MATHEMATICAL MODELING OF HIGH AND LOW TEMPERATURE HEAT PIPES

By

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Final Report to Goddard Space Flight Center, NASA
Grant No. NGR 09-010-070

December 1971

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ABSTRACT

After a review of heat and mass transfer theory relevant to heat pipe performance, math models are developed for calculating heat-transfer limitations of high-temperature heat pipes and heat-transfer limitations and temperature gradient of low temperature heat pipes. Calculated results are compared with the available experimental data from various sources to increase confidence in the present math models.

For the convenience of the users of the present theory, complete listings of two computer programs for high- and low-temperature heat pipes respectively are appended to the report. These programs enables the performance of heat pipes with wrapped-screen, rectangular-groove or screen-covered rectangular-groove wick to be predicted.
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\[ M \] Mach number, Eq. (2.54)
\[ n \] number of grooves, Eq. (3.19)
\[ N_f \] liquid transport factor, Eq. (3.6)
\[ P_f \] liquid pressure, Eq. (2.13)
\[ P_g \] vapor pressure, Eq. (2.9)
\[ P_o \] gas stagnation pressure, Eq. (2.51)
\[ P_{vap} \] vapor pressure, Eq. (2.44)
\[ P_{liq} \] liquid pressure, Eq. (2.44)
\[ \Delta P_c \] capillary pumping pressure, Eq. (2.1)
\[ \Delta P_f \] pressure drop due to liquid flow resistance, Eq. (3.1)
\[ \Delta P_g \] pressure drop due to vapor flow resistance, Eq. (3.15)
\[ \Delta P_{av} \] average pressure drop due to vapor flow resistance, Eq. (3.2)
\[ \Delta P_s \] pressure drop due to gravitational force, Eq. (3.14)
\[ P \] wetted perimeter, Eq. (2.4)
\[ q' \] heat transfer rate per groove per unit length, Eq. (2.36)
\[ q'' \] heat transfer rate per unit area, Eq. (2.48)
\[ Q \] heat transfer rate, Eq. (3.11)
\[ Q_{max} \] maximum heat transfer rate, Eq. (2.59)
\[ r \] radius, Eq. (2.3)
\[ r_b \] radius of vapor bubble, Eq. (2.24)
\[ r_e \] effective capillary radius for liquid flow, Eq. (2.13)
\[ r_i \] inner radius of heat pipes container, Fig. 2.1
\[ r_g \] radius of vapor flow passage, Fig. 2.1
\[ r_1, r_2 \] principle radius of curvature of meniscus, Eq. (2.1)
Nomenclature (continued)

R  radius of capillary tube, Eq. (2.5)

\( R_e \)  equivalent radius of capillary structure, Eq. (2.8)

\( Re \)  Reynolds number based upon axial velocity and radius, Eq. (2.11)

\( Re_w \)  Reynolds number based upon radial velocity at tube wall and radius, Eq. (2.11)

t  thickness of wick structure, Eq. (3.3)

T  Temperature, Eq. (2.29)

\( T_c \)  condenser wall temperature, Eq. (3.30)

\( T_e \)  evaporator wall temperature, Eq. (2.30)

\( T_g \)  vapor temperature, Eq. (2.31)

\( \Delta T_c \)  temperature difference defined as (\( T_e - T_g \)), Eq. (2.40)

\( \Delta T_e \)  temperature difference defined as (\( T_g - T_c \)), Eq. (3.30)

\( T_o \)  stagnation temperature, Eq. (2.52)

u  axial velocity, Eq. (2.9)

\( \bar{u} \)  bulk average axial velocity, Eq. (2.9)

v  radial velocity at tube wall, Eq. (2.11)

Wb  Weber number, Eq. (2.60)

x,y  co-ordinate axes, Eq. (2.29)

z  axial distance, Eq. (2.9)

\( z_a \)  length of adiabatic section, Eq. (3.4)

\( z_c \)  length of condenser, Eq. (3.4)

\( z_e \)  length of evaporator, Eq. (3.4)

\( z_t \)  total length of heat pipe, Eq. (3.3)
Nomenclature (continued)

\( \alpha \)  
defined as \((dp_f/dz)/\mu_f\), Eq. (2.14)

\( \gamma \)  
ratio of specific heats, Eq. (2.54)

\( \delta \)  
groove depth, Eq. (2.16)

\( \varepsilon \)  
porosity of capillary structure, Eq. (2.24)

\( \theta \)  
wetting angle, Eq. (2.5)

\( \mu_f \)  
liquid dynamic viscosity, Eq. (2.13)

\( \mu_g \)  
vapor dynamic viscosity, Eq. (2.18)

\( \rho_f \)  
liquid density, Eq. (2.13)

\( \rho_g \)  
vapor density, Eq. (2.20)

\( \rho_o \)  
gas density at stagnation state, Eq. (2.52)

\( \sigma \)  
surface tension, Eq. (2.1)

\( \tau_s \)  
shear stress at tube wall, Eq. (2.9)

\( \phi \)  
nondimensional temperature, Eq. (2.31)

\( \psi \)  
heat pipe elevation angle, Eq. (2.41)

\( \omega \)  
groove width, Eq. (2.22)
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I. INTRODUCTION

An Integrated Cryogenic Cooling Engine System (ICICLE) currently under study at Goddard Space Flight Center is an attractive concept for providing long life cryogenic cooling space. Fig. 1.1 is a schematic illustration of the ICICLE System concept. A miniature Vuilleunier (VM) cycle cryogenic engine is the central cooling refrigerator. The engine has three temperature regions: a hot volume (≈900° K), an ambient temperature volume (≈310° K) and a cold volume (≈75° K). The VM engine operates on thermal power which is provided by a radioisotope. Because safety considerations will dictate the location of the isotope and experiment requirements will dictate the location of spaces which must be cooled, the components of the ICICLE System will be distributed on a spacecraft. Consequently, heat pipes ranging from cryogenic to liquid-metal temperatures are used to couple the VM engine to other components of the system.

An objective of this research is to develop math models for predicting performance of heat pipes of various wick structures (e.g. wrapped-screen, open-groove and screen-covered-groove wicks) and temperatures ranging from cryogenic to liquid-metal temperatures. As far as possible, the confidence in theoretical predictions will be established by comparison of theory with available experimental data.
Section II below is devoted to a review of heat and mass transfer theory leading to the development of present math models which is described in Section III. Also included in Section III are comparisons of present predictions with available experiments. For the convenience of users of the present math models two complete program listings for calculating performance of low- and high-temperature heat pipes respectively are appended to the report.

Conclusions and recommendations for further work are given in Section IV.

Figure 1.1 Schematic illustration of the ICICLE system concept
II. REVIEW OF HEAT AND MASS TRANSFER THEORY

1. Introductory Remarks

A well-known method for the transfer of a large amount of heat with small temperature drop consists of the evaporation of a liquid, transport of the vapor through a duct, and subsequent condensation. In order to work continuously, the condensate must be returned to the evaporator. Ordinarily this step is accomplished by gravity or by a pump. In a heat pipe the return of liquid is accomplished by a wick of suitable capillary structure.

The principle of operation of a simple cylindrical heat pipe is shown schematically in Fig. 2.1. The wick is saturated with a wetting liquid. In the steady state the liquid temperature in the evaporator is maintained higher than in the condenser by an external heat source. The resulting difference in vapor pressure $\Delta p_g$ drives vapor from the evaporator to condenser. The depletion of liquid by evaporation causes the vapor-liquid interface in the evaporator to retreat into the wick surface and a capillary pressure is developed. This capillary pressure pumps condensate back to the evaporator. The heat pipe can operate without drying out the wick as long as the capillary pressure term is greater than or equal to the sum of pressure drops due to liquid and vapor flow resistance and gravitational forces acting on the liquid.
Figure 2.1 Schematic diagram of the principle of operation of a heat pipe
The temperature drop of a heat pipe equals to the sum of the temperature drops at the evaporator, vapor flow passage and condenser. Because of the usual thinness of wick structures and the small temperature drop at the vapor flow passages, the heat pipe exhibits a thermal conductance greatly in excess of that which could be obtained by the use of a homogeneous piece of any known metal.

However, unlike solid heat conductors, heat pipes cannot be characterized by a single property as equivalent thermal conductivity. Limitations and thermal conductance of a heat pipe are dependent upon not only the size, shape and materials of the heat pipe but also its heat transfer rate, wick structure and working fluid. Heat pipes using working fluids ranging from cryogens to liquid metals have been developed. Different types of wick structures have been used; several examples\textsuperscript{3} are shown in Fig. 2.2. Heat and mass transfer theory relevant to heat pipe performance are reviewed below.

2. Capillary Pressure in Wick Structures

The well-known Laplace and Young equation for the capillary pressure difference, $\Delta p_c$, established at any point of a liquid vapor interface is

$$\Delta p_c = \sigma \left(1/r_1 + 1/r_2\right),$$

in which $r_1$ and $r_2$ are the principal radii of curvature of meniscus and $\sigma$ the surface tension. The commonest method of measuring pressure differences across the meniscus in a
Figure 2.2 Cross section of various wick structures.
capillary is the observation of vertical liquid rise, \( h \). 
\( \Delta P_c \) can then be related to \( h \) by the equation

\[
\Delta P_c = (\rho_f - \rho_g)gh,
\]

where \( \rho_f \) is the liquid density, \( \rho_g \) is the vapor density and \( g \) is the gravitational acceleration. For an approximate treatment of the capillary rise, when the radius of curvature of the meniscus is much smaller than the capillary rise, Eq. (2.1) can be reduced to a simpler relation with one radius of curvature, \( r \), and the pressure-difference equation becomes

\[
\Delta P_c = 2\sigma/r = (\rho_f - \rho_g)gh.
\]

In the case of a perfectly wetting liquid in a capillary tube, the radius of curvature of the surface of the meniscus \( r \) approximates the hydraulic radius of the capillary, \( R \), defined as

\[
R = 2A/\varphi,
\]

where \( A \) is the cross-sectional area and \( \varphi \) the wetted perimeter. If the liquid, however, does not wet the capillary surface completely, an angle of contact between the liquid and the solid surface, \( \theta \), will be observed and the radius of curvature becomes \( r = R \cos \theta \). Then, the capillary pressure differences and the vertical liquid rises can be ascertained by the relation

\[
\Delta P_c = (\rho_f - \rho_g)gh = 2\sigma \cos \theta/R.
\]
Figure 2.3 Liquid rise in a capillary groove
Figure 2.4 Calculated fully wetted liquid rise in capillary grooves.
Figure 2.5 Measured dimensionless liquid rise in capillary grooves
The liquid-vapor interface in a capillary groove under the action of gravity is shown schematically in Fig. 2.3. Experimental and theoretical studies of liquid interfaces in different shaped grooves have been made by Bressler and Wyatt. It was found that both the measured and the calculated fully-wetted height can be represented accurately by a similar relation as given in Eq. (2.5)

\[ \Delta p_c = (\rho_f - \rho_g)g h_f = 2\alpha \cos \theta / R \]  

(2.6)
as shown in Figs. 2.4 and 2.5, when \( R \) for an open groove of cross-sectional area, \( A \), wetted perimeter, \( P \), and width, \( \omega \), is calculated by the equation

\[ R = 2A / (P - \omega) \]  

(2.7)

instead of Eq. (2.4).

