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Analysis of Capillary-Induced Rewetting in Circular Channels with Internal Grooves

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A combined theoretical and experimental investigation was conducted to determine the rewetting characteristics of capillary-induced liquid flow in circular channels with microgrooves on the inside surface. This investigation provides a theoretical description of the mechanisms that govern the rewetting of these surfaces, and presents an experimentally verified method by which the location of the rewetting front in the evaporator section of high capacity heat pipes can be determined as a function of the applied heat flux. The rewetting velocity was found to be a function of the thermal properties of the liquid and the channel, the input heat flux, the radii of the groove and the channel, and the location of the liquid front. The maximum heat flux under which rewetting would occur was found to be a function of all these factors plus the input heat flux distribution.

Nomenclature

c = specific heat of the channel
c' = experimental constant
F_{fric} = wall friction force
F_g = gravitational force
F_{surf} = surface tension force
g = gravitational acceleration
h = liquid rise height
h_f = latent heat of vaporization
k = thermal conductivity of the plate
$\dot{m}$ = mass
m_v = vaporization rate of the liquid
Pr = Prandtl number
Q = total heat transferred
q'' = heat flux
R = vapor channel radius
Re = Reynolds number
r = circumferential groove radius
T = temperature
T_s = liquid saturation temperature
T_w = surface temperature at the rewetting front
t = time
U = average velocity of the liquid
U_w = rewetting velocity
w = width of groove
α = thermal diffusivity
β = contact angle
γ = cone angle
δ = thickness of the channel wall
θ = angle
θ_w = local angle of the rewetting front
θ' = angle in moving coordinate frame
μ = absolute viscosity
σ = surface tension

Subscripts
e = effective
l = liquid
max = maximum
w = wetting

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Introduction

SEVERAL heat pipe configurations have been, or currently are under consideration for use as radiator elements in low Earth orbit, manned space platforms. Two of these, the Grumman monogroove heat pipe,[1] and the Lockheed graded-groove heat pipe,[2] shown in Figs. 1a and 1b, are comprised of two parallel circular channels, one for vapor flow and one for liquid flow. The liquid and vapor channels in these heat pipes are connected by a small longitudinal slot to provide axial pumping, while the small circumferential grooves on the inside of the vapor channel distribute the working fluid over the inner surface.

Under normal operating conditions, heat supplied to the horizontal fin of the evaporator is conducted around the circumference of the vapor channel and through the heat pipe walls to these circumferential grooves where the working fluid is vaporized. Because of the high pressure associated with the increased temperature at the evaporator, the vapor moves towards the cooler condenser where it condenses and gives up the latent heat. The liquid is then "pumped" back to the evaporator by the capillary forces in the longitudinal slot.

Under high thermal loads or adverse gravitational conditions, the longitudinal slot may not be capable of returning sufficient liquid to the circumferential wall grooves, resulting in dryout. When this occurs, it will be necessary to reduce the evaporator heat flux to allow the circumferential wall grooves to rewet. This rewetting process results from the high capillary pressure induced by the small characteristic radius of the wall grooves, and is extremely complex. The fluid flow in the grooves is further complicated by the physical geometry of the two heat pipes of interest here.

Several researchers, including Elliott and Rose,[3] Simopoules et al.,[4] Stroes et al.,[5] Ueda et al.,[6,7] Elias and Yadigaroglu,[8] and Raj and Pate,[9] have investigated the rewetting characteristics of liquid films on heated or hot rods, tubes, or flat surfaces. Saeed and Peterson[10] have reviewed the work of these and others and found that although these investigations have provided substantial experimental data and considerable insight into the behavior of these films on both circular tubes and flat plates, no general physical model exists which is capable of describing the governing phenomena or the behavior of the liquid on grooved surfaces.

More recently, Peng and Peterson[11–14] have proposed several models for investigating the rewetting behavior of specific cases, including the wetting and rewetting of heated and un-

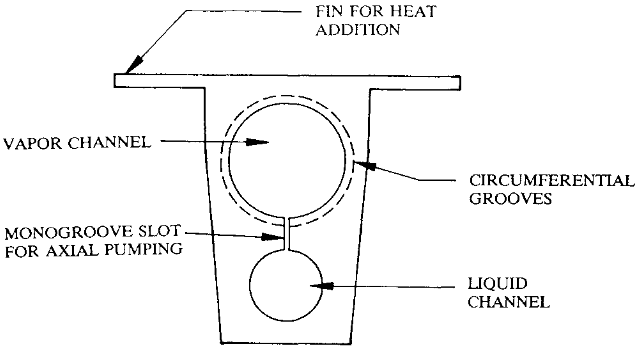

Fig. 1a Monogroove heat pipe.[1]

