N71-23923

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EFFECTS OF INTERPHASE TEMPERATURE DIFFERENCES AND WALL FRICTION IN HIGH-TEMPERATURE HEAT PIPES

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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • APRIL 1971

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
NASA TM X-2268		

4. Title and Subtitle EFFECTS OF INTERPHASE TEMPERATURE DIFFERENCES AND WALL FRICTION IN HIGH-TEMPERATURE HEAT PIPES	5. Report Date April 1971 6. Performing Organization Code
7. Author(s) Peter M. Sockol	8. Performing Organization Report No. E-6126
9. Performing Organization Name and Address Lewis Research Center National Aeronautics and Space Administration Cleveland, Ohio 44135	10. Work Unit No. 120-27 11. Contract or Grant No.
12. Sponsoring Agency Name and Address National Aeronautics and Space Administration Washington, D.C. 20546	13. Type of Report and Period Covered Technical Memorandum 14. Sponsoring Agency Code

15. Supplementary Notes

16. Abstract

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17. Key Words (Suggested by Author(s))	18. Distribution Statement
Heat pipes Heat transfer	Unclassified - unlimited

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price*
Unclassified	Unclassified	22	$3.00

*For sale by the National Technical Information Service, Springfield, Virginia 22151

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SUMMARY

From the analysis of a simple evaporation-heat-transfer system it is shown that the temperature drop required by the second law of thermodynamics occurs at the liquid-vapor interfaces. The temperature drop is estimated for conditions comparable to those in a 30-kilowatt, 1800 K lithium heat pipe. A simple one-dimensional treatment of the fluid dynamics in a high-temperature heat pipe is used to predict pressure drops in such a pipe. The effects of evaporation and condensation on wall friction in turbulent flow are included in the analysis. The additional friction due to condensation is shown to reduce the pressure recovery by a factor of 2. Detailed calculations are presented for two 1800 K lithium heat pipes.

INTRODUCTION

The typical heat pipe consists of a closed pipe, which serves as a containing envelope; an internal wick-like structure running the length of the pipe; and enough liquid to fill the wick. Heat transfer is achieved by the evaporation of liquid at one end of the pipe, transport of the vapor to the other end of the pipe, condensation of the vapor, and return of the liquid through the wick by capillary action. As the heats of vaporization of normal working fluids are very large, high heat-transfer rates can be obtained with relatively small temperature drops and fluid flow rates compared to those of other devices. In a particular application the choice of working fluid and the type of wick are determined by considerations such as the operating temperature range and the required heat flux. General review papers touching on many types of heat pipes are currently available (refs. 1 and 2).

It is relatively easy to demonstrate, using the second law of thermodynamics, that mass flow in a heat pipe requires a temperature drop over and above that required to conduct heat to and from the evaporating and condensing surfaces. It is not immediately obvious, however, where this temperature drop occurs. For this reason, in the present work a simple evaporation-heat-transfer system is analyzed, and the temperature drop is estimated for conditions comparable to those in an 1800 K lithium heat pipe.

The present fluid dynamic treatment is most appropriate to high-temperature heat pipes with high heat fluxes. Such a pipe finds its major application in the areas of high-temperature nuclear reactors and thermionic power systems, areas of current interest to the author. The normal working fluids are alkali metals which have vapor pressures of a few atmospheres at the temperatures of interest. The high vapor pressures permit high heat fluxes at low Mach numbers; this in turn means that the vapor flow can be considered to be incompressible. In addition, the high heat fluxes generally lead to turbulent vapor flow. These restrictions have been incorporated in a simple one-dimensional analysis of liquid and vapor flow in the evaporator, adiabatic, and condensor sections of a heat pipe. Measurements of the effects of injection and extraction of fluid on wall friction in turbulent flow have been reported in the recent literature. These results have been included for the first time in the present heat pipe analysis. Numerical calculations of pressure drops are presented for two lithium heat pipes.

TEMPERATURE DIFFERENCES

We consider a simple evaporation-heat-transfer system as sketched in figure 1. Liquid is evaporated at surface 1; the vapor flows at a low velocity, with constant pressure p_3 and temperature T_3, to surface 2 where it is condensed; and the liquid is returned to surface 1 by a separate channel (see appendix A for definition of symbols). The vapor passage is assumed to be large and thermally insulated so that frictional effects and heat transfer to or from the passage are negligible. Net heat conduction through the vapor is neglected. The frictional pressure drop in the liquid return passage is balanced by capillary forces produced by a fine mesh screen situated at surface 1. It is assumed that heat can be supplied directly to surface 1, so that no temperature drop across the liquid is required, and removed directly from surface 2.

