experiments could be performed. Thermal coupling with the source and sink were such that changes in heat pipe internal operating conditions could be brought about by altering one or more of the independent parameters which consisted of heater input, coolant flow rate, and coolant inlet temperature. Except where noted, where heat transfer rates are quoted in the following presentation of results, the rates refer to heat transferred into the cooling jacket.

THEORY

Mathematical models of heat pipes have been developed and studied parametrically by the authors.

In those cases where the capillary structure remains fully wetted, the following assumptions are made:

(1) heat is transferred through the liquid saturated circumferential capillary structure and through the heat pipe shell by conduction only;

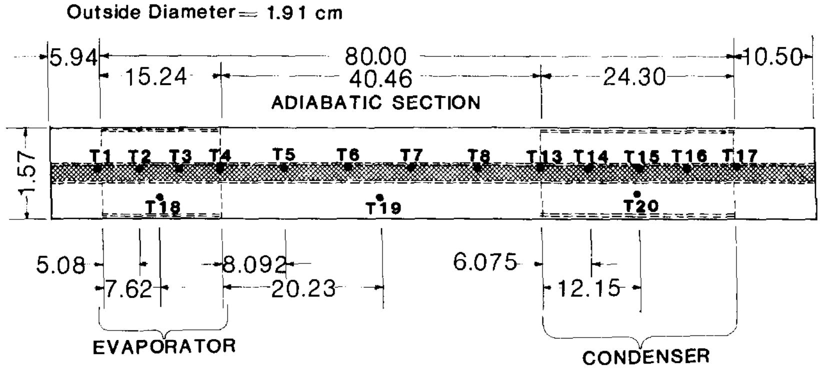

FIG. 2. Thermocouple locations within test section.

(2) thermal resistances associated with vaporization and condensation along the vapor–capillary structure interfaces are negligible; and

(3) temperature in the vapor space may change with time but is uniform with respect to geometry.

The transient energy equation for the present case is

$$\frac{\partial(\rho c T)}{\partial t} = \nabla \cdot (K \nabla T). \tag{1}$$

Applying this equation to the tube wall and the circumferential capillary structure and making radial and axial finite difference approximations about nodes located in each region as shown in Fig. 3 results in the following sets of linear algebraic equations. For the first half time step

$$\begin{bmatrix} B_1 & C_1 & 0 & 0 & \cdots & 0 & 0 \\ A_2 & B_2 & C_2 & 0 & \cdots & 0 & 0 \\ 0 & A_3 & B_3 & C_3 & \cdots & 0 & 0 \\ \cdot & \cdot & \cdot & \cdot & \cdots & \cdot & \cdot \\ \cdot & \cdot & \cdot & \cdot & \cdots & \cdot & \cdot \\ \cdot & \cdot & \cdot & \cdot & \cdots & \cdot & \cdot \\ 0 & 0 & 0 & \cdots & A_{i-1} & B_{i-1} & C_{i-1} \\ 0 & 0 & 0 & \cdots & 0 & A_i & B_i \end{bmatrix} \times \begin{bmatrix} T_{1,k}^{n+1/2} \\ T_{2,k}^{n+1/2} \\ T_{3,k}^{n+1/2} \\ \cdot \\ \cdot \\ \cdot \\ T_{i-1,k}^{n+1/2} \\ T_{i,k}^{n+1/2} \end{bmatrix} = \begin{bmatrix} D_1 \\ D_2 \\ D_3 \\ \cdot \\ \cdot \\ \cdot \\ D_{i-1} \\ D_i \end{bmatrix}, \tag{2}$$

and for the second half time step

$$\begin{bmatrix} B'_1 & C'_1 & 0 & 0 & \cdots & 0 & 0 \\ A'_2 & B'_2 & C'_2 & 0 & \cdots & 0 & 0 \\ 0 & A'_3 & B'_3 & C'_3 & \cdots & 0 & 0 \\ \cdot & \cdot & \cdot & \cdot & \cdots & \cdot & \cdot \\ \cdot & \cdot & \cdot & \cdot & \cdots & \cdot & \cdot \\ \cdot & \cdot & \cdot & \cdot & \cdots & \cdot & \cdot \\ 0 & 0 & 0 & \cdots & A'_{k-1} & B'_{k-1} & C'_{k-1} \\ 0 & 0 & 0 & \cdots & 0 & A'_k & B'_k \end{bmatrix} \times \begin{bmatrix} T_{i,1}^{n+1} \\ T_{i,2}^{n+1} \\ T_{i,3}^{n+1} \\ \cdot \\ \cdot \\ \cdot \\ T_{i,k-1}^{n+1} \\ T_{i,k}^{n+1} \end{bmatrix} = \begin{bmatrix} D'_1 \\ D'_2 \\ D'_3 \\ \cdot \\ \cdot \\ \cdot \\ D'_{k-1} \\ D'_k \end{bmatrix}, \tag{3}$$

