JOURNAL OF THERMOPHYSICS AND HEAT TRANSFER
Vol. 8, No. 2, April–June 1994

Thermodynamic Aspects of Heat Pipe Operation

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In this article the general operation and performance of heat pipes is being approached from fundamental thermodynamic considerations. This is in contrast to the classic heat pipe design analysis which equates an available "capillary pressure" with the two pressure drops associated with the circulation of the working fluid in the vapor and liquid phase and body forces where applicable, and erroneously attributing the circulation of the working fluid in a heat pipe to "capillary pumping." This article shows that the working fluid circulates in a heat pipe as the result of a thermodynamic cycle in which thermal energy is converted to kinetic energy. The basic analysis which is presented, and which can be extended in future research, identifies the total internal temperature difference over which the heat pipe operates and the heat pipe operating temperature as the two key operating parameters in full agreement with all observed heat transfer phenomena. For given physical characteristics of a heat pipe, i.e., length, internal and external diameters, wick, artery or groove structure, the thermal transport performance below its operating limits is shown to be entirely a function of these two parameters as it is for any other heat conductor. The thermodynamic cycle clarifies the often substantial discrepancy between the predicted and the actually realized thermal power transfer capacity of a heat pipe.

Nomenclature

C_d = configuration constant, m^2
C_v = constant
E_v = vaporization energy, J/mol
$k(T)$ = heat transfer coefficient, W/m^2-K
m = flow rate, kg/s
P = thermal transfer power, W
p_b = body forces on the fluid, N/m^3
p_c = available capillary pressure, N/m^2
p_{v2} = vapor pressure at maximum internal operating pressure of the heat pipe, T_2, as found from Eq. (9), N/m^2
Q = total heat tansfer, W
R = gas constant, 8.3144 J/mole-K
r_c = effective pore radius, m
T = temperature, K
T_2 = internal evaporator temperature, K
ΔH_{fg} = latent heat of vaporization, J/kg
Δp_{lf} = liquid pressure drop, N/m^2
Δp_{vf} = vapor pressure drop, N/m^2
ΔT = temperature difference, K
σ = surface tension, N/m
ϕ = attachment angle, deg

Introduction

THE actual performance of heat pipes of many varied designs has failed to match the expected performance predictions which were based on the currently accepted heat pipe design relations as compiled in Refs. 1 and 2 and many other publications. For a considerable number of years heat pipes have been tested at Wright Laboratory. Many heat pipes that were provided by outside contractors performed short of their design specifications. Some devices actually underperformed by an order of magnitude. For many years an intensive and systematic research program was conducted on a double wall artery heat pipe, which was an in-house design. The operation of this heat pipe generated some of the most complete and valuable data which finally shed new light on the actual performance parameters of heat pipes. This copper/water heat pipe demonstrated a thermal transport capacity of 2.8 kW-m at an operating temperature of about 210°C, although its theoretical capacity as based on the standard heat pipe design theory should have reached a performance of 6.0 kW-m at 100°C (Ref. 3). However, the experimental data showed that in every respect the heat transfer of this piece of material followed the laws of heat transfer of all other thermal conductors, i.e.

$$Q = C_d k(T) \Delta T \tag{1}$$

A casual review of the heat pipe literature reveals that large discrepancies between predicted performances based on the capillary pumping design analysis and experimental performances are no rare occurrences. Even the "monogroove" heat pipe as shown in Fig. 1 apparently carries only about 50% of its theoretically predicted heat load.[4] Other designs, such as the "cat's eye" artery concept (Fig. 1), failed to match theoretical predictions by over an order of magnitude. It was therefore necessary to arbitrarily multiply theoretically predicted thermal transport capacities by a factor that is considerably less than unity when sizing heat pipes for spacecraft applications.[5] A study of the initial failure to remove thermal waste heat by the 0-g Space Station heat pipe advanced radiator element (SHARE)[6] during a 1990 space flight experiment, and the subsequent modifications in the heat pipe design, as well as in the 0-g operation,[7] can illustrate clearly that currently applied heat pipe design techniques do not correctly predict either the performance or the operation of these high technology arterial heat pipes.

The theory of heat pipe design as presented in Refs. 1 and 2, and many other publications, bases the design of a heat pipe principally on the stipulation that the combined pressure drops of the vapor and the liquid while circulating in the heat

Presented as Paper 90-1772 at the AIAA/ASME 5th Joint Thermophysics and Heat Transfer Conference, Seattle, WA, June 18–20, 1990; received Oct. 18, 1991; revision received Aug. 3, 1992; accepted for publication July 28, 1993. This paper is declared a work of the U.S. Government and is not subject to copyright protection in the United States.