Application of the second law of thermodynamics to the above system shows that T_2 must be less than T_1. Work can be obtained from the system without altering T_1 or T_2 by inserting a small turbine in the liquid return passage and supplying an additional amount of heat to surface 1 to compensate for the work. Hence, if $T_2 = T_1$, the system could exchange heat with a single reservoir and produce work in violation of the second law. In the vapor, some distance away from surfaces 1 and 2, heat conduction can be neglected because of the low thermal conductivity of the vapor. From the continuity and steady-flow energy equations, in the absence of heat conduction and shaft work, the velocity and temperature of the vapor are constant. Therefore, the temperature drop required by the second law must occur in the vicinity of the liquid-vapor interfaces. The presence of interphase temperature differences has been used to interpret heat-transfer measurements in film condensation of liquid-metal vapors (ref. 3). For ordinary (nonboiling) evaporation, however, investigations of such temperature differences are not available. Evaluation of the vapor temperature T_3 for the present system is a difficult problem in the kinetic theory of gases. Nevertheless, the temperature drop $(T_1 - T_2)$ can be estimated quite well.

The treatment that follows is essentially that of references 3 and 4. It is assumed that the molecules leaving the surface have a half-Maxwellian velocity distribution, and those approaching the surface have a Maxwellian distribution with a small mass velocity toward or away from the surface. The condensation coefficient (ref. 3) is assumed to be unity. Evaluation of the difference in fluxes between these two distributions at surface 1 gives the net mass flux per unit area $\dot{m}$,

$$\dot{m} \approx (2\pi R T_1)^{-1/2} (p_1 - p_3) + \frac{1}{2} \dot{m}$$

or

$$\dot{m} \approx 2 (2\pi R T_1)^{-1/2} (p_1 - p_3) \tag{1}$$

where R is the gas constant for the vapor and p_1 is the saturation pressure at T_1. It is assumed that the relative temperature difference is small compared to the relative pressure difference. If the same expression is applied to surface 2 and p_3 is eliminated between the two equations, we obtain

$$p_1 - p_2 \approx (2\pi R T_1)^{1/2} \dot{m} \tag{2}$$

Finally, since p_1 and p_2 are both saturation pressures, the Clapeyron relation (ref. 5) can be used to obtain the temperature difference

$$T_1 - T_2 \approx (2\pi R T_1)^{1/2} \frac{T_1}{p_1} \frac{R T_1}{h_{fg}} \dot{m} \tag{3}$$

where h_{fg} is the heat of vaporization per unit mass. For the 30-kilowatt, 1800 K, lithium heat pipe described in appendix B, the heat flux to the liquid-vapor interface in the evaporator is 1.99×10^6 watts per square meter and $\dot{m} \approx 0.106$ kilogram per square meter per second. From equation (3) this gives $T_1 - T_2 \approx 0.26$ K.

The system of figure 1 can be modified to one that behaves much like a heat pipe by accelerating the vapor in a nozzle, passing it through a long passage with wall friction, and decelerating it in a diffuser before condensing it at surface 2 (see fig. 2). If the vapor passage is thermally insulated, by conservation of energy the temperature T_4 of the vapor adjacent to surface 2 will equal T_3. As the pressure p_4 is less than p_3 by the frictional drop in the intervening passage, the vapor at T_3 and p_4 will be superheated. Hence, there must be an additional temperature drop at surface 2 for condensation to occur. This frictional contribution ΔT_f to the temperature drop $(T_1 - T_2)$ can be calculated directly from the Clapeyron relation (ref. 5).

$$\Delta T_f \approx \frac{T_1}{p_1} \frac{RT_1}{h_{fg}} (p_3 - p_4) \tag{4}$$

For the same 30-kilowatt lithium heat pipe, the pressure drop in the 300-centimeter adiabatic section is 1.99×10^3 newtons per square meter. From equation (4) this gives $\Delta T_f \approx 1.32$ K. The temperature drop is now the sum of equations (3) and (4), $T_1 - T_2 \approx 1.58$ K.

In a real system the presence of a small amount of noncondensable gas can increase the temperature drop predicted by equation (3). The gas will tend to collect at the condensing surface and form a diffusion barrier to the vapor. In addition, the adsorption of the gas on the liquid surface can reduce the condensation coefficient. Both effects increase the pressure drop in the system and lead to a larger temperature difference. Finally, it must be repeated that all the temperature drops described above are in addition to the dominant ones required to conduct heat to and from the evaporating and condensing surfaces.

FLUID DYNAMICS

The pressure drop in a heat pipe should be calculated between the points of maximum and minimum curvature in the meniscus at the liquid-vapor interface. The former occurs at the beginning of the evaporator, and the latter somewhere in the condensor. At low vapor pressures the vapor recovery term, which varies as ρ_v^{-1} (see eq. (6)), can exceed the frictional pressure drop in the liquid. In this case the minimum difference between vapor and liquid pressures, and hence, the minimum curvature in the meniscus, occurs at the beginning of the condensor. For the higher vapor pressures of interest here, the recovery term is relatively small and the minimum occurs at the end of the condensor.