Hence capillary pumping pressure, when \( \rho_g \ll \rho_f \), can be calculated by the equation

\[ \Delta p_c = 2\alpha \cos \theta / R_E \]  

(2.8)

with \( R_E \) designated an equivalent radius defined as \( 2A/P \) for capillary tubes and \( 2A / (P - \omega) \) for open capillary grooves.

3. **Vapor Pressure Drop**

A stationary vapor control volume of diameter \( D_g \) and width \( dz \) with mass transfer per unit width \( m' \) is represented in Fig. 2.6 with the terms which appear in the momentum
Figure 2.6 An elementary control volume for vapor flow
equation, where $m'$ may be positive, zero or negative depending upon whether the evaporator, adiabatic section or condenser of a heat pipe is under consideration. The principle of conservation of axial momentum requires that

$$\frac{dp_g}{dz} = -2c_f \rho g \bar{u}^2 / D_g - 8(\frac{D_g}{2} \rho_g u^2 r dr/dz) / D_g^2, \quad (2.9)$$

in which $p_g$ is the vapor pressure, $z$ is the axial distance, $c_f$ is the skin friction coefficient $\tau_g / (\rho g \bar{u}^2 / 2)$, $Re$ is the Reynolds number $\rho g \bar{u} D_g / u_g u_g$, $\nu_g$ is the vapor dynamic viscosity, $u$ is the vapor velocity, and $\bar{u}$ is the vapor bulk average velocity.

Integration of Eq. (2.9) from $z = 0$ to $z = z_t (= z_e + z_a + z_c)$ yields a relation for the total vapor pressure drop

$$\Delta p_g = \int_0^{z_t} (2c_f \rho g \bar{u}^2 / D_g) dz. \quad (2.10)$$

It was noted in the above integration that $u = 0$ at both ends of a heat pipe. This Eq. (2.10) can be evaluated if the skin friction coefficient $c_f$ is known.

The values of $c_f$ for laminar flow may be obtained from numerical solution of complete Navier-Stokes equations for the tube flow with wall suction and injection. The dependence

*With negative $m'$ axial momentum flux associated with $m'(=m' u)$ may be subtracted from the righthand side of Eq. (2.9); but for heat pipe application condensation occurs at the vicinity of the liquid-vapor interface where $u$ is small, so it is neglected in the present analysis.*
Figure 2.7 Friction coefficient for the fully developed flow. o, exact numerical solution; _, empirical Eq. (1.11).
of the fully developed $c_f Re$ on the wall Reynolds number $Re_w = \rho_g v D_g / \mu_g$ from Yuan and Finkelstein's solution$^{5,6}$ is shown in Fig. 2.7. It can be seen in this figure that the empirical equation

$$c_f Re = \frac{1}{[0.048 + 0.0494/(4.7 + Re_w)^{0.8}]},$$  

(2.11)

accurately represents the exact numerical data for negative as well as positive $Re_w$. It can also be seen in this figure that at $Re_w = 0$ Eq. (2.11) reduces to

$$c_f Re = 16/Re,$$  

(2.12)

which is the well-known Hagen-Poiseuille solution for the fully developed tube flow without wall mass flux.

It should be noted that in the above derivation of Eq. (2.10) for the vapor-pressure drop complete recovery of dynamic pressure due to vapor velocity has been assumed. However, it can be seen in Fig. 2.7 that the friction coefficient $c_f$, and therefore the velocity gradient at the wall also, becomes zero at $Re_w = -4.5978$. At further increase in suction, negative velocity gradient exists at the wall; and reverse flow is likely to occur. When the flow reverse occurs, the recovery of dynamic pressure becomes difficult$^7$. Unfortunately, no quantitative information is available at the present time to account for the loss of dynamic pressure at reverse flows. This information is urgently needed, because most heat pipes have large $|Re_w|$ values at the condenser.
4. **Liquid Pressure Drop**

For the flow of liquid through capillary structures, the liquid pressure gradient in the axial direction \( \frac{dp_f}{dz} \) may be calculated by the equation

\[
\frac{dp_f}{dz} = \frac{c \mu_f m_f}{(Ar_e^2 \rho_f)},
\]

(2.13)

where \( c \) is a dimensionless constant depending on the detailed geometry of the capillary structure, \( \mu_f \) is the liquid dynamic viscosity, \( m_f \) is the liquid mass flow rate, \( A \) is the total cross-sectional area for liquid flow, \( r_e \) is the effective hydraulic radius for liquid flow, and \( \rho_f \) is the liquid density. For non-connected parallel circular cylinder \( c = 8 \), and for concentric annular \( c = 12 \). However, for wrapped screen the value of \( c \) is uncertain and dependent upon the screen mesh size and the wrapping tightness.

For liquid flow in passages of simple geometry such as those of open and closed grooves shown in Figs. 2.8, expressions for the pressure drop may be derived by theoretical consideration. The governing equation for the fully developed laminar flow of liquid in grooves can be written as

\[
\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} = \alpha,
\]

(2.14)

where \( \alpha \) is defined as \( \frac{(dp_f/dz)}{\mu_f} \). Eq. (2.14) can be solved with appropriate boundary conditions. For example, for flow
Figure 2.8 Cross section of various open and screen-covered grooves.
Figure 2.9 Co-ordinate system for liquid flow analysis
in an open rectangular groove of Fig. 2.9 the boundary conditions are:

\[ u = 0, \theta x = \pm a & y = 0; \text{ and } \partial u / \partial y = 0, \theta y = \delta; \]  

(2.15)

and solution of Eq. (2.14) may be sought in the form

\[ u = B(a^2 - x^2)y(\delta - y/2) \]  

(2.16)

which satisfies the boundary conditions (2.15). In accordance with the method of Galerkin, we obtain for determination of B the equation

\[ \int_{x=0}^{a} \int_{y=0}^{\delta} (\partial^2 u / \partial x^2 + \partial^2 u / \partial y^2 - a)(a^2 - x^2)y(\delta - y/2)dxdy = 0. \]  

(2.17)

whence

\[ B = \frac{5a}{4(a^2 + \delta^2)} \]  

(2.18)

and the required solution is

\[ u = -\frac{5a(a^2 - x^2)y(\delta - y/2)}{4(a^2 + \delta^2)}. \]  

(2.19)

Now mass flow rate in each groove \( m_1 \) may be determined by the equation

\[ m_1 = 2 \int_{x=0}^{a} \int_{y=0}^{\delta} \rho_f u dxdy. \]  

(2.20)

On substitution of \( u \) from Eq. (2.19) into the above Eq. (2.20) there is obtained
\begin{equation}
m_\perp = -5 \rho_f a^2 \delta^3 / 9 (a^2 + \delta^2) \tag{2.21}
\end{equation}

or

\begin{equation}
dp_f/dz = -18 (\omega^2 + 4 \delta^2) \mu_f m_\perp / (5 \omega^3 \delta^3 \rho_f), \tag{2.22}
\end{equation}

for the liquid pressure drop in an open groove of depth \(\delta\) and width \(\omega\).

If similar analysis is made for liquid flows in covered grooves of depth \(\delta\) and width \(\omega\), the resultant expression for the pressure gradient will be

\begin{equation}
dp_f/dz = -72 (\omega^2 + \delta^2) \mu_f m_\perp / (5 \omega^3 \delta^3 \rho_f). \tag{2.23}
\end{equation}

5. Heat Transfer

The primary heat-transfer mechanism for heat pipes is thermal conduction through the liquid saturated wick with surface evaporation at the evaporator section and film condensation with conduction through the liquid-saturated wick at the condenser section. Values of the thermal conductivity of heterogeneous materials such as the liquid-saturated wick are dealt with by Gorring and Churchill.9

The simplest configurations, as shown in Fig. 2.10, consist of arrangement of wick material and liquid in series or parallel. The exact solutions for these cases are respectively,

\begin{equation}
K_e = K_f K_w / [\epsilon K_w + K_f (1-\epsilon)] \tag{2.24}
\end{equation}
Figure 2.10  Heat transfer model for series or parallel arrangement of liquid-saturated wick structures
Figure 2.11 Heat transfer model for cubic array of truncated spheres
and

\[ K_e = \varepsilon K_f + (1-\varepsilon)K_w \]  \hspace{1cm} (2.25)

in which \( \varepsilon \) is the volume fraction of liquid, \( K_e \) is the thermal conductivity of the liquid saturated wick, \( K_f \) is the thermal conductivity of the liquid, and \( K_w \) is the thermal conductivity of the wick material. Thermal conductivities for the liquid saturated wick of distributed cylinders and spheres can be calculated respectively by the equations

\[ K_e = K_f \left[ K_f + K_w - (1-\varepsilon)(K_f-K_w) \right] / \left[ K_f + K_w + (1-\varepsilon)(K_f-K_w) \right], \]  \hspace{1cm} (2.26)

and

\[ K_e = K_f \left[ 2K_f + K_w - 2(1-\varepsilon)(K_f-K_w) \right] / \left[ 2K_f + K_w + (1-\varepsilon)(K_f-K_w) \right]. \]  \hspace{1cm} (2.27)

The thermal conductivity of a simple cubic array of adjoined, truncated spheres having circles of contact of radius \( r \) as illustrated in Fig. 2.11 is as follows:

\[ K_e = K_w / \left[ \left( R/r \right) + \left( \ln 2R/r \right) / \pi \right], \]  \hspace{1cm} (2.28)

when fluid surrounding the array is postulated to have zero conductivity and the radius of contact was assumed to be small relative to the radius of the spheres.