where the coefficients in these equations are functions of $T_{i,k}^{n}$ (for the first time step), $T_{i,k}^{n+1/2}$ (for the second time step), geometry, and thermal properties of solid materials and the working fluid. These equations were generated using the well-known alternating direction implicit method [11, 12]. Thermal conductivity in the capillary structure was computed from an expression developed by Williams [4]

$$\frac{K_w}{K_l} = \frac{1}{\left[\frac{2r_p}{d}+1\right]\left[\frac{2r_p}{d}-1+\frac{2K_l}{K_s}\right]} + \frac{2}{\left[\frac{2r_p}{d}+1\right]\left[\frac{d}{2r_p}+1\right]\left(\frac{K_l}{K_s}\right)} + \frac{1}{\left[\frac{d}{2r_p}+1\right]^2}. \tag{4}$$

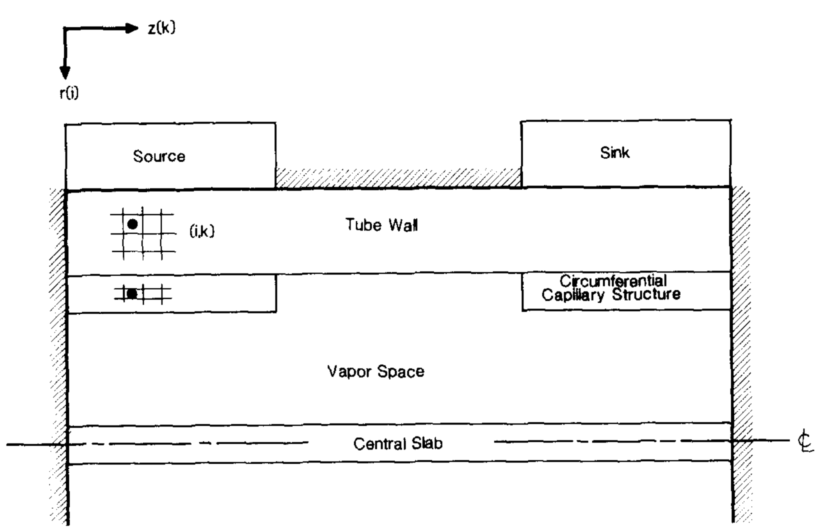

FIG. 3. Grid identification.

When the slab is fully wetted the vapor and slab regions are nearly at the same temperature at an instant. Temperature in this region may of course change with time. Under these conditions the temperature in the vapor space may be evaluated from

$$(Mc)_v \frac{dT_v}{dt} = 2\pi r_v L_e K_w \frac{\partial T}{\partial r}\bigg|_{\substack{\text{evap}\\ r=r_v}} + 2\pi r_v L_c K_w \frac{\partial T}{\partial r}\bigg|_{\substack{\text{cond}\\ r=r_v}}, \tag{5}$$

where $(Mc)_v$ is the heat capacity of the region including vapor, capillary structure and liquid. A heat balance on the cooling jacket at the condenser end may be written as

$$(Mc)_{cj}\frac{dT_{cj}}{dt} = 2\pi R_o L_c h_{cj}(\bar{T}_c - T_{cj}) - \rho_f \dot{V} c_f(T_{cj} - T_{in}), \tag{6}$$

where $(Mc)_{cj}$ is the heat capacity of the coolant in the cooling jacket and the jacket and $\bar{T}_c$ is the mean surface temperature of the heat pipe in contact with the coolant in the jacket. All properties throughout the heat pipe are re-evaluated after small changes in temperature.

A one-dimensional (1-D) momentum equation which holds for the liquid in the slab region is [13]

$$\frac{d}{dt}(Vl) = \frac{2\sigma_l}{r_p\rho_l} - \phi\frac{\mu_l}{\rho_l}Vl, \tag{6}$$

where V is the approach or superficial velocity for the porous structure, ϕ is the effective inverse permeability for the structure, and l is the wetted length in the composite slab. For the case of the fully wetted slab l becomes a constant with time and is approximately given by

$$L_{eff} = L_a + 0.5(L_e + L_c), \tag{7}$$

where it is assumed that on average condensation occurs at the mid-point of the condenser and evaporation occurs at the mid-point of the evaporator. The momentum equation can then be integrated directly to give

$$V = \frac{2\sigma_l}{r_p L_{eff}\phi\mu_l}\left[1 - \exp\left(-\phi\frac{\mu_l}{\rho_l}t\right)\right]. \tag{8}$$

This expression may be used to estimate response times to heat pipe transients for a fully wetted capillary structure. As an example, consider the case where the slab under consideration in this paper initially is isothermal at 300 K and is suddenly placed under conditions where the radii between vapor and liquid in the evaporator section approach the pore radius of the capillary structure, i.e. the condition where maximum capillary pumping occurs. The equation predicts that liquid velocity reaches 90% of its steady state value in about 0.01 s. A similar analysis of the vapor region shows that vapor response tends to be even more rapid than liquid response.