Consider a heat pipe with an annular liquid return passage as sketched in figure 3. The cross section need not be circular. The vapor flow is assumed to be everywhere turbulent and incompressible. The effects of the changing velocity profile in the beginning of the evaporator are neglected since the pressure drop in this low-velocity region should be small. Turbulent vapor flow, once established, should persist well into the condensor. The liquid flow in the thin annulus is assumed to be laminar.

A one-dimensional analysis is employed for the vapor flow. The pressure p_v is assumed to be constant over the cross section, and the difference between the mean square velocity $\overline{u^2}$ and the square of the mean velocity $\overline{u}^2$ is neglected. This is a reasonable approximation in turbulent flow where the velocity profile is somewhat flattened. Conservation of momentum for this model gives

$$- A \frac{dp_v}{dx} - C\tau_w = \rho_v A \frac{d\overline{u}^2}{dx} \tag{5}$$

where x is the coordinate along the pipe, τ_w the wall shear stress, ρ_v the vapor density, A the cross-sectional area, and C the wetted perimeter. Introduce the mass flow rate $w = \rho_v A\overline{u}$ and the hydraulic diameter $D = 4\ A/C$, and write equation (5) as

$$\frac{dp_v}{dx} = - \frac{4}{D}\tau_w - \frac{1}{\rho_v A^2}\frac{dw^2}{dx} \tag{6}$$

It has been noted before (ref. 6) that the fluid dynamics of vapor flow in the evaporator and condensor of a heat pipe is equivalent to flow in a porous tube with injection or extraction of fluid at the walls. Measurements of wall shear stress in turbulent flow have been made for both injection (ref. 7) and extraction (ref. 8). A simple theory based on a Reynolds flux analogy (ref. 9), which was developed to interpret the extraction data, has also been found to describe the injection data (ref. 10). Briefly, it is assumed that the fluid striking the wall carries with it the free-stream axial momentum, while that leaving the wall carries zero axial momentum. Superposition of an injection or extraction rate $\dot{m}$ per unit wall area on the normal turbulent flow leads to a reduction in the friction factor f in both cases. However, for extraction there is an additional term $\dot{m}\overline{u}$ in the wall shear stress. The results of the analysis may be written:

Injection: $$\tau_w = \frac{1}{8} f\rho_v\overline{u}^2 \tag{7}$$

Extraction:

$$\tau_w = \frac{1}{8} f \rho_v \bar{u}^2 + \dot{m}\bar{u} \tag{8}$$

with

$$f = f_0 \exp\left(-\frac{4\dot{m}}{f_0 \rho_v \bar{u}}\right) \tag{9}$$

where f_0 is the friction factor for $\dot{m} = 0$.

In the heat pipe the flow rate w varies linearly with x from zero to w_t if the wall heat flux is constant. Hence, $\dot{m} = w_t/Cl$, where l is the length of the appropriate section. It is assumed that the friction factor f_0 can be taken as constant. In the evaporator, equations (6), (7), and (9) with $\bar{u} = w/\rho_v A$ give

$$\frac{dp_v}{dx} = -\frac{1}{2} \frac{f_0 w^2}{\rho_v D A^2} \exp\left(-\frac{D}{f_0 l_e} \frac{w_t}{w}\right) - \frac{1}{\rho_v A^2} \frac{dw^2}{dx} \tag{10}$$

Integration of equation (10) from 0 to l_e with $dw/dx = w_t/l_e$ gives the pressure change in the evaporator

$$\Delta p_e = -\frac{1}{2} \frac{w_t^2}{\rho_v A^2}\left[2 + \frac{1}{z_e} E_4(z_e)\right], \quad z_e = \frac{D}{f_0 l_e} \tag{11}$$

where

$$E_4(z) = \int_1^\infty e^{-zt} \frac{dt}{t^4} \tag{12}$$

The function E_4 is tabulated in reference 11. In the condensor, equations (6), (8), and (9) give

$$\frac{dp_v}{dx} = -\frac{1}{2} \frac{f_0 w^2}{\rho_v D A^2} \exp\left(-\frac{D}{f_0 l_c} \frac{w_t}{w}\right) - \frac{1}{\rho_v A^2}\left(\frac{w_t}{l_c} w + \frac{dw^2}{dx}\right) \tag{13}$$

Integration of equation (13) from 0 to l_c with $dw/dx = -w_t/l_c$ gives the pressure change in the condensor

$$\Delta p_c = \frac{1}{2}\frac{w_t^2}{\rho_v A^2}\left[1 - \frac{1}{z_c} E_4(z_c)\right], \quad z_c = \frac{D}{f_0 l_c} \tag{14}$$

The pressure recovery in equation (14) is only one-half the acceleration pressure drop in equation (11).

