Status Report on Theory and Experiments on Heat Pipes at Los Alamos -

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Abstract

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A survey of the experiments on the characteristics of heat pipes is presented. The status of corrosion studies at elevated temperatures is reviewed. A quantitative engineering theory for the design and performance analysis of heat pipes is given.

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Introduction

The "heat pipe" [1,2] is a self-contained engineering structure which exhibits a thermal conductance greatly in excess of that which could be obtained by the use of a homogeneous piece of any known metal. This property is achieved within the containing envelope by the evaporation of a liquid, transport of the vapor to another part of the container, condensation of the vapor and return of the condensate to the evaporator through a wick of suitable capillary structure.

There are obviously many practical uses for a structure of extraordinarily large thermal conductance. The heat pipe principle is indeed applicable over a very wide range of sizes, shapes, temperatures and materials. Unlike solid heat conductors, however, heat pipes cannot be characterized by a single property (an "equivalent thermal conductivity," say), since the behavior and limitations of a heat pipe are largely integral properties of the device as a whole. Furthermore, even if the size, shape, temperature and materials of a heat pipe are specified, the mass, vapor volume fraction, thermal conductance and maximum heat flux are individually (though not independently) under the control of the designer. The particular application will determine which allowed combination of these properties is most desirable.

At the present time, certain quantitative features of heat pipe behavior have not in fact been verified experimentally, though they can be predicted with some confidence. A few properties cannot yet even be treated with any conviction. The analysis presented here is an abridgement of LA-3246-MS, "Theory of Heat Pipes", and is intended simply to offer some initial orientation in the quantitative principles of heat pipes, and to serve as a stimulus for further experimentation, applications, and improvement of the theory.

Theory

The advantages of heat pipes are best realized when they are long and thin, that is, take the form of long cylinders or extended thin planar structures. For definiteness the discussion here will be confined to right circular cylinders of large length-to-diameter ratio. The course of analysis for other shapes will be evident, though not always straight-forward in detail. As shown in Fig. 1, such a heat pipe consists of a containing tube of length l with outer radius r_p , an annular capillary structure saturated with a wetting liquid, with outer radius r_w , and a vapor space of radius r_v .

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Since heat is added to and removed from the heat pipe through the container wall by ordinary thermal conduction, this should be as thin as other considerations permit, in order to minimize radial temperature differences. The container wall must of course sustain the difference between the internal and the ambient pressure. Heat pipes become effective at internal vapor pressures as low as a hundredth of an atmosphere, and improve with increasing pressure. It will ordinarily be possible nearly to match the ambient pressure by choice of a working fluid with an appropriate vapor pressure at the desired operating temperature.

The details of the wick admit of wide variation. It need not be disposed against the inside surface of the container as shown, though this will ordinarily be the best place for it for several reasons. Since evaporation and condensation take place at the vapor-liquid interface, this disposition of the wick allows the necessary radial heat transfer to occur through the medium of highest thermal conductance and thus minimizes radial temperature differences. This also makes the hydraulic diameter of the vapor space as large as possible which minimizes axial pressure gradients in the flowing vapor. The wick may be a woven cloth, roving, felt, sinter, etc., or even simply slots or grooves in the container wall [3]. The capillary structure will be characterized by its mean pore radius, permeability and liquid volume fraction.

The working fluid must wet the wick material; that is, the contact angle (the angle formed by a wedge of liquid in equilibrium contact with the solid substrate) must be less than $\pi/2$. While not essential, it is desirable that the fluid also wet the container wall as this improves the heat transfer. There is little penalty for a modest excess of liquid over the amount required to saturate the wick. A deficiency on the other hand can be expected to reduce the maximum heat transport by reducing the effective wick volume in the evaporator section of the heat pipe.

Suppose, first, that there is no heat addition or removal and that the pipe is at equilibrium with its length, z , at an angle ϕ to a gravitational field of acceleration, g . The pressure distribution in the liquid phase, $p_l(z)$, obeys the usual hydrostatic law for an incompressible fluid:

$$p_l(z) = p_l(0) + \rho_l g \sin\phi \quad (1)$$

where ρ_l is the density of the liquid. The pressure in the vapor phase $p_v(z)$, assumed an ideal gas, has a Boltzmann distribution in the gravitational field, but the variation of pressure is entirely negligible and we may take the pressure to be constant. The interface between the liquid in the capillary structure and the adjacent vapor must assume a local mean radius of curvature, $r(z)$, so that surface tension, γ , supports the difference in the pressure between the liquid and the vapor. Thus

$$p_v(z) - p_l(z) = \frac{2\gamma}{r(z)} \quad (2)$$

The vapor pressure of the liquid depends not only on the temperature T , but also somewhat on the radius of curvature of the liquid-vapor interface. For capillary pore sizes of practical interest, however, we may neglect this dependence. In the present

equilibrium case the vapor pressure of the liquid, $p(T)$, must be equal to the pressure in the adjacent vapor, $p_v(z)$.