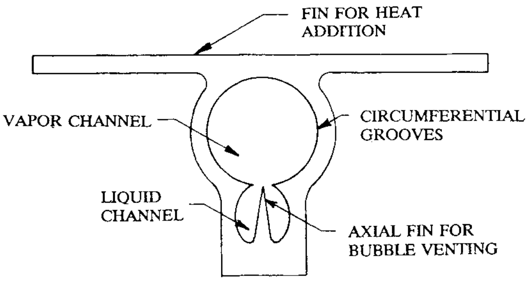

Fig. 1b Graded groove heat pipe.[2]

heated flat surfaces, and the rewetting of porous cover layers. In all of these models, the vaporization of the liquid at the rewetting front was assumed to be an important aspect of rewetting. Consequently, the rewetting velocity was first related to the liquid flow velocity, then exact analytical solutions were obtained to determine the rewetting velocity for a thin liquid film as a function of the bulk liquid flow velocity, the applied heat flux, the thermophysical properties of the liquid, and the surface over which the liquid was flowing. These theoretical analyses were then compared with several different sets of experimental data.[15] As a result of this work, a generalized model which described the rewetting of flat heated plates was presented and discussed.[16]

All of these investigations have focused on the rewetting behavior of thin liquid films on flat heated surfaces with either a porous cover layer or a series of relatively large parallel grooves, and are significantly different from the case of a circular surface with circumferential grooves.

Development of the Physical Model and Theoretical Solution

In the evaporator section of the high-capacity heat pipes (shown in Figs. 1a and 1b) the liquid flow is driven by surface tension and, therefore, for a wetting fluid, the liquid film will never be higher than the upper surface of the top-most groove. This simplifies the problem considerably in that it allows each parallel groove to be evaluated independently. To further simplify the problem, assume initially that the solid surface is at ambient conditions (i.e., not heated) and that the liquid wets the surface (i.e., the wetting angle is zero) forming a continuous film. For this case, the liquid layer is subjected to a wall shear stress, a gravitational force, which may either assist or hinder the rewetting depending upon the orientation of the channel with respect to gravity, and a capillary driving force. If an integral analytical method is utilized with the entire liquid layer taken as the control volume, and the liquid mass is assumed to increase as the length of the liquid layer increases, the length of the liquid layer for a given time (t) is fixed and the liquid velocity is the same for any point in the control volume (by continuity). As a result, the liquid film advances at an average tangential velocity (U) which will vary with respect to the length of the liquid film. The physical model for this case is shown in Fig. 2. As illustrated, the flow is assumed to be one-dimensional curvilinear flow, and the

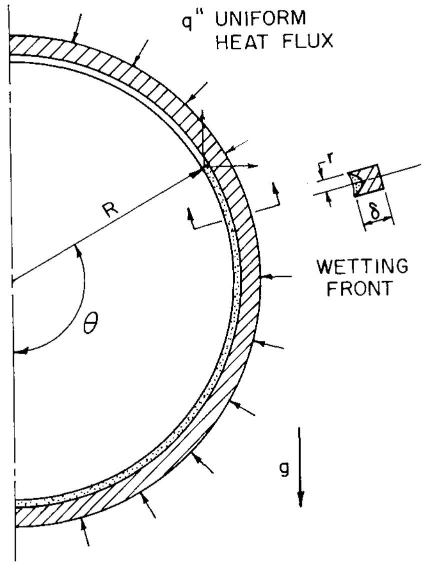

Fig. 2 Physical model.

liquid film thickness is assumed to be constant along the groove, as are the thermophysical properties of both the liquid and the channel. In addition R is assumed to be very large compared with r and δ.