Finally, the pressure change in the adiabatic section is readily obtained from equation (6) with $dw/dx = 0$ and

$$\tau_w = \frac{1}{8} f_0 \rho_v \bar{u}^2 = \frac{1}{8}\frac{f_0 w_t^2}{\rho A^2} \tag{15}$$

Integration of equations (6) and (15) from 0 to l_a gives the pressure change in the adiabatic section

$$\Delta p_a = -\frac{1}{2}\frac{w_t^2}{\rho_v A^2}\frac{1}{z_a}, \quad z_a = \frac{D}{f_0 l_a} \tag{16}$$

The total pressure change in the vapor is the sum of equations (11), (14), and (16),

$$\Delta p_v = -\frac{1}{2}\frac{w_t^2}{\rho_v A^2}\left[1 + \frac{1}{z_e} E_4(z_e) + \frac{1}{z_c} E_4(z_c) + \frac{1}{z_a}\right] \tag{17}$$

The laminar flow in the liquid-return annulus can be treated in an equally simple fashion. From the two-dimensional analysis of fully developed laminar flow in a porous channel (ref. 12), it is seen that the velocity profile is almost parabolic when the wall Reynolds number, $R_w = \dot{m}d/\mu_l$ with d the annular spacing and μ_l the viscosity, is less than 1 in magnitude. This is normally the case. For a parabolic profile we have

$$\tau_w = 6\frac{\mu_l \bar{u}}{d} \tag{18}$$

and

$$\overline{u^2} = \frac{6}{5} \overline{u}^2 \tag{19}$$

where the bar again indicates the mean value. Conservation of momentum requires

$$- A \frac{dp_l}{dx} - C\tau_w = \rho_l A \frac{d\overline{u^2}}{dx} \tag{20}$$

where the parameters have the same meaning as in equation (5). Combining equations (18) to (20) and setting $\overline{u} = w/\rho_l A$ and $A = 1/2\ Cd$ gives

$$\frac{dp_l}{dx} = - 24 \frac{\mu_l}{\rho_l C d^3} w - \frac{6}{5} \frac{1}{\rho_l A^2} \frac{dw^2}{dx} \tag{21}$$

When equation (21) is integrated over the entire length of the liquid-return channel, the second term cancels, and the total pressure change in the liquid is given by

$$\Delta p_l = - 24 \frac{\mu_l w_t \hat{l}}{\rho_l C d^3} \tag{22}$$

where

$$\hat{l} = \frac{1}{2} (l_e + l_c) + l_a$$

The expressions for Δp_v and Δp_l are valid only if the heat pipe has the same cross section throughout. When this is not the case, the cancelation of terms between evaporator and condensor will not occur, and there will be additional frictional losses in the transition regions between the sections. An example of this type is considered later.

The total frictional pressure drop $(\Delta p_v + \Delta p_l)$ must be balanced by the rise in pressure across the capillary screen in the evaporator. The maximum flow rate, and therefore the heat flux, in a high-temperature heat pipe is thus limited by the requirement

$$2 \frac{\sigma}{r} + \Delta p_v + \Delta p_l > 0 \tag{23}$$

where σ is the liquid surface tension and r the pore radius of the capillary screen. In practice one would like to operate well within this limit as a safety factor.

RESULTS OF PRESSURE DROP CALCULATIONS

The equations of the previous section have been used to calculate the pressure drops in two lithium heat pipes. The details of the calculations, including checks on the validity of some of the basic assumptions, are presented in appendix B. These could be of use to a designer. The results of the calculations are presented in this section.

The first heat pipe might be used to cool a small compact nuclear reactor. It is to be 1.0 centimeter in outside diameter by 35 centimeters long and to carry 10 kilowatts thermal power at an operating temperature of 1800 K. The internal dimensions are evaporator length, $l_e = 15$ centimeters; adiabatic section length, $l_a = 5$ centimeters; condensor length, $l_c = 15$ centimeters; wall thickness, 0.75 millimeter; annular spacing, d = 0.25 millimeter; screen thickness, t = 0.15 millimeter. A realistic pore size for the screen is given by r = 50 micrometers. Hence, the limiting capillary pressure rise is $2\sigma/r = 7.5\times10^3$ newtons per square meter.

For this case equations (17) and (22) give

$$\Delta p_v = -6.25\times10^2 \text{ N/m}^2$$

$$\Delta p_l = -1.66\times10^3 \text{ N/m}^2$$

and the total pressure drop of -2.29×10^3 newtons per square meter is well within the capillary pressure rise. Although Δp_v is only 27 percent of the total pressure drop, the second term in the wall shear stress for the condensing vapor (eq. (8)) accounts for 72 percent of Δp_v.

Next consider a long heat pipe which might be used to carry heat from a nuclear reactor to a string of thermionic converters. It is to be 1.9 centimeters in outside diameter by 450 centimeters long, with a 300-centimeter adiabatic section, and to carry 30 kilowatts thermal power at an operating temperature of 1800 K. In the evaporator and condensor a thin liquid-return annulus is desirable to reduce the radial temperature drop produced by heat conduction into and out of the pipe. In the long adiabatic section, however, a thin annulus produces an excessive pressure drop, proportional to d^{-3}, in the liquid (see eq. (22)). One solution is to select a larger value of d for this section by minimizing the sum of the vapor and liquid pressure drops with respect to d. The screen is taken to have the same pore size as in the previous case.