In a capillary structure of minimum pore radius, r_c , containing a liquid for which the contact angle is θ , the smallest radius of curvature that the meniscus can achieve is $r \sec \theta$. The liquid-vapor interface under some circumstances may be at the surface of the capillary structure or even outside it, so the maximum radius of curvature may be at least as large as the radius of the vapor space. If enough liquid is present to form a pool in which the gravitational force dominates surface tension, then the radius of curvature of the interface may be essentially infinite. Thus r must lie in the range: $\infty \geq r \geq r_c \sec \theta$. Using these limiting values in (2) and combining with (1) yields for the maximum height of capillary rise, $z_{\max} \sin \phi$, the well-known result

$$z_{\max} \sin \phi = \frac{2\gamma \cos \theta}{\rho_l g r_c} \quad (3)$$

In order to work properly the length of the heat pipe should not exceed this $z_{\max}$.

The quantitative details of the start-up transient are of minor interest. We need describe only qualitatively how it is accomplished quite automatically.

Beginning with the equilibrium condition for which $r(z) > r_c \sec \theta$, we consider then what happens when heat is added to the pipe in the evaporator section, $0 \leq z \leq l$ and removed in the condenser section, $l \leq z \leq L$. The temperature, and consequently the vapor pressure, of the liquid rises in the evaporator and falls in the condenser. This gives rise to a pressure difference in the vapor which drives it from the evaporator to the condenser. The driving pressure difference in the vapor is somewhat less than the difference of the liquid vapor pressures, since, in order to maintain continued evaporation, the vapor pressure of the liquid in the evaporator must exceed the pressure in the adjacent vapor. Similarly, in order to continue condensing, the pressure in the condenser vapor must exceed the vapor pressure of the adjacent liquid. As a result of evaporation the liquid-vapor interface in the evaporator recedes somewhat into the capillary structure and the radius of curvature of the meniscus consequently decreases there. Condensation of vapor increases the radius of curvature of the meniscus in the condenser, if it is not already essentially infinite. Thus, according to (2), the pressure distribution in the liquid changes in the direction which drives liquid from the condenser to the evaporator. The resulting distribution of pressures is shown in Fig. 2.

We consider now the quantitative dynamics of the long cylindrical heat pipe in steady-state operation.

Conservation of matter requires that the total axial flow of vapor, $\dot{m}_v$, and of liquid, $\dot{m}_l$, at each axial position z satisfy

$$\dot{m}_v(z) + \dot{m}_l(z) = 0 \quad (4)$$

The pressure distribution in the liquid phase is determined by the gravitational and viscous forces. For long thin pipes, that is as

long as $r_v \ell \gg r_w^2$, the radial pressure gradient will be negligible, and we may assume that both the flow velocity and pressure in the liquid depend only on z . As an approximation valid for the flow of liquid through the wick we use the following version of Darcy's law:

$$\frac{dp_\ell}{dz} = \rho_\ell g \sin\phi - \frac{b \eta_\ell \dot{m}(z)}{\pi (r_w^2 - r_v^2) \rho_\ell e r_c^2} \quad (5)$$

Here η_ℓ is the liquid viscosity, e the fraction of wick volume occupied by liquid, and b is a dimensionless constant depending on the detailed geometry of the capillary structure. For nonconnected parallel cylindrical pores, $b = 8$. For realistic capillary structures with tortuous and interconnected pores, b is approximately 10-20.

The dynamics of the vapor flow is decidedly more complex, partly because in general an equation like (5) relating the local pressure gradient with the local mass flow does not even exist, and partly because the inertial contribution to the pressure gradient is often not negligible in cases of interest. The vapor flow in the evaporator and condenser of a heat pipe is dynamically identical to pipe flow with injection or suction through a porous wall. This problem has been studied by Yuan and Finkelstein [4] for cylindrical pipes, and by Knight and McInteer [5] for flow between plane parallel walls. We summarize and quote the partial results of these authors.

