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The thermal performance of heat pipes with localized heat input

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Abstract—The performance of heat pipes with localized heat input including the effects of axial and circumferential heat conduction under high and low working temperatures is investigated. The numerical results show that when heat pipes are spot heated, the peak temperature of the wall is greatly reduced and the surface can be protected from being burned out by the high heat flux. The boiling limitation becomes the most important limitation for this type of heat pipe. Numerical results for block heating a heat pipe with low working temperatures indicate a good agreement with existing experimental data. It is also shown that most of the input heat passes through the wall beneath the heated block.

INTRODUCTION

SINCE the publication of the first paper on heat pipes, various kinds of heat pipes have been manufactured, tested and put into operation. In the meantime, thousands of theoretical and experimental analyses dealing with the characteristics of heat pipes have been published. Among the special types of heat pipes, localized heat input or spot heated heat pipes have not been extensively studied. This is surprising since many high performance heat pipes are subjected to localized heating for a variety of applications. The study of spot heating heat pipes is important in research areas such as the cooling of leading edges on hypersonic aircraft, the protection of special surfaces from being attacked by very high heat flux sources such as a laser beam, cooling of microelectronic elements, etc.

According to the working conditions and the application, spot heated heat pipes can be classified into two major types, namely, heat pipes with low or moderate working temperatures which are mainly used for the purpose of energy conservation or electronic cooling, and heat pipes with high working temperatures which are used to protect a surface from being burned out by a very high heat flux. Rosenfeld [1] studied the performance of line heated heat pipes with low working temperatures analytically with a one-dimensional model (circumferential direction) and numerically with a two-dimensional model (radial and circumferential directions).

For the present analysis of spot heating or block heating, axial conduction needs to be considered, which has a much more pronounced effect than conduction in the radial direction. Furthermore, the effect of radiation is included in the analysis of spot heated pipes with high working temperatures. In addition, the operating temperature of the heat pipe should be obtained by an overall energy balance rather than an input condition as carried out by Rosenfeld [1] for the line heated heat pipe.

ANALYSIS

Spot heated heat pipes with high working temperatures

Spot heated heat pipes are to be used as a means of protecting a surface from being burned out by a very high heat flux by using the working fluid within the porous media inside the pipe as an evaporator to absorb the heat energy. The vapor flows to the rest of the porous surface and releases latent heat energy as it condenses. The energy is dissipated into space or to the environment by radiation from the outer surface. Because of the high latent heat of the working fluid, a large amount of incoming heat can be absorbed by evaporation, and spread to the surrounding surface of the wall to be dissipated into space, without causing the temperature in the wall to become too high and burn out. The positions of the evaporator and condenser are not fixed nor are their areas. These depend on where the pipe is hit by the incoming heat flux, and how large the surface area is that is being hit by the heat flux. Also, this kind of heat pipe has no adiabatic section. This is illustrated in Fig. 1(a). The end caps have been removed to demonstrate the typical interior structure and the vapor flow pattern. The origin of the coordinate system is set at the center of the heated spot. Figure 1(b) shows a typical wall temperature profile along the x-axis at $y = 0$ and the vapor temperature T_s. If the temperature of the wall is higher

NOMENCLATURE

A	radiation area [m^2]
A_H	heated area [m^2]
a	coefficient in the power-law equation for boiling
b	exponent in the power-law equation for boiling
H	heat transfer coefficient [W m^{-2} K^{-1}]
h	height of the spot heated heat pipe [m]
k	thermal conductivity [W m^{-1} K^{-1}]
L_C	condenser length [m]
L_E	evaporator length [m]
L_H	length of heated block [m]
L_1, L_2	length of spot heated heat pipe [m]
Q_H	input heat [W]
Q_R	radiation heat transfer to surroundings [W]
q	heat flux [W m^{-2}]
q_H	input heat flux [W m^{-2}]
R	radius of heat pipe [m]
s	circumferential distance coordinate [m]
S_H	$W_H/2\pi R$
T	wall temperature [K]
T_H	reference temperature [K]
T_0	temperature of environment [K]
T_s	vapor temperature [K]
T^*	$(T-T_s)/T_H$
ΔT	$T-T_s$ [°C]
ΔT_c	critical temperature difference [K]
W_H	width of heated block [m]
W_1, W_2	width of spot heated heat pipe [m]
X	$x/\pi R$
x_1	distance of the edge of the heated block from the origin [m]
x, y	Cartesian coordinates [m]
Y	$s/\pi R$.