In situations where the capillary structure may be drying or rewetting, equation (6) holds but both V and l vary with time. In addition, in many rewetting cases the capillary structure ahead of the liquid front may be at a higher temperature than the liquid moving through it. For these cases vaporization will occur at the liquid front and energy and continuity balances must then be made at the liquid front.

Chang [1] discusses in detail the theory and solution techniques which have been only briefly outlined above.

RESULTS

Relatively detailed internal measurements were made while operating the heat pipe under a variety of normal and adverse transient conditions. When operating at modest heat transfer rates with coolant inlet temperatures well below the critical value for refrigerant 11 the heat pipe operated very smoothly easily accommodating abrupt changes in heating rates, coolant flows, and inlet coolant temperatures. Figure 4 shows typical startup from room temperature. On this figure only a single inside temperature is given at each instant of time since the interior space was nearly isothermal. For example, in one of the tests shown the entire test section was initially isothermal at 297 K. Suddenly the temperature of the silicone fluid entering the cooling jacket was raised to 310 K and heat was added at the evaporator end. All internal temperatures increased smoothly and after about 1.5 h the operation was nearly steady. At that time all slab readings were very close to 363.1 K, the vapor temperature varied from 363.8 K in the heated region to 363.4 K in the cooled region, 67.8 W was being transferred into the cooling jacket, and vapor pressure was about 7 bar. Many other experiments were conducted where internal temperatures and pressures were increased or decreased with time and where initial temperatures were above or below room temperature. In all cases where heat transfer was modest and coolant inlet temperature was well below the critical value, the heat pipe approached steady operation in a smooth fashion similar to that indicated in Fig. 4.

The solid lines in Fig. 4 show results of computations made using the approach described in the theory section of the present paper. The assumed gap between layers of material in the circumferential capillary structure is β. Gaps which actually occur in a well constructed layered capillary structure are typically within the values shown. Agreement between computed and measured values as shown in Fig. 4 is relatively good.

When the heat pipe was operated under more severe conditions partial drying of the capillary structure in the evaporator zone occurred. As an example of this type of behavior consider the data presented in Figs. 5–7. In this experiment the heat pipe was initially isothermal at 294 K. Heating was suddenly started and silicone fluid was introduced to the cooling jacket at 346 K. It is clear that partial drying occurred after about 100 min. However, under these conditions the drying

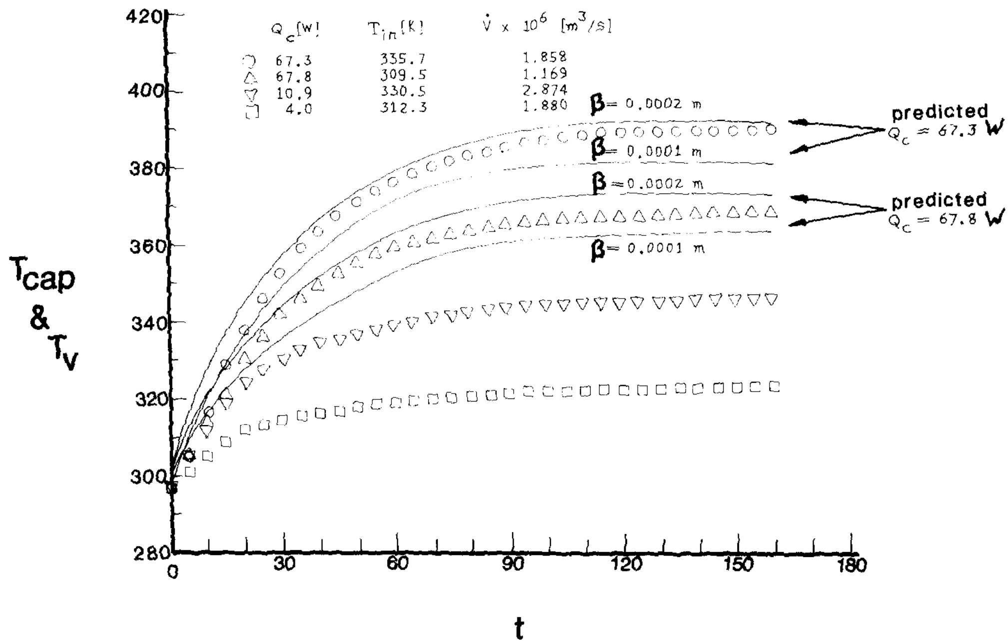

FIG. 4. Transient temperature responses under various operating conditions.

was contained and steady operation was reached. The vapor temperature at steady state varied from 453.7 to 437.7 K, the temperature in the dried section of the slab varied from 449.8 to 440.3 K and the wetted portion of the slab was nearly at a uniform temperature of 437.2 K. The pressure increased smoothly to a steady state value of 31 bar. Saturation temperatures are shown on Fig. 5 which correspond to measured pressures as shown on Fig. 7.

