# United States Patent

[11] 3,598,180


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[54] **HEAT TRANSFER SURFACE STRUCTURE**
**28 Claims, 21 Drawing Figs.**


[52] U.S. Cl. ........ 165/133, 165/105, 165/180, 165/181
[51] Int. Cl. ........ F28f 13/06
[50] Field of Search ........ 165/105, 133, 180, 181



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ABSTRACT: A heat transfer surface structure is described wherein both heat and vaporizable liquid are conveyed through a capillary material to a free vaporizing surface of the capillary material where the liquid vaporizes. Heat is conducted from a heat source wall through a portion of the capillary material to the vaporizing surface where it escapes as heat of vaporization along with the vapor. The liquid flows through the pores of the capillary material from a liquid source to the vaporizing surface under the influence of capillary forces. The vaporizing surface is divided into a large number of regions which are close to the heat source wall and are connected by way of vapor passages to a region external to the capillary material. Thus, the heat conducting paths through the capillary material are very short, and vapor can escape freely through relatively large passages rather than having to force its way through the pores of the capillary material where it would interfere with the liquid flow. Such a vented capillary vaporizer is capable of handing much higher heat flux densities than previous capillary vaporizers. Four examples of capillary vaporizer are set forth, two of these for operation where the liquid and vapor are comingled as in a boiler tube or evaporator tube. One of these has separated areas of capillary material in thermal contact with the heat source surface, thereby defining passages therebetween. The other is similar with added portions of porous materials to form a manifold having a hierarchy of vapor passages of decreasing number and increasing cross section, thus increasing the separation between the regions of liquid input and vapor output. The third example accepts liquid from a capillary structure or wick as in a heat pipe, and also has a pair of manifolds in the form of a hierarchy of passages for vapor flow and capillary paths for liquid flow. The fourth example receives bulk liquid through a channel, and delivers the vapor through a separate passage. Thermal insulation maintains the bulk liquid relatively cool, and active cooling may be provided. This latter embodiment is unique in its ability to pump the heat transfer fluid since the output vapor can be at a higher pressure than the incoming bulk liquid.

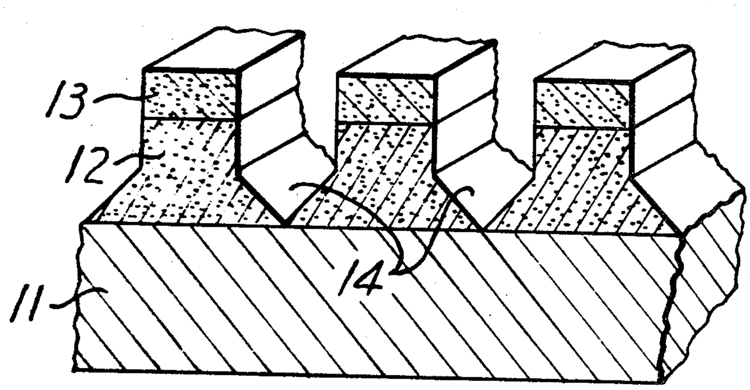



FIG. B

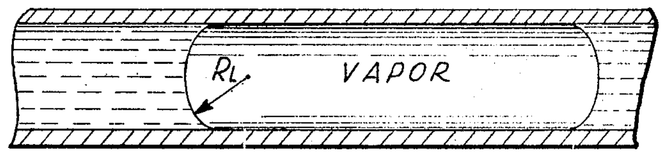



FIG. 1

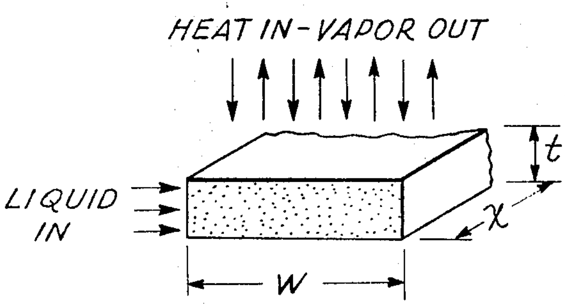
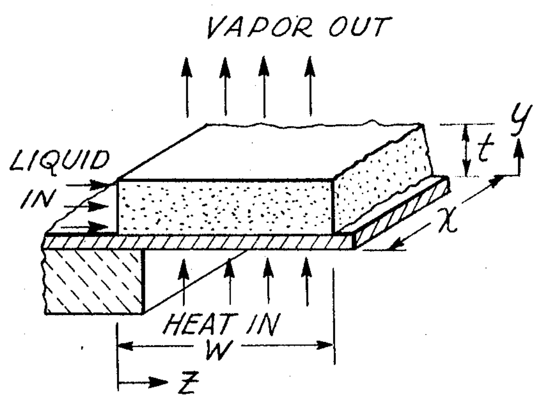



FIG. 2



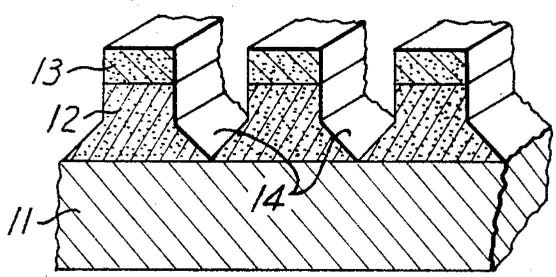
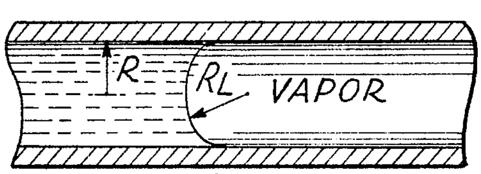




FIG. 3

FIG. A

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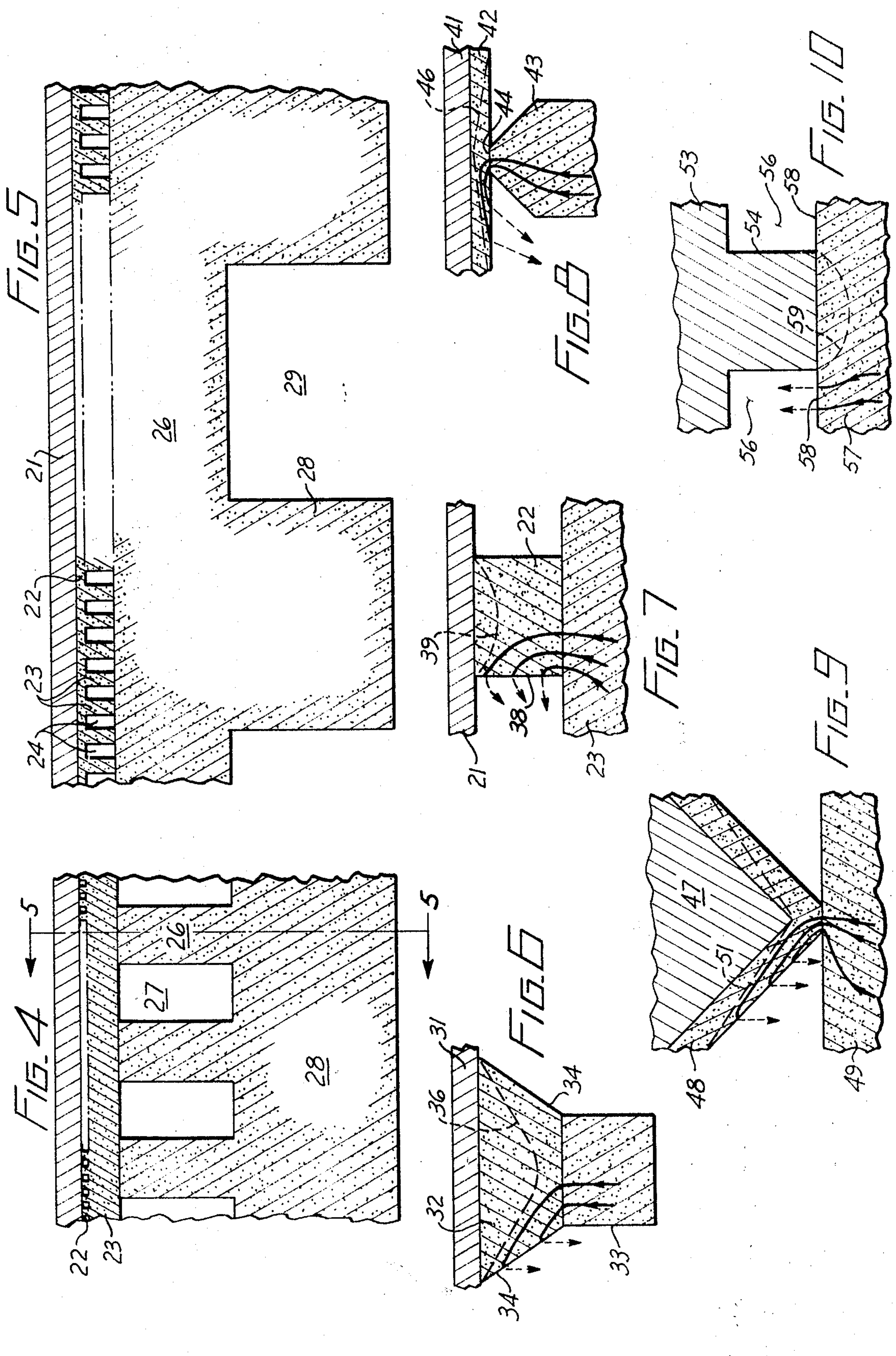