Both analyses assume incompressible laminar flow and uniform injection or suction. Several regimes must be distinguished, depending on the magnitude of a Reynolds number, R_r , based on the radial flow velocity at the channel wall, v_r ; the channel radius, r_v ; the vapor density, ρ_v ; and viscosity, η_v :

$$R_r = - \frac{\rho_v r_v v_r}{\eta_v} = \frac{1}{2\pi\eta_v} \frac{d\dot{m}_v}{dz} \quad (6)$$

Note that R_r is positive for evaporation and negative for condensation. For all values of R_r solutions are found for which the axial velocity profiles are symmetric about the channel axis, with the profiles at different axial stations differing only by a velocity scale factor proportional to the distance from the axial origin of the flow. For $|R_r| \ll 1$, viscous effects dominate and the axial velocity profile is close to the usual parabolic shape for Poiseuille flow. The pressure decreases in the direction of flow, with a gradient larger than that of Poiseuille flow in the case of evaporation, and smaller if vapor is condensing. In this regime the flow properties can be calculated by a straightforward perturbation expansion in powers of the Reynolds number. The resulting pressure gradient is given approximately by [4].

$$\frac{dp_v}{dz} = - \frac{8\eta_v \dot{m}_v}{\pi \rho_v r_v^4} \left(1 + \frac{3}{4} R_r - \frac{11}{27} R_r^2 + \dots \right) \quad (7)$$

This expression is derived on the assumption that R_r as defined in (6) is a constant, independent of z . This will often be the case in practical application of heat pipes.

When $|R_r|$ is large the evaporation and condensation cases become qualitatively different. Knight and McInteer [5] show this in theory for flow between plane parallel walls. Wageman and Guevara [6] have verified the following description experimentally for cylindrical pipe flow. For high evaporation rates, $R_r \gg 1$, the radial dependence of the velocity is not parabolic but is proportional to $\cos \pi/2 (r/r_v)^2$. The pressure decreases in the direction of flow. The flow properties can be calculated by a perturbation expansion in powers of $1/R_r$. With high condensation rates on the other hand, the flow is of boundary layer type. The axial velocity is constant across most of the channel, with the transition to zero velocity occurring in a thin layer at the wall. The pressure increases in the direction of fluid motion as a consequence of partial dynamic recovery in the decelerating flow. In this regime only the limiting behavior can at present be described analytically, as perturbation expansions cannot be made self-consistently. In either limit, $|R_r| \rightarrow \infty$, the pressure gradient is given by [5]

$$\frac{dp_v}{dz} = - \frac{s \dot{m}_v}{4 \rho_v r_v^4} \frac{d\dot{m}_v}{dz} \quad (8)$$

where the difference in the flows enters only in the numerical coefficients: for evaporation, $s = 1$; for condensation, $s = 4/\pi^2$. As before, (8) is strictly correct only for constant R_r .

Nothing appears to be known about the stability of these flows, so that nothing definite can be said about the onset of turbulence.

To complete the discussion of the flow dynamics we must state the connections between the vapor and liquid pressures and the vapor and liquid mass flows. As in the equilibrium case the interface meniscus assumes a radius of curvature satisfying (2), except that in the steady-state p_v also depends on z . The two mass flows are coupled with the liquid temperature at the interface, $T(z, r_v)$, which in turn determines the vapor pressure of the liquid, p . The local condensation rate $d\dot{m}_l/dz$ is given by the gas kinetic formula

$$\frac{d\dot{m}_l}{dz} = - \frac{d\dot{m}_v}{dz} = \frac{\alpha r_v (p_v - p)}{\sqrt{RT/2\pi M}} \quad (9)$$

The numerical factor $\alpha \approx 1$ includes both the probability of condensation of an impinging vapor molecule and the "roughness" of the meniscus interface formed on the capillary structure. Equation (9) also applies for surface evaporation, but not for boiling evaporation, i.e., the formation of vapor bubbles within the capillary structure.

We now discuss the transport of energy. The device may be said to be operating in the heat pipe regime when, though the heat flow may be very large, the axial and radial temperature gradients throughout are very small, excepting only the radial temperature gradient in the container wall and wick. In this condition it may readily be shown that the transport of energy is essentially entirely accomplished by radial conduction through the container wall and wick, and the axial convection of latent heat of condensation. The total axial heat flux is then simply

$$Q(z) = L \dot{m}_v(z) \quad (10)$$

where L is the heat of evaporation of the working fluid.