Based on the considerations above, Newton's law for the liquid film in the groove can be written as

$$F_{\text{surf}} - F_g - F_{\text{fric}} = \frac{\mathrm{d}}{\mathrm{d}t}(mU) \tag{1}$$

For a wetting fluid, the capillary driving force can be expressed as

$$F_{\text{surf}} = (2\sigma/r)(\pi r^2/2) = \pi r \sigma \tag{2}$$

The gravitational force is much more complex since the orientation of the groove with respect to the gravitational vector changes for different positions. However, an expression for the gravitational force as a function of the angular position can be expressed as

$$F_g = \int_0^\theta \frac{\pi}{2} r^2 \rho_l g \sin\theta R \,\mathrm{d}\theta = \frac{1}{2}\pi r^2 R \rho_l g \int_0^\theta \sin\theta \,\mathrm{d}\theta$$

$$= \frac{1}{2}\pi r^2 R \rho_l g (1 - \cos\theta) \tag{3}$$

For Newtonian fluids in laminar flow, the shear stress at the wall can be expressed as

$$\tau_w = \mu_l \left.\frac{\partial u}{\partial r}\right|_{\text{at wall}} \approx \mu_l \frac{U}{r/2} = \frac{2\mu_l U}{r} \tag{4}$$

Using this value, an expression for the wall friction can be derived as

$$F_{\text{fric}} = \tau_w \theta R \pi r = (2\mu_l U/r)\theta R \pi r = 2\pi \mu_l R U \theta \tag{5}$$

The differential time, dt, in Eq. (1) can be replaced by

$$\mathrm{d}t = (R\,\mathrm{d}\theta/U) \tag{6}$$

$$m = \tfrac{1}{2}\pi r^2 R \theta \rho_l \tag{7}$$

so

$$\frac{\mathrm{d}}{\mathrm{d}t}(mU) = \frac{1}{2}\pi r^2 R \rho_l \left(U \frac{\mathrm{d}\theta}{\mathrm{d}t} + \theta \frac{\mathrm{d}U}{\mathrm{d}t}\right) \tag{8}$$

Substituting Eqs. (2), (3), (5), and (8) into Eq. (1) yields

$$\pi r\sigma - \frac{1}{2}\pi r^2 R\rho_l g(1 - \cos\theta) - 2\pi\mu_l RU\theta = \frac{1}{2}\pi r^2\rho_l\left(U^2 + \theta U\frac{dU}{d\theta}\right) \tag{9}$$

or

$$\frac{dU}{d\theta} = \frac{2\sigma - rR\rho_l g(1-\cos\theta) - (4/r)\mu_l RU\theta - r\rho_l U^2}{r\rho_l\theta U} \tag{10}$$

with boundary conditions

$$\theta = 0, \qquad U = 0 \tag{11}$$

Equation (10) represents an expression for U as a function of the angular position of the wetting front θ, and assumes that the surface is at ambient conditions, i.e., uniform temperature, with no heat addition. Because of the functional dependence of the velocity on position, a direct analytical solution is not possible, and therefore, a numerical technique must be employed.

Of greater interest in heat pipe applications is the case of rewetting of a surface after dryout, i.e., with heat addition. For this case, the liquid front is assumed to be driven by capillary pressure and to advance along the grooves in the channel wall to which a uniform circumferential q'' has been applied, as shown in Fig. 2. Because the surface is at an elevated temperature (at or near the Leidenfrost temperature, that temperature where the liquid will not remain in contact with the surface due to rapid vaporization), some of the liquid at the leading edge of the advancing liquid front is vaporized. Assuming that vaporization occurs only at the leading edge of the wetting front, the remaining liquid advances with a velocity U_w, referred to as the wetting front velocity. The heat required to vaporize this liquid is supplied by conduction from the dry hot zone of the channel surface. When dryout occurs, the heat supplied to the wetting front exceeds that required to vaporize all of the liquid. For the case of no heat addition previously described, no vaporization occurs and the wetting front velocity and liquid velocity are equal. However, for a heated plate, the liquid velocity is higher than the wetting front velocity, since some of the liquid is vaporized, thereby reducing the liquid mass flow rate.

To determine the amount of heat absorbed by the vaporization process, the conduction equation for the channel wall can be transformed to a coordinate system moving with a velocity equal to that of the wetting front, U_w. In this analysis, a curvilinear coordinate system was employed and the conduction through the walls was assumed to be one-dimensional. Several other fundamental assumptions were made and can be summarized as follows:

1) The grooves on the surface were assumed to be located immediately adjacent to one another, and the radius of the grooves were assumed to be much smaller than the thickness of the wall (i.e., $r << \delta$).

2) The liquid temperature at the rewetting front T_l was assumed to be different from the rewetting or Leidenfrost temperature T_w, which was assumed to be constant and equal to T_s.

3) The convective heat transfer between the plate and vapor along with radiation from the heated surface and vapor was neglected.

