Flooding and Dry-Up Limits of Circumferential Heat Pipe Grooves

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Theme

HIGH-performance heat pipes using axial arteries to minimize viscous pressure losses often employ circumferential vee-grooves to provide capillary paths from the artery to the heat transfer surface in the evaporator and condenser regions.[1] The condenser grooves serve to gather up condensate and conduct it to the axial artery, whereupon it is transported the length of the pipe to the evaporator. Various artery configurations have employed spirals with and without tunnels, pedestals, and diametral or bottom slab wicks.[1,2] The evaporator grooves conduct liquid from the artery (or a webbing contact) around the evaporator wall.

In a zero-g field when the heat pipe is off, the liquid and vapor in the pipe equilibrate and cease to flow, and the pressure is constant throughout the liquid, as in the vapor phase. With constant vapor-liquid pressure difference or "stress," all the grooves fill with liquid uniformly and adopt an everywhere-equal curvature. When a heat load is imposed, evaporation from the heated grooves depletes them of liquid so that the radius of curvature is decreased. The decrease is accompanied by an increase in stress. The evaporated vapor condenses in the cooled grooves and fills them more plentifully with liquid thus decreasing the stress and raising the liquid pressure there. The liquid pressure differential then drives the condensate around the grooves into and down the axial artery and into the evaporator grooves.

If the stress approaches zero in the condenser, the grooves fill and overflow; hence, "flooding" occurs. Such flooding is undesirable, because it raises the condenser thermal resistance. In the evaporator section of the pipe the stress may rise to such a high level that the groves dry out, and that too is undesirable.

The action of gravity at right angles to the pipe axis shifts the flooded region into the lower portion of the condenser. Gravity heightens the danger of dry-out in the upper regions of the evaporator.

An analysis of circumferential groove flow[3] was based upon theory for a triangular groove.[4] Isothermal externally-grooved tubes fed from above as in a condenser, where the bottom tubes receive liquid from overhead dripping, were treated. Gravity aided the groove flow. This paper presents an analysis of the state of stress in a circumferential groove having an imposed uniform heat flux. Flow in the evaporator against gravity is considered, and zero-gravity conditions are treated. When vee-grooves approach dry-out, a singularity in the stress occurs, and it is necessary to have a theoretical basis from which to start numerical calculations. This basis is developed, and numerical calculations are made to find optimum groove sizes.

Contents

A bottom slab-wick artery[2] was assumed (Fig. 1). Circumferential grooves with a sharp vee profile and a uniform heat flux in the interval $\theta=0$ to $\theta=\theta_w$ and over length L_e in the evaporator or length L_c in the condenser were assumed. In Ref. 3, the momentum equation for flow in the groove was derived under the assumptions that groove depth is small compared to the tube radius R_T and groove flow is laminar and quasi-established with negligible vapor shear. The friction factor coefficient K is a function of groove half angle θ_g and contact angle θ_c.[4] Stress P then is given by

$$\frac{\mathrm{d}P}{\mathrm{d}\theta}+R_T\rho_{\ell v}g\sin\theta-\frac{R_TK\mu V}{2D^2}=0 \tag{1}$$

where $\rho_{\ell v}$ is the liquid-vapor density difference, g = gravity, μ = liquid viscosity, and V = liquid velocity in the groove. Geometric relations[3] give area A and hydraulic diameter D as a function of P. Two cases are distinguished: 1) when the meniscus is attached to the groove tip; and 2) when the meniscus recedes, upon the contact angle θ_c falling to a breakaway value θ_{ba}.

Flooding: What is desired is the stress P_w where the grooves join the slab wick as a function of heat load $\dot{Q}$. The relationship arises from the flooding requirement that P go to zero somewhere in the domain $0\le\theta\le\theta_w$. One could find the relation numerically by fixing $\dot{Q}$ and assuming P_w and numerically integrating Eq. (1) backwards from θ_w. If P were everywhere positive, a lower value of P_w could be assumed, and the procedure repeated. However, this procedure is avoided as follows: "Flooding" is taken to occur not just where $P=0$, but where the liquid meniscus starts to lift off the groove tips, while remaining flat. Hence $\mathrm{d}P/\mathrm{d}\theta$ is zero. Equation (1) then gives the flooding angle θ_f. A parameter γ is defined with A, D, and K evaluated for $P=0$.