In the evaporator and condensor the internal dimensions are evaporator length, l_e = 30 centimeters; condensor length, l_c = 120 centimeters; wall thickness, 0.75 millimeter; annular spacing, d = 0.50 millimeter; and screen thickness, t = 0.25 millimeter. For this case, equations (17) and (22) give

$$\Delta p_v = -3.43\times10^2 \ \mathrm{N/m^2}$$

$$\Delta p_l = -1.14\times10^3 \ \mathrm{N/m^2}$$

and the total pressure drop is -1.48×10^3 newtons per square meter. Here the second term in the wall shear stress for the condensing vapor is 64 percent of Δp_v and 15 percent of the total drop.

In the adiabatic section, minimizing the total pressure drop gives $d \approx 1.0$ millimeter. This of course means that the diameter of the vapor passage is reduced from its value in the evaporator and condensor sections by 1.0 millimeter. The resulting pressure drops are

$$\Delta p_v = -1.99\times10^3 \ \mathrm{N/m^2}$$

$$\Delta p_l = -5.85\times10^2 \ \mathrm{N/m^2}$$

and the total drop is -2.58×10^3 newtons per square meter. Here Δp_v is 77 percent of the total pressure drop. Finally, the total pressure drop for the heat pipe is -4.06×10^3 newtons per square meter which is again well within the capillary pressure rise of 7.5×10^3 newtons per square meter. Note that if the transitions from the 0.5 millimeter annulus in the evaporator and condensor to the 1.0-millimeter annulus in the adiabatic section are sufficiently gradual, the flow of liquid and vapor through the transitions will produce negligible pressure drop.

An alternative means of reducing the pressure drop in the adiabatic section is to return the liquid in a porous artery. The artery is made just large enough to keep the liquid flow laminar. For the present case this requires an inside diameter of 0.5 centimeter. The resulting pressure drops are

$$\Delta p_v = -1.99\times10^3 \ \mathrm{N/m^2}$$

$$\Delta p_l = -1.64\times10^2 \ \mathrm{N/m^2}$$

and the total drop is -2.15×10^3 newtons per square meter. The pressure drop for the entire heat pipe is now -3.63×10^3 newtons per square meter.

CONCLUDING REMARKS

A simple evaporation-heat-transfer system has been studied to determine the location and magnitude of the temperature drop required by the second law of thermodynamics. The drop occurs at the liquid-vapor interfaces and is quite small for a high-temperature heat pipe. In the heat pipe the increased wall friction associated with the condensation process in turbulent flow reduces the pressure recovery by a factor of 2. Finally, for a heat pipe with a long adiabatic section, the pressure drop can be minimized by a proper selection of the ratio between the liquid and vapor cross sections.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, February 3, 1971,
120-27.

APPENDIX A

SYMBOLS

A cross-sectional area

C wetted perimeter

D hydraulic diameter, $4A/C$

D_l mean diameter of annulus

D_v diameter of vapor passage

d annular spacing

E_4 exponential integral, eq. (12)

f friction factor

f_0 friction factor for $\dot{m} = 0$

h_{fg} heat of vaporization

l length of passage

$\hat{l}$ effective length, eq. (22)

M Mach number

$\dot{m}$ mass injection or extraction rate per unit wall area

p pressure

Δp pressure change

Q heat-transfer rate

R gas constant for vapor

R_w wall Reynolds number, $\dot{m}d/\mu$

R_x axial Reynolds number, $\rho\bar{u}D/\mu$

r pore size of capillary screen

s_v speed of sound in vapor

T absolute temperature

ΔT_f temperature drop due to wall friction

t screen thickness

u axial velocity

w local mass flow rate

w_t total mass flow rate

x axial coordinate

z dimensionless parameter, $D/f_0 l$

μ viscosity

ρ density

σ surface tension

τ_w wall shear stress

Subscripts:

a adiabatic section

c condensor section

e evaporator section

l liquid

v vapor

1, 2, 3, 4 positions in evaporation-heat-transfer system (figs. 1 and 2)

APPENDIX B

PRESSURE DROP CALCULATIONS

The following properties of lithium at 1800 K were extracted from reference 13:

$$h_{fg} = 1.88\times10^{7}\ \text{J/kg}$$

$$p_v = 3.11\times10^{5}\ \text{N/m}^2$$

$$\rho_v = 1.44\times10^{-1}\ \text{kg/m}^3$$

$$\mu_v = 2.10\times10^{-5}\ \text{N-sec/m}^2$$

$$\rho_l = 3.87\times10^{2}\ \text{kg/m}^3$$

$$\mu_l = 2.04\times10^{-4}\ \text{N-sec/m}^2$$

$$\sigma = 1.87\times10^{6}\ \text{N/m}$$

The speed of sound in monatomic lithium vapor is

$$s_v = \left(1.67\,\frac{p_v}{\rho_v}\right)^{1/2} = 190\ \text{m/sec}$$

For a pore size $r = 50\ \mu\text{m}$, the limiting capillary pressure rise is given by $2\ \sigma/r = 7.5\times10^{3}\ \text{N/m}^2$.