Utilizing these assumptions, the conduction equation for the wall can be written as

$$\frac{k}{R^2}\frac{\partial^2 T}{\partial\theta'^2} + \frac{q''}{\delta} = \rho c\frac{\partial T}{\partial t} \tag{12}$$

or

$$k\frac{d^2T}{d\theta'^2} + \frac{q''R^2}{\delta} = \rho cRU_w\frac{dT}{d\theta'} \tag{13}$$

The general solution of Eq. (12) is

$$T = C_1 + C_2\exp(\rho cU_w R\theta'/k) + (q''R\theta'/\rho cU_w\delta) \tag{14}$$

For individual grooves of finite length, the condition at the end of the grooves are very difficult to predict. However, the issue of interest here is the determination of the conditions at the wetting front. Therefore, it is reasonable to assume grooves of infinite length, to obtain an approximate solution. Differentiating Eq. (14) with θ' yields

$$\frac{dT}{d\theta'} = C_2\left(\frac{\rho cU_w R}{k}\right)\exp\left(\frac{\rho cU_w R\theta'}{k}\right) + \left(\frac{q''R}{\rho cU_w\delta}\right) \tag{15}$$

When $\theta' \to \infty$, $dT/d\theta' \to \infty$ this is physically impossible which implies that $C_2 = 0$. At the rewetting front, however, the plate temperature can be assumed to be equal to the Leidenfrost or rewetting temperature, i.e., $\theta' = 0$, $T = T_w$ and using this condition a solution for Eq. (15) can be derived as

$$T(\theta') = T_w + (q''R/\rho cU_w\delta)\theta' \tag{16}$$

From this expression, the total heat conduction at $\theta' = 0$ can be found as

$$Q = \left(2r\delta - \frac{1}{2}\pi r^2\right)k\frac{\partial T}{R\, d\theta'}\bigg|_{\theta'=0} \approx 2r\delta k\frac{dT}{R\, d\theta'}\bigg|_{\theta'=0} \tag{17}$$

or

$$Q = (2r\delta kq''/\rho cU_w\delta) = (2rq''\alpha/U_w) \tag{18}$$

Since the total heat conduction in the region of the wetting front is equal to the energy absorbed by the liquid vaporization

$$Q = \tfrac{1}{2}\pi r^2(U - U_w)\rho_l h_f \tag{19}$$

Combining Eqs. (16) and (18) yields

$$U_w^2 - UU_w + (4q''\alpha/\pi r\rho_l h_f) = 0 \tag{20}$$

Solving for the rewetting velocity yields

$$U_w = \tfrac{1}{2}[U \pm \sqrt{U^2 - (16q''\alpha/\pi r\rho_l h_f)}] \tag{21}$$

As noted in Ref. 11, the correct solution occurs when the second term of Eq. (21) is positive. Hence, the rewetting velocity is

$$U_w = \tfrac{1}{2}[U + \sqrt{U^2 - (16q''\alpha/\pi r\rho_l h_f)}] \tag{22}$$

where, the velocity of liquid flowing in the groove is given by Eq. (10).

Analysis and Discussion

Clearly, the liquid velocity U, as determined from Eq. (10), is dependent upon the thermophysical properties of the liquid, the radii of the vapor channel and the grooves, and the location of the liquid front. The rewetting velocity as determined by Eq. (22) is also a function of these parameters, along with the applied heat flux and the thermophysical properties of the channel. As was the case for the liquid velocity, the rewetting velocity can only be obtained numerically.

In order to solve for the real root which describes the rewetting velocity given in Eq. (20), the following relation must hold true:

$$U^2 - (16q''\alpha/\pi r\rho_l h_f) \geq 0 \tag{23}$$

Upon rearranging, this yields

$$q''_{max} \leq (\pi r\rho_l h_f/16\alpha)U^2 \tag{24}$$

This expression implies that for a specified liquid and channel, (i.e., given the channel and groove radii, and the liquid and solid properties) the applied heat flux cannot exceed the value predicted by Eq. (24). This heat flux is the maximum heat flux under which rewetting can occur. In addition, since the liquid velocity U is a function of the angular position θ, the maximum steady-state location to which the liquid front will advance can be determined.

As noted in several previous investigations[11–14] liquid sputtering and variations in the liquid/solid contact angle may have significant effects on the rewetting of hot surfaces. Although completely incorporating these factors into the equations developed here is difficult, some simplifying assumptions may provide insight into how these factors affect the rewetting mechanisms.