Greek symbols

δ	wall thickness [m]
ε	emissivity
σ	Stefan–Boltzmann constant [W m^{-2} K^{-4}].

Subscripts

C	condensation
E	evaporation.

than T_s, it serves as the evaporator; if the wall temperature is lower than T_s, it serves as the condenser. This particular phenomenon is not observed for conventional heat pipes where it is operated at a nearly constant wall temperature.

The present analysis is based on a number of physical assumptions. The emphasis is put on the wall temperature in the analysis because of the need to prevent the melting of the heat pipe wall. The heat pipe is assumed to contain only one fluid and is operating under steady-state conditions. This means that the vapor region is nearly isothermal, and the total energy input and output are equal. The wick–liquid matrix is assumed to be very thin, and its existence is represented by the evaporation heat transfer coefficient, H_E, and the condensation heat transfer coefficient, H_C. This is a good approach for the purpose of designing a heat pipe because in most cases, evaporation and condensation heat transfer coefficients in porous media can only be obtained from experimental data

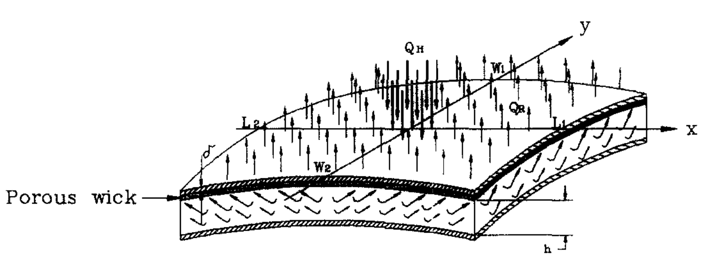

FIG. 1(a). Typical configuration of a spot heated heat pipe.

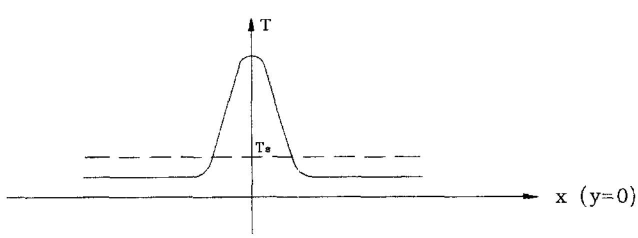

FIG. 1(b). Typical wall temperature profile at $y = 0$.

[2–4]. The common measurable parameters in experiments on heat pipes are T_s, H_E, H_C, and the wall temperature. The results of the analysis based on the above concept can be readily used as a guide when designing heat pipes. Finally, it is assumed that the wall thickness is thin, and the radius of curvature of the surface is much larger than h, so that the wall temperature does not vary substantially with radial position in the wall.

In the Cartesian coordinate system shown in Fig. 1(a), the two-dimensional governing equations and boundary conditions for the wall can be written by an energy balance of the differential elements in each section of the heat pipe. For the evaporation section, except the surface beneath the heated spot, the governing equation is

$$\frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right)-\frac{\varepsilon\sigma}{\delta}(T^4-T_0^4)-\frac{q_E}{\delta}=0 \tag{1}$$

where $T = T_E$ at the junction of the evaporator and the condenser segments, ε is the emissivity, δ the thickness of the wall, T_0 the temperature of the environment, and q_E the evaporation heat flux. It has been shown that the evaporation heat transfer coefficient varies with heat flux because the meniscus in a capillary evaporator recedes as the heat flux increases. An examination of the reported data shows that a power-law boiling relation is appropriate for relating the heat flux to the evaporating temperature drop in a heat pipe [1, 2], i.e.

$$q_E = a(T-T_s)^b. \tag{2}$$

The most common values of the exponent vary from 1.0 to 2.0, with that of liquid metal heat pipes remaining 1.0 and the coefficient $a = H_E$ in many cases [2, 3, 5].

For the wall beneath the heated spot, the corresponding equation is

$$\frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right)+\frac{q_H}{\delta}-\frac{q_E}{\delta}=0 \tag{3}$$

where q_H is the incoming heat flux at the heated spot.