$$\gamma=K\mu\dot{Q}/4D^2\rho_{\ell v}g\rho\hat{o}_{\nu\xi}\mathrm{K}\Psi_{\xi}\Sigma_{\phi}\theta_w \tag{2}$$

Here ρ is liquid density, $\hat{h}_{fg}$ = latent heat of vaporization, and N_g = the number of grooves per unit length. The equation giving θ_f is then a transcendental one

$$\sin\theta_f-\gamma\theta_f=0 \tag{3}$$

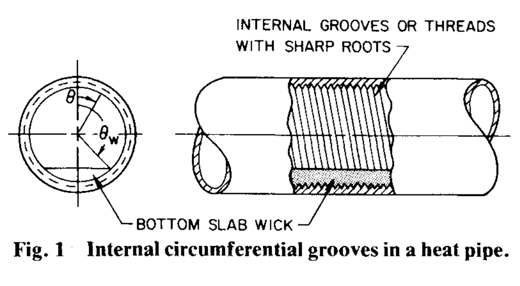

Fig. 1 Internal circumferential grooves in a heat pipe.

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When γ exceeds unity (for example, in the zero-g case, $\gamma=\infty$), there is no root to Eq. (3) other than $\theta_f=0$. When γ is less than unity, another root exists which fixes the position of flooding.

Equation (1) can then be integrated forward from θ_f, starting with $P=0$, to θ_w where $P=P_{wf}$. For values of P_w above P_{wf}, flooding will not occur, and for values below it, flooding will occur. Special design would be required for a P_w which permits high values of $\dot{Q}/L_c$ without flooding even when N_g is high. Either the pipe must be slightly "starved" for or "underfilled" with working fluid, a condition impossible to maintain under differing pipe temperatures, or the pipe must have an "excess fluid reservoir"[5] with the right configuration to provide P_{wf}, but this back stress constitutes a drain of axial pumping capacity. Thus coarse grooves are desirable in the condenser while fine grooves are used in the evaporator.

Dry-up: The point of incipient dry-up is always $\theta=0$ opposite the bottom slab wick. What is desired is the relation between $\dot{Q}$ and P_w when $P\to\infty$ at $\theta=0$. There the gravitational term in Eq. (1) is negligible, and V may be eliminated to give

$$\frac{dP}{d\theta}+\frac{R_T K\mu\dot{Q}}{4D^2A\rho\hat{h}_{fg}N_gL_e\theta_w}\theta=0 \tag{4}$$

The quantity AD^2 goes like R^4, where R is the meniscus radius of curvature, $R=\sigma/P$ and σ is surface tension; that is, $AD^2=GR^4=G\sigma^4/P^4$, where

$$G(\theta_g,\theta_{ba})=\frac{4\sin^2\theta_g}{\sin^2\psi_{\max}}\times[\sin\psi_{\max}(\sin\psi_{\max}\cot\theta_g+\cos\psi_{\max})-\psi_{\max}]^3 \tag{5}$$

and $\psi_{\max}$ is $\pi/2-\theta_g=\theta_{ba}$. Let a dry-up scaling pressure be defined

$$P_{du}=\left[\frac{8\theta_wG\sigma^4\rho\hat{h}_{fg}N_gL_e}{3R_TK\mu\dot{Q}}\right]^{1/3} \tag{6}$$

Equation (4) then becomes

$$\frac{dP}{d\theta}+\frac{2\theta}{3P_{du}^3}P^4=0,\qquad P=P_{du}\theta^{-2/3} \tag{7}$$

A starting angle θ_0 and pressure P_0 is desired beyond which the full Eq. (1) is numerically integrated. There is adopted

$$R_T\rho_{\ell v}g\sin\theta_0=\epsilon_0\frac{2\theta_0}{3P_{du}^3}P_0^4 \tag{8}$$

where ϵ_0 is arbitrarily 0.001. Equations (7) and (8) are solved with $\sin\theta_0\doteq\theta_0$.

$$\theta_0=\left[\frac{2\epsilon_0P_{du}}{3R_T\rho_{\ell v}g}\right]^{3/4} \tag{9}$$

$$P_0=P_{du}\theta_0^{-2/3} \tag{10}$$

In addition to its utility as a starting condition, Eq. (7) possesses value for indicating the number of watt-in. per groove beyond the break-away point under zero-g conditions. At the angle $\theta=\theta_r$ where meniscus recession ends, $P=P_r$ is given by the geometry, but Eq. (7) applies too,

$$P_r=\frac{2\sin\psi_{\max}\sigma}{W}=P_{du}\theta_r^{-2/3},\quad \sin\psi_{\max}=\cos(\theta_g+\theta_{ba}) \tag{11}$$