The heat pipe is coupled to the external environment through the net rate of heat addition per unit length of pipe, $H = H(z, T_p, Q)$. As indicated, H may depend on: z explicitly, when heat is added P with a known distribution, as might be the case with electron bombardment or induction heating of the pipe surface; $T_p = T(z, r_p)$, the temperature of the external surface of the pipe, as in the case of radiation or conduction to a reservoir of specified temperature; and $Q = Q(z)$, as in the case of heating or cooling by means of parallel forced convection by an external fluid. In any case a prescription of the environment determines H as a known function of its arguments. We then have

$$\frac{dQ(z)}{dz} = H(z, T_p, Q). \quad (11)$$

The radial heat flux through the container wall and wick to the vapor-liquid interface in the heat pipe regime, H , also determines the difference between the temperature at the outside of the container, T_p , and the temperature at the vapor-liquid interface by the standard P result for radial thermal conduction in a composite cylinder:

$$T_p = T_v + H/K \quad (12)$$

where K is the integral radial conductance of the cylinder.

Rather accurate approximate solutions of the foregoing system of equations can be obtained fairly simply. In the heat pipe regime the temperature is nearly uniform throughout the whole vapor space and the distributions of axial heat and mass flows differ little from what they would be if the vapor temperature was exactly a constant, T_v . If (12) is solved for T_p , we may express H in (11) as a function of z , T_v and Q .

$$\frac{dQ}{dz} = H(z, T_v, Q) \quad (13)$$

The heat flows through the two ends of the pipe either are negligible or at worst may be known functions of the local vapor temperature, which we denote by $F_o(T_v)$ and $F_l(T_v)$ respectively. The effective average temperature, T_o , and its associated axial heat flux distribution, $Q_o(z)$, are then obtained as the solution of

$$\frac{dQ_o}{dz} = H(z, T_o, Q_o) \quad (14)$$

$$Q_o(0) = F_o(T_o); \quad Q_o(L) = F_l(T_o)$$

Since this is a two-point boundary value problem on a first order differential equation it can in general only be satisfied for particular values of T_o . In a physically well defined problem $H(z, T_o, Q_o)$ will depend explicitly on T_o , and the value of T_o satisfying (14) will be unique.

With this good approximation to the heat flux we may obtain the vapor and liquid mass flows from (10) and (4)

$$\dot{m}_v(z) = -\dot{m}_l(z) = Q_0(z)/L(T_0) \quad (15)$$

The vapor mass flow in turn determines the distribution of pressure in the vapor, to within a constant, by integration of (7) or (8) as appropriate. Using $\dot{m}_v(z)$ and $p_v(z)$ in (9) then determines the vapor pressure of the liquid to within an additive constant. Consistent with the accuracy of the calculation, this constant may be taken as $p(T_0)$. Since the vapor pressure is a known function of the liquid surface temperature, this determines $T_v(z)$. The self-consistency of the approximate calculation is verified if the total variation of $T_v(z)$, found in this way, is small compared to T_0 . Finally, the liquid mass flow determines the axial distribution of pressure in the liquid by integration of (5). Throughout the foregoing all the temperature dependent properties, with the exception of the vapor pressure, are sufficiently slowly varying that they may be taken as constants evaluated at T_0 .

As an illustration of the magnitudes of pressure and temperature drops typically encountered, we cite an experimental horizontal liquid sodium heat pipe which was reported in our original description of these devices [2]. The relevant specifications are given in the left column and various calculated properties in the right:

$Q(\ell_e) = 500$ watts	$\dot{m}_v(\ell_e) = 0.1$ gm/sec
$T_0 = 920^\circ\text{K}$	$p(T_0) = 50$ mm Hg
$\ell = 90$ cm	$\Delta p_v = -0.2$ mm Hg
$\ell_e = 13$ cm	$\Delta p = -0.5$ mm Hg
$r_v = 0.64$ cm	$\Delta p_\ell \cong 2$ mm Hg
$r_w = 0.80$ cm	$\Delta T_v = -0.7^\circ\text{K}$
$r_c = 0.012$ cm	

The main features of a working heat pipe are evident here. The transport of considerable heat is accomplished by the circulation of a small amount of working fluid. This circulation requires but small pressure differences, and the accompanying temperature difference is so small that its precise magnitude is not of importance in practical applications.

While the thermal conductance of a heat pipe is very large, there are, however, limitations on the magnitudes of both the total and local energy fluxes.