FIG. 4
FIG. 5
FIG. 6
FIG. 7
FIG. 8
FIG. 9
FIG. 10

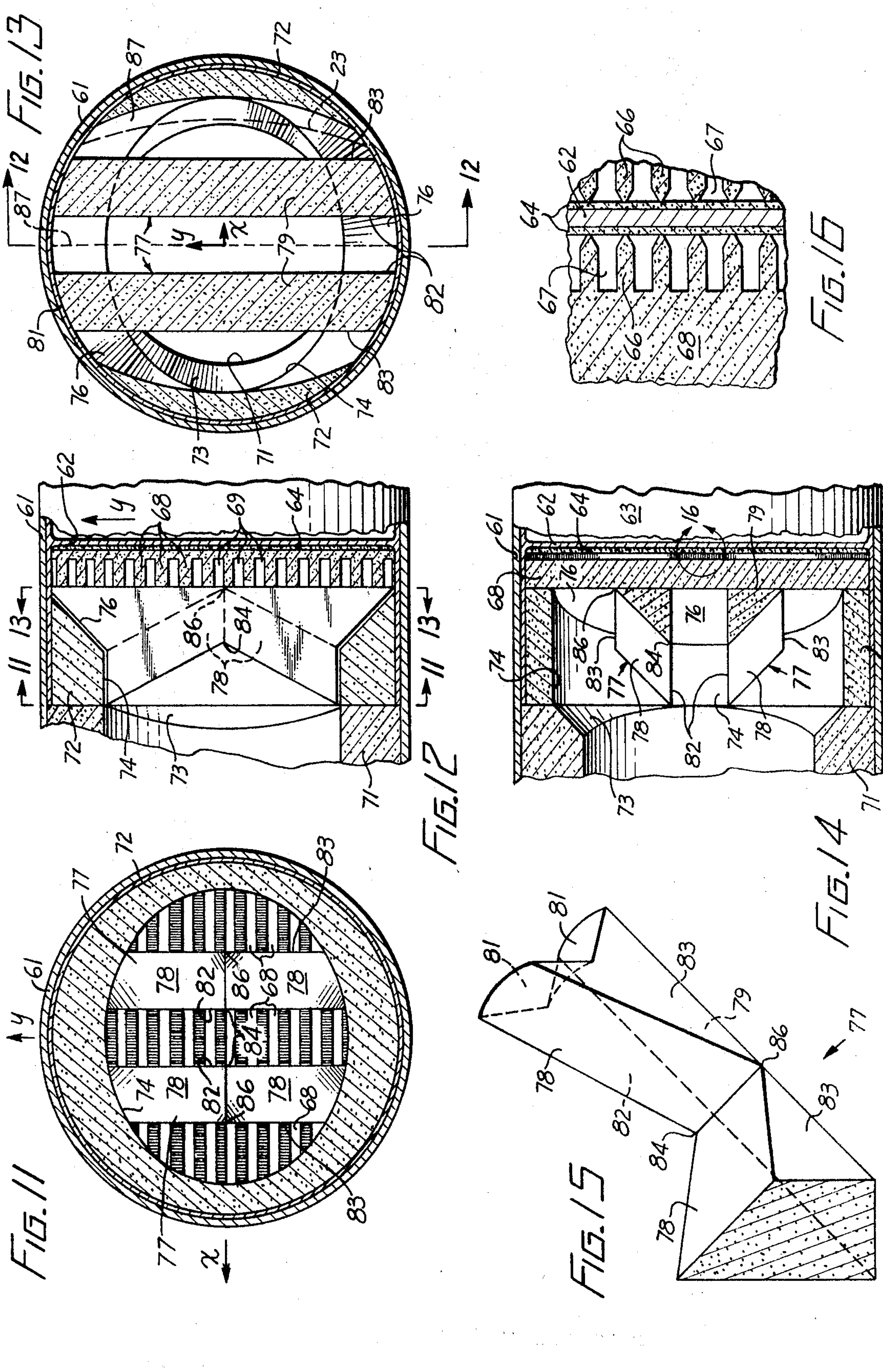


FIG. 11
FIG. 12
FIG. 13
FIG. 14
FIG. 15
FIG. 16

FIG_17

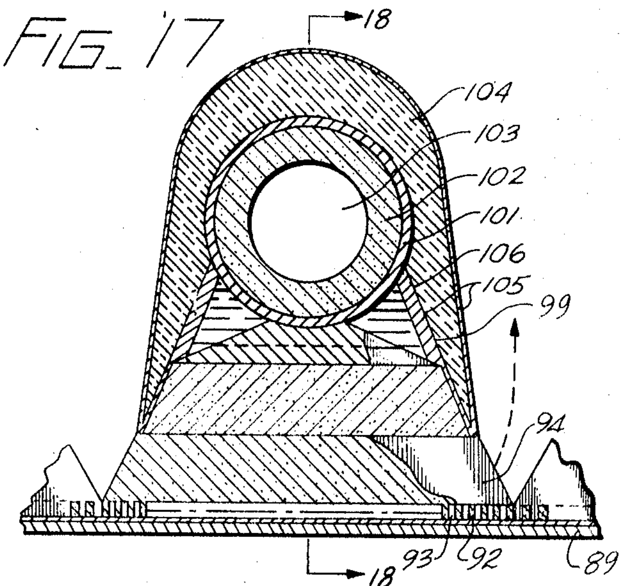



FIG_18

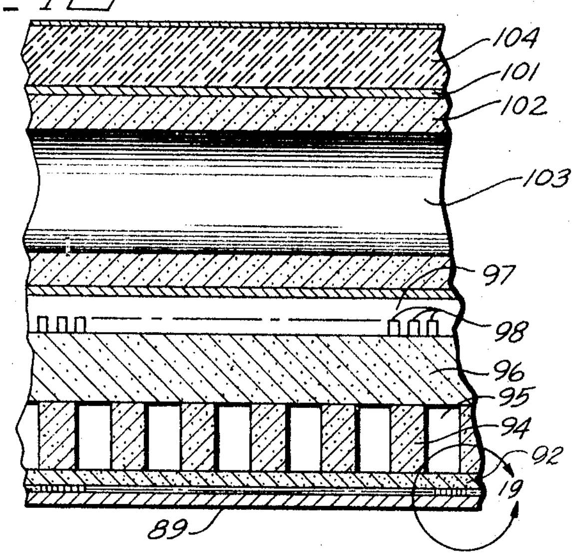
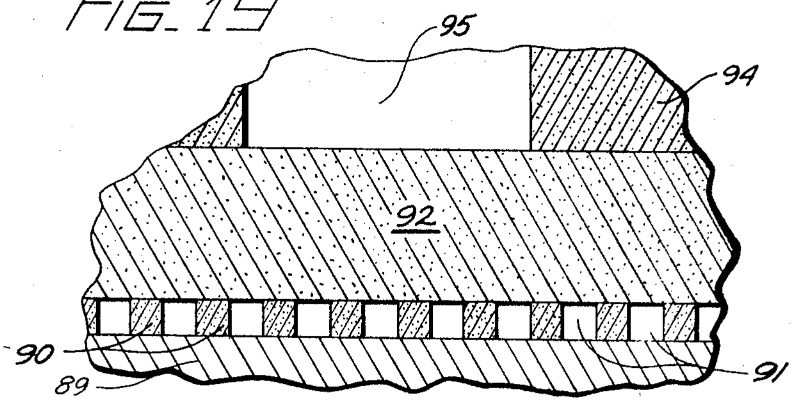



FIG_19



# HEAT TRANSFER SURFACE STRUCTURE

## BACKGROUND

Filed simultaneously herewith are two closely related patent applications by Robert David Moore, Jr., entitled "Segmented Heat Pipe", Ser. No. 52,249 and "The Heat Link, A Heat Transfer Device With Isolated Fluid Flow Paths" Ser. No. 52,642, each of which describes and claims heat transfer apparatus, including heat transfer surface structures incorporating principles of this invention. The content of these copending patent applications is hereby incorporated by reference for full force and effect as if set forth in full herein.