For some simple geometries, analytical expression for film evaporation and condensation can be derived in a straightforward manner. For example, for the case of film evaporation from rectangular grooved wick as shown in Fig. 2.12 with \( K_f \ll K_w \),
if liquid depletion is neglected, the conduction equation for the liquid in grooves may be written as:

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} = 0, \quad (2.29)$$

and the approximate boundary conditions may be written as:

$$\begin{align*}
\text{at } x = 0, x = w & \text{ and } y = 0: T = \text{evaporator wall temperature}, T_e; \\
\text{at } y = \delta: T = \text{vapor temperature}, T_g. \quad (2.30)
\end{align*}$$

On the introduction of $\phi$ defined as:

$$\phi = \frac{T - T_e}{T_g - T_e}, \quad (2.31)$$

Eq. (2.31) becomes

$$\frac{\partial^2 \phi}{\partial x^2} + \frac{\partial^2 \phi}{\partial y^2} = 0 \quad (2.32)$$

with boundary conditions

$$\begin{align*}
\text{at } x = 0, x = w & \text{ and } y = 0: \phi = 0; \text{ and } \text{at } y = \delta: \phi = 1. \quad (2.33)
\end{align*}$$

A solution of Eq. (2.32) with boundary conditions (2.33) by the separation-of-variables method is:

$$\phi = \frac{2}{\pi} \sum_{n=1}^{\infty} \frac{((-1)^n + 1)/n}{\sinh(n\pi\delta/\omega)} \sin(n\pi x/\omega) \sinh(n\pi y/\omega) / \sinh(n\pi\delta/\omega), \quad (2.34)$$

Hence the temperature of the liquid is given by the equation

$$T = T_e + (T_g - T_e) \frac{2}{\pi} \sum_{n=1}^{\infty} \frac{((-1)^n + 1)/n}{\sin(n\pi x/\omega) \sinh(n\pi y/\omega) / \sinh(n\pi\delta/\omega)}. \quad (2.35)$$
Figure 2.12 Co-ordinate system for analysis of heat transfer in rectangular grooves
Figure 2.13 $f(\delta/\omega)$ versus $\delta/\omega$. $x$, exact $f(\delta/\omega)$ Eq. (1.38); , approximate $f(\delta/\omega)$ Eq. (1.39).
The heat transfer per unit length of the rectangular groove $q'$ can then be calculated by the equation:

$$q' = -2K_f \int_0^{\omega/2} (\partial T/\partial y)_{y=0} \, dx,$$  \hspace{1cm} (2.36)

that is

$$(T_e - T_g) = q'/(K_f(4/\pi) \sum_{n=0}^{\infty} [2/(2m+1)\coth((2m+1)\pi \delta/\omega)]).$$  \hspace{1cm} (2.37)

Now let

$$f(\delta/\omega) \equiv \pi/[4 \sum_{m=0}^{\infty} [(2/(2m+1)\coth((2m+1)\pi \delta/\omega))].$$  \hspace{1cm} (2.38)

Fig. 2.13 shows that the exact values of $f(\delta/\omega)$ can be represented accurately by the equation

$$f(\delta/\omega) = 0.185 \text{th}[5.4(\delta/\omega)].$$  \hspace{1cm} (2.39)

Hence the evaporator temperature drop for the rectangular grooved wick with $K_f \ll K_w$ can be calculated accurately by the equation

$$\Delta T_e = 0.185 q' \{\text{th}[5.4(\delta/\omega)]\}/K_f.$$  \hspace{1cm} (2.40)

6. Miscellaneous Topics

(i) Gravitational force. The pressure drop due to gravity $\Delta p_s$ in a column of liquid as shown in Fig. 2.14 can be calculated simply by the equation

$$\Delta p_s = \rho_f g z_t \sin \psi,$$  \hspace{1cm} (2.41)

where $\Delta p_s$ is defined as $(p_2 - p_1)$ and $g$ is the gravitational acceleration.
Figure 2.14 Pressure drop due to gravitational force, 
\[ \Delta P_s = P_2 - P_1 \]
(ii) **Vapor temperature drop.** The Clausius-Clapeyron equation with the perfect gas approximation relates $T_g$ and $p_g$ along the saturation line

$$\frac{dp_g}{dT_g} = \frac{h_{fg} \rho_g}{T_g}$$  \hspace{1cm} (2.42)

where $h_{fg}$ is the heat of vaporization. Hence the vapor temperature drop $\Delta T_g$ is related to the vapor pressure drop $\Delta p_g$ by the equation

$$\Delta T_g = T_g \frac{\Delta p_g}{(\rho_g h_{fg})}.$$  \hspace{1cm} (2.43)

(iii) **Nucleate boiling.** Imagine a spherical vapor bubble of radius $r_b$ in a liquid; at equilibrium

$$\pi r_b^2 (p_{vap} - p_{liq}) = 2\pi r_b \sigma.$$  \hspace{1cm} (2.44)

For heat pipe application $p_{vap}$ corresponds to the saturation vapor pressure at the evaporator wall temperature $T_e$ and $p_{liq}$ corresponds to the pressure of the vapor at the evaporator $p_g$ Eq. (2.44) can be written as

$$p_e - p_g = 2\sigma/r_b.$$  \hspace{1cm} (2.45)

Since $(p_e - p_g) \sim (T_e - T_g)(dp/dT)$, we can combine Eqs. (2.42) and (2.45) to get

$$T_e - T_g = 2T_g \sigma/(h_{fg} \rho_g r_b).$$  \hspace{1cm} (2.46)

Hence if

$$T_e - T_g > 2T_g \sigma/(h_{fg} \rho_g r_b).$$  \hspace{1cm} (2.47)
a bubble of radius \( r_b \) will grow and nucleate boiling may occur.

A large number of experimental data for nucleation boiling have been reported; and a number of correlation equations for empirical data have been derived. Rohsenow\(^{10}\) reports the following correlation equation which satisfactorily represents the pool nucleate boiling data of a number of investigators:

\[
\frac{1}{3} \frac{c_p f (T_e - T_g)}{h_{fg}} = c_{sf} \left\{ \frac{q''}{\sqrt{\frac{\mu h_{fg}}{\sigma/[g(p_f - p_g)]}}} \right\} (\mu_f c_{pf}/K_f)
\]

(2.48)

in which \( c_{pf} \) is the liquid specific heat, \( q'' \) is the heat flux density, and \( c_{sf} \) is the correlation constant ranging from 0.002 to 0.015 dependent upon surface-fluid combinations. For example, for water-platinum combination \( c_{gf} = 0.013 \) and for water-nickel \( c_{sf} = 0.006 \).

Allingham and McEntire\(^{11}\), made both theoretical and experimental studies of nucleate boiling from water-saturated wick material. They recommend the following correlation equation:

\[
\frac{q''}{c_{pf} G(T_e - T_g)} = 0.072 (c_{pf} \mu_f/K_f)^{-0.6} (\rho_f c/\rho_g^2)^{-0.21} (D_e G/\mu_f)^{-0.77},
\]

(2.49)

30
where $G$ is defined as $\dot{q}''/\varepsilon h_{fg}$, $\varepsilon$ is the wick porosity and $D_e$ is the effective diameter of the capillary pores.

(iv) **Sonic velocity.** For gas flow with wall injection and suction as shown in Fig. 2.15, the maximum axial mass flux density $\dot{m}''$ occurs at the downstream end of the injection section. On consideration of a control volume for the injection section, the conservation of mass, momentum and energy requires that

$$ \dot{m}'' = \rho u, \quad (2.50) $$

$$ p_0 = p + \rho u^2, \quad (2.51) $$

$$ T_0 = T + u^2/(2C_p); \quad (2.52) $$

and the ideal gas law states that

$$ p_0/(\rho_0 T_0) = p/(\rho T). \quad (2.53) $$

In the above Eqs. subscript $o$ indicates stagnation state of the injected gas. In terms of Mach number $M$, Eqs. (2.50) through (2.52) may be rewritten as:

$$ \dot{m}'' = \rho M \sqrt{\gamma RT}, \quad (2.54) $$

$$ p_0/p = 1 + \gamma M^2, \quad (2.55) $$

$$ T_0/T = 1 + (\gamma-1)M^2/2. \quad (2.56) $$

In the above Eqs. $\gamma$ is the ratio of specific heats, and $R$ is the gas constant. Combination of Eqs. (2.53) and (2.55) and (2.56) yields an expression for the density ratio
Figure 2.15  Schematic diagram of tube flow with wall injection and suction
\[ \rho_0/\rho = (1+\gamma M^2)/(1+(\gamma-1)M^2/2). \] (2.57)

From Eqs. (2.54), (2.56) and (2.57), the following expression for \( m'' \) can be derived

\[ m'' = \rho_0 \sqrt{\gamma R T_0} M[1+(\gamma-1)M^2/2]^{1/2}/(1+\gamma M^2). \] (2.58)

The maximum mass flux density possible \( \dot{m}_{\text{max}}'' \) for a given stagnation condition is at \( M \) equal to unity. \( \dot{m}_{\text{max}}'' \) can therefore be calculated by Fig. (2.58) with \( M = 1 \), i.e.

\[ \dot{m}_{\text{max}}'' = \rho_0 \sqrt{\gamma R T_0} / [2(\gamma+1)] \] (2.59)

(V) Entrainment. For a vapor flow over a liquid surface, the force which tends to tear the liquid apart is proportional to the product of dynamic pressure \( \rho g \bar{u}^2/2 \) and square of a characteristic length \( a^2 \), and the force which holds the surface liquid is proportional to the product of surface tension of the liquid \( \sigma \) and a characteristic length \( a \). At high velocity, some liquid may be entrained by the vapor. Weber number \( Wb \) defined as the ratio of the two forces mentioned above, namely:

\[ Wb \equiv \rho g \bar{u}^{-2} a/\sigma, \] (2.60)

is used as a criterion for the liquid entrainment.
III. PERFORMANCE OF HEAT PIPES

Performance of a heat pipe is dependent upon the properties of its working fluid. An approximate analysis of heat pipes with various working fluids will first be made below and then mathematical models for steady operation of low-temperature as well as high-temperature heat pipes will be developed.