The total axial heat transport may increase only if the force of capillary origin can sustain the required circulation of fluid. As previously noted, the maximum difference in pressure between vapor and adjacent liquid that can be supported by surface tension in the capillary structure is $(2\gamma\cos\theta)/r_c$. It is therefore necessary that

$$p_v(z) - p_\ell(z) \leq \frac{2\gamma\cos\theta}{r_c} ; 0 \leq z \leq \ell \quad (16)$$

In general, the largest pressure difference will occur at the beginning of the evaporator section $z = 0$. If the heat pipe has been prepared with a fully saturated wick, the vapor-liquid interface meniscus will have large radius of curvature at the terminal end of

the condenser and, therefore, $p_v(\ell) \approx p_\ell(\ell)$. Thus if (16) is satisfied at $z = 0$, it will be satisfied for all z .

There is a further limitation on the local radial heat flux in the evaporator section of the heat pipe. The liquid in the interior of the wick here is necessarily superheated. We should therefore expect that the limitation will be closely connected with the conditions for the onset of boiling in the capillary structure and the quantitative properties of the subsequent evaporation and heat transfer. The problems here are more complex even than those encountered in pool boiling of liquids, for in addition to the local limitations of boiling heat transfer, the undoubtedly deleterious interaction of boiling with the overall circulation of liquid throughout the capillary structure will be important. There is no generally useful experimental information available yet on this problem.

The steady-state behavior of a heat pipe in which a mixture of liquids rather than a single pure compound is used as the working fluid is of some interest. It may readily be shown that nearly complete fractionation of the components is to be expected. The steady-state consists of a series of segments each containing a pure component which is refluxing as an independent heat pipe. The components are arranged in order of increasing volatility with the most volatile at the terminal end of the condenser. The temperature distribution forms a series of plateaus, with the plateau temperatures decreasing in order of increasing volatility of the local component, in such a way that the pressure within the vapor is nearly constant throughout the entire pipe. Between the segments there are short transition zones of rapidly varying temperature within which there is no refluxing, the entire axial heat transport occurring by ordinary thermal conduction, mainly through the container wall and wick. Along each temperature transition zone the stagnant two-component liquid and vapor phases vary in equilibrium concentrations in a way consistent with constancy of total vapor pressure. Since the distribution of density is not uniform, gravitationally induced convection, particularly of the vapor phase, may modify this distribution considerably.

If the heat pipe contains some non-condensable gas of low solubility in the working fluid, the foregoing applies equally well. It has been demonstrated [2] that in the steady-state the gas is driven to the terminal end of the condenser where it forms a stagnant zone. The length of this zone is proportional to the mass of gas and to the mean temperature in the zone, and is inversely proportional to the pressure of the vapor in the refluxing section of the heat pipe. This feature might be useful. In such a heat pipe the working length of the condenser increases as the temperature is increased.

A heat pipe will commonly be required to transport the largest possible amount of heat, subject to whatever subsidiary constraints arise in the particular application. In this case the maximum heat flux criterion (16) applies, in the form

$$\Delta p_\ell - \Delta p_v - \frac{2\gamma \cos \theta}{r_c} = 0 \quad (19)$$

If the subsidiary constraints do not involve the capillary pore size, r_c , then its optimum value may be deduced from (19) alone. According to (5) the viscous contribution to Δp_ℓ is inversely proportional to

r_c^2 . The standard extremizing procedure then yields the result that r_c should be chosen so that the viscous contribution to Δp_l is one-half the magnitude of the capillary pressure term, $(2\gamma \cos \theta)/r_c$. If the hydrostatic contribution to Δp_l is absent, and if the ratio r_v/r_w is not constrained by the subsidiary conditions, then the optimum value of r_v/r_w is $\sqrt{2/3}$. This follows by noting that if Δp_v is obtained either from (7) or (8) as appropriate, and the optimal value of r_c is used for Δp_l in (19), then the greatest heat transport corresponds to the maximum value of $r_v^4 (r_w^2 - r_v^2)$. A wick of uniform thickness is not optimal, but is not much inferior to the best distribution of wick thickness. If, for example, a fixed total volume of wick is distributed in a uniformly heated evaporator, so as to minimize the liquid viscous pressure drop in this section of the pipe, the cross-sectional area of wick should be proportional to $\sqrt{z}$, and the pressure drop is 8/9 of that of the wick of constant thickness.