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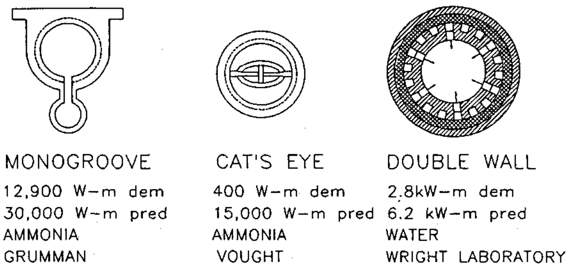

Fig. 1 Heat pipe designs.

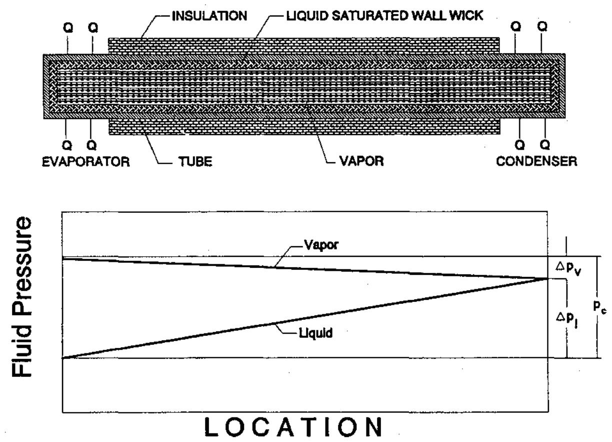

Fig. 2 Generic heat pipe operation.

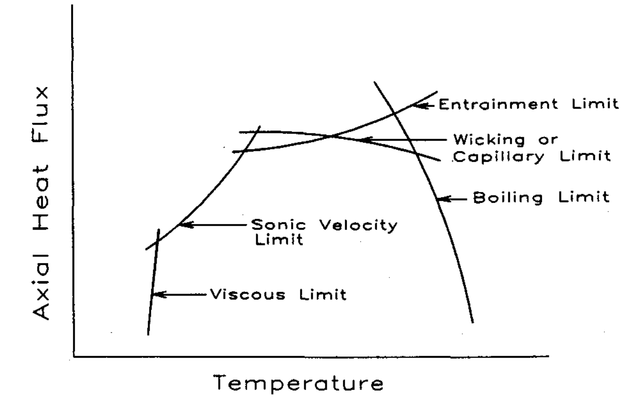

Fig. 3 Operating limits of heat pipes.

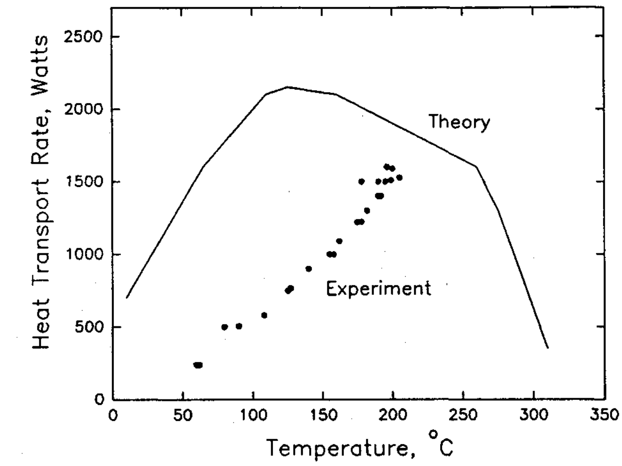

Fig. 4 Experimentally determined heat transport capacity of a heat pipe.

pipe and the pressure associated with body forces are equal to the available capillary pressure which is produced in a porous structure. The operating temperature enters into the analysis only for the determination of the temperature-dependent property values of the working fluid. No consideration is given to temperature differences at all. The circulation of the working fluid is attributed to capillary pumping. Special variations of heat pipes are actually called "capillary pumped loops."

Thus, the fundamental heat pipe design relation is expressed by

$$p_c = p_b + \Delta p_{lf} + \Delta p_{vf} \tag{2}$$

The available capillary pressure is assumed to be equal to the pressure as determined by the Laplace relation

$$p_c = (2\sigma/r_c)\cos\phi \tag{3}$$

Furthermore, this heat pipe design analysis must postulate that at the same axial location along a heat pipe the pressure of the liquid that returns in the wick, arteries, or grooves is less than the pressure of the vapor as shown in Fig. 2.

While the capillary pressure is accepted as the driving force for the circulation of the operating fluid, it is equally considered an operating limit with respect to the total pressure drop. No relations are presented for predicting heat pipe operation below this capillary pressure limit. Four additional limits for the operation of a heat pipe have been recognized that can cause a heat pipe to operate in certain temperature regimes below its thermal transport capability as determined by the capillary pressure. The four limits that are shown in Fig. 3—as taken from Ref. 2—are given as viscous limit, sonic velocity limit, entrainment limit, and boiling limit. Although these limits are based on valid physical phenomena, experimental data have rarely fallen on any of these limits. Typically, measured thermal transport rates are found located as indicated in Fig. 4, which was taken from Ref. 8. It is the objective of this article to show that the operation of the heat pipe within those limits can be determined by the thermodynamic aspect of heat pipe operation.