For the condenser segment, the governing equation is

$$\frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right)-\frac{\varepsilon\sigma}{\delta}(T^4-T_0^4)+H_C(T_s-T)=0. \tag{4}$$

With $T = T_C = T_E$ at the junction of the evaporator and the condenser sections, and $\partial T/\partial n = 0$ at the peripheral boundary of the heat pipe where n is the normal direction of the periphery.

The vapor temperature, T_s, is an unknown in equation (4). The fact that the vapor temperature must be determined requires one more equation. This equation is provided by an energy balance over the entire heat pipe. In a steady-state operation, there are no energy and mass accumulations, and the vapor temperature will be adjusted according to the heat input and the ambient conditions. This means that the condensing heat transfer is equal to that of evaporation, and all the heat input is rejected to the ambient, i.e.

$$\iint_{A_E} H_E(T-T_s)\,dA = \iint_{A_C} H_C(T_s-T)\,dA \tag{5}$$

$$Q_H = \iint_A \varepsilon\sigma(T^4-T_0^4)\,dA \tag{6}$$

where Q_H is the heat input through the spot; A_E and A_C are the evaporator surface area and the condenser surface area inside the heat pipe, respectively; A is the outer radiation surface area. It should be noted that A is not necessarily equal to A_C. When spot heating heat pipes, the evaporator and condenser are not prescribed. Usually, the heated area is small, and a large amount of heat needs to be spread to the surrounding surfaces to be dissipated. Therefore, A_E is larger than the spot heating area A_H and as a result, $A > A_C$.

Block heated heat pipes with low or moderate working temperatures

Block heated heat pipes are normally used to transport energy from one place to another for the purpose of energy conservation or electronic component cooling. The evaporator and condenser segments are normally separated, which is similar to conventional heat pipes except for the localized heating. The working temperature is comparatively low or moderate, which means that radiation heat transfer is not important in the analysis of these heat pipes.

Consider the heat pipe shown in Fig. 2. It has an evaporator section of length L_E, a wall thickness δ, an outside radius R and a block heated area of width W_H and length L_H.

The analysis here is based on similar assumptions used for spot heated pipes with high operating temperatures. With a thin wall and a large pipe diameter, the problem can be investigated in Cartesian coordinates.

In the evaporator section, the governing equation for the pipe wall that is not underneath the heated block is

$$\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial s^2}-\frac{a}{k\delta}(T-T_s)^b=0 \tag{7}$$

and the governing equation for the wall beneath the heated block is

$$\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial s^2}-\frac{a}{k\delta}(T-T_s)^b+\frac{q_H}{k\delta}=0. \tag{8}$$

In equation (8), we have employed the power-law relation for the boiling heat flux. In this situation, $b > 1.0$. The boundary conditions for this problem are

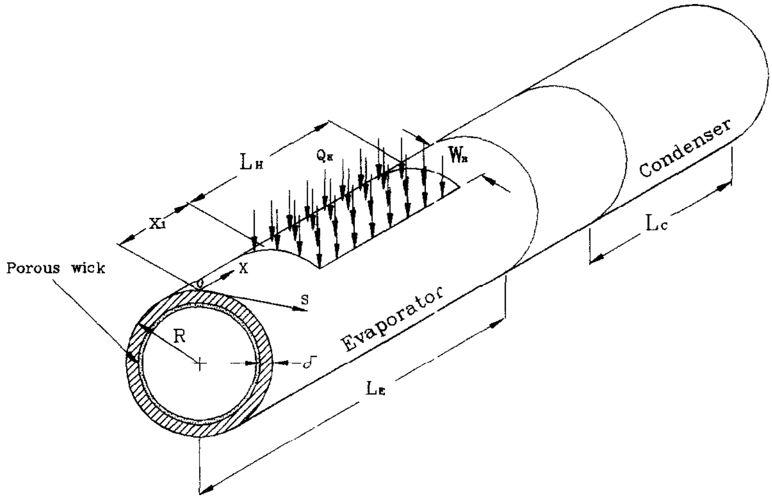

FIG. 2. A general configuration of a block heated heat pipe.

$$\frac{\partial T}{\partial x} = 0 \quad \text{at } x = 0 \text{ and } L_E \tag{9}$$

$$\frac{\partial T}{\partial s} = 0 \quad \text{at } s = 0 \text{ and } \pi R. \tag{10}$$

Equation (9) defines an insulated boundary condition, and equation (10) is true due to symmetry.