1. Approximate Analysis

For qualitative comparison of heat-pipe performance with various working fluids, an approximate theory is to be developed for a simple cylindrical heat pipe at horizontal orientation as shown in Fig. 3.1. The following simplifying assumptions are made:

(i) Liquid wets the wick completely,

(ii) Vapor pressure losses are negligible,

(iii) Wick thickness $t$ is much smaller than the radius of the vapor flow passage,

(iv) Heat flux density is uniform at the evaporator and condenser surface,

(v) Thermal conductivity of the liquid-saturated wick is proportional to that of the liquid.

The pressure balance for steady operation of this heat pipe can then be written

$$\Delta p_c = \Delta p_f.$$  (3.1)
At the maximum heat flux $Q_{\text{max}}$, capillary pressure $\Delta p_c$ can be evaluated by Eq. (2.8) with $\theta = 0$, i.e.

$$\Delta p_c = 2\sigma/R_E$$

(3.2)

and liquid pressure drop due to flow resistance $\Delta p_f$ can be evaluated approximately by integrating Eq. (2.13), namely:

$$\Delta p_f = \int_0^{z_e} \left[ c \mu_f m_f / (2\pi t r_i e^2 \rho_f) \right] dz.$$  

(3.3)

In this Eq. (3.3) the axial distribution of $m_f$ in terms of $Q_{\text{max}}$ can be written as:

- $0 \leq z \leq z_e$: $m_f = Q_{\text{max}} z / (z_e h_{fg})$
- $z_e \leq z \leq (z_e + z_a)$: $m_f = Q_{\text{max}} / h_{fg}$
- $(z_e + z_a) \leq z \leq z_t$: $m_f = Q_{\text{max}} \{1 - [z - (z_e + z_a)] / z_e\} / h_{fg}$

(3.4)

Substitution from Eq. (3.4) followed by integration results in the following expression for $\Delta p_f$:

$$\Delta p_f = c Q_{\text{max}} u_f (z_e + 2z_a + z_c) / (4 \rho_f h_{fg} \pi r_i r_e^2 t \varepsilon).$$

(3.5)

Combination of Eqs. (3.2) and (3.5) yields an expression for $Q_{\text{max}}$:

$$Q_{\text{max}} = N_f t (8 \pi r_i r_e^2 \varepsilon / [c R_E (z_e + 2z_a + z_c)]).$$

(3.6)
Where $N_f$ is the liquid transport factor defined as $\sigma \rho_f h_{fg} / \mu_f$, and it is a function of the working-fluid properties. All other terms on the right hand side of Eq. (3.5) depend on the heat pipe design. Fig. 3.2 shows the values of the liquid transport factor, $N_f$, for several heat-pipe working fluids.

The temperature gradient of a heat pipe is determined by the heat flux density and the radial thermal conductance of the wick material saturated with the working fluid in the liquid state. Based upon the simplifying assumptions (iii) and (v) described above, the conductance of the liquid-saturated wick $U$ is seen to be proportional to the thermal conductivity of the liquid divided by the wick thickness, i.e.,

$$U \propto K_f / t \text{ or } \Delta T \propto Qt / K_f.$$  

Now for a heat pipe of fixed physical dimensions, its approximate operating characteristics with different working fluid may be derived from Eqs. (3.6) and (3.7). These are: (a) the maximum heat-transfer capability of the heat pipe is directly proportional to the values of liquid transport factor of the working fluid, and (b) the temperature drop at equal heat-transfer rate is inversely proportional to the liquid thermal conductivity of the working fluid. Fig. 3.3 shows the values of the liquid thermal conductivity for several fluids.
Figure 3.2: Liquid transport factor of several heat-pipe working fluids.

Liquid transport factor (N J ergs/cm² sec) vs. Temperature (°K).
Figure 3.3 Liquid thermal conductivity of several heat-pipe working fluids
Figure 3.4 Liquid thermal conductance factor of several heat pipe working fluids
Similarly, the approximate operating characteristics of heat pipes of equal maximum heat transfer capability may also be derived from Eqs. 3.6 and 3.7. These are: (a) the wick thickness required is inversely proportional to the value of the liquid transport factor of the working fluid; and (b) the temperature drop at equal heat transfer rate is inversely proportional to the product of the liquid thermal conductivity and the liquid transport factor. For the convenience of future reference, this product is named the liquid thermal conductance factor. Its values for several fluids are plotted in Fig. 3.4.

The foregoing are the relative operating characteristics of heat pipes in terms of the liquid transport factor, liquid thermal conductivity and liquid thermal conductance factor. The high values for the liquid-metal fluids, as can be seen in Figs. 3.2 through 3.4, indicate large heat transfer capability and small temperature drop for the high-temperature heat pipes. It is often possible to consider the high-temperature heat pipe as an isothermal device. On the other hand the low values for the cryogenic- and ambient-temperature fluids indicate that the temperature gradients is required to be considered for the low-temperature heat pipes.

2. Performance of High-Temperature Heat Pipes

As seen in the above section, the temperature gradient for high-temperature heat pipe is expected to be small. For
Figure 3.5 Heat pipe limitations

(1-2) sonic
(2-3) entrainment
(3-4) wicking
(4-5) boiling
the performance, its heat-transfer capability is of main concern. Theories for the calculation of heat-transfer limitations have been developed by several authors, for example, see Refs. 2 and 3. Individual limitations indicated in Fig. 3.5 are considered by Kemme. Following the work of Kemme, we derive a mathematical model for cylindrical high-temperature heat pipes at various wick structures as shown in Fig. 3.6.

Sonic limitation. The maximum Mach number at the evaporator exit is unity. For given values of vapor temperature and density at the upstream end of the evaporator \( T_g \) and \( \rho_g \), the maximum mass flux density \( \dot{m}_g^\text{max} \) at the exit of the evaporator, assuming that vapor behaves like an ideal gas, can be calculated by Eq. 2.59, namely:

\[
\dot{m}_g^\text{max} = \rho_g \sqrt{\frac{\gamma R T_g}{2(\gamma+1)}}.
\]  \( (2.59) \)

where the value of \( \gamma \) for monatomic gases is 1.667 and that for diatomic gases is 1.4. As the value of \( \dot{m}_g^\text{max} \) is related to the maximum heat transfer rate by the equation:

\[
\dot{m}_g^\text{max} = Q_{\text{max}}/(\pi r_g^2 h_{fg}).
\]  \( (3.8) \)

Combination of Eqs. (2.59) and (3.9) results in an expression for sonic limitation:

\[
Q_{\text{max}} = \rho_g h_{fg} \pi r_g^2 \sqrt{\frac{\gamma R T_g}{2(\gamma+1)}}.
\]  \( (3.9) \)
Entrainment limitation. If the dynamic pressure of the vapor is excessive, some liquid from the wick-return system may be entrained. Once entrainment begins in a heat pipe, fluid circulation increases until the liquid-return path cannot accommodate the increased flow. This causes dryout and overheat of the evaporator. Weber number defined by Eq. (2.60) i.e., \( W_b = \rho_g \bar{u}^2 a / \sigma \), can be used as a criterion for the entrainment limitation. Magnitude of the characteristic length 'a' in the calculation of the Weber number is dependent upon the wick structures, which can be empirically determined for different structures once and for all. The entrainment limitation can then be determined by the equation

\[
\rho_g \bar{u}^2 a / \sigma = 1. \tag{3.10}
\]

As \( \bar{u} \) is related to \( Q \) by the equation

\[
\bar{u} = Q / (\pi r^2 g \rho h_{fg}) \tag{3.11}
\]

Substitution of Eq. (3.11) into (3.10) yields the following equation for entrainment limitation:

\[
Q_{\text{max}} = \pi r^2 h_{fg} \sqrt{\sigma \rho_g / a}. \tag{3.12}
\]

Wicking limitation. When the sum of pressure drops due to gravitational force and the resistance for liquid and vapor flows balances the maximum capillary pressure, wicking limitation is reached.
The maximum capillary pressure for a given liquid-wick combination can be calculated by Eq. (2.8), i.e.

$$\Delta p_c = 2\sigma \cos \theta / R_E$$  \hspace{1cm} (3.13)

where \( \theta \) is the liquid-wick wetting angle, and \( R_E \) is the effective capillary radius which equals to \( 2\lambda / \varphi \) for the meshed-screen wick and equals to \( 2\lambda / (\varphi - \omega) \) for the wick of open grooves.