The thermal transport limit of a heat pipe is manifested by the "dry-out" of the evaporator. Dry-out of the evaporator is the final, but not necessarily the immediate, manifestation of working fluid returning to the evaporator in the liquid state at a lower rate than leaving in the vapor state. The delay between the onset of the imbalance between working fluid flowing into and out of the evaporator and the failure of the heat pipe is determined by the amount of fluid that can be contained in the evaporator. Since failure of a heat pipe operation will not coincide instantaneously with a change in operating conditions, a correlation between failure and the detrimental operating condition is illusive and almost impossible to establish.

The accepted heat pipe operating model that postulates the capillary pressure of a porous structure to be the driving force for circulating the working fluid in a heat pipe has failed to predict the performance of heat pipes correctly regardless of their design as shown in Fig. 1. Artery heat pipes that were sized to carry relatively large amounts of thermal power as needed in future thermal management space applications have especially fallen short of their expected performance. The observed shortcomings have stimulated this re-evaluation of the accepted heat pipe design theory.

Analysis

A heat pipe is designed to permit circulation of a working fluid in its vapor and liquid phase for the transport of thermal energy in the form of latent heat of vaporization over a desired distance. Three main processes must occur internally of the heat pipe: 1) evaporation, 2) mass flow in the vapor and liquid state, and 3) condensation of the working fluid. In every aspect of engineering it is accepted that fluid circulates only if and when potential energy is converted to kinetic energy. Since heat is the only available potential energy for pumping

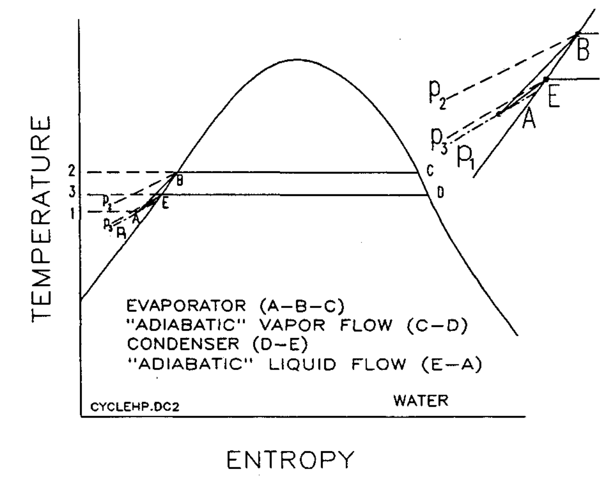

Fig. 5 Basic thermodynamic cycle in a heat pipe.

the working fluid in a heat pipe, a thermodynamic cycle must be an integral aspect of the operation of a heat pipe, a consideration that is entirely absent from the current heat pipe design theory. The current heat pipe design theory solely stipulates that the vapor and the liquid flow are the result of "capillary suction" of a liquid film stretched over a porous structure that can be a wick, groove, or artery. The suction pressure is assumed to be equal to the capillary pressure as determined by the effective pore size and the surface tension of the fluid.

In reality, the circulation of the working fluid in a heat pipe is the consequence of a pumping process that results from the conversion of potential thermal energy into kinetic energy in a well-defined thermodynamic cycle. The fluid membrane stretched across the porous structure solely facilitates this pumping process by separating the liquid from the vapor as needed for the flow in the liquid and the vapor flow passages. For the thermodynamic cycle to produce the pumping power commensurate with the product of the flow rate and the total pressure drop of the vapor and liquid within the confines of the heat pipe, a specific minimum temperature difference has to exist at a given operating temperature.

The most basic thermodynamic cycle that can support heat pipe operation is shown in Fig. 5. The working fluid enters the evaporator at a temperature T_1 and is raised to the upper operating temperature T_2 along the path A-B. It should be clear that the maximum internal temperature T_2 is lower than the externally applied and measured temperature of a heat pipe evaporator. In the detailed analysis of a specific heat pipe which starts with the basic thermodynamic cycle analysis presented in this article the temperature drops at the evaporator as well as at the condenser across the heat pipe structures have to be accounted for. Evaporation and expansion from the liquid volume to the vapor volume occurs along path B-C. The vapor flows under the influence of the pressure difference between the evaporator and the condenser through the "adiabatic" section of the heat pipe along path C-D. The vapor condenses in the condenser section along path D-E. The lowest temperature measured on the outer surface of a heat pipe must naturally be lower than the lowest internal temperature that establishes the thermodynamic cycle because of the temperature drop across the heat pipe structure at the condenser. Finally, the liquid working fluid returns to the condenser along path E-A. The cycle shown in Fig. 5 is the most basic thermodynamic cycle and is subject to variations as imposed by the hardware design and the application like any other thermodynamic cycle. The location of path C-D depends on the radial heat fluxes that might occur along the assumed adiabatic section. The pressure difference