Equations (9) and (10) can be nondimensionalized with the following variables:

$$T^* = (T - T_s)/T_H, \quad X = x/\pi R, \quad Y = s/\pi R.$$

The resulting non-dimensional equations are

$$\frac{\partial^2 T^*}{\partial X^2} + \frac{\partial^2 T^*}{\partial Y^2} + C_1 T^{*b} = 0 \tag{9a}$$

$$\frac{\partial^2 T^*}{\partial X^2} + \frac{\partial^2 T^*}{\partial Y^2} + C_1 T^{*b} + C_2 q_H = 0 \tag{10a}$$

where

$$C_1 = -a(\pi R)^2 T_H^{b-1}/\delta k, \quad C_2 = (\pi R)^2/T_H \delta k$$

and T_H is the reference temperature.

The above model is justified due to the fact that since the main purpose is to transport energy from one place to another, the heat dissipation to the ambient from the evaporator section is negligible. Also, because of the large evaporation heat transfer coefficient, very little heat is transported from the evaporator to the condenser through the pipe wall. This will be shown in the numerical results in the next section.

NUMERICAL RESULTS AND DISCUSSION

The governing equations for spot heated and block heated heat pipes are nonlinear and nonhomogeneous and require an iterative procedure. The problems were solved by utilizing the finite-difference method based on the control-volume formulation [6]. A combination of the direct method (TDMA) and the Gauss–Seidel method was employed to solve the discretization equations. In some special cases, under-relaxation parameters were used to control the advancement of the solutions. The grid size employed in the program varies from 20×70 to 70×300 depending on the computational domain. To determine a suitable grid size, the computed temperature profiles are compared for a number of different grid sizes for the same problem. The maximum errors among these grid sizes are less than 1.0%. Also, an energy balance over the entire computational domain was checked for every computed temperature field, the maximum error of which was at most 0.1%.

Figure 3 shows the numerical results for spot heated heat pipes with high working temperatures. Like conventional heat pipes, the heat input has a strong influence on the heat pipe performance. As the heat input increases, the peak wall temperature increases. For example, the peak temperature with $Q_H = 8000$ W is more than twice as high as that with $Q_H = 2000$ W.

In conventional heat pipes, the area available for heat input is comparatively large, the input heat flux is comparatively low, and the working limitation is mainly determined by the total energy input. With spot heated heat pipes, the situation is different. Figure 4 shows the temperature distribution of the wall with a very high incoming heat flux (as high as 10 kW cm^{-2}). Comparing the curve of $H = H_E = H_C = 10\,000$ W m^{-2} K^{-1} and $Q_H = 2750$ W in Fig. 4 with that of $Q_H = 4000$ W in Fig. 3, it can be seen that even though the heat input $Q_H = 2750$ W in Fig. 4 is less than that of $Q_H = 4000$ W in Fig. 3, its peak temperature is still several hundred kelvin higher because of the smaller area that is heated. On the other hand, Fig. 4 shows that a higher heat transfer coefficient can reduce the peak temperature, as is expected.

Also shown in Fig. 4 is the influence of A_H on the

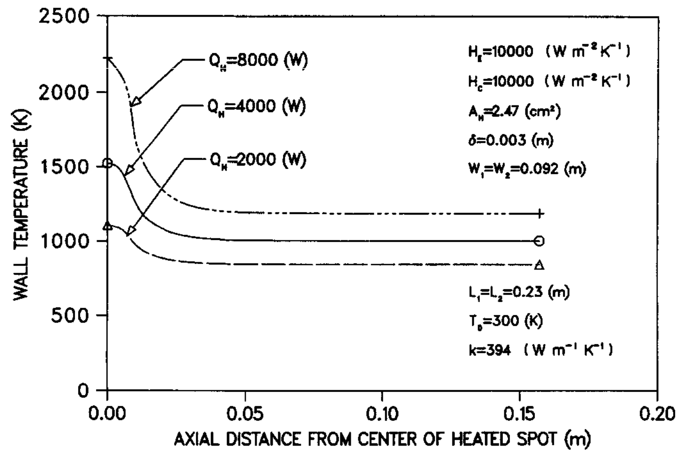

FIG. 3. The variation of the wall temperature with input heat.

peak wall temperature. The curve with a much lower peak wall temperature in Fig. 4 is subject to the same heat flux as that of the other three curves ($q_E = 10$ kW cm^{-2}), but a smaller heating area (Q_H is also smaller). The resultant peak wall temperature is almost one thousand kelvin lower than that with the larger heating area. In this study, the heating area is square. Because of the small heating areas, its shape is less important.