When operating temperatures are allowed to reach high levels, the capillary structure will become completely dried in the evaporator section and the heat pipe will fail as exhibited by very large temperature gradients, low overall conductance, and large internal pressures. Figures 8 and 9 show transient slab temperatures and internal pressures taken in an experiment where the heat pipe failed. In this experiment the heat pipe was initially at room temperature. Suddenly heat was added and coolant was introduced at 375.3 K. After about 50 min the capillary

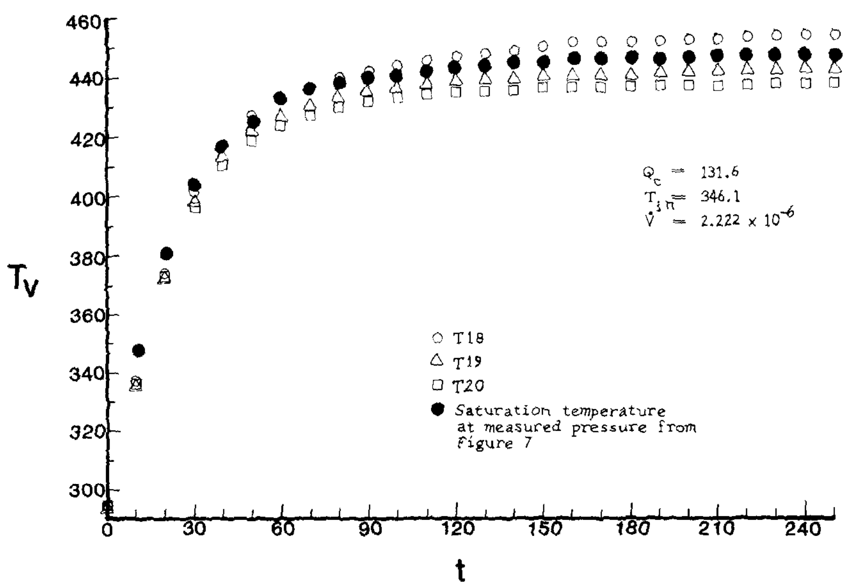

FIG. 5. Transient vapor temperature response.

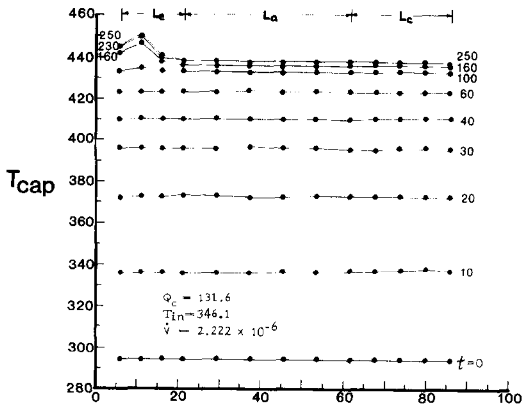

FIG. 6. Slab temperature distribution.

structure began drying and continued to do so until the device failed. For example, after about 85 min of operation, temperatures within the slab varied from 578 to 418 K while only 47.7 W was delivered to the cooling jacket. The pressure rose continuously to very high levels and clearly was not near a maximum value. Under these conditions the heated and cooled portion of the heat pipe was internally nearly thermally uncoupled and consequently the overall thermal conductance became quite small. In those cases where the working fluid may pass near the thermodynamic critical point during transient operation rather strange behavior may be observed. This is probably due to the fact that fluid properties such as thermal conductivity, density, and heat capacity may change dramatically in the near critical region. It is also important to point out that the amount of working fluid introduced into the interior of the heat pipe may greatly affect the rate of pressure rise which occurs as the heat pipe fails.