Vaporization heat transfer is widely employed in a broad variety of boilers, heat exchangers, heat pipes and the like. In the ordinary boiler, where bulk liquid and vapor are comingled for contact with the heat transfer surface, vaporization takes place in three well-known stages as the energy input through the solid surface into the liquid is increased. Initially, the liquid is warmed and vaporization takes place from the liquid surface without the formation of bubbles. In the second stage, so-called nucleate boiling occurs wherein vapor bubbles form at the heated surface and pass through the bulk liquid to give a very high heat transfer rate. As the heat flux through the surface into the liquid rises, the vapor bubbles form at a higher rate and at closer spacing so that eventually they form a substantially continuous film of vapor over the surface. The thermal transfer characteristics of the vapor are appreciably lower than that of the liquid, and this so-called film boiling results in a sharp decrease in heat transfer away from the heat input surface. Thus causes a sharp increase in temperature of the surface, and not only is the heat transfer rate decreased, but also there is a danger of damage to the surface due to excessive temperatures. In order to maximize heat flux, it is desirable to prevent the formation of a continuous vapor film at the heat transfer surface.

In a conductively heated capillary vaporizer as in a heat pipe a somewhat different arrangement is employed than in a boiler, since in a boiler vaporization takes place at the heat input surface which is normally just a sheet of metal or the like, and in a heat pipe a porous capillary material is employed for conveying both the liquid and the heat to the surface or region of vaporization. Although the heat flux rates obtainable with a conductively heated capillary vaporizer are quite high, they are limited by the rate at which vapor can escape from the surface or region of vaporization or the tendency of the vapor to drive the liquid out of the hotter portions of the porous matrix, thus cutting off the supply of liquid to the surface or region at which vaporization occurs. These limitations are serious since the volume of vapor formed is quite high compared with the volume of liquid, and the vapor must either be formed at the surface of the porous capillary material following heat conduction through the capillary material, in which case a considerable portion of the capillary material becomes hotter than the vapor, or the vapor must pass through the pores of the capillary material with substantial flow resistance. It is, therefore, desirable to provide a heat transfer surface structure which will deliver heat and liquid to a surface of vaporization and remove vapor from the surface with minimized thermal and fluid flow resistances.

## BRIEF SUMMARY OF THE INVENTION

Thus, in practice of this invention according to a preferred embodiment, there is provided a heat transfer surface on a heat source surface. A capillary matrix, wet by a vaporizable liquid, has a first surface portion in thermal contact with the heat source, a second surface portion in contact with the vaporizable liquid, and a third surface portion from which the principal vaporization of liquid from the capillary matrix takes place. The third surface portion is arranged as a multiplicity of regional areas sufficiently close to each other and to the heat source surface that the volume of the capillary matrix, through which the liquid must pass between the second surface portion and the third surface portion, remains liquid filled. A multiplicity of vapor passages sufficiently larger than the capillary size of the capillary matrix to remain vapor filled are in fluid communication between the regional areas and another region external to the capillary matrix to which vapor is free to flow.



## DRAWINGS

The above mentioned and other features and advantages of the present invention will be better understood by reference to the following detailed description of a presently preferred embodiment when considered in connection with the accompanying drawings wherein:

FIG. A illustrates schematically a vapor-liquid interface at the end of a capillary;

FIG. B illustrates a vapor bubble in a liquid-filled capillary;

FIG. 1 illustrates schematically a porous body for fluid and heat transfer;

FIG. 2 illustrates schematically a conductively heated capillary body for fluid transfer and heat transfer;

FIG. 3 illustrates a simple heat transfer surface structure incorporating principles of this invention;

FIG. 4 illustrates in cross section a compound boiling surface incorporating principles of this invention;

FIG. 5 is another cross section of the structure of FIG. 4;

FIG. 6 illustrates schematically a fragment of heat transfer surface incorporating principles of this invention;

FIGS. 7, 8, 9 and 10 illustrate other embodiments of heat transfer surface structure related to that of FIG. 6;

FIG. 11 illustrates in transverse cross section a wick-fed vaporizer incorporating principles of this invention, such as may be used in a heat pipe;

FIG. 12 is a longitudinal cross section of the vaporizer of FIG. 11;

FIG. 13 is another transverse cross section of the vaporizer of FIG. 11;

FIG. 14 is another longitudinal cross section of the vaporizer of FIG. 11;

FIG. 15 is a perspective view of a fragment of the vaporizer of FIG. 11;

FIG. 16 is a magnified view of a portion of the vaporizer of FIG. 11;

FIG. 17 is a transverse cross section of a fragment of a capillary pump vaporizer structure incorporating principles of this invention;

FIG. 18 is a cross section transverse to that of FIG. 17; and

FIG. 19 is a magnified view of a portion of the structure of FIG. 17.

Throughout the drawings, like reference numerals refer to like parts.

In order to obtain maximum heat flux rates from a vaporizer, it is necessary to allow heat and the vaporizable liquid to reach the liquid-vapor interface at which vaporization of the liquid takes place as easily as possible, and also allow the vapor formed to escape easily. In a boiler or the like where the surface is exposed to bulk liquid, the formation of a continuous vapor film inhibits the flow of vaporizable liquid to the heat transfer surface and the flow of heat to the liquid. Previous arrangements for conductively heated capillary vaporizers have been limited in either the ability of the heat to reach the interface, or the liquid to reach the interface, or in the ability of vapor to leave the interface.

In a typical conventional heat pipe, the capillary vaporizer has a relatively thick layer of porous material adjacent the wall through which heat enters the heat pipe in order to obtain an adequate liquid flow. Both heat and liquid must flow through this material. If the layer of porous material is too thin the resistance to liquid flow is high. At high heat flux the liquid may not reach across the entire porous vaporizer, and excessive temperatures may be encountered in "dry" regions of the vaporizer. If, on the other hand, the porous material is too thick, heat entering the heat pipe through the impermeable wall heats the porous material and the liquid therein in the region adjacent the wall. When the heat flux becomes high, the liquid adjacent the wall may vaporize, but the vapor is effectively trapped by the liquid-filled porous material, and a situation somewhat analogous to film boiling is encountered. In this situation, heat must be conducted through both the vapor-filled porous material and the liquid-filled porous material to evaporate liquid from the free surface. Thus, in either a free boiling situation or in a heat pipe, it is desirable to have means for conveying liquid and heat to the surface where vaporization occurs with as little resistance as possible, and also provide means for removing vapor from the vaporization region with minimum resistance.

In order to thoroughly appreciate the principles and operating advantages of the heat transfer surface structure provided in practice of this invention and to define and develop the art sufficiently that practical use can be made of the invention, the operating principles of conventional and improved capillary vaporizers are discussed herein. Emphasis is given to the limitations on the operation of a capillary vaporizer, particularly the maximum heat flux rates obtainable, in order to confirm the relative and absolute performance of the improved capillary vaporizer and to provide the theoretical basis needed for optimal engineering design. In one of the improved vaporizers, "pumping" can be obtained wherein the vapor leaving the capillary vaporizer is at a higher pressure than the liquid reaching the vaporizer.

The flow of liquid through a porous material under the influence of surface tension forces is analyzed first and the resulting equation applied to demonstrate the performance of a radiantly heated vaporizer. The limiting heat flux of a simple conductively heated vaporizer is then shown to be far lower, on the order of about 1 percent of that of the radiantly heated vaporizer, due to the small temperature differential that can be maintained across the porous material without vapor displacing the liquid from the pores. A generalized improved vaporizer is then described that has numerous vapor passages close to the heat source and close to each other so as to greatly reduce the distance between the heat source and those surfaces of the porous material from which vapor can freely escape. Scaling laws and formulas for the maximum heat flux rate are presented which illustrate the extremely high heat flux capability of the improved vaporizer which, for example, is often over an order of magnitude better than previously available with conventional conductively heated capillary vaporizers or even the maximum available from nucleate boiling. Various geometries of the improved vaporizer are then described and illustrated which allow the new heat transfer surface to be used to advantage in such diverse applications as boiler tubes, refrigerator evaporators, high heat flux heat pipes, and advanced heat transfer systems such as described in the aforementioned copending patent applications.