Unlike conventional heat pipes, the thermal conductivity of the wall influences the wall temperature distribution greatly for spot heated heat pipes, especially in the case of a very high incoming heat flux, as shown in Fig. 5. When the thermal conductivity of the wall is small, the peak temperature would jump intolerably high and the surface would be burned out. On the other hand, with a large wall thermal conductivity, the peak temperature decreases sharply. This is not surprising because a large amount of heat needs to be transferred to the surrounding wall through a very small area by conduction.

In conventional heat pipes, better cooling conditions and a larger cooling surface can ameliorate the performance and result in a lower working temperature. This is also true for spot heated heat pipes. The trend is well illustrated in Fig. 6, where a larger surface area (radiation heat transfer area) corresponds to a lower working temperature T_s. Also, the peak wall temperature is decreased accordingly. But, if we pay attention to the area around the center of the heated spot, we may notice that the temperature difference between the peak temperature and the working temperature for different curves almost remains the same. This local phenomenon obviously results from the localized heat input characteristics of the heat pipe. The most important factor which determines the performance is the working condition at the heated spot.

To estimate the validity of spot heating heat pipes to reduce the wall temperature, a comparison is made between the wall temperature of a plate that is not a heat pipe and that of a plate heat pipe. The curve with the solid circle legend in Fig. 7 is the wall temperature profile of a spot heated heat pipe under normal working conditions, while the curve indicated with $H = 0$ is the temperature profile of a plate that is not a heat pipe ($H_E = H_C = H = 0$), with other conditions being the same. It can be seen that the peak wall temperature

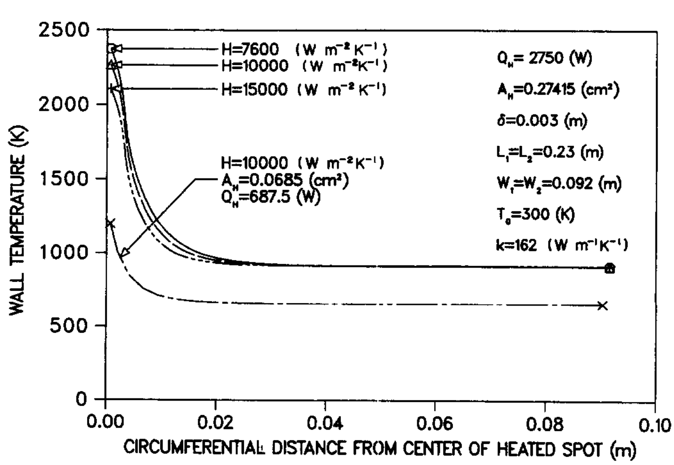

FIG. 4. Variation of wall temperature with H and A_H.

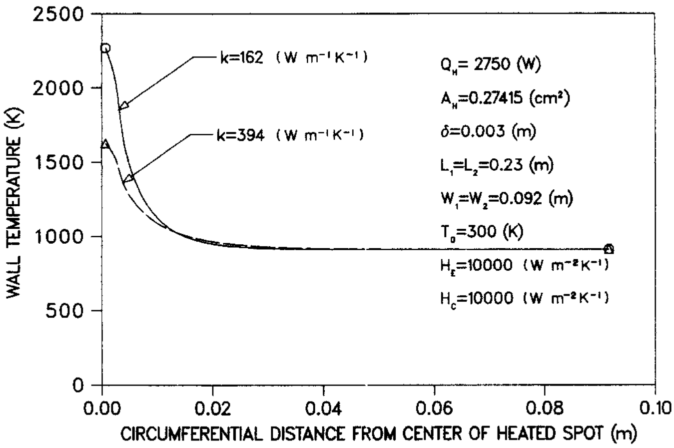

FIG. 5. Variation of wall temperature with thermal conductivity k.