In a continuation of the experiment shown in Figs. 8 and 9 an attempt was made to restart the heat pipe after failure. The results of the rewetting process are shown in Figs. 10–12. After failure the coolant flow and heat inputs were stopped for a period of 15 min and as expected during this time temperatures in the capillary structure, as shown in Fig. 10, and temperatures in the vapor space, as shown in Fig. 11, began to approach an isothermal condition and internal pressure approached a steady value. After 15 min with no coolant flow and no heat input, coolant was suddenly introduced at 357 K and as Figs. 10–12 show the heat pipe responded rather quickly. After a few minutes temperatures had become much lower in both the vapor and liquid regions and pressure had decreased smoothly to a much lower value. From a detailed study of the large volume of experimental data gathered during this experiment, it seems clear that rewetting had occurred in the capillary structure and that thermal coupling between evaporator and condenser ends had occurred. Notice, however, that after about 6 min of coolant flow, temperature gradients in both vapor and liquid regions were far greater than before failure occurred. It has been found during this study that very long times may be required for internal temperature gradients, particularly in the liquid region, to approach normal values during restart after a dryout.

In another experiment where the heat pipe was operated at high heat fluxes with large inlet coolant temperature, the capillary structure completely dried and extremely large gradients developed within. Figures 13 and 14 show conditions during restart. After failure heat input and coolant flow were stopped for a period of 15 min and then coolant was introduced into

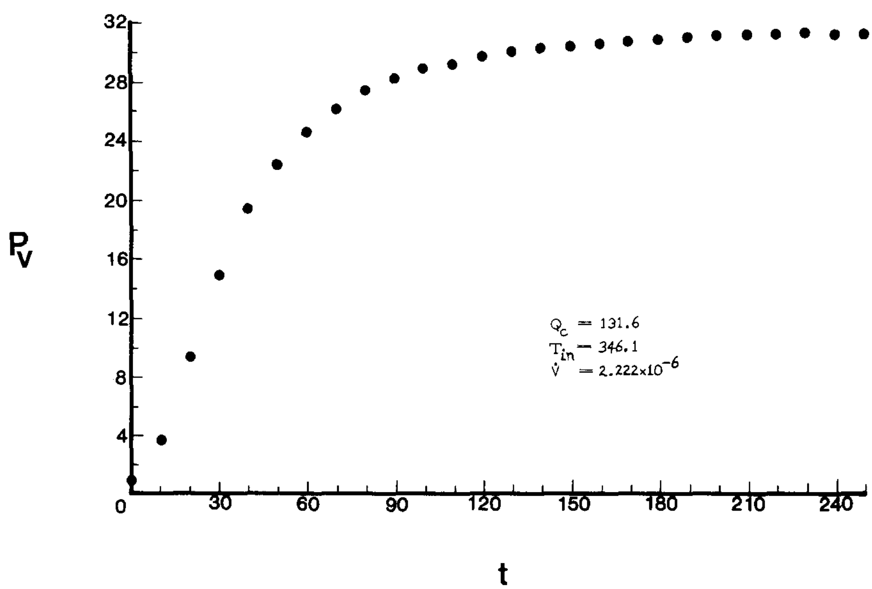

FIG. 7. Transient pressure response.

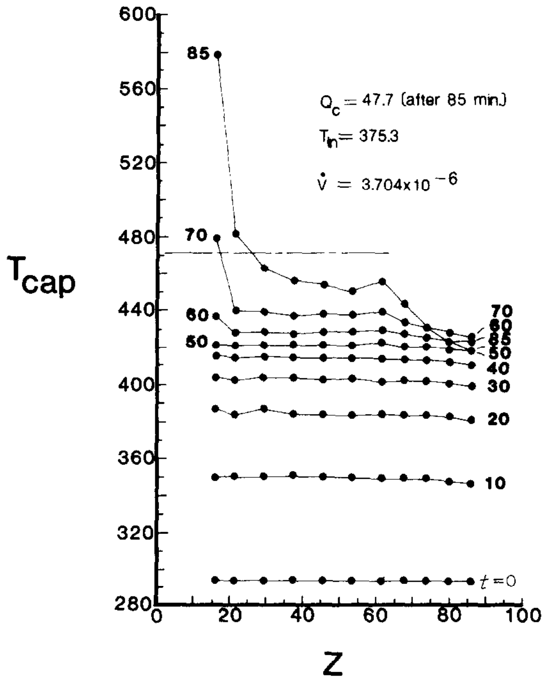

FIG. 8. Slab temperature.

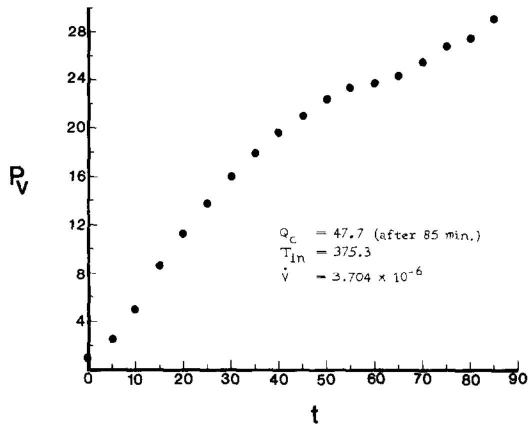

FIG. 9. Measured vapor pressure.

the coolant jacket at a temperature of 371 K. The internal pressure quickly and smoothly decreases when cooling is initiated as shown in Fig. 14. However, as Fig. 13 indicates, while temperatures quickly fall within the capillary structure as external cooling is started, extremely large gradients still exist after 45 min of cooling with no heat input. Unfortunately some thermocouples failed during the extreme conditions which existed during failure and therefore less detailed information in the heated section is available than in previous runs. The data is sufficiently detailed to show that a long time period will be required to approach normal heat pipe operation even with small or no heat input.