$$\Delta p_{vf} = p_2 - p_3 \tag{4}$$

is the result of a temperature difference between the evaporator and the condenser

$$\Delta T = T_2 - T_3 \tag{5}$$

The pressure drop of the liquid returning to the evaporator in its flow passage is

$$\Delta p_{lf} = p_3 - p_1 \tag{6}$$

Several processes are associated with the flow of the liquid: 1) friction, 2) conduction of heat into the liquid from the adjacent vapor, 3) heat conduction from the evaporator structure to the incoming liquid, and 4) the heat conduction from the liquid to the heat pipe wall and from the wall to the environment. Processes 1, 2, and 3 cause a rise in the temperature of the liquid unless rejection of heat in process 4 is possible. Effects which increase the entropy in the liquid will force the liquid towards the two phase region unless the liquid is cooled.

The capillary pressure p_c, Eq. (3), enters into the heat pipe operation solely, but importantly, in that it has to be equal to or larger than the sum of the pressure drops associated with the liquid and vapor flow so that the required pressure difference between the liquid and vapor phase

$$\Delta p_f = p_2 - p_1 \tag{7}$$

as shown in Fig. 2 can be maintained. The enclosed area of the T-S diagram represents naturally the work performed on the working fluid necessary for its circulation in the heat pipe. It should be clear that the required work for pumping the working fluid around the entire cycle is not solely a function of the total heat transfer, but also of the physical design of the heat pipe, i.e., its length, bends, internal and external diameters, wick, artery or groove structure, and whatever other aspects cause pressure drops in fluid circulation. These physical parameters determine the value of C_d of Eq. (1) for a heat pipe.

The discussion of the thermodynamic cycle clearly indicates that with the circulation of the working fluid, pressure drops will occur as has been the primary consideration of the currently accepted heat pipe design theory. But pressure differences can only be established by compatible temperature differences that have to be allowed to exist in the heat pipe. The work performed on the working fluid in the form of pumping energy is directly a function of the temperature difference $T_2 - T_3$ that exists inside the heat pipe as can be seen from Fig. 5. Like in any other thermodynamic cycle, conversion of thermal energy to kinetic energy is associated with heat rejection at a temperature below the upper temperature of the cycle, and occurs with a conversion efficiency of less than 100%.

From the above thermodynamic considerations it becomes clear that a heat pipe cannot operate at a uniform temperature. The higher the thermal transport, the larger the internal temperature difference has to be between the evaporator and the condenser at a given operating temperature in order to achieve the fluid circulation that is commensurate with the desired thermal transfer power as given by

$$P = m \times \Delta H_{fg} \tag{8}$$

With the understanding of the interrelation between total pressure drop and temperature difference in the heat pipe, the minimum temperature difference can be determined that has to be maintained for a given pressure difference. Figure 6 shows the correlation between saturation temperature and vapor pressure for water. The vapor pressure is related to the saturation temperature by an exponential function (Clausius-Clapeyron Equation)

$$p_v = C_v \times \exp(-E_v/RT) \tag{9}$$

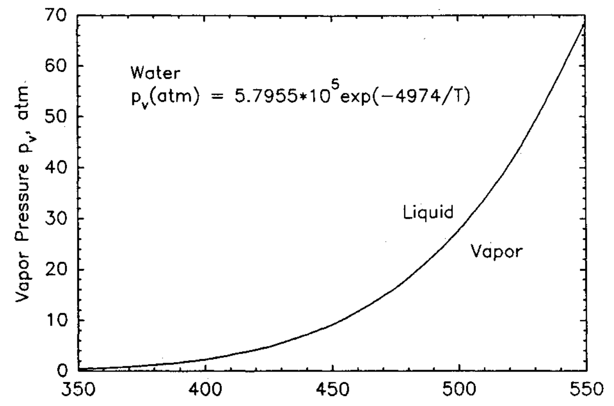

Fig. 6 Vapor pressure of water.

By differentiation of Eq. (9) it is easily seen that a vapor pressure difference Δp proves to be a function of ΔT as well as the absolute temperature T:

$$\Delta p_v = C_v \times (E_v/RT^2)\exp(-E_v/RT) \times \Delta T \qquad (10)$$

The ΔT required for providing a desired vapor pressure difference Δp_v can therefore be closely approximated for small differences by the relation

$$\Delta T = \Delta p_v \times (RT^2/E_v C_v) \times \exp(E_v/RT) \qquad (11)$$

For larger pressure differences the needed pressure difference Δp for the desired flow rate is

$$\Delta p = p_{v2} - p_{v1} \qquad (12)$$

and the actual temperature difference is found from

$$\Delta T = (E_v/R) \times [1/\ln (C_v/p_{v2}) - 1/\ln (C_v/p_{v1})] \qquad (13)$$

or

$$\Delta T = T_2 - (E_v/R)/\ln (C_v/p_{v1}) \qquad (14)$$

However, the pressure difference cannot be larger than that which can be tolerated across the fluid membrane that separates the liquid from the vapor in the evaporator. The pressure difference is limited by the capillary pressure, which is a function of the characteristics of the porous structure and the temperature dependent surface tension as determined by Eq. (3).