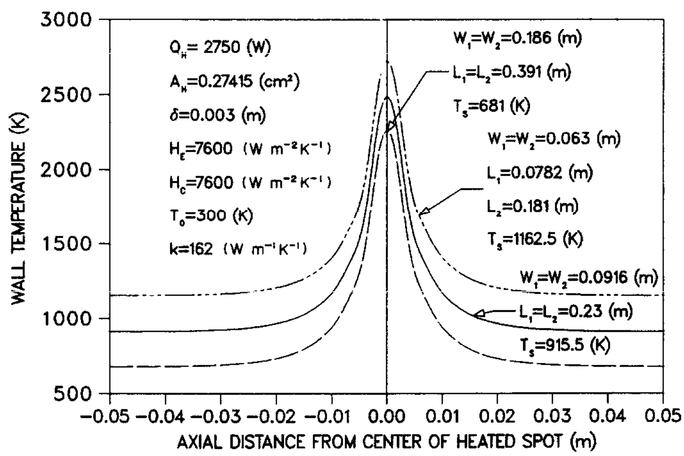

FIG. 6. Variation of wall temperature with heat pipe surface area.

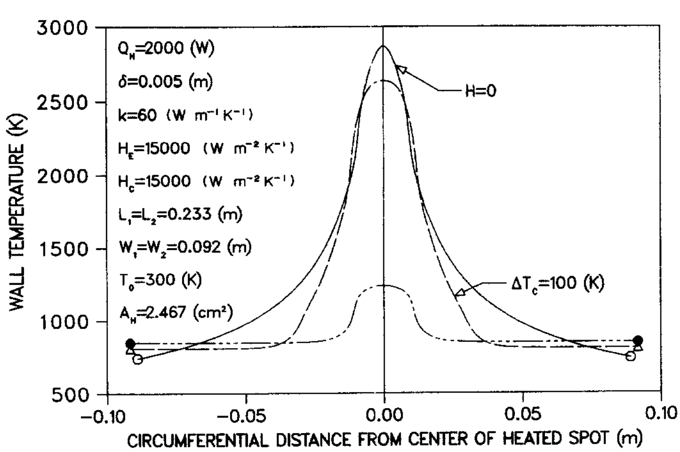

FIG. 7. Temperature profiles with different working conditions.

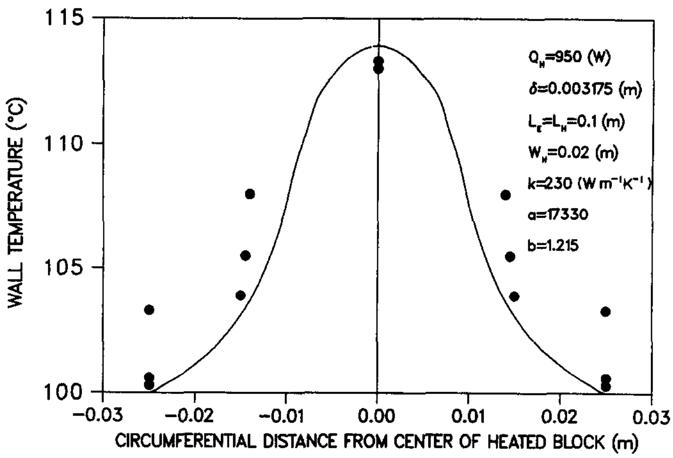

FIG. 8. Comparison of the numerical results with experimental data.

of the surface adopting heat pipe technology is reduced significantly.

When a comparison is made with conventional heat pipes, it is clear that spot heated heat pipes have small evaporator surfaces, very high evaporation heat fluxes, larger condenser surfaces and vapor passages. Because of these factors, the boiling limit becomes the most important operating limit of these heat pipes. In Fig. 7, the curve indicated with $\Delta T_c = 100$ K assumes that the boiling limitation is reached for the local evaporating surface when $T - T_s > \Delta T_c = 100$ K. When this occurs, the porous media at that point is assumed to be completely dry, and no evaporation takes place. For the curve with the solid circle legend, this restriction has not been imposed on ΔT, and the pipe works properly with the other conditions being the same. Once the boiling limitation is reached and the evaporating surface becomes dry, the wall temperature at the heated spot will jump thousands of kelvin higher than that of heat pipes under normal working conditions. From the viewpoint of heat pipe design, most difficulties arise from the avoidance of the incipience of boiling in the porous wick of the pipe. Special care must be taken to properly design the structure of the porous media, to choose suitable working fluids, and to insure the wettability of the wick and the wall to increase the boiling limitation.