If the liquid does not fully wet the wall, the effect of the wetting angle must be included in the driving force term given in Eq. (1) or

$$F_{surf} = (2\sigma \cos \beta/r)(\pi r^2/2) \tag{25}$$

With this modification, Eq. (10) can be modified such that

$$\frac{dU}{d\theta} = \frac{2\sigma \cos \beta - rR\rho_l g(1 - \cos \theta) - (4/r)\mu_l RU\theta - r\rho_l U^2}{r\rho_1 \theta} \tag{26}$$

From this expression, the liquid layer velocity U can be obtained, and therefore, U_w as a function of the contact angle can also be predicted.

Evaporation of the liquid will occur only after the liquid has been heated to a level somewhat higher than the liquid saturation temperature. At the wetting front, only a thin layer of the liquid immediately adjacent to the plate will reach a sufficiently high temperature to be vaporized. Because of the rapid boiling that occurs in this thin layer, the liquid above it may be carried off by sputtering. To compensate for the liquid carried off by sputtering, an expression for the thermal boundary layer thickness for laminar flow is presented by Kays and Crawford[17]

$$\delta_t/x = 4.64Re_x^{-1/2}Pr^{-1/3} \tag{27}$$

can be modified to obtain

$$\delta_t/r = c'Re_r^{-1/2}Pr^{-1/3} \tag{28}$$

where for the case of interest here, $Re_r = Ur\rho_1/\mu_l$, c' is a constant determined experimentally, and δ_t is the thickness of the evaporated liquid layer. With these modifications, Eq. (19) becomes

$$Q = \pi r\delta_t(U - U_w)\rho_l h_f \tag{29}$$

Combining Eqs. (29) and (18) yields

$$\pi r\delta_t(U - U_w)\rho_l h_f = (2rq''\alpha/U_w) \tag{30}$$

or

$$U_w^2 - UU_w + (2q''\alpha/\pi\rho_l h_f\delta_t) = 0 \tag{31}$$

Solving for the rewetting velocity yields

$$U_w = \tfrac{1}{2}[U + \sqrt{U^2 - (8q''\alpha/\pi\rho_l h_f\delta_t)}] \tag{32}$$

Substituting the expression in Eq. (28) yields a rewetting velocity of

$$U_w = \tfrac{1}{2}[U + \sqrt{U^2 - (8U^{1/2}Pr_l^{1/3}q''\alpha/c'\pi\rho_l^{1/2}\mu_l^{1/2}h_f r^{1/2})}] \tag{33}$$

and a maximum heat flux for which complete rewetting will occur of

$$q''_{max} = [c'\pi(\rho_l\mu_l)^{1/2}h_f r^{1/2}U^{3/2}/8Pr_l^{1/3}\alpha] \tag{34}$$

Although the solution presented here is for the simplest case of a circular channel with uniform heat flux on the outer surface, solutions for other more complex geometries can be obtained for different types of heat flux distributions. These solutions require only that the geometry be well-defined and that the heat flux distribution at the outer surfaces be known. For example, in the case illustrated in Figs. 1a and b where heat is transferred through the top surface, a two-dimensional computational technique must be developed to determine the circumferential heat flux distribution in the vapor channel. Once this has been done, Eqs. (10) and (33) can be used to determine the liquid velocity and rewetting velocity as a function on θ and/or the applied heat flux, and Eq. (34) can be used to predict the maximum sustainable heat flux which would allow complete rewetting.

Experimental Investigation

To further investigate the rewetting characteristics of this particular problem, verify the preceding theoretical analyses, and determine experimentally the constant c' included in Eqs. (33) and (34), an experimental investigation was conducted. A schematic of the experimental apparatus utilized in this investigation is shown in Fig. 3. The test section was made of stainless steel plate with sharp pointed V-shaped grooves machined into the surface with a horizontal milling machine. The radius of the circular test section was 12.7 mm and the width and apex angle of the circumferential grooves was 0.635 mm and 60 deg, respectively. The test sections were insulated on the backside and heated by directly applying a dc current through it.