For the first heat pipe $Q = 10\ \text{kw} = 10^4\ \text{J/sec}$, and

$$w_t = \frac{Q}{h_{fg}} = 5.32\times10^{-4}\ \text{kg/sec}$$

The internal dimensions are

$$l_e = 15\ \text{cm},\quad l_a = 5\ \text{cm},\qquad l_c = 15\ \text{cm}$$

$$D_v = 0.77\ \text{cm},\ t = 0.15\ \text{mm},\ d = 0.25\ \text{mm}$$

Hence,

$$C_v = \pi D_v = 2.42 \text{ cm}, \quad A_v = \frac{1}{4} \pi D_v^2 = 0.465 \text{ cm}^2$$

$$C_l = 2\pi \; (D_v + 2t + d) = 5.18 \text{ cm}$$

Before calculating the pressure drops a few checks are desirable:

$$\bar{u}_v = \frac{w_t}{\rho_v A_v} = 79 \text{ m/sec}$$

$$M = \frac{\bar{u}_v}{s_v} = 0.042 << 1$$

$$R_{xv} = \frac{4w_t}{\mu_v C_v} = 4.2 \times 10^3$$

$$f_0 \approx 0.042 \qquad \text{(ref. 14)}$$

hence, the vapor flow is incompressible and turbulent;

$$R_{xl} = \frac{4w_t}{\mu_l C_l} = 2.0 \times 10^2 < 10^3$$

$$R_{wl} = \frac{\dot{m}d}{\mu_l} \approx \frac{1}{2} R_{xl} \frac{d}{l_e} = 0.17 < 1$$

and the liquid flow is laminar with a parabolic velocity profile. The pressure drops are obtained from equations (17) and (22):

$$\frac{1}{2}\frac{w_t^2}{\rho_v A_v^2} = 4.53\times10^2 \text{ N/m}^2$$

$$z_e = z_c = \frac{D_v}{f_0 l_e} = 1.22$$

$$2 z_e^{-1} E_4(z_e) = 0.107 \qquad \text{(ref. 10)}$$

$$z_a^{-1} = \frac{f_0 l_a}{D_v} = 0.273$$

$$\Delta p_v = -4.53\times10^2 \times (1 + 0.107 + 0.273) = -6.25\times10^2 \text{ N/m}^2$$

$$\hat{l} = 20 \text{ cm}$$

$$\Delta p_l = -24 \frac{\mu_l w_t \hat{l}}{\rho_l C_l d^3} = -1.66\times10^3 \text{ N/m}^2$$

and

$$\Delta p_v + \Delta p_l = -2.29\times10^3 \text{ N/m}^2$$

For the second heat pipe $Q = 30$ kw $= 3\times10^4$ J/sec, and

$$w_t = \frac{Q}{h_{fg}} = 1.60\times10^{-3} \text{ kg/sec}$$

The internal dimensions for the evaporator and condensor are

$$l_e = 30 \text{ cm}, \quad l_c = 120 \text{ cm}$$

$$D_v = 1.60 \text{ cm}, \quad t = 0.25 \text{ mm}, \quad d = 0.50 \text{ mm}$$

Hence,

$$C_v = \pi D_v = 5.03 \text{ cm}, \; A_v = \frac{1}{4} \pi D_v^2 = 2.01 \text{ cm}^2$$

$$C_l = 2\pi \left(D_v + 2t + d\right) = 10.7 \text{ cm}$$

The same checks as before are desirable:

$$\bar{u}_v = \frac{w_t}{\rho_v A_v} = 55.3 \text{ m/sec}$$

$$M = \frac{\bar{u}_v}{s_v} = 0.029 << 1$$

$$R_{xv} = \frac{4w_t}{\mu_v C_v} = 6.06 \times 10^3$$

$$f_0 \approx 0.036 \qquad \text{(ref. 14)}$$

Hence, the vapor flow is incompressible and turbulent;

$$R_{xl} = \frac{4w_t}{\mu_l C_l} = 2.9 \times 10^2 < 10^3$$

$$R_{wl} = \frac{\dot{m} d}{\mu_l} \approx \frac{1}{2} R_{xl} \frac{d}{l_e} = 0.24 < 1$$

and the liquid flow is laminar with a parabolic velocity profile. The pressure drops are obtained from equations (17) and (22):