In addition, the emissivity of the surface and the thickness of the wall have strong effects on the temperature profile of the wall. Increasing the emissivity will reduce the working temperature, and therefore the peak wall temperature. The value of ε in this numerical calculation was taken as 0.8. Also, a larger thickness of the wall will alleviate the temperature jumps at the center of the heated spot, but this is not practical in many applications.

Figures 8–10 show the numerical results for heat pipes with localized heat input working under low or moderate temperatures. In this situation, the heated area is comparatively large, and the heat flux is comparatively low, so that the temperature jump is not so severe as that for spot heating heat pipes with high working temperatures.

Figure 8 shows the comparison of the numerical results of the circumferential wall temperature profile with the experimental data from the paper by Rosenfeld [1]. The experiment was conducted with a narrow line heater at the evaporation section of the heat pipe. The evaporation heat flux relation is also taken from that experiment, with

$$q_E = a(T - T_s)^b$$

where $a = 17\,330$ and $b = 1.215$.

The agreement between the numerical results and the experimental data is excellent. The line width surely influences the performance of the heat pipe. With the width of the heated line becoming larger, the temperature distribution along the circumferential direction becomes smoother, as shown in Fig. 9. Among the different heating widths, half heating ($S_H = W_H/\pi R = 0.5$) is of special interest in many applications. It can be seen that with a uniform input heat flux and a large evaporation heat transfer coefficient, the temperature profile of the wall beneath the heated block is nearly smooth, and the working conditions of this half of the evaporator are nearly the same as that of heat pipes with a uniform heat input.

Figure 10 shows the performance of block heated heat pipes for different values of the wall thermal conductivity. Unlike the spot heated heat pipes shown in Fig. 5, the thermal conductivity of the wall has little effect on the temperature distribution. The reason is that, because of the large boiling heat transfer coefficient, most of the input heat was absorbed by the evaporating surface beneath the heated block, and only a small amount of heat is spread to the wall that is not heated. This is more pronounced as the wall thermal conductivity decreases. For the working conditions indicated in Fig. 10, with $k = 394$ W m^{-1} K^{-1}, 86.2% of the heat passes through the wall under the heated block, while with $k = 45$ W m^{-1} K^{-1}, 96% of

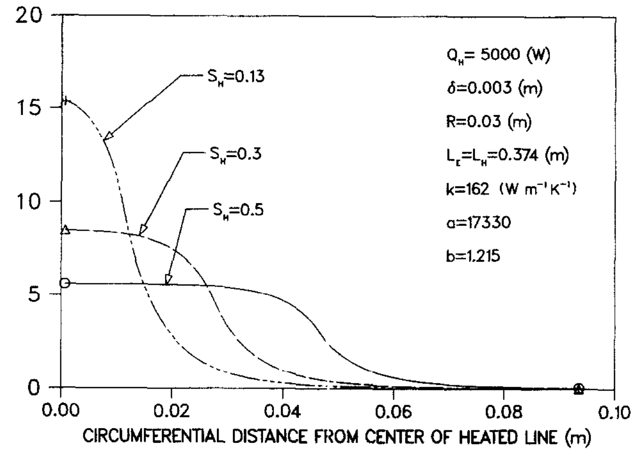

FIG. 9. Variation of the temperature difference $T-T_s$ with the dimensionless width S_H.

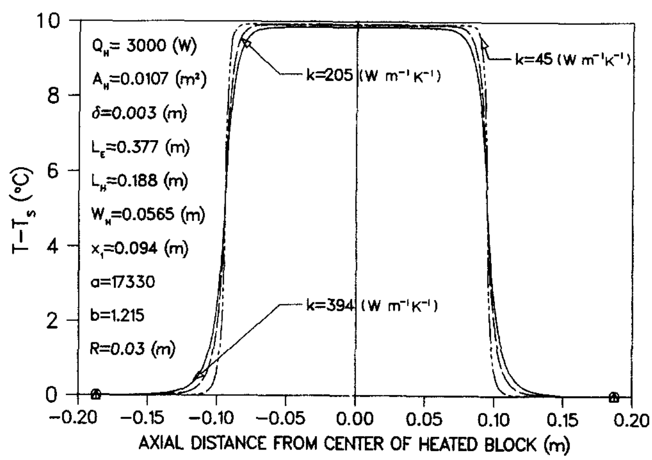

FIG. 10. Variation of $T-T_s$ with thermal conductivity k.

the heat passes through the wall under the heated block.