In this experiment, the liquid height or liquid front location was measured using an angular scale attached directly to the test section. This scale provided a method by which the liquid height or front location could be measured with an experimental uncertainty of ±0.5 deg. The heating current and voltage were measured to determine the heat flux supplied

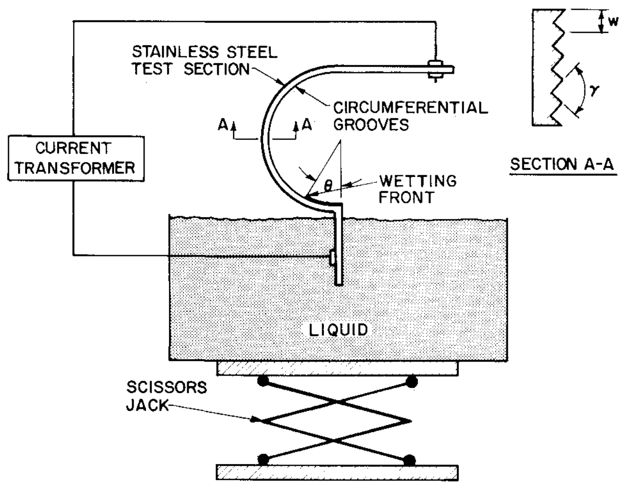

Fig. 3 Experimental test facility.

to the test section. Three different liquids, R11, methanol, and acetone, were used as the working fluid for the experimental investigation.

The experiment was conducted for three different liquids over a wide range of applied heat fluxes. When the wall temperature of the test section was higher than the saturation temperature corresponding to the ambient pressure (approximately 60–100°C), the test section was lowered into the liquid until the liquid level reached the location where $\theta = 0$, as shown in Fig. 3. After the liquid rise height had reached a maximum steady-state value, the angular position of the wetting front and heat flux were measured. No attempt was made to measure the rewetting velocity in the experiment due to the difficulty associated with this measurement.

As mentioned above, the rewetting is induced by the surface tension forces occurring in the circumferential wall grooves. Therefore, the effective capillary radius of the grooves is an extremely important factor in this problem. Here, the effective capillary radius for the three different test plates was experimentally determined from

$$(2\sigma/r_e) = \rho_l g h \tag{35}$$

where the effective capillary radius is the groove width.

For different liquids, the liquid rise height without heating could be measured. The effective capillary radius was therefore calculated by Eq. (35), and the average capillary radius from the data obtained for all three liquids, was 0.26 mm with a variation of less than $\pm 1\%$.

Results and Discussion

To analyze the theoretical and experimental results, a numerical technique was used to find the average tangential liquid velocity as a function of θ, assuming the test section was at ambient temperature (i.e., not heated). Although the analytical expression Eq. (10) derived previously was for semicircular grooves, not the V-shaped grooves used in this investigation, V-shaped cross-sectional area and hydraulic diameter were approximately equivalent to semicircular grooves with a radius equal to the experimentally determined radius. The resulting tangential velocities are illustrated in Fig. 4 as a function of angular position θ for Liquid R11, methanol, and acetone. The numerically determined maximum angular position, corresponding to that point where the velocities are equal to zero, are in good agreement with the measured values of 1.52, 2.21, and 2.36 rad for R11, methanol, and acetone, respectively.

The experimental data for the test conducted on the three heated plates are shown in Fig. 5, where the applied heat flux is plotted as a function of the liquid rise angle. Based on the experimental data in Fig. 5, c' was estimated to be 0.0021 for all three liquids. In addition to the experimental data, the theoretical results of maximum liquid rise angle for specified heat flux, predicted by Eq. (34) with $c' = 0.0021$, are also presented in Fig. 5. As illustrated, the theoretical results agree quite well with the experimental data. This, at least, implies that the physical model, mathematical description, and the solution methods are effective and useful in the analyses of this type of problem.

It is worthy to note that the modification developed to account for the effect of sputtering is very important. The experimental observation clearly indicates that the sputtering phenomena existed in all test cases. As previously described in Ref. 12, the computed results from Eq. (24), in which the modification for sputtering is not included, are significantly higher than those obtained from Eq. (34).

The heat pipes described earlier and depicted in Figs. 1a and 1b have $R = 7.6$ mm and $r = 0.067$ mm. Using these dimensions, the resulting liquid velocity can be expressed as a function of location of the wetting front using Eq. (10) and can be determined numerically. Once this has been done, the variation of maximum sustainable heat flux with respect to the angular position can be found from Eq. (34) using $c' = 0.0021$. Figure 6 illustrates the results of this process for several different liquid/solid material combinations. These results indicate that as the heat flux increases, θ which describes the location of wetting front, decreases. Clearly, when the liquid rewets the entire channel surface, the applied heat flux is less than or equal to the value at that point where $\theta = \pi$. For different liquid/material combinations, the applied heat flux under which liquid can rewet the entire channel surface is different, ammonia/copper, $q'' = 3150 w/m^2$, R11/copper, $q'' = 73.4 w/m^2$, ammonia/aluminum, $q'' = 3925 w/m^2$, and R11/aluminum, $q'' = 73.4 w/m^2$.