$$\frac{1}{2}\frac{w_t^2}{\rho_v A_v^2} = 2.20\times10^2 \text{ N/m}^2$$

$$z_e = \frac{D_v}{f_0 l_e} = 1.48$$

$$z_e^{-1} E_4(z_e) = 0.032 \qquad \text{(ref. 10)}$$

$$z_c = \frac{D_v}{f_0 l_c} = 0.370$$

$$z_c^{-1} E_4(z_c) = 0.533$$

$$\Delta p_v = -2.20\times10^2 \times (1 + 0.03 + 0.53) = -3.43\times10^2 \text{ N/m}^2$$

$$\hat{l} = 75 \text{ cm}$$

$$\Delta p_l = -24\frac{\mu_l w_t \hat{l}}{\rho_l C_l d^3} = -1.14\times10^3 \text{ N/m}^2$$

and

$$\Delta p_v + \Delta p_l = -1.48\times10^3 \text{ N/m}^2$$

In the adiabatic section of the long heat pipe the sum of the vapor and liquid pressure drops is given by equations (16) and (22)

$$\Delta p_a = -\frac{8}{\pi^2}\frac{w_t^2 f_0 l_a}{\rho_v D_v^5} - \frac{12}{\pi}\frac{\mu_l w_t l_a}{\rho_l D_l d^3}$$

where $D_v = D - 2t - 2d$, $D_l = D - d$, and D is the inside diameter of the pipe. If $|\Delta p_a|$ is minimized with respect to d, one finds $d \approx 1.0$ mm, $D_v = 1.50$ cm, and $D_l = 1.65$ cm. Hence,

$$R_{xv} = \frac{4w_t}{\mu_v C_v} = 6.5\times10^3$$

$$f_0 \approx 0.035 \qquad \text{(ref. 14)}$$

and with $l_a = 300$ cm,

$$\Delta p_v = -\frac{8}{\pi^2}\frac{w_t^2 f_0 l_a}{\rho_v D_v^5} = -1.99\times10^3 \text{ N/m}^2$$

$$\Delta p_l = -\frac{12}{\pi}\frac{\mu_l w_t l_a}{\rho_l D_l d^3} = -5.85\times10^2 \text{ N/m}^2$$

$$\Delta p_a = \Delta p_v + \Delta p_l = -2.58\times10^3 \text{ N/m}^2$$

Finally, the total pressure drop for the long heat pipe is -4.06×10^3 N/m^2.

If an arterial liquid return passage is used in the adiabatic section of the long heat pipe, laminar flow requires

$$R_{xl} = \frac{4w_t}{\pi \mu_l D_l} \approx 2\times10^3$$

$$f_{0l} = \frac{64}{R_{xl}} \approx 0.032$$

Hence, $D_l \approx 0.5$ cm and the liquid pressure drop is

$$\Delta p_l = -\frac{8}{\pi^2}\frac{w_t^2 f_{0l} l_a}{\rho_l D_l^5} = -1.64\times10^2 \text{ N/m}^2$$

If the porous artery has the same wall thickness as the capillary screen,

$$C_v = 7.23 \text{ cm}, \; A_v = 2.17 \text{ cm}^2$$

$$R_{xv} = \frac{4w_t}{\mu_v C_v} = 4.2\times10^3$$

$$f_{0v} \approx 0.042 \qquad \text{(ref. 14)}$$

The pressure drop in the vapor is given by

$$\Delta p_v = -\frac{1}{8}\frac{w_t^2 f_{0v} C_v l_a}{\rho_v A_v^3} = -1.99\times10^3 \text{ N/m}^2$$

and

$$\Delta p_a = \Delta p_v + \Delta p_l = -2.15\times10^3 \text{ N/m}^2$$

The total pressure drop for the long heat pipe is now -3.63×10^3 N/m^2.