In the following discussion, an appreciable use of mathematical equations is required for a full understanding of the subject matter to calculate the performance of the improved vaporizer. In the following discussion, symbols for the various quantities and parameters as set forth in the following table are employed. This table not only sets forth the symbols employed and their nature, but also typical units as employed in examples herein.

TABLE OF SYMBOLS

| Symbol | Name or Description | Typical Units |
|---|---|---|
| $\eta$ | viscosity | poise |
| $\sigma$ | surface tension | dyne/cm. |
| $h$ | heat of vaporization/unit volume of liquid | joule/cm.$^3$ |
| $P_v$ | vapor pressure of liquid | dyne/cm.$^2$ |
| $\rho$ | density | gm./cm.$^3$ |
| L | as subscript—liquid | — |
| $v$ | as subscript—vapor | — |
| $w$ | as subscript—wick | — |
| E | as subscript—evaporator or vaporizer | — |
| $k_M$ | heat conductivity of capillary matrix | watts/cm. K |

TABLE OF SYMBOLS—Continued

| Symbol | Name or Description | Typical Units |
|---|---|---|
| $k_S$ | heat conductivity of solid substance forming capillary matrix | watts/cm. K |
| $\alpha$ | fluid conductivity of capillary matrix with unit viscosity fluid | cm.$^2$ |
| $\delta$ | effective matrix pore surface/volume ratio | 1/cm. |
| $\epsilon_w$ | wick microstructure efficiency | — |
| $\epsilon_E$ | vaporizer microstructure efficiency | — |
| G | vaporizer geometrical factor | — |
| P | pressure | dyne/cm.$^2$ |
| $\Delta P_M$ | pressure drop in pores of capillary due to viscous drag | dyne/cm.$^2$ |
| $P_B$ | bubble pressure of liquid filled capillary material | dyne/cm.$^2$ |
| $\Delta P_F$ | pressure differential available to drive liquid through porous material | dyne/cm.$^2$ |
| $\Delta P_G$ | gravitational head at evaporator | dyne/cm.$^2$ |
| $\Delta P_R$ | other pressure differential to overcome viscous drag of fluids | dyne/cm.$^2$ |
| $\Delta P_x$ | sum of pressure drops external to capillary matrix | dyne/cm.$^2$ |
| T | temperature | °K. |
| $\theta$ | contact angle liquid to vaporizer | radians |
| $g$ | acceleration of gravity | cm./sec.$^2$ |
| F | fluid flow rate | cm.$^3$/ sec. |
| H | heat flux | watts |
| $z$ | height of surface of vaporization above bulk liquid surface | cm. |
| R | radius | cm. |
| $R_p$ | pore radius in capillary matrix | cm. |
| A | area | cm.$^2$ |
| $A_p$ | total cross-sectional area of pores | cm.$^2$ |
| $w$ | width of single vaporizer slab | cm. |
| $t$ | thickness of single vaporizer slab | cm. |
| L | length of heat pipe | cm. |
| $x$ | length or distance | cm. |
| $n_p$ | number of vapor passages/unit length | no./cm. |
| $a$ | thickness of microslabs composing vaporizer matrix | cn |
| $b$ | thickness of microchannels in vaporizer matrix | cm. |
| $d$ | distance between vaporizing surface regions | cm. |
| S | length of vaporizer strip-unit area of surface | 1/cm. |

Certain simplifying assumptions are also made throughout the following discussion, and generally speaking the assumptions do not change the results appreciably in practical situations, including the example set forth hereinafter. As with almost all simplifying mathematical assumptions, extreme examples can be found where gross error would result from the assumptions. The typical conventional capillary vaporizers and the improved heat transfer structures hereinafter described are not such examples.

It is assumed that liquid within the porous matrix wets the surface with a contact angle $\theta$ equal to 0. This results only in a mathematical simplicity since if the contact angle is other than 0 the surface tension $\sigma$ is merely replaced by $\sigma'$ which is equal to $\sigma \cos \theta$ in all of the formulas.

The heat conductivity $k_M$ of the whole porous matrix is assumed to be the same whether the pores are filled with liquid or vapor. This is a good approximation leading only to minor inaccuracies for water or organic fluids in a matrix of high thermal conductivity metal such as silver, aluminum, copper or gold. Where this approximation is poor, as in the case where the fluid is a high-heat conductivity liquid metal, the parametric constants obtained when solving equations for limiting heat flux in a conductively heated vaporizer may change, but the mathematical form of the equations does remain the same.

The porous material is assumed to have uniform pore size, fluid conductivity, bubble pressure, heat conductivity and the like throughout its defined boundaries, though in some circumstances, as pointed out hereinafter, two or more porous materials with different parameters are employed for improved performance. Practical, presently available porous materials rarely have uniform pore size, but calculations based on statistically varying pore size distributions are unnecessarily complicated and in more circumstances would merely introduce a geometric factor that would not alter the comparative merit of one structure as compared with another. If it is desired to obtain a more exact solution, the fluid conductivity $\alpha$ may be measured empirically as a function of the difference between vapor pressure and the fluid pressure in the matrix with the results employed directly in a numerical solution rather than calculating the fluid conductivity from pore size distributions.

Two idealized geometrical structures are utilized in the mathematical treatment, a wick matrix having a honeycomb like structure that is essentially of a bundle of uniform parallel tubes with very thin walls, and a vaporizer matrix comprising closely spaced parallel plates of high thermal conductivity which when optimized turn out to have a plate thickness the same as the width of the gaps or channels between adjacent plates. While most presently available capillary matrix structures for either wicks or vaporizers do not approach these ideal structures closely, calculations based on the ideal structures determine the relationships between fluid conductivity $\alpha$, effective matrix pore surface to volume ratio $\delta$, and the ratio of the heat conductivity of the capillary matrix to the heat conductivity of the solid substance forming the capillary matrix ($k_M/k_S$), and also identify ideal performance figures with which other real matrix structures can be compared in order to rate their "efficiency".

Analyzing first the flow through a porous material induced by surface tension forces as applied to capillary vaporizers, the flow rate per unit area F/A of a fluid of viscosity $\eta$ through a porous material having fluid conductivity $\alpha$ under a pressure gradient $dp/dx$ is

$$F/A=\frac{\alpha}{\eta}\frac{dP}{dx} \quad (1)$$

Also the maximum static pressure differential or bubble pressure $\Delta P_B$ that can be supported across a liquid vapor interface in a porous material having an effective pore surface to pore volume ratio $\delta$ and filled with a liquid wetting the porous material with a zero contact angle, and having a surface tension $\sigma$ is

$$\Delta P_{\mathrm{B}}=\sigma\delta \quad (2)$$

The meaning of the bubble pressure $\Delta P_B$ and the effective pore surface to volume ratio $\delta$ is appreciated by consideration of FIGS. A and B depicting a liquid having a surface tension $\sigma$ at pressure $P_L$ in a pore of radius R which is terminated, as at the surface of a capillary material, in the illustration of FIG. A, or which has a vapor bubble in the pore as in the illustration of FIG. B. In both cases the pore is assumed to have been evacuated prior to filling with liquid so that no gasses are present except for the vapor from the liquid. The liquid is assumed to wet the pore wall with a zero contact angle and the pore, liquid, vapor and surroundings are all at temperature T. The liquid has a vapor pressure $P_v$ at temperature T so that vapor at this pressure fills all spaces not occupied by the liquid.

It is obvious that if the pressure in the liquid $P_L$ is greater than the vapor pressure $P_v$, the liquid will flow to the right in FIG. A and will collapse the vapor bubble in FIG. B, in both cases replacing the vapor and forcing it to condense. What is less obvious is that the liquid will remain in the capillary tube of FIG. A and the vapor bubble will collapse in FIG. B even when the vapor pressure $P_v$ is greater than the liquid pressure $P_L$ so long as the sum of the liquid pressure $P_L$ and the bubble pressure $P_B$ is greater than the vapor pressure $P_v$. This is due to the pressure exerted by the surface tension $\sigma$ in the curved liquid-vapor interface, which acts like a stretched elastic membrane. The pressure difference $\Delta P$ across the liquid-vapor interface is

$$\Delta P=\sigma\left(\frac{1}{R_1}+\frac{1}{R_2}\right)$$

where $R_1$ and $R_2$ are the principal radii of the curvature of the interface. For a circular pore as illustrated in FIGS. A and B, $R_1=R_2=R_L$, the radius of the spherical interface.