The pressure drop due to gravitational force is calculated by Eq. (2.41), i.e.,

$$\Delta p_s = \rho_f g z_t \sin \psi.$$  \hspace{1cm} (3.14)

The vapor pressure drop can be evaluated by integrating Eq. (2.9) with values of \( c_f \) for the evaporator and adiabatic section being calculated by Eq. (2.11). For the condenser at \( |Re_w| < 4.5978 \), \( c_f \) values can also be calculated by Eq. (2.11). However, \( |Re_w| \) values at the condenser of heat pipes operating at this maximum heat-transfer capability are usually greater than 4.5978. Under this circumstance the reverse flow is likely to be developed at the condenser and the recovery of dynamic pressure at the condenser becomes difficult. At the present time, no quantitative information of pressure recovery at the condenser is available. It will be assumed that the dynamic pressure at the down-steam end of the adiabatic section will be lost at the condenser. The total pressure
can then be evaluated by integrating Eq. (2.9) from \( z = 0 \) to \( z = z_e + z_a \) with \( c_f \) values for the evaporator and adiabatic section being calculated by Eq. (2.11), whence

\[
\Delta p_g = \int_0^{z_e+z_a} \frac{2\rho_g u^2}{D_g Re[0.0481 + 0.0494/(4.7+Re_w)]} \, dz \\
+ \left( 8 \int_0^{D_g/2} \rho_g u^2 rdr/D_g^2 \right)_{z=z_e+z_a}^{z=z_e+z_a} \tag{3.15}
\]

The first integral of this Eq. (3.15) can be integrated in a straight-forward manner; and the second integral was investigated by Bohdansky et al.\textsuperscript{12} and Busse\textsuperscript{13}, whose results in the present notation may be written as:

\[
\left( 8 \int_0^{D_g/2} \rho_g u^2 rdr/D_g^2 \right)_{z=z_e+z_a}^{z=e+z_a} = F(2\rho_g \bar{u}^2) \tag{3.16}
\]

Where \( F \) in first approximation is equal to unit. For the case of uniform wall heat-flux density at the evaporator, \( Re_w \) for the evaporator may be written as \( Q_{\text{max}}/(\pi h_f g z_e u_g) \), and \( \bar{u} \) for the evaporator, and the adiabatic section can also be written in terms of \( Q_{\text{max}} \) as follows:

\[
\begin{align*}
\text{if } Q \leq z \leq z_e: \quad & \bar{u} = 4Q_{\text{max}}/(\pi h_f g \rho g D_g^2 z_e) \\
\text{if } a \leq z \leq (z_e + z_a): \quad & \bar{u} = 4Q_{\text{max}}/(\pi h_f g \rho g D_g^2) \tag{3.17}
\end{align*}
\]

Hence the following equation for \( \Delta p_g \) can be derived:

\[
\Delta p_g = \left( Q_{\text{max}}/h_f g \right) \left( (4\rho g/\pi h_f g D_g^4) [z_e/(0.0481 + 0.0494/(4.7 + \frac{Q_{\text{max}}}{h_f g z_e u_g})^0.8 + 32z_a] + 32Q_{\text{max}}/(\pi^2 \rho g h_f g D_g^4) \right) \tag{3.18}
\]
The liquid pressure drop due to flow resistance can be evaluated by integration of Eq. (2.13), (2.22) or (2.23) depending on whether the capillary wick is made of wrapped screens, open grooves, or screen covered grooves. The resultant $\Delta p_f$ expressions written in terms of $Q_{\text{max}}$ are as follows:

For wrapped-screen screen wick:

$$
\Delta p_f = \frac{cQ_{\text{max}} u_f (z_e + 2z_a + z_c)}{[2\pi \rho_f h_f \varepsilon e (r_i - g^2)]}. \quad (3.19a)
$$

For open-groove wick:

$$
\Delta p_f = 9Q_{\text{max}} u_f (\omega^2 + 4\delta^2) (z_e + 2z_a + z_a) / (5n \rho_f h_f \omega^3 \delta^3). \quad (3.19b)
$$

For screen-covered groove wicks:

$$
\Delta p_f = \frac{36 Q_{\text{max}} u_f (\omega^2 + \delta^2) (z_e + 2z_a + z_c)}{5n \rho_f h_f \omega^3 \delta^3}. \quad (3.19c)
$$

In Eqs. (3.19b) and (3.19c), $n$ is the number of grooves.

On the substitution of Eqs. (3.13), (3.14), (3.18) and (3.19) for the pressure differences $\Delta p_c$, $\Delta p_s$, $\Delta p_g$ and $\Delta p_f$ respectively into the following pressure-balance equation,

$$
\Delta p_c = \Delta p_s + \Delta p_g + \Delta p_f', \quad (3.20)
$$

the maximum heat-transfer rate due to wick limitation can be calculated.
Boiling limitation. Nucleate boiling in the wick structure of a heat pipe may interfere with the liquid return. The limiting temperature drop at the evaporator for the onset of nucleation can be calculated by Eq. (2.46), namely:

\[ T_e - T_g = \frac{2T_g \sigma}{(h_f g g r_b)}, \]  

(3.21)

where the value of the effective bubble radius is dependent upon the wick structure and it has to be determined empirically at the present time. \((T_e - T_g)\) may be written, in terms of the heat transfer rate, as follows:

\[ (T_e - T_g) = \frac{Q_{\text{max}} (r_i - r_g)}{\pi K e z (r_i + r_g)} \]  

(3.22)

with the effective thermal conductivity, \(K_e\), being evaluated by Eq. (2.26) for the wrapped-screen wicks and by Eq. (2.25) for the groove wicks. The value of porosity, \(\varepsilon\), for the wrapped screen wick is dependent upon the wrapping tightness and the value of \(\varepsilon\) for the groove wick may be calculated by the equation

\[ \varepsilon = n \omega / [\pi (r_i - r_g)] \]  

(3.23)

Combination of Eqs. (2.26), (3.21) and (3.22) yields the following expression of boiling limitation for the wrapped-screen heat pipe:

\[ Q_{\text{max}} = 2\pi T_g \sigma \{z_e (r_i + r_g) K_f [K_f + K_w (1-\varepsilon) (K_f - K_w)]/[\pi (r_i - r_g) (K_f + K_w + (1-\varepsilon) (K_f - K_w)) + (K_f - K_w)] h_f g g r_b}\]  

(3.24)
Fig. 3.7 Comparison of experimental and theoretical heat-pipe limitations. •, screen-covered groove wick; x, open-groove wick
and combination of Eqs. (2.25) and (3.21) through (3.23) gives the following expression of boiling limitation for the groove heat pipes:

$$Q_{\text{max}} = 2T_g \sigma z_e [n\omega K_f + \pi (r_i + r_g) - n\omega K_w] / \{(r_i - r_g) h_f \rho_f \gamma}$$ (3.25)

Computer program and comparison with experiments. A computer program has been written for the math model developed above for the calculation of heat transfer limitation. A program listing together with users instruction is appended to the manual, Appendix A.

An example of the calculated results using this program is shown in Fig. 3.7 for comparison with Kemme's experiments under the same conditions. It can be seen in this figure that the experimental data are well represented by the theoretical prediction. However, it should be mentioned that values of the liquid-wick wetting angle \(\theta\), and the characteristic lengths 'a' for entrainment have been so chosen to give this good agreement.

3. Performance of Low-Temperature Heat Pipes

It was shown in Section I of this chapter that the temperature drops for the ambient- and low-temperature heat pipes are often important. Complete performance of these heat pipes will include not only the maximum heat transfer capability but also the temperature drops at various operating conditions. Accordingly a math model is developed to calculate these quantities.
**Maximum heat-transfer capability.** Because of the low values of liquid transport factor for the working fluids of the low temperature heat pipes, the most likely heat-transfer limitation is the wick limitation. Procedure for the calculation of wick limitation described in the above section for the high-temperature heat pipe is also applicable to the calculation of wick limitation for the low-temperature heat pipes. Thus the wick limitation is calculated by equating maximum capillary pressure $\Delta p_c$ of Eq. (3.13) to the sum of the hydrostatic pressure drop $\Delta p_s$ of Eq. (3.14), the vapor-flow pressure drop $\Delta p_g$ of Eq. (3.18) and the liquid-flow pressure drop $\Delta p_f$ of Eq. (3.19).

**Temperature drop.** The total temperature drop of a heat pipe is equal to the sum of temperature drops at the evaporator, vapor flow passage and condenser. Conduction model described in Section 5 of Chapter II can be used to calculate the temperature drops at the evaporator and condenser, respectively. The vapor pressure drop $\Delta T_g$ can be calculated by Clausius-Clopeyron Eq. (2.43)

$$\Delta T_g = T_g \frac{\Delta \overline{p}_g}{(\rho_g h_{fg})},$$  \hspace{1cm} (3.26)

where the average vapor pressure drop $\Delta \overline{p}_g$ is the difference of the average vapor pressures at the evaporator and the condenser; it can be calculated by Eq. (3.27a) or (3.27b) below
depending on whether $|Re_w|$ at condenser is less or greater than 4.5975:

$$\Delta P_g = (2\mu g Q/\pi h_{fg} \rho g D_g)^4 \left\{ \frac{z_e}{(0.0481 + 0.0494/(4.7 + \frac{Q}{\pi h_{fg} \rho g})^{0.8})} \right\}$$

$$+ 64z_a + \left[ \frac{z_e}{(0.0481 + 0.0494/(4.7 - \frac{Q}{\pi h_{fg} \rho g})^{0.8})} \right]$$

(3.27a)

or

$$\Delta P_g = (Q/h_{fg}) \left\{ (2\mu g/\pi \rho g D_g)^2 \left[ \frac{z_e}{(0.0481 + 0.0494/(4.7 + \frac{Q}{\pi h_{fg} \rho g})^{0.8})} \right] \right\}$$

$$+ 64z_a + 32Q/(\pi \rho g h_{fg} D_g^4)$$

(3.27b)

These Eqs. (3.27a) and (3.27b) for $\Delta P_g$ are obtained by integration of Eq. (2.9) using the procedure described in Section 2 of this chapter.

The temperature drop at the evaporator $\Delta T_e$ for thin wick structures can be calculated by the equation

$$\Delta T_e = Q(r_i - r_g)/[\pi z_e K_e (r_i + r_g)]$$

(3.28)

where $K_e$ is the effective thermal conductivity of the liquid saturated wick which may be approximated by Eq. (2.26) for the wrapped screen wick and by Eq. (2.28) for the sintered-porous-metal wick. For wick structures composed of rectangular
grooves with $K_f \ll K_w$ evaporation occurring at the liquid-vapor interface but not at the groove fin tip, $\Delta T_e$ expression can be derived from Eq. (2.40), i.e.,

$$\Delta T_e = 0.185 Q_{th} [5.4(\delta/\omega)]/(n z_e K_f)$$

(3.29)

The temperature drop at the condenser $\Delta T_c$ for thin wick structures can be calculated by the equation

$$\Delta T_c = Q(r_i-r_g)/[\pi z_e K_e (r_i+r_g)]$$

(3.30)

where the effective thermal conductivity $K_e$ for the liquid-saturated wrapped-screen and sintered-porous-metal wicks can again be calculated by Eqs. (2.30) and (2.28), respectively. For wicks of rectangular grooves condensation occurs at the liquid-vapor interface as well as the groove fin tips; and liquid and groove fin conduct heat in parallel. $K_e$ can be calculated by Eq. (2.25), namely:

$$K_e = \varepsilon K_f + (1-\varepsilon) K_w$$

(3.31)

where $\varepsilon$ is equal $n \omega / [\pi (r_i+r_g)]$.