Table 1 Heat flow beyond meniscus detachment

Fluid	ω [watt-in.]	ωN_g [watt-in./in.]
Water at 200°F	0.45	45.0
Ammonia at 80°F	0.066	6.6
Methanol at 80°F	0.038	3.8

Substituting Eq. (6) and defining ω, the watt-inches per groove, gives

$$\omega\equiv\left[\frac{\dot{Q}}{2R_T\theta_wN_gL_e}\right]R_T^2\theta_r^2=\left[\frac{G(\theta_g,\theta_{ba})}{6\cos^3(\theta_g+\theta_{ba})K(\theta_g,\theta_{ba})}\right]\left[\frac{\sigma\rho\hat{h}_{fg}}{\mu}\right]W^3 \tag{12}$$

The first grouping on the right-hand side of Eq. (12) depends on groove and meniscus shape fixed by θ_g and θ_{ba}. The second grouping is the heat pipe fluid figure of merit. The final term is W^3, giving insight into what is shown by numerical results, that decreasing W to obtain more *axial* pumping P_r decreases the ability of the grooves to pump circumferentially, despite the larger $N_g\doteq 1/W$.

The information in Table 1 is based on Eq. (12) where $W=0.008$ in. $=$ 0.20 mm, $\theta_g=30°$, $\theta_{ba}=0°$, and $N_g=100$ grooves per in. The quantity ωN_g is half the number of watt-inches per unit length of evaporator which can be pumped at zero-g around the circumferential length $R_T\theta_w$ when $P_w=P_r$.

To summarize, analysis has yielded a single dimensionless parameter γ, defined by Eq. (2), which, with Eq. (3), indicates the circumferential position most susceptible to flooding. Numerical calculations have been made commencing at that position and stepping around to the wick-groove-contact point. The results indicate that there is no optimum groove size; the coarser the grooves the higher the condenser heat flux without flooding. Gravity acting on a horizontal pipe raises the no-flooding condenser capacity, so that a one-g test of a zero-g device is not conservative.

In the case of evaporation, Eq. (12) gives ω, the watt-inches per groove which can be pumped beyond the point of meniscus detachment from the groove tips. The stress at meniscus detachment is defined in Eq. (11).

Results were presented for water, and they can be scaled for other fluids. Numerical calculations show that evaporator heat flow at incipient dry-out scales as $(\rho\sigma\hat{h}_{fg}/\mu)$, and that there is an optimum number of grooves per inch approximately proportional to the wick plus gravity stress $[P_w+R_T\rho_{\ell v}g(1-\cos\theta_w)]$, which scales with σ. For the situations considered, approximately 200 grooves per inch (80 grooves per cm) gave maximum capacity. Gravity acts to reduce capacity, so that a one-g test of a zero-g evaporator is conservative (overly severe).

References

[1] Edelstein, F. and Hembach, R.J., "Design, Fabrication, and Testing of a Variable Conductance Heat Pipe for Equipment Thermal Control," *Progress in Astronautics and Aeronautics: Fundamentals of Spacecraft Thermal Design,* Vol. 29, 1972, pp. 482-504.

[2] Kirkpatrick, J.P. and Marcus, B.D., "A Variable Conductance Heat Pipe Flight Experiment," *Progress in Astronautics and Aeronautics: Fundamentals of Spacecraft Thermal Design,* Vol. 29, 1972, pp. 505-527.

[3] Edwards, D.K., Gier, K.D., Ayyaswamy, P.S., and Catton, I., "Evaporation and Condensation in Circumferential Grooves on Horizontal Tubes," ASME Paper 73-HT-25, presented at the National Heat Transfer Conference, Atlanta, Ga., Aug. 5-8, 1973.

[4] Ayyaswamy, P.S., Catton, I., and Edwards, D.K., "Capillary Flow in a Triangular Groove," *Journal of Applied Mechanics,* Vol. 41, 1974, pp. 332-336.

[5] Marcus, B.D., "Theory and Design of Variable Conductance Heat Pipes," CR-2018, April 1972, pp. 80-82, NASA.