Figure 15 shows predicted vapor temperatures when the heat pipe was initially isothermal at a supercritical temperature of 480 K. At time zero, heat is suddenly introduced at a constant rate of 10 W at the evaporator surface and coolant suddenly flows into the coolant jacket at 360 K. At any instant of time the value of vapor temperature shown in Fig. 15 is a mean value determined from the temperature distribution within the vapor space at that instant of time. Direct comparison can not be made between predictions in this figure and the measurements discussed earlier since initial and operating conditions are quite different in

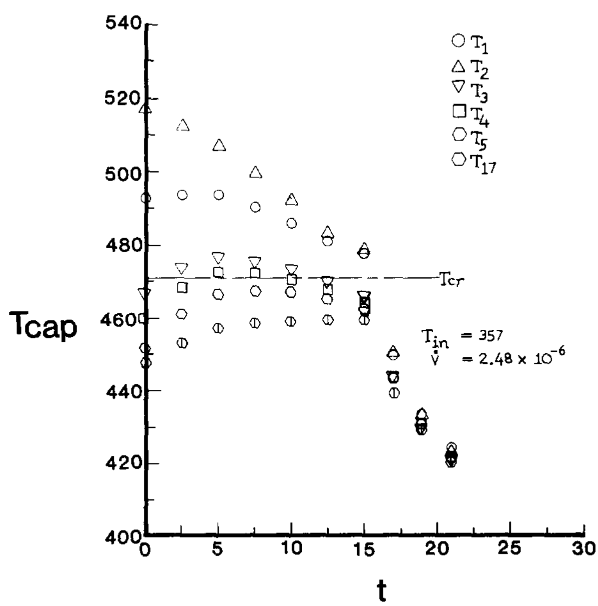

FIG. 10. Slab temperature during restart.

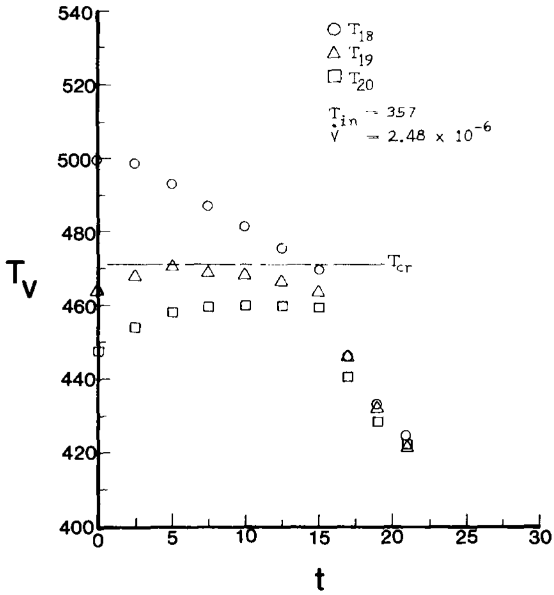

FIG. 11. Vapor temperature during restart.

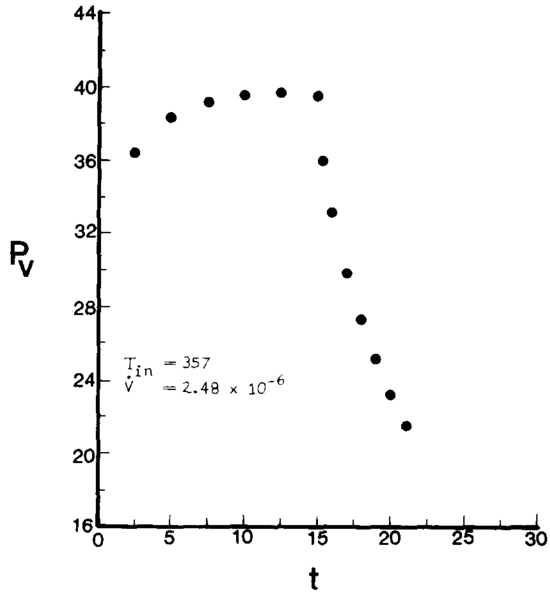

FIG. 12. Vapor pressure during restart.