CONCLUSIONS AND REMARKS

(1) The use of a heat pipe is an excellent method to protect a surface from burning out when the surface is spot heated. The peak wall temperature is greatly reduced due to the operation of the heat pipe. The parameters which influence the performance are Q_H, k, H_E, H_C, A_H, ε and δ. For a fixed heat input Q_H, larger values of k, A_H, H_E, H_C, or δ can reduce the peak wall temperature, and larger values of ε and the heat pipe surface area result in a lower working temperature. Because of the localized heating characteristics of heat pipes, a temperature jump is inevitable at the center of the heated spot, and results in a large ΔT and a high q_E at the heated location. Special measures must be taken to increase the boiling limitation of the heat pipe. Otherwise, the temperature at the heated location will jump intolerably high. More work needs to be done on the structure of the porous media, the wettability of the wick and wall, and the vapor flow pattern in the pipe for this special kind of heat pipe.

(2) The numerical results for localized heat input heat pipes working under low or moderate temperatures agree well with the existing experimental data, and can be used to improve the predictions of heat pipe performance under localized heating. With a large evaporation heat transfer coefficient, most of the input heat passes through the wall under the heated block.

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PERFORMANCE THERMIQUE DES CALODUCS AVEC SOURCE DE CHALEUR LOCALISEE

Résumé—On étudie la performance des caloducs avec source de chaleur localisée, conduction thermique axiale et circonférentielle, pour des températures de fonctionnement faibles ou élevées. Les résultats numériques montrent que quand les caloducs sont chauffés ponctuellement, le pic de température de la paroi est fortement réduit et la surface peut être protégée du brûlage par les flux de chaleur élevés. La limitation par ébullition devient la limitation la plus importante pour ce type de caloduc. Des résultats numériques pour le chauffage par bloc d'un caloduc, avec des températures faibles de fonctionnement, indiquent un bon accord avec les données expérimentales existantes. On constate aussi que la plus grande part de la chaleur entrante passe à travers la paroi sous le bloc chauffé.

DAS THERMISCHE VERHALTEN VON WÄRMEROHREN BEI ÖRTLICHER WÄRMEZUFUHR

Zusammenfassung—Es wird das Verhalten von Wärmerohren bei örtlicher Wärmezufuhr für hohe und niedrige Arbeitstemperaturen untersucht. Die numerisch ermittelten Ergebnisse zeigen bei einer punktuellen Wärmezufuhr, daß die Maximaltemperatur in der Rohrwand bei Berücksichtigung axialer und tangentialer Wärmeleitung stark abnimmt. Die Oberfläche ist dadurch vor einem Durchbrennen bei hohen Wärmestromdichten geschützt. Die Siedegrenze ist bei dieser Art Wärmerohr die wichtigste Leistungsbegrenzung. Die numerischen Ergebnisse für flächige Beheizung des Wärmerohres bei niedriger Arbeitstemperatur stimmen gut mit experimentellen Daten überein. Dabei zeigt sich auch, daß unter diesen Bedingungen der größte Teil der zugeführten Wärme im beheizten Bereich durch die Rohrwand dringt.

ТЕМПЕРАТУРНАЯ ХАРАКТЕРИСТИКА ТЕПЛОВЫХ ТРУБ С ЛОКАЛИЗОВАННЫМ ПОДВОДОМ ТЕПЛА

Аннотация—Исследуются характеристики тепловых труб с локализованным подводом тепла, включающие эффекты осевой периферической теплопроводности при высокой и низкой рабочих температурах. Численные результаты показывают, что при локализованном нагреве тепловых труб максимальная температура стенки значительно снижается и поверхность может быть защищена от пережега при высоком тепловом потоке. Ограничение кипения является наиболее важным для данного типа тепловых труб. Численные результаты для блока, нагревающего тепловую трубу при низких температурах, показывают хорошее соответствие с имеющимися экспериментальными данными. Показано также, что основная часть подводимого тепла проходит через стенку под нагретым блоком.