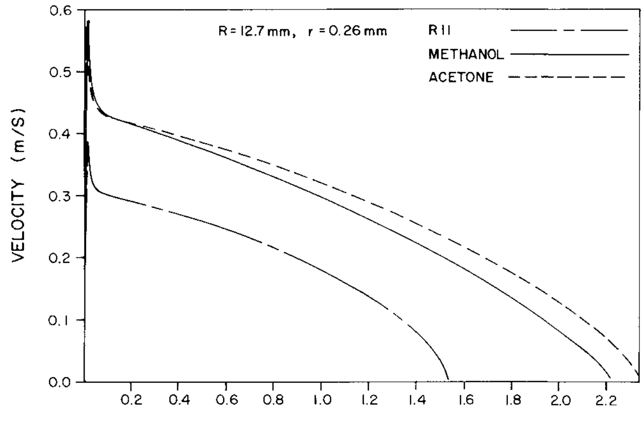

Fig. 4 Liquid velocity as a function of radial position.

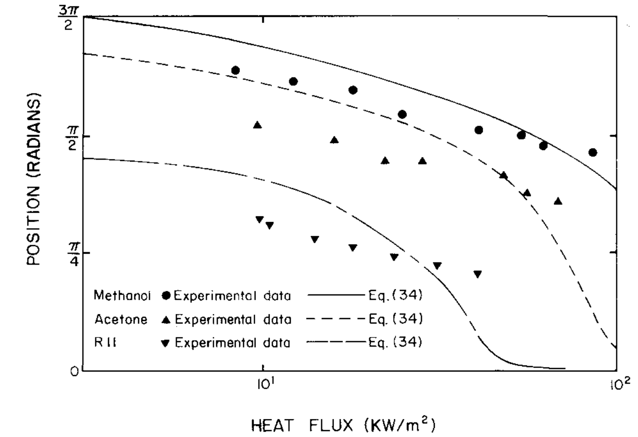

Fig. 5 Comparison of a predicted and experimentally measured maximum angular position as a function of applied heat flux.

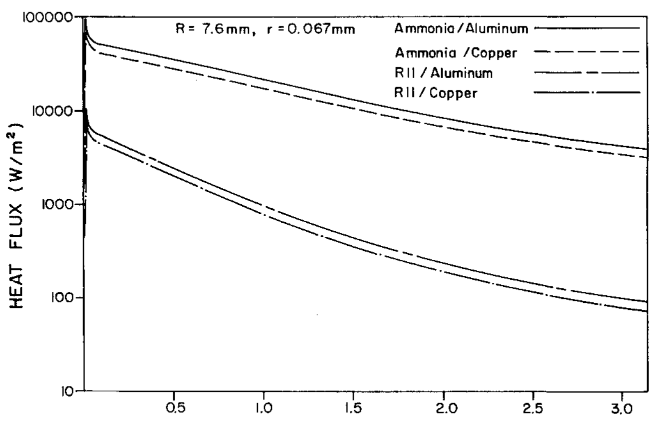

Fig. 6 Predicted maximum radial position as a function of applied heat flux for heat pipes with $R = 7.6$ mm and $r = 0.067$ mm.

Conclusions

A theoretical investigation has been conducted and a physical model has been developed to determine the rewetting characteristics of capillary-induced liquid flow in circular channel with microgrooves on the inside surface. This investigation provides a theoretical description of the mechanisms that govern the rewetting of these surfaces. The rewetting velocity was found to be a function of thermal properties of liquid and the channel, the input heat flux, the radii of the groove and the channel, and the location of the liquid front. The maximum heat flux under which rewetting would occur was found to be a function of all these factors plus the input heat flux distribution.

An experimental investigation was conducted in parallel to verify the modeling technique and to determine the maximum rewetting position as a function of the applied heat flux. Comparison of the analytical and experimental results indicated good agreement and demonstrated that the rewetting front position as a function of the applied heat flux could be determined for the two proposed heat pipe configurations with a reasonably high degree of accuracy.

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