Equation (11) indicates that the higher the operating temperature, the lower the ΔT will be to achieve a required Δp. This relation between pressure difference and temperature difference is shown graphically in Figs. 7 and 8 for water. From the very beginning of heat pipe development it was observed that for the same working fluid the thermal transport capacity of a given heat pipe improved with increasing temperature. The test data which were presented in Ref. 3 and are reproduced in Fig. 4, substantiate this observation.

Instead of being a capillary pressure driven device, clearly a heat pipe operates under the influence of a thermodynamic process, the two principle parameters of the cycle being the total temperature difference and the operating temperature. The fluid membrane drawn across the pores of a porous structure or a small groove solely functions as a diode, preventing vapor from expanding into the liquid volume while permitting liquid to evaporate from its surface. The capillary pressure sets only an upper limit to the thermal transport capacity. It does not determine or influence the heat transfer rate below

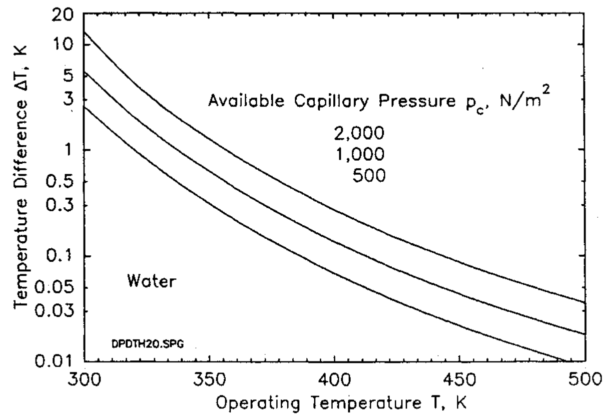

Fig. 7 Temperature difference requirement for water heat pipes to operate at their capillary limit.

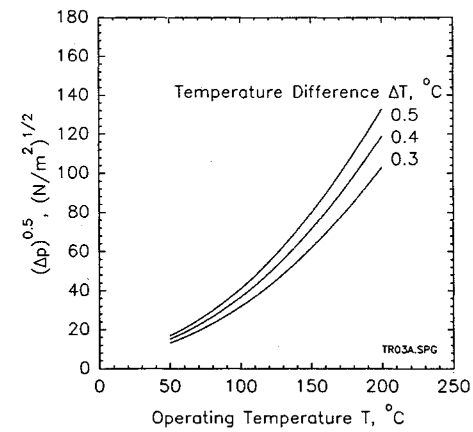

Fig. 8 Generated pressure difference as a function of operating temperature and total temperature difference.

this limit which is entirely a function of the thermodynamic parameters, operating temperature, and temperature difference.

Discussion

Mercury has been considered quite often as the working fluid for applications in the 440 K temperature regime, because its temperature-vapor pressure relationship appeared to be more favorable than water, i.e., water having an excessively high vapor pressure at that operating temperature. However, it has also been observed that a correctly filled mercury heat pipe, i.e., having no liquid working fluid in the vapor flow volume that permits the heat pipe to operate in the reflux mode in a 1g environment, would have to reach a minimum operating temperature before heat transport could be detected, and a considerably higher operating temperature than its design temperature, to come close to its design capacity. It can now easily be shown that the limitation on the lower operating temperature is imposed by the thermodynamic cycle. By the currently accepted heat pipe design theory that is based on capillary pumping, a 1.41-m-long mercury heat pipe with a 1.3-cm o.d. and a wall wick made of three layers of 90 Mesh wire cloth should transfer 52 W of thermal power when operating at 440 K, at which the vapor pressure would be 7.0×10^{-3} atm. Based on the surface tension of mercury at that operating temperature, a "suction pressure" of $p_c = 6435$ N/m^2 would be achieved. The design of this heat pipe satisfies fully the standard pressure balance as required by the accepted design theory, i.e., the combined pressure drops of the vapor and the fluid is equal to the suction pressure.