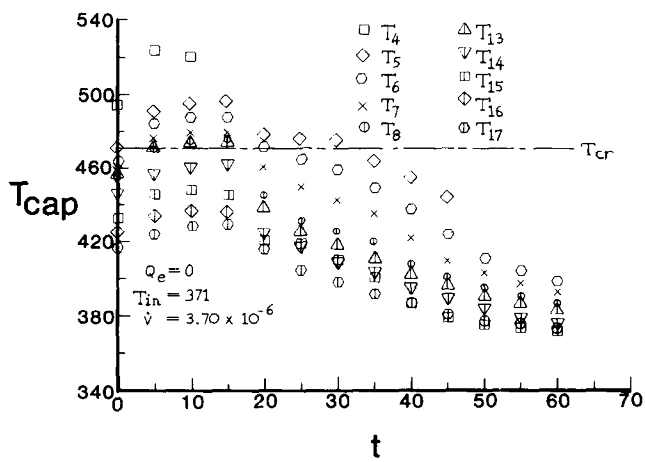

FIG. 13. Slab temperatures during restart.

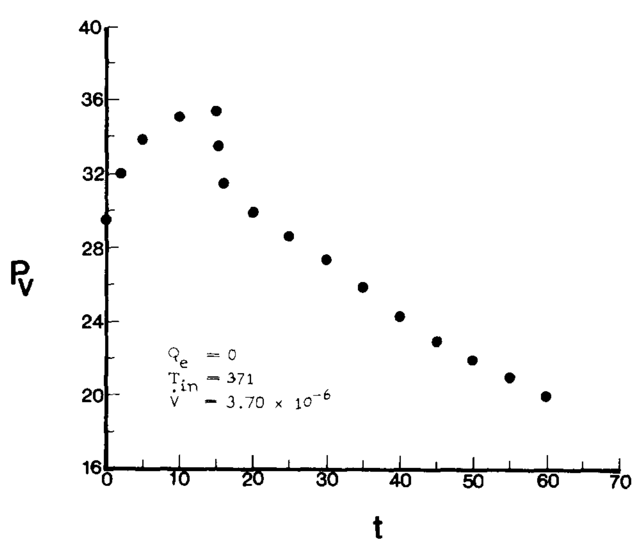

FIG. 14. Vapor pressure during restart.

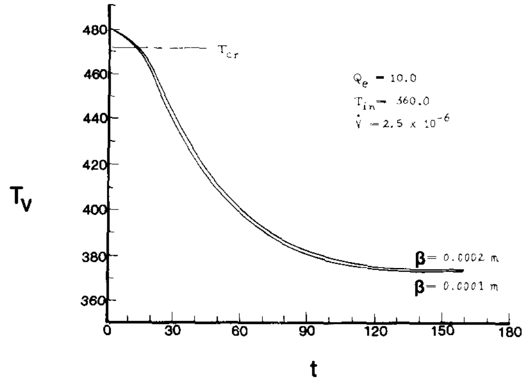

FIG. 15. Predicted startup from the supercritical state.

the two cases. However, it is clear that the models described correctly predict final steady conditions and that shapes of measured and predicted curves are similar. In order to accurately check the predictions of startup from the supercritical state it will be necessary to construct a different test section which can operate at higher temperature and pressure levels.

Based on theoretical and experimental results obtained in this study it is clear that many parameters affect heat pipe performance. Performance may be measured by overall conductance, internal temperature and pressure distributions, and internal liquid and vapor velocity distributions. This performance is affected by both design parameters and operating parameters. Design parameters include mass and arrangement of components, capillary structure geometry, working fluid, component materials and methods of supplying and removing heat. Operating parameters include volume flow through the cooler, inlet temperature to the cooler and power input to the heated section. When design and operating parameters are selected, performance becomes dependent.

CONCLUSIONS

Heat pipe dependent operating parameters such as vapor pressure, vapor temperature, working fluid flow rate, and overall conductance change smoothly in response to changes in independent parameters such as temperature of the external surface at the heated end and inlet coolant temperature and flow rate at the cooled end when conditions within the pipe are normal, i.e. when the capillary structure is fully wetted, the working fluid is not frozen or in the thermodynamic supercritical region, vapor choking does not occur, and boiling does not occur in the evaporator section.

During transient operation under adverse conditions a heat pipe may fail as exhibited by rapid large increases in internal pressure and temperature and decrease in overall conductance or it may require a long time for startup from a frozen state or a supercritical state. Transient normal operation can be predicted accurately using transient conduction equations applied to the shell and to the combination fluid—capillary structure with appropriate initial and boundary conditions. Under adverse conditions, account must be taken of internal fluid dynamics and predictions become much more difficult.