**Computer program and comparison with experiments.** A computer program for the above described math model for the low temperature heat pipe has been developed; both maximum heat transfer capability and total temperature drops are calculated at various operating conditions. The program is applicable to the heat pipe with wrapped-screen wick, open-rectangular-groove wick or screen-covered-rectangular-groove wick. The
program listing together with users instruction is given in Appendix B.

The calculated evaporator temperature drop for a liquid nitrogen heat pipe with wrapped-screen wick is plotted versus heat transfer rate in Fig. 3.8 for comparison with Raskin's experimental data under the same operating conditions. Predictions based upon the present conduction model closely represents the empirical data. Also shown in Fig. 3.8 are the evaporator temperature drops calculated by the theories of nucleate pool boiling Eq. (2.48), and nucleate boiling in wick structures Eq. (2.49). It can be seen in this Fig. 3.8 that nucleate boiling heat transfer theory underestimate the evaporate temperature drop. For further improvement on the present conduction model, liquid property variation with respect to temperature and convection heat transfer should be accounted, as can be seen in Figs. 3.9 and 3.10. Details of conduction evaporation model with convection and property variation considered are described in Ref. 14.

The low temperature heat pipes with the wrapped screen wicks have rather poor performance from the temperature-drop point of view as can be seen in Fig. 3.10. The grooved heat pipe offers improvement in both the heat transfer capability and the effective thermal conductance, because the grooves provide a small liquid flow resistance for liquid return and
Figure 3.8 Comparison of evaporation heat transfer theories with experiments
Figure 3.9 Evaporation based upon conduction model taking into account of convection and fluid property variations
Figure 3.10 Condenser, evaporator and total temperature drops of a LN$_2$ heat pipe -- A comparison between theory and experiments
the highly conductive groove fins insure low resistance for heat flow.

The present predictions for the heat transfer capability and temperature drops of a Dynatherm liquid nitrogen heat pipe with groove wicks are shown in Figs. 3.11 and 3.12 for comparison with the experiments\textsuperscript{16} by Dynatherm under the same conditions.

The above Dynatherm nitrogen cryopipe is used as an example to illustrate the use of the present computer program to generate complete performance of the low-temperature heat pipes. This heat pipe has the following specifications:

\begin{itemize}
  \item Mean groove height, $\delta$ \hspace{1cm} 0.089 cm
  \item Mean groove width, $\omega$ \hspace{1cm} 0.064 cm
  \item Inner diameter, $D_i$ \hspace{1cm} 0.97 cm
  \item Outer diameter, $D_o$ \hspace{1cm} 1.27 cm
  \item No. of grooves, $n$ \hspace{1cm} 30
  \item Evaporator length, $z_e$ \hspace{1cm} 10 cm
  \item Adiabatic-section length, $z_a$ \hspace{1cm} 120 cm
  \item Condenser length, $z_c$ \hspace{1cm} 10 cm
  \item Empirical capillary radius, $R_E$ \hspace{1cm} 0.0354 cm
  \item Container material \hspace{1cm} 6061-76Al
\end{itemize}

The predicted complete performance of this heat pipe is shown in Fig. 3.13 in which the locus of the maximum heat-transfer capability and the lines of constant rate of heat transfer are mapped on the total temperature-drop versus condenser-temperature coordinates.
Figure 3.11 Comparison of theoretical heat transfer limitation of a grooved LNG heat pipe with experimental data.
Figure 3.12 Comparison of theoretical temperature drop of a grooved IN2 heat pipe with experiments
Figure 3.13 Predicted performance of a grooved LN$_2$ heat pipe
IV. CONCLUSIONS AND RECOMMENDATIONS

1. Conclusions

In conclusion the results of this research can be summarized as follows:

(i) Fundamentals of heat and mass transfer theory are reviewed with a view of application to theoretical analyses of heat pipes of various wick structures and working fluids.

(ii) A math model for high-temperature heat pipes has been developed by means of which it is possible to predict the heat-transfer limitations of heat pipes with wrapped-screen, rectangular-groove or screen-covered-rectangular-groove wick.

(iii) A math model for low temperature heat pipes with wrapped-screen, rectangular-groove or screen-covered-rectangular-groove wick has also been developed by means of which complete performance of heat pipes (e.g., see Fig. 3.3) including both heat transfer limitations and temperature gradients at different operating conditions can be predicted.

(iv) The extent to which the present prediction correlates the existing experimental data can be judged by inspection of Figs. 3.7, 3.11 and 3.12.

(v) The theory is capable of greater refinement when more comprehensive experimental data become available.
2. **Recommendations**

It is recommended that the present theory should be extended and developed in the following ways:

(i) There is a need for comprehensive experimental data in particular those for the low temperature heat pipes. Initial work confined to a single low-temperature heat pipe at different operating conditions would be fruitful, particularly with the aim of obtaining complete performance of a heat pipe as shown in Fig. 3.13. The next step would carry out tests on the same heat pipe with a variety of working fluids and combinations of evaporator, adiabatic-section and condenser lengths.

(ii) Laminar and turbulent tube flows with mass injection have been reasonably well understood. By comparison, knowledge of laminar and turbulent tube flows with large wall suction is limited at the present time. Both theoretical and experimental data on tube flows with large suction are urgently needed to increase confidence in calculating pressure drops due to vapor flow resistance.

(iii) Mechanisms of heat and mass transfer in groove wick are not well understood at the present time. Detailed studies should now be made.
(iv) When the data recommended above become available, complete performance of heat pipe will be able to be developed with greater assurance of its accuracy.
V. REFERENCES


APPENDIX A

A COMPUTER PROGRAM FOR LIMITATIONS OF HIGH TEMPERATURE HEAT PIPES

A complete listing of the computer program in Fortran IV language for calculation of sonic, entrainment, wick and boiling limitations is given in this Appendix. This program is written for application to the wrapped-screen-wick, open rectangular-groove and screen-covered rectangular-groove heat pipes. Also given in this Appendix are the flow diagram of the program (Fig. A1), and the data input sheets (Table A1). All the integers, beginning with I through N, at the input are 5-digit figures (I5); and all other inputs are 15-digit figures (E15.6). The main Fortran input and output symbols are defined as follows:

Input:

- **GRF**: gravitational constant, 981 cm/sec²
- **ND**: number of sets of heat-pipe physical dimensions
- **NTHP**: type of heat pipes; i.e., 1 for wrapped screen, 2 for open groove, and 3 for screen covered groove
- **RG**: radius of vapor flow passage, cm
- **RI**: inner radius of heat-pipe container, cm
- **ZE**: length of evaporator, cm
- **ZA**: length of adiabatic section, cm
- **ZC**: length of condenser, cm
- **PSI**: inclination of heat pipe, radian
- **RB**: effective bubble radius for boiling, cm
- **THETA**: liquid wetting angle, radian
- **A**: characteristic length for entrainment, cm
**Appendix A - Continued**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>CDWK</td>
<td>thermal conductivity of wick material, erg/cm sec °K</td>
</tr>
<tr>
<td>RF</td>
<td>effective hydraulic radius for liquid flow, cm</td>
</tr>
<tr>
<td>C</td>
<td>permeability factor for capillary structure, a non-dimensional constant</td>
</tr>
<tr>
<td>EPSIL</td>
<td>porosity of capillary structure</td>
</tr>
<tr>
<td>AREA</td>
<td>cross-sectional area of capillary pores or capillary grooves, cm</td>
</tr>
<tr>
<td>PERI</td>
<td>wetted perimeter of capillary pores or capillary grooves, cm</td>
</tr>
<tr>
<td>GRVN</td>
<td>number of capillary grooves</td>
</tr>
<tr>
<td>WIDTH</td>
<td>width of capillary grooves</td>
</tr>
<tr>
<td>DEPTH</td>
<td>depth of capillary grooves</td>
</tr>
<tr>
<td>NP</td>
<td>number of sets of fluid property input</td>
</tr>
<tr>
<td>TSAT</td>
<td>fluid saturation temperature, °K</td>
</tr>
<tr>
<td>PSAT</td>
<td>saturation vapor pressure, dyne/cm²</td>
</tr>
<tr>
<td>CPF</td>
<td>saturation liquid specific heat, ergs/gm °K</td>
</tr>
<tr>
<td>DENF</td>
<td>saturation liquid density, gm/cm³</td>
</tr>
<tr>
<td>VSF</td>
<td>saturation liquid viscosity, poise</td>
</tr>
<tr>
<td>CDF</td>
<td>saturation liquid thermal conductivity</td>
</tr>
<tr>
<td>SFT</td>
<td>saturation liquid surface tension, dyne/cm</td>
</tr>
<tr>
<td>HFG</td>
<td>heat of vaporization, erg/gm</td>
</tr>
<tr>
<td>DENG</td>
<td>saturation vapor density, gm/cm</td>
</tr>
<tr>
<td>VSG</td>
<td>saturation vapor viscosity, poise</td>
</tr>
<tr>
<td>GAMMA</td>
<td>vapor specific heats ratio</td>
</tr>
</tbody>
</table>

**Output**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>ZE</td>
<td>length of evaporator, cm</td>
</tr>
<tr>
<td>ZA</td>
<td>length of adiabatic section, cm</td>
</tr>
<tr>
<td>ZA</td>
<td>length of condenser, cm</td>
</tr>
<tr>
<td>PSI</td>
<td>heat-pipe inclination, radian</td>
</tr>
<tr>
<td>A</td>
<td>characteristic length for entrainment, cm</td>
</tr>
<tr>
<td>THETA</td>
<td>liquid wetting angle, radian</td>
</tr>
<tr>
<td>RB</td>
<td>effective bubble radius for boiling, cm</td>
</tr>
<tr>
<td>TSAT</td>
<td>heat-pipe vapor temperature, °K</td>
</tr>
</tbody>
</table>
Appendix A - Continued

QS  sonic limitation, erg/sec
QEL  entrainment limitation, erg/sec
QWL  wick limitation, erg/sec
QBL  boiling limitation, erg/sec
QMAX  ultimate heat-pipe limitation (i.e., smallest of QSL, QEL, QWL, QBL), erg/sec.