The bubble pressure $P_B$ is the maximum pressure that can be supported across the interface

$$P_{\mathrm{B}}=\sigma\left(\frac{1}{R_{1\,\mathrm{min}}}+\frac{1}{R_{2\,\mathrm{min}}}\right)$$

In a pore of radius R, $R_{1min}=R_{2min}=R$, so that the bubble pressure, $P_B=2\sigma/R$. In an infinite plane slot between parallel surfaces spaced apart a distance $b$, $R_{1min}=b/2$ and $R_{2min}$ equals infinity so that the bubble pressure $P_B=2\sigma/b$.

The surface to volume ratio $\delta$ of the circular pore is $\delta_{pore}=2\pi R/\pi R^2=2/R$. The surface to volume ratio of the infinite slot is $\delta_{slot}=2/b$. Thus, in equation 2, $P_{Btube}=\sigma\delta_{tube}$ and $P_{Bslot}=\sigma\delta_{slot}$. The relation of equation 2 is valid, in general, for most uniform rounded cross sections without sharp corners or reverse curved regions. Even in structures not fitting these assumptions, an effective pore surface to volume ratio $\delta$ defined as $\delta=P_B/\sigma$ is useful. Both $P_B$ and $\sigma$ are easily measured and $\delta$ is dependent only on the shape and size of the pores, that is, calculation of $\delta$ from measurements of $P_B$ using fluids of different $\sigma$ will always result in the same value of $\delta$ for a given capillary material.

Since the liquid-vapor interface can support a pressure differential up to the bubble pressure $P_B=\sigma\delta$, a pore will remain filled with liquid and any bubble formed in it will collapse so long as $P_L+P_b>P_v$ or $P_L+\sigma\delta>P_v$. This is the prime condition for stable existence of liquid in the pores.

The heat flow per unit area H/A through a porous matrix of heat conductivity $k_M$ with a temperature gradient $dt/dx$ is

$$H/A=k_{\mathrm{M}}\frac{dT}{dx}=\left(\frac{k_{\mathrm{M}}}{k_{\mathrm{S}}}\right)k_{\mathrm{S}}\frac{dT}{dx} \quad (3)$$

where $k_S$ is the heat conductivity of the solid substance out of which the capillary matrix is constructed.

The parameters $\eta$, $\sigma$ and $k_S$ are properties of the materials used in a particular embodiment, while the parameters $\alpha$, $\delta$ and $k_M/k_S$ are properties of the microstructure of the capillary matrix. The first set of parameters is determined by the choice of materials employed in a particular example. It is, however, feasible to determine that matrix structure giving the best possible combination of $\alpha$, $\delta$ and where necessary $k_M/k_S$ for a particular example chosen. In order to approach this analysis on a stepwise basis, the matrix structure for a wick will be analyzed first since it does not involve the additional complicating factor of heat conduction as is present in a vaporizer.

The ideal wick matrix, as mentioned hereinabove, comprises a bundle of hexagonal tubes with infinitesimally thin walls forming a honeycomb structure. The fluid conductivity of each hexagonal tube is approximately the same as that of a circular tube of the same area, so that for this analysis the matrix will be treated as a bundle of circular tubes of radius $R_P$ having a total internal area A equal to the cross-sectional area of the matrix. It might be noted that a hexagonal tube has only about 5 percent more actual wall area and about 2 ½ percent more effective wall area for determining the effective matrix pore surface to volume ratio $\delta$ so that this is not a damaging assumption. The ratio of wall area to volume $\delta$ of a circular pore is

$$\delta=\frac{2\pi R_{\mathrm{p}}}{\pi R_{\mathrm{p}}^2}=\frac{2}{R_{\mathrm{p}}} \quad (4)$$

It should be noted that the ratio $\delta$ for a bundle of N pores is the same as that for a single pore since both the surface and volume (actual cross-sectional area) are multiplied by N.

The fluid conductivity $\alpha$ of the matrix for a fluid of unit viscosity is the fluid conduction of a single pore divided by the area of the pore

$$\alpha=\frac{F_{\mathrm{p}}}{\pi R^2\frac{dP}{dx}}=\left(\frac{1}{\pi R_{\mathrm{p}}^2\frac{dP}{dx}}\right)\frac{\pi R_{\mathrm{p}}^4}{8}\frac{dP}{dx}=\frac{R_{\mathrm{p}}^2}{8} \quad (5)$$

Optimally both the fluid conductivity $\alpha$ and the effective pore surface to volume $\delta$ should be as great as possible; however, there is a limitation independent of the radius of the pore that

$$\epsilon_{\mathrm{w}}\equiv 2\alpha\delta^2=\frac{2R_{\mathrm{p}}^2}{8}\cdot\frac{4}{R_{\mathrm{p}}^2}=1 \quad (6)$$

For other microstructures than the thin-wall hexagonal tubes $\epsilon_w$ is always less than 1. Thus, while the wick matrix microstructure efficiency is independent of a change of scale or size, that is $\epsilon_w$ will remain the same if the entire matrix is uniformly expanded or shrunk, this is not true of the surface to volume ratio $\delta$ or the fluid conductivity $\alpha$. Thus, within the limit imposed by the wick matrix efficiency $\epsilon_w$ remaining constant, it is possible to optimize the fluid conductivity $\alpha$ and surface to volume ratio $\delta$ for any given matrix microgeometry by simply changing the scale since if $x$ is an arbitrary dimension in an arbitrary microgeometry, then $\delta$ equal $C_1/x$ and $\alpha$ equal $C_2x^2$, where $C_1$ and $C_2$ are constants dependent only on the shape and independent of the size $x$.

To illustrate the usefulness of the wick matrix efficiency $\epsilon_w$ and how the parameters introduced so far determine the operation of a simple type of heat pipe, the maximum heat flux capacity of such a heat pipe is determined. It is assumed that this heat pipe comprises a cylinder of capillary material of length L and cross section A with the bottom surface just touching the surface of a pool of liquid of density $\rho$, surface tension $\sigma$, and viscosity $\eta$ which wets the capillary material with a zero contact angle. The top end of the cylinder lies a distance $z$ above the liquid surface (the cylinder is not necessarily vertical), and the top end is radiantly heated so as to evaporate liquid from this surface. All of this above structure is enclosed and purged so as to be free from any residual gases other than vapor of the liquid. The pressure differential $\Delta P_F$ necessary to drive a flow F of fluid through the capillary material is

$$\Delta P_F=\frac{FL\eta}{\alpha A} \qquad (7)$$

and the pressure differential $\Delta P_g$ necessary to overcome the gravitational head is

$$\Delta P_g=\rho g z \qquad (8)$$

Thus, the pressure differential $\Delta P_B$ that must be provided by the capillary forces is

$$\Delta P_B=\sigma\delta=\Delta P_F+\Delta P_g=\frac{FL\eta}{\alpha A}+\rho g z \qquad (9)$$

or solving for the fluid flow rate F

$$F=\frac{\alpha A}{L\eta}(\sigma\delta-\rho g z) \qquad (10)$$

The maximum heat flux capacity of the heat pipe given a fluid heat of evaporation per unit volume of $h$ is

$$H=hF=\frac{\alpha A h}{L\eta}(\sigma\delta-\rho g z) \qquad (11)$$

Assuming that the scale factor of the microgeometry of the matrix, which has an efficiency factor of $\epsilon_w$, can be adjusted so as to maximize the heat flux H, since

$$\alpha=\frac{\epsilon_w}{2\delta^2} \qquad (12)$$

$$H=\frac{\epsilon_w A h}{2L\eta\delta^2}(\sigma\delta-\rho g z) \qquad (13)$$

The value of the effective matrix pore surface to volume ratio $\delta$ that maximizes the heat flux is found by setting $dH/d\delta$ equal 0. Finding

$$\delta_{maxH}=\frac{2\rho g z}{\sigma} \qquad (13a)$$

and

$$H_{max}=\frac{\epsilon_w h\sigma^2 A}{8\eta\rho g z L} \text{ or } \frac{H_{max}}{A}=\frac{\epsilon_w h\sigma^2}{8\eta\rho g z L} \qquad (14)$$