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REFERENCES

1. W. S. Chang, Heat pipe startup from the supercritical state, Ph.D. thesis, Georgia Institute of Technology, March (1981).
2. G. T. Colwell, Transient heat pipe operation in the near-critical region, *Advances in Heat Pipe Technology*, pp. 289–296. Pergamon Press, New York (1981).
3. D. E. Priester, Transient response of a cryogenic heat pipe, M.S. thesis, Georgia Institute of Technology, August (1976).
4. C. L. Williams, Correlation of heat pipe parameters, Ph.D. thesis, Georgia Institute of Technology, March (1973).
5. C. L. Williams and G. T. Colwell, Heat pipe model accounting for variable evaporator and condenser lengths, *AIAA J.* **12** (9), 1261–1267 (1974).
6. M. Groll and P. Zimmermann, Das Stationäre und Instationäre Betriebsverhalten von Wärmerohren, *Wärme- und Stoffübertragung* **4,** 39–47 (1971).
7. A. V. Luikov, Systems and differential equations of heat and mass transfer in capillary-porous bodies, *Int. J. Heat Mass Transfer* **18,** 1–14 (1975).
8. M. S. Plesset and A. Prosperetti, Flow of vapour in a liquid enclosure, *J. Fluid Mech.* **78,** 433–444 (1976).
9. L. L. Vasilév and S. V. Konev, Heat pipes, *Heat Transfer–Soviet Res.* **6** (1), 1–97 (1974).
10. W. B. Hall, Heat transfer near the critical point, *Advances in Heat Transfer*, Vol. 7. Academic Press, New York (1971).
11. D. U. von Rosenberg, *Methods for the Numerical Solution of Partial Differential Equations*. Elsevier, New York (1969).
12. B. Carnahan, H. A. Luther and J. O. Wilkes, *Applied Numerical Methods*. Wiley, New York (1969).
13. A. E. Scheidegger, *The Physics of Flow through Porous Media* (3rd edn.). University of Toronto Press, Toronto (1974).

MESURE DU COMPORTEMENT VARIABLE D'UNE STRUCTURE CAPILLAIRE SOUS DIFFERENTES CHARGES THERMIQUES

Résumé—Les caractéristiques oprératoires variables d'une structure capillaire de type lame dans un caloduc à R11 sont étudiées sous des conditions qui occasionnent un assèchements partiel ou complet et un remouillage. On présente des changements dans le temps dans les distributions des températures du liquide et de la vapeur, dans les pressions internes et les capacités de transfert thermique. On trouve que l'équipement doit, sous certaines conditions, approcher le fonctionnement permanent avec une portion de structure capillaire sèche dont le remouillage peur être réalisé rapidement. Néanmoins quand toute la structure est sèche, le remouillage est beaucoup plus difficile à réaliser.

MESSUNG DES INSTATIONÄREN VERHALTENS EINER KAPILLARSTRUKTUR UNTER HOHER THERMISCHER BELASTUNG

Zusammenfassung—Das instationäre Betriebsverhalten einer plattenförmigen Kapillarstruktur in einem mit R11 gefüllten Wärmerohr wurde unter Bedingungen untersucht, bei denen partielles oder vollständiges Austrocknen und anschließende Wiederbenetzung auftraten. Die zeitlichen Veränderungen der Temperaturverteilung in Flüssigkeit und Dampf, des Drucks im Inneren und des Wärmetransportvermögens werden eingehend dargestellt. Dabei zeigt sich, daß der Apparat unter bestimmten Umständen einen stationären Betriebszustand erreicht, bei dem ein Teil der Kapillarstruktur ausgetrocknet ist, und daß ein Wiederbenetzen schnell bewirkt werden kann. Es ist jedoch sehr viel schwieriger, ein Wiederbenetzen hervorzurufen, wenn die gesamte Struktur ausgetrocknet ist.

ИЗМЕРЕНИЕ ХАРАКТЕРИСТИК КАПИЛЛЯРНОЙ СТРУКТУРЫ В ПЕРЕХОДНОМ РЕЖИМЕ ПРИ БОЛЬШОЙ ТЕПЛОВОЙ НАГРУЗКЕ

Аннотация—Рабочие характеристики переходного процесса в капиллярной структуре, имеющей форму пластины и помещенной в тепловую трубу с фреоном 11 в качестве рабочей жидкости, исследовались в условиях частичного или полного высыхания и повторного смачивания. Представлены подробные изменения во времени распределений температуры жидкости и пара, внутреннего давления и теплопередающей способности. Найдено, что в определенных условиях тепловая труба может выйти на стационарный режим работы, когда произошло высыхание только части капиллярной структуры и возможно быстрое повторное смачивание. Однако в случае полного высыхания повторное смачивание в значительной степени затруднено.