Experimental Summary

The experimental investigations of the heat pipe at the LASL have been divided into three temperature regimes which reflect the application areas of interest. The lowest temperature regime is of little concern for thermionic applications. Water and the alcohols are used as the working fluids, and control of satellite component temperatures is one objective. The next temperature regime is from 700 to 1400°K for heat removal from the collectors of thermionic converters. While the heat transfer rates are not excessive in most applications, the lifetime requirement is long. Therefore life testing of suitable material combinations is of importance. The high temperature regime of 1400 to 2100°K presents the most challenging area of heat pipe application. The capability to move heat in this range of temperatures would allow a measure of freedom of design in thermionic cells heretofore impossible to consider.

Several liquid metals are being investigated for use in heat pipes at high temperatures. The purpose of this investigation is to find compatible liquid-container combinations for long-term operation over a wide temperature range.

Compatibility must include consideration of mass transfer in a liquid-vapor cycle such as exists in an operating heat pipe. Tests were first made in simple isothermal capsules. These were made with silver, lead and indium in tantalum capsules. After 1000 hours at 1800°C, the capsules were opened. No indications of interaction were obtained but the tests available at that time for determination of the tantalum solubility in these liquid metals were not sufficiently sensitive to detect less than 100 ppm. Since this is far greater than can be tolerated, static compatibility testing was discontinued in favor of circulating two-phase flow test capsules.

The construction features of the heat pipes used in this investigation have been described in previous publications [2,7]. Thin-walled container tubes are internally lined with one or more layers of fine-mesh screen. This screen surrounds a vapor passage and provides the capillary return for liquid from the condenser to the evaporator. After addition of enough liquid to saturate the capillary structure, the container tube is evacuated and sealed by electron-beam welding.

Test heat pipe sizes range from 1 to 2 cm in diameter and from 10 to 43 cm in length. The capillary structure is formed from 100-mesh screen with 0.003" diameter wire.

A completed heat pipe assembly is first heated uniformly to assure filling and wetting of the capillary structure by the liquid metal. Heat is then applied locally through the container wall near one end of the tube either by induction or electron-bombardment heating. The heat pipes are operated in a vacuum chamber and heat is rejected from the condenser by radiation. Heat removal rates were verified by measuring power input from a focused electron-beam filament. The mass flow of vapor and liquid could then be calculated using latent heat of vaporization data.

Lithium is an ideal heat pipe fluid because of its high latent heat of vaporization and its high surface tension. Lithium has been operated in Nb-1 w/o Zr tubes lined with Nb-1 w/o Zr screen at temperatures from 900°C to 1350°C. One pipe has operated for over 4000 hours at 1100°C without a detectable change in performance. The total mass of lithium circulated during this period was over 720 kgm. Another example has over 2000 hours of operation but it will be several months before the tests are completed and detailed examination is possible.

Lithium has been operated in a tantalum - 5 w/o tungsten-yttrium alloy pipe for a few hours at temperatures as high as 1875°C. Some bulging of the half-millimeter thick wall occurred because of the internal pressure.

Tantalum and tantalum alloys are used as heat pipe container tubes at temperatures above 1500°C. Fine-wire tantalum screen lines the inside of these tubes and provides the capillary return for the circulating fluid. All of these pipes, when operated with liquid silver, indium or lead, show a tantalum mass transfer pattern peculiar to a two-phase circulating system. The screen is dissolved in the condenser by freshly purified liquid. This liquid is returned to the evaporator and repurified by vaporization. The dissolved tantalum remains in the evaporator and eventually fills the capillary pores. With the pores closed, the wall of the evaporator overheats and localized melting and leakage occurs.

Examination of a sectioned heat pipe after operation with silver for 100 hours at 1900°C has been made. The total mass of silver circulated during this period was 200 kgm. Although it was difficult to determine the exact amount of tantalum transferred, estimates based on three pipes operated at 1900°C give mass transport rates between 2 and 4×10^{-9} grams per joule of heat transferred. These rates indicate a tantalum solubility in silver on the order of 10 ppm.

In one test, a tungsten screen was substituted for the tantalum screen but containment was in a tantalum tube. This tungsten screen appeared unchanged after circulation of 200 kgm of silver at 1900°C. Some deposit was found in the evaporator section, but it appears that this was dissolved tantalum from the container tube wall. We are obtaining vapor-deposited tungsten heat pipes having many axial channels integral with the wall to provide the necessary capillary flow. It is planned to use these pipes to determine the compatibility of silver, indium and lead with tungsten.

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Figure 1. Cylindrical Heat Pipe Structure

Figure 2. Distribution of Pressures in a Heat Pipe

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