REFERENCES

1. Cheung, Henry: A Critical Review of Heat Pipe Theory and Applications. Rep. UCRL-50453, California Univ., Lawrence Radiation Lab., July 15, 1968.
2. Busse, C. A.: Heat Pipe Research in Europe. Proceedings of the Second International Conference on Thermionic Electrical Power Generation. Rep. EUR-4210 f, e, EURATOM, 1969, pp. 461-475.
3. Sukhatme, S. P.; and Rohsenow, W. M.: Heat Transfer During Film Condensation of a Liquid Metal Vapor. J. Heat Transfer, vol. 88, no. 1, Feb. 1966, pp. 19-28.
4. Schrage, Robert W.: A Theoretical Study of Interphase Mass Transfer. Columbia Univ. Press, 1953.
5. Keenan, Joseph H.: Thermodynamics. John Wiley & Sons, Inc., 1957, p. 347.
6. Cotter, T. P.: Theory of Heat Pipes. Rep. LA-3246-MS, Los Alamos Scientific Lab., Mar. 26, 1965.
7. Olson, Reuben M.; and Eckert, E. R. G.: Experimental Studies of Turbulent Flow in a Porous Circular Tube with Uniform Fluid Injection Through the Tube Wall. J. Appl. Mech., vol. 33, no. 1, Mar. 1966, pp. 7-17.
8. Wallis, G. B.: Pressure Gradients for Air Flowing Along Porous Tubes with Uniform Extraction at the Walls. Proc. Inst. Mech. Eng., vol. 180, pt. 1, no. 1, 1965-1966, pp. 27-35.
9. Silver, R. S.; and Wallis, G. B.: A Simple Theory for Longitudinal Pressure Drop in the Presence of Lateral Condensation. Proc. Inst. Mech. Eng., vol. 180, pt. 1, no. 1, 1965-1966, pp. 36-40.
10. Wallis, G. B.: Discussion on Paper by Olson and Eckert. J. Appl. Mech., vol. 33, no. 4, Dec. 1966, pp. 950-953.
11. Abramowitz, Milton; and Stegun, Irene A.; eds.: Handbook of Mathematical Functions with Formulas, Graphs, and Mathematical Tables. Appl. Math. Ser. 55, National Bureau of Standards, June 1964, pp. 245-248.
12. White, F. M., Jr.; Barfield, B. F.; and Goglia, M. J.: Laminar Flow in a Uniformly Porous Channel. J. Appl. Mech., vol. 25, no. 4, Dec. 1958, pp. 613-617.
13. Weatherford, W. D., Jr.; Tyler, John C.; and Ku, P. M.: Properties of Inorganic Energy-Conversion and Heat-Transfer Fluids for Space Applications. Southwest Research Inst. (WADD TR-61-96), Nov. 1961.
14. Hunsaker, J. C.; and Rightmire, B. G.: Engineering Applications of Fluid Mechanics. McGraw-Hill Book Co., Inc., 1947, p. 126.

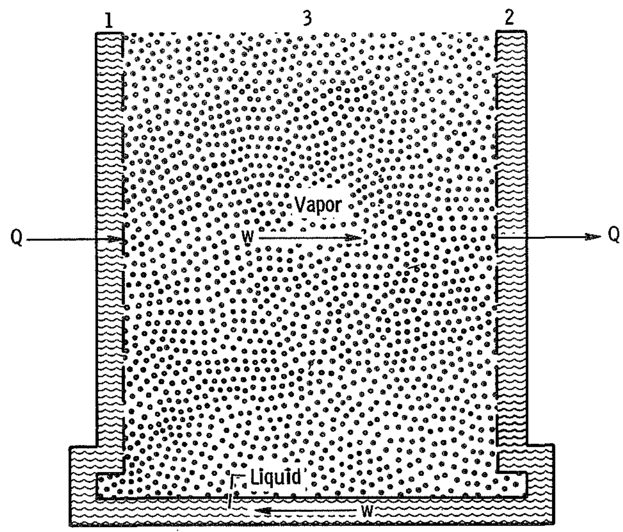

Figure 1. - Evaporation-heat-transfer system.

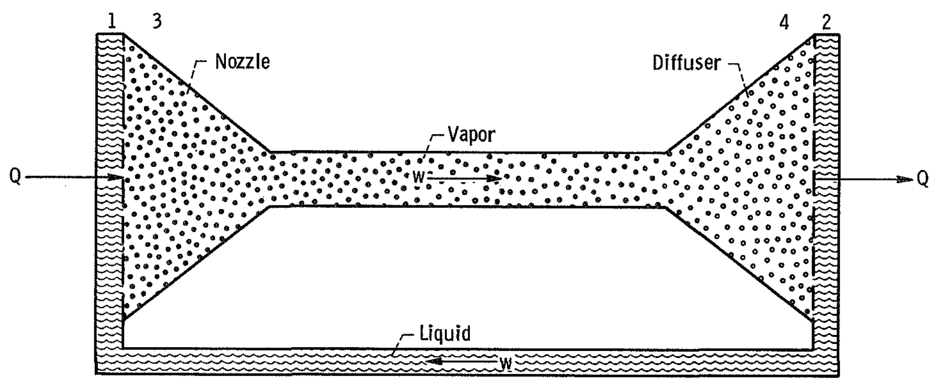

Figure 2. - Heat-pipe-like system.

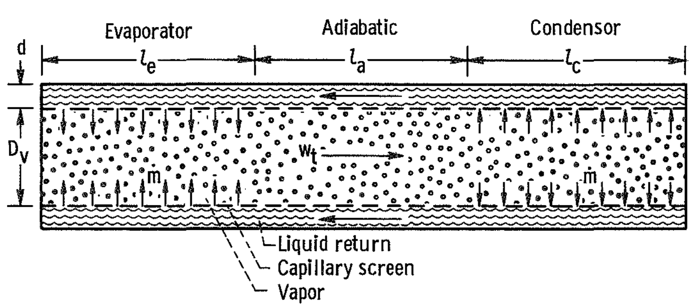

Figure 3. - Heat pipe with annular liquid return.

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