The correlation of vapor pressure and temperature for mercury is presented in Fig. 9. It shows that at an operating temperature of 440 K, a pressure difference of 6435 N/m^2

could be achieved only by a temperature difference considerably higher than the absolute operating temperature. This means that at an operating temperature of 440 K no thermodynamic cycle can be established to generate a sufficient pumping pressure differential for circulating the working fluid, mercury, at a rate commensurate with the predicted heat transport of 52 W. Figure 9 illustrates that to achieve the design heat transfer rate, the minimum operating temperature would have to be at least $T_2 = 506$ K. The thermodynamic analysis makes it now quite obvious that at the design operating temperature the temperature difference for generating the pumping power needed to circulate the amount of working fluid at a rate commensurate to a heat transfer of 52 W could not be achieved; although the porous structure provides sufficient capillary suction pressure. Naturally, at an operating temperature of 440 K, the heat pipe will operate below its operating limit according to relation 1. This relation between heat transfer, temperature difference, and operating temperature was well demonstrated in Ref. 9.

The thermodynamic considerations point to additional design and operating conditions that must be observed for a common cylindrical heat pipe whose liquid flow passages are located along the inner wall. To maintain the flow, a pressure gradient has to exist in the vapor flow as well as in the liquid return flow as was shown in Fig. 2. Thus, a pressure difference between vapor and liquid has to exist at the very same axial location including the assumed adiabatic section of the heat pipe. If the adiabatic section were truly devoid of heat transfer, no radial heat flux would be present. The vapor and liquid would be of the same temperature unless the condensate had been highly supercooled at the condenser. A uniform temperature of vapor and liquid at the same axial location, however, would violate one of the thermodynamic prerequisites for the circulation of the working fluid. Vapor would form in the porous structure, arteries, or grooves that could reduce or even totally block the return flow of liquid to the evaporator. Before the argument can be advanced that superheating of the fluid is a factual possibility in the return flow, it has to be convincingly shown that indeed the fluid can remain superheated while flowing through the tortuous path of a wick. Furthermore, the allowable level of the superheating would have to be sufficient to prevent vapor development. Naturally, nobody has ever been able to measure superheating of a flowing liquid inside a wick structure located inside a hermetically sealed heat pipe.

In the real thermodynamic world, the liquid can be prevented from vaporizing in the wick by permitting a radial temperature gradient commensurate with the pressure gradient. This condition requires radial heat transfer in the adiabatic section of the heat pipe. Fortunately, the insulation material used for wrapping the adiabatic section of heat pipes was most of the time limited in its effectiveness, thereby allowing the radial heat flux needed for maintaining the essential radial temperature gradients.

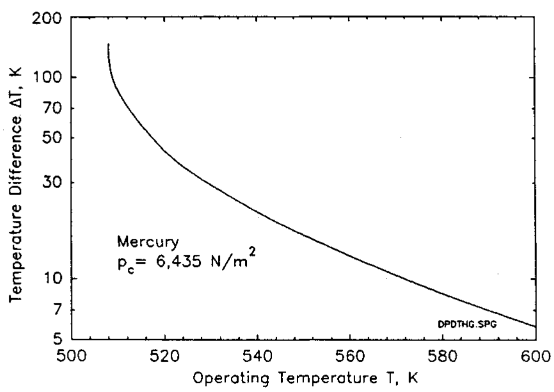

Fig. 9 Temperature difference requirement for a mercury heat pipe.

As an example, the senior author of this article operated a 4.572-m (15-ft) sodium heat pipe that was insulated with MinK® at a nominal operating temperature of 1113 K in the laboratory environment, 20°C.[10] Of a total of 6 kW of input thermal power at the evaporator, about 1 kW was lost along the adiabatic section. A 3 K temperature must have existed between the vapor and the liquid to achieve this heat loss. This temperature difference was sufficient to maintain the return fluid in the wall wick in the liquid phase, and thus permitted the heat pipe to operate to its full design capacity of 6 kW.

The same temperature distribution that was shown to be required for wall wick heat pipes must be imposed on heat pipes which are designed with arteries. Whenever heat pipe action failed in artery heat pipes, "depriming" of the arteries has usually been blamed as the cause of failure. As previously noted, there are effects that tend to increase the entropy of the liquid in return passages. These can cause the generation of vapor in the artery with a partial or total vapor block of the liquid flow. Depriming, therefore, is not the displacement of the working fluid by a gas, but is the blockage of the liquid flow by the vapor of the working fluid. The vapor does not flow into the liquid flow passages, but is generated in it. The vapor generation that causes failure is the result of a local mismatch between temperature and pressure in the liquid that is incompatible with the thermodynamic cycle that supports the heat pipe operation.

The thermodynamic principle of heat pipe operation makes it difficult to consider designs in which arteries are located in the center of a heat pipe, like in the once proposed cat's eye artery design (Fig. 1). In this configuration the liquid return passage is entirely surrounded by vapor in the adiabatic section, resulting in heat transfer from the vapor to the liquid with no cooling of the liquid possible. This design greatly enhances the probability of vapor generation in the artery, causing burnout of the pipe at a relatively low power transfer level unless a high degree of subcooling of the liquid is achieved at the condenser, as indicated by path E-A in Fig. 5. This evaluation has been fully borne out by the low performance of this type of heat pipes,[4] and other similar designs.