(NOTE: 1 watt = $10^7$ ergs/sec)
Fig. A1 Flow diagram of program for performance of high-temperature heat pipe

Start

Input magnitude of gravitational acceleration, GRF

Input number of sets of heat pipe dimensions, ND

Input value of NTHP for type of heat pipe: 1 for wrapped screen, 2 for open groove, 3 for covered groove

Input heat pipe dimensions: RG, RI, ZE, ZA, ZC, PSI, RB, THETA, A, CDW

no

NTHP=1?

yes

Input further dimensions: RF, G, EPSIL, AREA, PERI

no

NTHP=2?

yes

Input further dimensions: GRVN, WIDTH, DEPTH

Input further dimension: GRVN, WIDTH, DEPTH, AREA, PERI

Input number of sets of fluid properties, NP

Input property values: TSAT, PSAT, CPF, DENF, VSF, CDF, SFT, HFG, DENG, VSC, GAMMA

Calculate sonic, entrainment, wick, boiling and overall limitations

Output Tsat and heat pipe limitations

Stop
<table>
<thead>
<tr>
<th>GRF</th>
<th>ND</th>
<th>NTHP (=1)</th>
<th>RG</th>
<th>RI</th>
<th>ZE</th>
<th>ZA</th>
<th>ZC</th>
<th>PSI</th>
<th>RB</th>
<th>THETA</th>
<th>A</th>
<th>CDWK</th>
<th>RF</th>
<th>C</th>
<th>EPSIL</th>
</tr>
</thead>
<tbody>
<tr>
<td>AREA</td>
<td>NP</td>
<td>TSAT</td>
<td>PSAT</td>
<td>CPF</td>
<td>DENF</td>
<td>VSF</td>
<td>CDF</td>
<td>SFT</td>
<td>HFG</td>
<td>DEVG</td>
<td>VSG</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>GAMMA</td>
<td></td>
<td></td>
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</tr>
</tbody>
</table>
**TABLE A1 (b)**

**INPUT DATA FOR OPEN-GROOVE HEAT PIPES**

<table>
<thead>
<tr>
<th>Input Data</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>GRF</td>
<td></td>
</tr>
<tr>
<td>ND</td>
<td></td>
</tr>
<tr>
<td>NTHP (=2)</td>
<td></td>
</tr>
<tr>
<td>RG</td>
<td></td>
</tr>
<tr>
<td>ZE</td>
<td></td>
</tr>
<tr>
<td>RB</td>
<td></td>
</tr>
<tr>
<td>GRVN</td>
<td></td>
</tr>
<tr>
<td>NP</td>
<td></td>
</tr>
<tr>
<td>TSAT</td>
<td></td>
</tr>
<tr>
<td>DENF</td>
<td></td>
</tr>
<tr>
<td>HFG</td>
<td></td>
</tr>
<tr>
<td>GAMMA</td>
<td></td>
</tr>
</tbody>
</table>

- **GRF**: GRF
- **ND**: ND
- **NTHP (=2)**: Number of heat pipes
- **RG**: Radius of Groove
- **ZE**: ZE coordinate
- **RB**: Radius of Base
- **GRVN**: Groove Number
- **NP**: Number of Points
- **TSAT**: Temperature Saturation
- **DENF**: Density of Fluid
- **HFG**: Heat Flux
- **GAMMA**: Gamma
**TABLE A1 (c)**

**INPUT DATA FOR SCREENED-COVERED-GROOVE HEAT PIPES**

<table>
<thead>
<tr>
<th>GRF</th>
<th>ND</th>
<th>NTHP ( = 3 )</th>
<th>RG</th>
<th>RI</th>
<th>ZE</th>
<th>ZA</th>
<th>ZC</th>
<th>PSI</th>
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<tr>
<td></td>
<td></td>
<td></td>
<td>RB</td>
<td>THETA</td>
<td></td>
<td>A</td>
<td>DEPTH</td>
<td>CDWK</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>GRVN</td>
<td>WIDTH</td>
<td></td>
<td>PERI</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>AREA</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>NP</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>TSAT</td>
<td>PSAT</td>
<td></td>
<td>CPF</td>
<td></td>
<td>SFT</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>DENF</td>
<td>VSF</td>
<td></td>
<td>CDF</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>HFG</td>
<td>DENG</td>
<td></td>
<td>VSG</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>GAMMA</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
C PERFORMANCE OF HIGH-TEMPERATURE HEAT PIPES

1 20 FORMAT (//43H PERFORMANCE OF HIGH-TEMPERATURE HEAT PIPES)
2 WRITE (6,20)
3 1 FORMAT (15)
4 21 FORMAT (///SGH ZF ZA ZC 1 PSI)
5 22 FORMAT (/3ZH A THETA RB)
6 11 FORMAT (E16.6)
7 12 FORMAT (2E15.6)
8 13 FORMAT (3E15.6)
9 14 FORMAT (4E15.6)
10 16 FORMAT (6E15.6)
11 READ (5,11)GRF
12 READ (5,11) NO
13 DO 101 MD=1,ND
14 READ (5,1) NTHP
15 READ (5,12) PG,RI
16 RG=2.*RG
17 READ (5,14) ZE,ZA,ZC,PSI
18 READ (5,14) RB,THETA,ACDWP
19 IF (NTHP - 1) 35,31,35
20 35 IF (NTHP - 2) 33,32,33
21 31 READ (5,13) RF,CG,FPISIL
22 READ (5,12) AREA,PERI
23 RE=2.*AREA/PERI
24 GO TO 34
25 32 READ (5,13) GRVN, WIDTH, DEPTH
26 RE=WIDTH
27 GO TO 34
28 33 READ (5,13) GRVN, WIDTH, DEPTH
29 READ (5,12) AREA, PERI
30 RE = 2.*AREA/PERI
31 34 READ (5,11) NP
32 WRITE (6,21)
33 WRITE (6,14) ZE,ZA,ZC,PSI
34 WRITE (6,22)
35 WRITE (6,13) A,THETA,RB
36 WRITE (6,23)
37 DO 102 MP=1,NP
38 READ (5,13) TSAT,PSAT,CPF
39 READ (5,14) DNF,VSF,CDP,SFT
40 READ (5,13) HFG,DENG,VS
41 READ (5,11) GAMMA
42 QSL=DENG*HFG*3.14159*RG*RG*SQF(T(GAMMA*PSAT/(2.*DENG*(GAMMA+1.)))
43 QEL=3.14159*RG*PS*SQF(T(SFT*DENG/T))
44 DPC=2.*SFT*COS(THETA)/RE
45 DPS=DFS*GRF*(ZE*ZC)*SIN(PSI)
46 IF (NTHP-1) 45,41,45
47 45 IF (NTHP = -2) 43,42,43
48 41 DPF=DFS*GRF*(ZE*ZC)/(1.*3.14159*DNF*HFG*EPSIL*RE*RF)
49 DPF=DPF/(RI*RI-DENG)
50 QBL=3.14159*TSAT*ST*ZE*CDF(RI*RG)
51 JBL=QBL*(DCF*CDFW0.1.*EPSIL)*CDW
52 QBL=QBL*(HFG*DENG*RF*(RI*RG))
53 QBL=QBL*(DCF*CDFW0.1.*EPSIL)*CDW
54 QBL=2.*QBL
55 GO TO 44
56 42 DPF=9.*VSF*(WIDTH**2+0.*DEPTH**2)*Z(E*ZC)
57 DPF=DPF/(5.*GRVN*DENG*HFG*(WIDTH**3)*(DEPTH**3))
QBL = 2. * TPSAT * SFT * (GRVN * WIDTH * CD + (3.14159 * (RI + RG) - GRVN * WIDTH) * CDW):
1 * ZF / (RI - RG) * HG * DENGE * R8)
69 GC TO 44
60 DPF = 36.0 * VSF * (WIDTH ** 2 + DEPTH ** 2) * (ZI + 2. * ZA + 2C)
61 DPF = DPF / (S * GRVN * DENGE * HG * (WIDTH ** 3) * (DEPTH ** 3))
62 QBL = 2. * TPSAT * SFT * (GRVN * WIDTH * CD + (3.14159 * (RI + RG) - GRVN * WIDTH) * CDW):
1 / (RI - RG) * HG * DENGE * R8)
63 DPG2 = 32. / (3.14159 * 3.14159 * DENGE * HG * DG ** 4)
64 DPGX = 4. * VSF / (3.14159 * DENGE * HG * DG ** 4)
65 DQ TO 103 M = 1, 5
66 IF (M - 1) 52, 51, 52
67 DPG1 = DPGX * (21. * ZE + 32. * ZA)
68 GO TO 53
69 DNM = EXP (0.8 * ALQG (4.7 + QWL / (3.14159 * HG = ZE * VSF)))
70 DNM = 0.00481 * 0.0494 / DNM
71 DPG1 = DPGX * (ZE / DNM + 32. * ZA)
72 IF (DPG1 - OPC) 61, 60, 60
73 60 AQ = 0
74 GO TO 103
75 DQ = DPG2
76 EQ = (DPF - DPG1) / 2.
77 DQ = DPG1
78 103 QWL = (Q0 + SQRT (80 * Q0 - AQ * 0)) / AQ
79 FORMAT (1 / 87M, TSAT, CS, QEL
80 1 QWL, QEL, QWMAX)
81 IF (QSL - QEL) 71, 71, 72
82 71 QMAX = QSL
83 72 QMAX = QEL
84 73 IF (QMAX - QWL) 74, 74, 75
85 75 QMAX = QWL
86 74 IF (QMAX - QWL) 76, 76, 77
87 77 QMAX = QWL
88 WRITE (6, 16) TSAT, QSL, QEL, QWL, QRL, QMAX
89 102 CONTINUE
90 101 CONTINUE
91 CALL EXIT
92 STOP
93 END
A complete listing of the computer program in Fortran IV language for calculation of complete performance of cryogenic and ambient-temperature heat pipes is given in this Appendix; both the maximum heat transfer capability and the evaporator— as well as the condenser-wall temperatures at various heat transfer rates are predicted. This program is applicable to the wrapped-screen-wick, open rectangular-groove and screen-covered rectangular-groove heat pipes. Also given in this Appendix are the flow diagram of the program (Fig. B1) and the data input sheets (Table B1). All the integers, beginning with I through N, at the input are 5-digit figures (I5); and all other inputs are 15-digit figures (E15.6). The main FORTRAN input and output symbols are defined as follows:

**Input**

- **NQ**: number of heat transfer rates at which the evaporator and condenser temperatures are calculated, ergs/sec.
- **QI**: initial heat transfer rates for temperature calculations, ergs/sec.
- **DQ**: stepwise increase of heat transfer rates for temperature calculations, ergs/sec.