Thus, when one sets the scale of the matrix microstructure so as to maximize the heat flux H, it is the matrix wick efficiency $\epsilon_w$ that remains to determine the maximum heat flux capability $H_{max}$ of the heat pipe. This is true not only in this particular example but in all cases where the capillary matrix acts purely as a wick for liquid flow, that is where it does not also transmit heat as part of its function. In the more general case where a pressure differential $\Delta P_R$ must also be present for the circulation of vapor within the heat pipe and possibly for circulating liquid in a portion external to the wick equation 14 is readily modified by adding $\Delta P_R$ to the $\rho g z$ term and letting $\Delta P_x=\Delta P_R+\rho g z$, that is $\Delta P_x$ is set equal to all of the "external" pressure differences in the total fluid flow system, that is, all the pressure differences except that required to overcome the viscous drag in driving the liquid through the capillary matrix being considered. Then

$$H_{max}=\frac{\epsilon_w h\sigma^2 A}{8\eta L\Delta P_x} \text{ or } \frac{H_{max}}{A}=\frac{\epsilon_w h\sigma^2}{8\eta L\Delta P_x} \qquad (15)$$

where $\delta_{maxH}=2\Delta P_x/\sigma$. Since $\Delta P_R$ is generally a function of the vapor flow rate in the heat pipe and hence of the total heat flux, (15) is generally at least cubic in terms of the heat flux H, and the equation is most easily solved by numerical approximation rather than an analytic solution.

In the example just set forth, the liquid is vaporized from a surface opposite to the surface from which liquid is introduced to the wick. An almost identical set of formulas results when the liquid is evaporated from a side adjacent to the side where the liquid is introduced. FIG. 1 illustrates a rectangular slab of capillary material with a width $w$ and thickness $t$ with $w$ much greater than $t$ and of indefinite length $x$. Liquid is introduced to the capillary material at the side having a cross sectional area $tx$ and evaporated from the adjacent face having an area $wx$, with the heat for vaporization being supplied to that face by radiant heating, for example. Thus, the maximum heat flux capability of the capillary material is

$$H=\frac{\epsilon_w h t x}{\eta w\delta^2}(\sigma\delta-\Delta P_x) \text{ or } \frac{H}{x}=\frac{\epsilon_w h t}{\eta w\delta^2}(\sigma\delta-\Delta P_x) \qquad (16)$$

When the effective matrix pore surface to volume ratio $\delta$ is selected to maximize the heat flux H

$$H_{max}=\frac{\epsilon_w h t\sigma^2 x}{4\eta w\Delta P_x} \text{ or } \frac{H_{max}}{x}=\frac{\epsilon_w h t\sigma^2}{4\eta w\Delta P_x} \qquad (17)$$

where once again $\delta_{maxH}=2\Delta P_x/\sigma$. Equations 16 and 17 are only strictly accurate when $w$ is very much greater than $t$ and can, where more accuracy is required, incorporate a geometrical shape factor $G_R$ which is always less than 1 but approaches 1 as $t/w$ approaches zero. The shape factor $G_R$ must in general be calculated numerically for each particular case and when included in the equations the heat flux is found by

$$\frac{H}{x}=\frac{G_R\epsilon_w h t}{\eta w\delta^2}(\sigma\delta-\Delta P_x) \qquad (18)$$

$$\frac{H_{max}}{x}=\frac{G_R\epsilon_w h t\sigma^2}{4\eta w\Delta P_x}<\frac{\epsilon_w h\sigma^2}{4\eta\Delta P_x}\leq\frac{h\sigma^2}{4\eta\Delta P_x} \qquad (19)$$

The function of the capillary matrix is considerably different and somewhat more complicated when it is used as a conductively heated vaporizer rather than merely a simple wick since it must conduct both liquid and heat to the surface at which the liquid vaporizes so that the heat conductivity $k_M$ of the matrix becomes important. The surface tension forces between the liquid and matrix must also support an additional pressure differential since portions of the matrix must be hotter and therefore the liquid vapor pressure in that portion greater than at a surface of vaporization. This is so for causing heat to flow to the surface of vaporization from the hotter region.

The only exception to this situation in a conductively heated vaporizer (that is, where the heat is transferred to the surface of vaporization by conduction through the porous matrix and not by radiation or convection) is when the surface of vaporization lies within the capillary matrix between the heat and liquid sources. Under these circumstances, the vapor must escape from the capillary matrix by flowing through the pores in the matrix. This is a severe restriction on operation as can be seen from the situation for water and water vapor at 100° C where the mass flow rate of liquid water through a porous material under a selected pressure differential is about seventy times the mass flow rate of water vapor through the same porous material under the same pressure differential. Thus is due to the much lower density of the water vapor and thus the much higher flow velocity required to give the same mass flow rate. The higher velocity far outweighs the effect of the lower viscosity of the vapor. Also, when the vaporization surface is between the heat source and the liquid source, the vapor produced may be largely blocked from escaping since it cannot flow through the same capillaries occupied by the liquid. Hence, such an arrangement is not practical structure for most situations. The examples set forth herein are thus limited to structures where the surface of vaporization coincides with a surface of the porous matrix from which vapor may easily escape.

The simplest structure of this type comprises a substantially rectangular strip of capillary material of width $w$, thickness $t$ and length $x$ which is fed with liquid through one face of area $tx$ and heated through one of the adjacent two faces having area $wx$. So far this is exactly the same as the case of the radiantly heated vaporizer strip hereinabove described. In this latter case, however, the face $wx$ is heated conductively through an impervious wall adjacent the face so that vapor can no longer escape from that face but must escape from the opposite face as illustrated in FIG. 2.

A heat flux density $H/A_v=H/wx$ must pass through the strip normal to the face wx requiring a temperature gradient $dT/dy$ to cause heat flow, where

$$\frac{dT}{dy}=\frac{H}{k_M wx}=\left(\frac{H}{x}\right)\frac{1}{k_M w} \qquad (20)$$

The increase in vapor pressure $\Delta P_v$ of the liquid between $y=0$ and $y$ where $y$ is the distance of the liquid-vapor interface from the free or vaporizing surface, is

$$\Delta P_v=\left(\frac{dP_v}{dT}\right)\Delta T=\frac{H}{x}\,\frac{y(dP_v/dT)}{k_M w} \qquad (21)$$

The pressure gradient $dP_M/dz$ in the liquid necessary to force the liquid through the matrix is

$$\frac{dP_M}{dz}=\frac{-H(1-z/w)\eta}{xh\alpha y}=\frac{-2H\delta^2\eta(1-z/w)}{xh\epsilon_w y} \qquad (22)$$

$$\Delta P_M=-\int_0^z \frac{dP_M}{dz}dz=\frac{-2H\delta^2\eta}{xh\epsilon_w}\int_0^z \frac{(1-z/w)}{y}dz \qquad (23)$$

The pressure in the matrix $P_M$ is assumed to be a function of only $z$, that is the liquid flows only in the $x$ direction. This is an approximation that becomes exact as the thickness decreases and width increases, that is $t/w$ approaches zero.

The position $y$ of the internal liquid-vapor interface for maximum heat flux is determined from equations 20 and 22 since along the liquid-vapor interface $\Delta P_v-\Delta P_M$ equals a constant. AT maximum heat flux $H_{max}$ the liquid filled portion of the capillary will just reach the end in a thin wedge. That is, the thickness $y=0$ at a distance $z=w$. This can be seen since if $y>0$ at the far edge of the matrix, then the heat flux could be increased and, on the other hand, if $y<0$ at the far edge, then the heat flux density must be decreased until the liquid extends across the entire capillary matrix. Also $dP_M/dz=dP_v/dz$. Therefore equating the derivitives and solving for $dy/dz$

$$\frac{dy}{dz}=-\frac{2\delta^2\eta k_M}{h\epsilon_w\frac{dP_v}{dT}}\,\frac{(w-z)}{y} \qquad (24)$$

thus,

$$\int y\,dy=\frac{2\delta^2\eta k_M}{h\epsilon_w\frac{dP_v}{dT}}\int(w-z)dz \qquad (25)$$

Integrating Equation 25, solving for $y$,

$$\frac{y^2}{2}=\frac{2\delta^2\eta k_M}{h\epsilon_w\frac{dP_v}{dT}}\left(wz-\frac{z^2}{2}+C\right) \qquad (26)$$