The effect of subcooling of the liquid return flow was investigated during the performance testing of a sideflow heat pipe.[11] The heat pipe was tested with and without subcooling, with the artery in various attitudes, and at various adverse tilts. The test results are shown in Fig. 10. The performance at 90-deg rotation, and with a 0.635-cm ($\frac{1}{4}$-in.) tilt, are particularly interesting. Without subcooling, the heat pipe reached 1.2 kW of thermal power transfer just prior to burnout. With 10°C subcooling, which was achieved by extracting 100 W of thermal power from the liquid return flow, the heat pipe transferred 2.1 kW of thermal power without impending burnout. The 2.1-kW thermal power transfer limit was imposed by the cooling capacity of the laboratory heat sink. Thus, 100 W of subcooling provided an increase of at least 900 W in heat transfer capacity to the heat pipe. After subcooling was removed, the heat pipe failed within minutes due to a vapor lock in the liquid return tube.

From the above discussion it becomes clear that cooling and/or subcooling of the liquid return has to be sufficient in any heat pipe configuration anywhere in the liquid return path—whether the return path is located internally or externally of the vapor path—so that the liquid is prevented from moving into the vapor region not only in the return path itself—whether this path be coupled or decoupled thermally from the vapor flow path—but also at the entrance to the evaporator. For the latter condition the required amount of subcooling is very much a function of the heat conduction from the evaporator structure through the container wall to the liquid return path.

The understanding of the thermodynamic aspects of heat pipe operation can now explain the difficulty in correlating test data and evaluating heat pipe operation. The thermo-

SUMMARY OF SIDEFLOW HEAT PIPE TEST RESULTS
TRANSPORT CAPACITY (kW)

Tilt[b] (in.)	Rotation[c] 0° w/o SC	90° w/o SC	90° with SC	180° with SC
1/8	2.1[a]	2.1[a]	—	1.8
1/4	2.3[a]	1.2	2.1[a]	—
1/2	2.3[a]	—	—	—
2	2.1	—	—	—
2-1/2	1.8	—	—	—
2-3/4	1.5	—	—	—

NOTE:

a) Heat sink limited

b) Tilt

c) Rotation

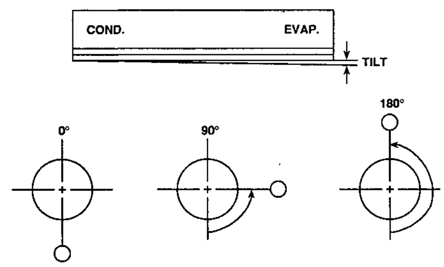

Fig. 10 Sideflow heat pipe subcooling data.

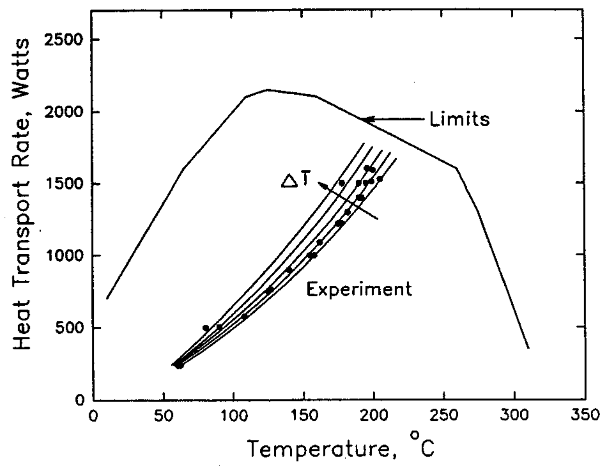

Fig. 11 Presentation of experimental heat pipe data based on thermodynamic considerations.

dynamic analysis has identified the two principle parameters that determine the thermal transport capacity, the total temperature difference, and the operating temperature. The capillary pumping pressure, which previously was considered the parameter which establishes the operation of the heat pipe, obviously defines only one of the operating limits. Neither one of the two thermodynamic parameters alone is sufficient to characterize the heat pipe transport capacity. With the presented thermodynamic relations, the test data of Fig. 4 can now correctly be interpreted. The authors of Ref. 8 suggested that these data were located on the viscous and/or sonic velocity limits as shown in Fig. 3, with some undetermined offset due to the idealization of the theoretical model. However, a comparison between Figs. 4 and 8 indicates that the test data follow the thermodynamically imposed relation between the actually generated pressure difference, operating temperature, and temperature difference.

As in all flow devices, the flow rate of the working fluid, which is equivalent to heat transport in a heat pipe, is directly a function of the available pressure difference. The exact relation between flow rate and pressure drop is naturally determined by the physical design parameters of a heat pipe as mentioned previously. Thus, the test data of Ref. 8 would have been properly plotted as indicated in Fig. 11, accounting for the additional parameter ΔT. Since, during testing, many different condenser chill block configurations were used, the heat pipe was exposed to a large range of temperature differences. Unfortunately, the temperature differences were not recorded. At the time of testing the total temperature difference had not been identified yet as one of the two major operating parameters.