(All other input symbols are same as those defined in APPENDIX A.)
Appendix B - Continued

Output

ZE     length of evaporator, cm
ZA     length of adiabatic section, cm
ZC     length of condenser, cm
PSI    angle of inclination of heat pipe, radian
THETA  liquid wetting angle, radian
TSAT   saturation vapor temperature, °K
QMAX   maximum heat transfer capability, ergs/sec
TE     evaporator wall temperature, °K
TC     condenser wall temperature, °K
DT     temperature difference between the evaporator and the condenser walls, °K
Fig. B1 Flow diagram of program for performance of low-temperature heat pipe

Start

Input magnitude of gravitational acceleration, CRF

Input number of sets of heat pipe dimensions, ND

Input value of NTHP for type of heat pipe: 1 for wrapped screen, 2 for open groove, 3 for covered groove

Input heat pipe dimensions: RG, RI, ZE, ZA, ZC, PSI, THETA, CDWK

Input number & magnitude of heat transfer rates, NQ, QI, DQ

NTHP=1?

no

NTHP=2?

yes

yes

Input further dimensions: RF, C, EPSIL, AREA, PERI

no

Input further dimension: GRVN, WIDTH, DEPTH

Input further dimensions: GRVN, WIDTH, DEPTH, SREA, PERI

Input number of sets of fluid properties, NP

Input property values: TSAT, PSAT, CPF, DENF, VSF, CDF, SFT, HFC, DENG, VSC

Calculate and output of heat pipe limitations at various operating temperatures

Calculate and output condenser and evaporator wall temperatures and total temperature drop at various heat transfer rates

Stop

End of Document
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<tr>
<td>ZE</td>
<td>ZA</td>
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<tr>
<td>THETA</td>
<td>CDWK</td>
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<td>PSAT</td>
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<tr>
<td>DENF</td>
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<tr>
<td>HFG</td>
<td>DENG</td>
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### TABLE B1 (b)

**INPUT DATA FOR OPEN-GROOVE HEAT PIPE**

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<thead>
<tr>
<th>Column</th>
<th>Abbreviation</th>
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<td>ND</td>
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</tr>
<tr>
<td>NTHP</td>
<td>(= 2)</td>
</tr>
<tr>
<td>RG</td>
<td>RI</td>
</tr>
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<td>ZE</td>
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<td>CDWK</td>
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</tr>
<tr>
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<tr>
<td>GRVN</td>
<td>Sidth</td>
</tr>
<tr>
<td>NP</td>
<td></td>
</tr>
<tr>
<td>TSAT</td>
<td>PSAT</td>
</tr>
<tr>
<td>DENF</td>
<td>VSP</td>
</tr>
<tr>
<td>HFG</td>
<td>Deng</td>
</tr>
</tbody>
</table>

**Additional Units:**
- PSI
- DEPTH
- CPF
- CDF
- SFT
- VSG
## TABLE B1 (c)

INPUT DATA FOR SCREEN-COVERED-GROOVE HEAT PIPES

| GRF  | ND  | NTHP (= 3) | RG  | RI  | ZE  | ZA  | CDWK | THETA  | NQ  | QI  | GRVN | WIDTH | AREA | PERI | NP  | TSAT | PSAT | DENF | VSF  | HFG  | DENG | VSG  | PSI  |
|------|-----|------------|-----|-----|-----|-----|------|--------|-----|-----|------|-------|------|------|-----|------|------|------|------|------|------|-----|-----|------|------|------|------|------|------|------|
|      |     |            |     |     |     |     |      |        |     |     |      |       |      |      |     |      |      |     |     |     |     |     |     |      |      |      |      |      |      |
$JOB (7352, F1, 'CHL, J')

1 C PERFORMANCE OF LOW TEMPERATURE HEAT PIPES
2 20 FORMAT (///42H PERFORMANCE OF LOW-TEMPERATURE HEAT PIPES)
3 WRITE (6,20)
4 1 FORMAT (15)
5 11 FORMAT (E15.6)
6 12 FORMAT (2E15.6)
7 13 FORMAT (3E15.6)
8 14 FORMAT (4E15.6)
9 15 FORMAT (5E15.6)
10 READ (5,11) GRF
11 READ (5,1) NO
12 READ (5,1) NTMP
13 READ (5,12) RG,RF
14 RG=2.*RG
15 READ (5,14) ZE,ZA,ZC,PSI
16 READ (5,12) THETA,COWK
17 READ (5,1) NO
18 READ (5,12) Q1,QD
19 IF (NTMP=1) 35,31,35
20 35 IF (NTMP=2) 33,32,33
21 31 READ (5,13) RF,C,EPISL
22 READ (5,12) AREA,PERI
23 RF=2.*AREA/PERI
24 GO TO 34
25 32 READ (5,13) GRVN,WIDT, DEPT
26 RF=WIDT
27 GO TO 34
28 33 READ (5,13) GRVN,WIDT, DEPT
29 READ (5,12) AREA,PERI
30 IF (NTHP=2) 45,41,45
31 34 IF (NTHP=1) 45,41,45
32 21 FORMAT (///72H ZF ZA ZC
1 PSI THETA)
33 WRITE (6,21)
34 WRITE (6,15) ZE,ZA,ZC,PSI,THETA
35 DO 102 MP=1,NP
36 READ (5,13) TSAT,PSAT,GRF
37 READ (5,14) DENF,VSF,COF,SFT
38 READ (5,13) HFG,DENG,VSF
39 OPC=2.*SFT*COS(THETA1)/RF
40 OPC=OPF*(ZE+ZA+ZC)*SIN(P1)
41 IF (NTHP=1) 45,41,45
42 45 IF (NTHP=2) 43,42,43
43 41 OPC=OPF*(ZE+2.*ZA+ZC)/(2.*3.14159*DENG*EPSIL*RF*RF)
45 GO TO 44
48 GO TO 44
53 DO 103 M=1,5
54 103 IF (M=1) 52,51,52
55 51 OP3=OPF*EXP((9.*3+3.14159+2*VSF)/(3.14159*DENG*2*VSF))
FCO = DPGX*(ZI 53 IF (OPS-OPC) = 0
GO TO 103
AO=DPG2
B/ONM*32.*ZT) M/6C
3Q=(DPF*DPG1
3Q=OPS-OPC
DC 1C4 M0=1,10
IF (MQ-1) 72,71,72
Q=OWL
GO TO 73
72 Q=Q+DQ
72 DTGE=E XP(0.3*ALOG(4.7*Q/(3.14159*Z/EQ*VS.G))
74 DTGE=2*X*Z*VS.G/13.14159+EPS*EPS=EPS
75 DTGA=128*VSG*Q/X/13.14159+ENG*(ENG**4)*HFG
76 DTGC=32*Q/X/13.14159+ENG*ENG=ENG**4)
77 DTGE=TSAT*DTGE/(ENG*HFG)
78 DTGA=TSAT*DTGA/(ENG*HFG)
79 DTGC=TSAT*DTGC/(ENG*HFG)
80 IF (YTHP-1) 82,81,82
81 CDH = COF*(COF*COF-(1.-FPS/R)) (C.OF-CFOK)/(C.OF^2-CFOK)
82 DTWC=Q*(QI-RG) / (3.14159*QDE*ZE*(R1+RG))
83 DTWC=Q*(R1-RG) / (3.14159*QDE*ZE*(R1+RG))
GO TO 83
85 92 EPP=GRVN*WIDTH/ (3.14159*(R1+RG))
86 CDK=FPP*COF*(1.-EPP)*CDWK
87 DTWC=Q*(R1-PG) / (3.14159*QDE*ZE*(R1+RG))
88 DTWE=EXP(5.*DEPTH/WIDTH)-EXP(-5.*DEPTH/WIDTH)/(EXP(5.*DEPTH/ WIDTH)+EXP(-5.*DEPTH/ WIDTH))
89 DTWE=0.185*D*TWC/GRVN*ZE*COF
90 83 TE=TSAT-DTGE-DTWE
91 TC=TSAT-2.*DTGE-DTGA-DTGC-DTWC
92 DTEC=DTGE+DTGA+DTGC+DTWC+DTWE
93 22 FORMAT (/69H
TSAT QMAA TE TC
1 DT)
94 IF (MQ-1) 92,91,92
95 23 FORMAT (/53H Q TE TC DT)
96 91 WRITE (6,221)
97 WRITE (6,15) TSAT, Q, TE, TC, DTEC
98 WRITE (6,23)
99 Q=01-DQ
GO TO 93
91 WRITE (6,14) 2, TE, TC, DTEC
93 CONTINUE
94 CONTINUE
95 CONTINUE
96 CALL EXIT
97 STOP
98 END