When $z=w$, and $wz-\frac{z^2}{2}+C=0$, so that $C=-w^2/2$,

$$y=w\delta\left(\frac{2\eta k_M}{h\epsilon_w\frac{dP_v}{dT}}\right)^{1/2}\left(1-\frac{z}{w}\right) \qquad (27)$$

The thickness $y$ is then substituted back into equations 21 and 23 to find

$$\Delta P_v=\frac{H\frac{dP_v}{dT}w\delta}{xk_M w}\left(\frac{2\eta k_M}{h\epsilon_w\frac{dP_v}{dT}}\right)^{1/2}\left(1-\frac{z}{w}\right)=\frac{H\delta}{x}\left(\frac{2\eta\frac{dP_v}{dT}}{h\epsilon_w k_M}\right)^{1/2}\left(1-\frac{z}{w}\right) \qquad (28)$$

$$\Delta P_M=-\frac{2H\delta^2\eta}{xh\epsilon_w w\delta}\left(\frac{h\epsilon_w\frac{dP_v}{dT}}{2\eta k_M}\right)^{1/2}\int_0^z\frac{\left(1-\frac{z}{w}\right)}{\left(1-\frac{z}{w}\right)}dz=-\frac{H\delta z}{xw}\left(\frac{2\eta\frac{dP_v}{dT}}{h\epsilon_w k_M}\right)^{1/2} \qquad (29)$$

$$\Delta P_v-\Delta P_M=\frac{H\delta}{x}\left(\frac{2\eta\frac{dP_v}{dT}}{h\epsilon_w k_M}\right)^{1/2}\left[1-\frac{z}{w}+\frac{z}{w}\right]=\frac{H\delta}{x}\left(\frac{2\eta\frac{dP_v}{dT}}{h\epsilon_w k_M}\right)^{1/2} \qquad (30)$$

In some situations, the liquid must be lifted into the vaporizer region of the capillary matrix against a gravity head $\Delta P_g=\rho gz$, and also against a pressure differential $\Delta P_R$ necessary to cause liquid and vapor flow around the remaining fluid flow circuit of a heat pipe or the like. Then assuming that $\Delta P_x$ is the sum of the gravity head, driving pressure differential, and other pressure drops external to the capillary matrix, there is a pressure balance at maximum heat flux where

$$P_B=\sigma\delta=\Delta P_x+\Delta P_v-\Delta P_M \qquad (31)$$

$$\sigma\delta-\Delta P_x=\frac{H_{max}\delta}{x}\left(\frac{2\eta\frac{dP_v}{dT}}{h\epsilon_w k_M}\right)^{1/2} \qquad (32)$$

$$\frac{H_{max}}{x}=\left(\frac{h\epsilon_w k_M}{2\eta\frac{dP_v}{dT}}\right)^{1/2}\left(\frac{\sigma\delta-\Delta P_x}{\delta}\right)=\sigma\left(\frac{h\epsilon_w k_M}{2\eta\frac{dP_v}{dT}}\right)^{1/2}\left(1-\frac{\Delta P_x}{\sigma\delta}\right) \qquad (33)$$

Equation 31 is true when the maximum value of $y$ is equal to or less than the thickness $t$ so that

$$t\geq w\delta\left(\frac{2\eta k_M}{h\epsilon_w\frac{dP_v}{dT}}\right)^{1/2}\ \text{or}\ \frac{w\delta}{t}\left(\frac{2\eta k_M}{h\epsilon_w\frac{dP_v}{dT}}\right)^{1/2}\leq 1 \qquad (34)$$

It should be recognized that equations 33 and 34 are approximations since the pressure in the matrix $P_M$ is assumed to be a function of $z$ only, that is $dP_M/dy=0$. This is strictly true only when the liquid flow is normal to the heat flow at all points, which is the case when the ratio of thickness to width $t/w$ is very small. For structures where $t/w$ is relatively large, geometrical correction factors $G_1$ and $G_2$ can be added to the equations. These correction factors are dependent only on the geometry and approach 1 when the liquid flow becomes nearly perpendicular to the heat flow. Generally $G_1$ and $G_2$ must be calculated numerically or by analog means.

Equations similar to Equations 33 and 34 can be derived for any strip geometry (at least where the faces are not extremely concave) that can be formed from the rectangular geometry hereinabove described by stretching or shrinking the strips in such a manner that they are stretched or shrunk in the same amount in the $y$ direction as in the $z$ direction at each point. The same limitation that the heat and liquid flows are substantially normal to each other still exists. The resulting equations, including the geometric factors, are

$$\frac{H_{max}}{x}=G_1\sigma\left(\frac{h\epsilon_w k_M}{2\eta\frac{dP_v}{dT}}\right)^{1/2}\left(1-\frac{\Delta P_x}{\sigma\delta}\right) \quad (35)$$

$$G_2\delta w\left(\overline{\frac{1}{t}}\right)\left(\frac{2\eta k_M}{h\epsilon_w\frac{dP_v}{dt}}\right)^{1/2}\leqq 1 \quad (36)$$

where

$$\left(\overline{\frac{1}{t}}\right)\equiv\frac{1}{w}\int_0^w\left(\frac{1}{t}\right)dx \quad (36A)$$

replaces $1/_t$ in equation 34.

A truncated wedge geometry formed from a sector of a flat circular disk of radius $R_1$ and truncated at radius $R_0$ with liquid applied to the arcuate face and the heating and vaporizing surfaces being the two radial faces forms a particularly useful example of the generalized geometry. Here letting $dy=d\theta$ and $dx=dR/R$ so that

$$\left(\overline{\frac{1}{y}}\right)=\left(\overline{\frac{1}{\theta}}\right)=\frac{1}{\theta} \quad (37A)$$

and

$$w=\int_{R_0}^{R_1}dx=\int_{R_0}^{R_1}\frac{dR}{R}=\ln\,(R_1/R_0) \quad (37B)$$

where $\overline{1/t}$ is the mean value of $1/t$.

Thus equation 35 remains unchanged and equation 36 becomes

$$\frac{G_2\delta\,\ln\,[R_1/R_0]}{\theta}\left(\frac{2\eta k_M}{h\epsilon_w\frac{dP_v}{dT}}\right)^{1/3}\leqq 1 \quad (38)$$

The general equation 35 for maximum heat flux as limited in range by equation 36, is applicable to substantially all of the structures of interest. Equation 36 limits the ratio of effective matrix pore surface to volume ratio $\delta$ which can be considered and thus limits the pressure difference $\Delta P_x$ external to the vaporizer against which the vaporizer can operate since $\Delta P_x<\sigma\delta$. This is not a significant limitation in most vaporizer designs, such as in heat pipes or surfaces exposed directly to bulk liquid, since in these cases $\Delta P_x$ is small.

A general equation similar to equation 36 for the situation where

$$G_2\delta w\left(\overline{\frac{1}{t}}\right)\left(\frac{2\eta k_M}{h\epsilon_w\frac{dP_v}{dT}}\right)^{1/2}\geq 1 \quad (39)$$

is

$$\frac{H}{x}=G_1\sigma\left(\frac{h\epsilon_w k_M}{2\eta\frac{dP_v}{dT}}\right)^{1/2}\frac{\left(1-\frac{\Delta P_x}{\sigma\delta}\right)}{\frac{1}{2}\left(C\frac{\sigma\delta}{\Delta P_x}+\frac{1}{C}\frac{\Delta P_x}{\sigma\delta}\right)} \quad (40)$$

where

$$C=\frac{G_2 w\left(\overline{\frac{1}{t}}\right)\Delta P_x}{\sigma}\left(\frac{2\eta k_M}{h\epsilon_w\frac{dP_v}{dT}}\right)^{1/2} \quad (40A)$$

and when equation 40 is maximized with respect to $\delta$,

$$\frac{H_{max}}{x}=G_1\sigma\left(\frac{h\epsilon_w k_M}{2\eta\frac{dP_v}{dT}}\right)^{1/2}\left(\frac{1}{C+\sqrt{1+C^2}}\right) \quad (41)$$

$$\delta_{max}=\frac{\Delta P_x}{\sigma C}\left(C+\sqrt{1+C^2}\right) \quad (42)$$

The geometrical factors $G_1$ and $G_2$ in these equations, while both approaching 1 as the liquid flow direction approaches solely dependent upon the geometry but are also somewhat dependent on C (equation 40A). This is true since the internal liquid-vapor interface position for maximum heat flux varies with C in this case, thus changing the localized flow geometry. Thus is to be contrasted with equations 35 and 36 wherein the flow geometry for maximum heat flux is independent of any other variables.