Conclusions

The previously accepted heat pipe design theory cannot predict the performances of heat pipes. The fluid circulation in a heat pipe is the result of a thermodynamic cycle that operates over a temperature difference. The capillary pressure supported by the porous structure in a heat pipe and the temperature dependent surface tension of the working fluid only facilitates the conversion of thermal energy to kinetic energy. It provides the fluid diode action which prevents vapor from penetrating into the liquid return passages while permitting liquid to be vaporized and expand into the vapor space. The rate at which the working fluid can be circulated, and thus thermal power can be transferred, is determined by the internal temperature difference that can be achieved and the absolute operating temperature.

A heat pipe has been shown to obey all the observed laws of heat transfer like any other heat conductor. Its high heat conductivity (super thermal conductivity) is a very strong function of the operating temperature in a limited temperature range as determined by the working fluid. When the operating range is exceeded either at the lower end or at the upper end, the working fluid loses its ability to contribute to the total heat conduction of the device by circulating in its two phases, liquid and vapor. At the low temperature end the vapor pressure of the working fluid is insufficient, while above the upper temperature limit the capillary pressure is inadequate to sustain the pressure difference. This variation in thermal conductivity of a heat pipe structure is almost exactly the same as that found in solid material structures, where in different temperature ranges different physical phenomena contribute to the total heat transfer to different degrees.[12,13]

This understanding of heat pipe operation should not only lead to better predicting heat pipe performances, but might point to new thermal transport concepts that can carry thermal energy over longer distances with components that weigh less and perform with a lower loss in thermal potential than the present heat pipes. It is hoped that the theoretical considerations that were presented will stimulate their full experimental verification and further refinement.

Acknowledgments

The research described in this article was carried out by the Jet Propulsion Laboratory, California Institute of Technology and by Wright Laboratory, Wright Patterson Air Force Base. It was sponsored by the U.S. Air Force and the National Aeronautics and Space Administration.

References

[1]Chi, S. W., "Heat Pipe Theory and Practice, A Source Book," Hemisphere, New York, 1976, pp. 33–77.

[2]Dunn, P. D., and Reay, D. A., *Heat Pipes*, Pergamon, New York, 1978, pp. 19–98.

[3]Ponnappan, R., Ramalingam, M. L., Beam, J. E., and Mahefkey, E. T., "Remedial Measures for Condenser Quenching in a Long Water Heat Pipe," AIAA Paper 87-1646, June 1987.

[4]Alario, J. P., Haslett, R., and Kosson, R., "The Monogroove High Performance Heat Pipe," AIAA Paper 81-1156, June 1981.

[5]Benner, K., and Feild, Z., private communication, Lockheed Missiles and Space Co., discussion at Wright Research and Development Center, Nov. 1989.

[6]Kosson, R., Brown, R., and Unger, E., "Space Station Heat Pipe Advanced Radiator Element (SHARE) Flight Test Results and Analysis," AIAA Paper 90-0059, Jan. 1990.

[7]Brown, R., Unger, E., and Cornwell, J., "Flight Test Results of the SHARE II Monogroove Heat Pipe," AIAA Paper 92-2886, July 1992.

[8]Ponnappan, R., and Mahefkey, E. T., "Development of a Double-Wall Artery High Capacity Heat Pipe," *Spacecraft Thermal Control, Design and Operation*, edited by P. E. Bauer and H. E. Collicott, Vol. 86, Progress in Astronautics and Aeronautics, AIAA, New York, 1983, pp. 202–221.

[9]Deverall, J. E., "Mercury as a Heat Pipe Fluid," Los Alamos Scientific Lab., LA-4300-MS, Los Alamos, NM, October 1969.

[10]Richter, R., "Solar Collector Thermal Power System, Volume III Basic Study and Experimental Evaluation of Thermal Train Components," Air Force Applied Physics Lab., AFAPL-TR-74-89-3, Wright-Patterson Air Force Base, OH, Nov. 1974.

[11]Fleischman, G. L., Baseules, A., and Gottschlich, J. M., "Advanced Heat Pipe Components for High Power Spacecraft Thermal Management," AIAA Paper 90-0058, Jan. 1990.

[12]Touloukian, Y. S., et al., "Thermophysical Properties of Matter Volume 1: Thermal Conductivity Metallic Elements and Alloys," IFI/Plenum, New York-Washington, 1970

[13]Touloukian, Y. S., et al., "Thermophysical Properties of Matter Volume 2: Thermal Conductivity Nonmetallic Solids," IFI/Plenum, New York-Washington, 1970.