When C is small the values for $H_{max}/x$ given by equation 41 closely approach those given by Equation 35. For the case when C is very large, the maximum heat flux $H_{max}/X$ approaches

$$G_1\sigma\left(\frac{h\epsilon_w k_M}{2\eta\frac{dP_v}{dT}}\right)^{1/2}\left(\frac{1}{2C}\right)=\left(\frac{G_1}{G_2}\right)\frac{\epsilon_w h\sigma^2}{4\eta\Delta P_x w\left(\overline{\frac{1}{t}}\right)} \quad (43)$$

which is the same as Equation 19 for the radiantly heated evaporator where $G_1/G_2=G_R$ and $(\overline{1/t})=1/t$. This is to be expected since the liquid fills almost the entire porous matrix at high values of C.

Due to its relative simplicity, Equation 35 for maximum heat flux $H_{max}/x$ is employed hereinafter although equation 41 can be substituted in its place as desired in applications involving high values of $\Delta P_x$ or C.

The temperature drop $\Delta T_v$ across the vaporizer capillary matrix between the heated surface and the surface where liquid is vaporizing is

$$\Delta T_v=\frac{H}{x}\left(\frac{1}{k_M w\left(\overline{\frac{I}{t}}\right)}\right) \quad (44)$$

This temperature drop is generally quite small, being on the order of only a fraction up to several degrees centigrade, with the higher values corresponding to higher values of heat flux $H/x$ and external pressure drop $\Delta P_x$.

In order to provide comparisons of performance parameters in a vaporizer surface structure, an efficiency factor $\epsilon_E$ for the matrix structure is desirable. The efficiency $\epsilon_E$ of a selected capillary matrix microstructure is defined as the ratio of the maximum heat flux $H_{max}/x$ for a vaporizer constructed with the capillary matrix microstructure relative to that of a vaporizer constructed with an ideal capillary matrix microstructure. Both matrices are made of the same material and all the other conditions are identical except for the microstructure. To implement this an ideal matrix microstructure geometry must be selected.

In equation 35, the only parameters dependent on the capillary matrix microstructure are $\epsilon_w$ and $k_M$. Further $\epsilon_w$ is dependent only on the microstructure, while $k_M$ is divisible into two factors, the heat conductivity $k_S$ of the material the matrix is constructed of, and the relative heat conductivity of the matrix $k_M/k_S$ which is dependent only on the matrix structure. Thus, the ideal geometry is one that maximizes the factor $\epsilon_w k_M/k_S$ which is proportional to $\alpha\delta^2k_M/k_S$. A suitable standard or "ideal" vaporizer microstructure is one composed of parallel sheets of heat conducting material of thickness "$a$" mutually spaced apart to form channels therebetween of width "$b$." For this structure,

$$\frac{k_M}{k_S}=\frac{a}{a+\beta} \quad (45)$$

$$\delta=2/b \quad (46)$$

$$\alpha=\frac{b^2}{12}\left(\frac{b}{a+b}\right) \quad (47)$$

$$\delta=\left(\alpha\frac{k_M}{k_S}\right)^{1/2}=2\left(\frac{ab}{12(a+b)^2}\right)^{1/2} \quad (48)$$

The right-hand term in equation 48 is maximized when $a=b$ and then

$$\left(\frac{k_M}{k_S}\right)_{maxH}=1/2 \quad (49)$$

$$\delta_{maxH}=2/b \quad (50)$$

$$\alpha_{\max H}=b^2/24 \tag{51}$$

$$\epsilon_w=2\alpha\delta^2=1/3 \tag{52}$$

$$\delta\left(\alpha\frac{k_M}{k_X}\right)^{1/2}=\left(\frac{1}{12}\right)^{1/2} \tag{53}$$

The evaporator matrix efficiency factor $\epsilon_E$ is then defined so as to equal 1 for the "ideal" vaporizer matrix microstructure

$$\epsilon_E=\delta\left(\frac{12\alpha k_M}{k_S}\right)^{1/2}=\left(\frac{6\epsilon_w k_M}{k_S}\right)^{1/2} \tag{54}$$

Using equation 54, equations 35 and 36 for maximum heat flux are rewritten in terms of the efficiency factor $\epsilon_E$ as

$$\frac{H_{\max}}{x}=G_1\epsilon_E\sigma\left(\frac{hk_S}{12\eta\frac{dP_v}{dT}}\right)^{1/2}\left(1-\frac{\Delta P_x}{\delta\sigma}\right)\leq G_1\epsilon_E\sigma\left(\frac{hk_S}{12\eta\frac{dP_v}{dT}}\right)^{1/2} \tag{55}$$

$$\frac{G_2\delta w\left(\frac{\bar{1}}{y}\right)k_M}{\epsilon_E k_S}\left(\frac{12\eta k_S}{h\frac{dP_v}{dT}}\right)^{1/2}\leqq 1 \text{ or } \sigma\delta\leqq\frac{G_2\epsilon_E\sigma k_S}{w\left(\frac{\bar{1}}{t}\right)}\left(\frac{h\frac{dP_v}{hT}}{12\eta k_S}\right)^{1/2} \tag{56}$$

It is instructive to compare the maximum possible heat flux per unit length of the radiantly heated vaporizer (equation 19) with that of the conductively heated vaporizer, assuming ideal situations where $\epsilon_w=\epsilon_E=1$; $G_C=1$; $G_R t/w=1$

$$\frac{\left(\frac{H_{\max}}{x}\right)_{\text{radiant}}}{\left(\frac{H_{\max}}{x}\right)_{\text{conduct}}}=\left(\frac{h\sigma^2}{4\eta\Delta P_x}\right)\frac{1}{\sigma}\left(\frac{12\eta\frac{dP_v}{dT}}{hk_S}\right)^{1/2}$$

$$=2\frac{\sigma}{\Delta}P_x\left(\frac{3h\frac{dP_v}{dT}}{\eta k_S}\right)^{1/2} \tag{57}$$

Substituting the appropriate numerical values for water and copper at 100° C.

$$\frac{\left(\frac{H_{\max}}{x}\right)_{\text{radiant}}}{\left(\frac{H_{\max}}{x}\right)_{\text{conduction}}}=\frac{5.9\times 10}{2(\Delta P_x)}\left(\frac{3\times 2.16\times 10^3\times 3.68\times 10^4}{2.8\times 10^{-3}\times 3.8}\right)^{1/2}$$

$$=\frac{4.4\times 10^6}{\Delta P_x} \tag{58}$$

In many cases, the total pressure head $\Delta P_x$ is in the order of from about $10^4$ to $10^5$ dynes per square centimeter. That is equivalent to a pressure head of from about 10 to 100 centimeters of water and

$$\frac{\left(\frac{H_{\max}}{x}\right)_{\text{radiant}}}{\left(\frac{B_{\max}}{x}\right)_{\text{conduction}}}\cong 440 \text{ to } 44 \text{ respectively} \tag{59}$$

Thus, except at unusually high gravity heads or other external pressure drops, a conductively heated vaporizer strip is limited to far lower heat flux density than a comparable radiantly heated vaporizer strip. Since in most cases of interest, the object of the heat transfer surface is to remove heat from a hot surface or body where the heat radiated is far less than the heat it is necessary to remove, the heat usually must be transferred to the vaporizer conductively. Thus, any means for overcoming the heat flux limitation of the conductively heated vaporizer is desirable as leading to greatly increased utility for the structure.

The new type of conductively heated capillary vaporizer surface, described herein, avoids the heat flux limitation caused by overheating of the porous matrix and the consequent displacement of the liquid from it. This is accomplished by keeping the heat from having to be conducted very far through the matrix and particularly from having to be conducted across the principal width of the matrix supplying the liquid. On the other hand, as pointed out hereinabove, it is desirable to have the surface of vaporization coincide with a surface of the porous matrix from which the vapor may escape with minimum restriction.

It is, therefore, an important feature of the capillary vaporizer surface structure described herein that passages are provided in the vaporizer structure which form, or connect to, cavities forming surfaces in the capillary matrix material near the heat source where the liquid is vaporized and from which the resulting vapor may escape from the capillary matrix. The passages may lie in either or both the capillary matrix material or the material between the heat source and the capillary material so long as the passages expose, or connect to cavities exposing, surfaces of capillary material and provide for the escape of vapor formed at these exposed surfaces.

In a preferred embodiment for large areas and high heat flux densities, there are a hierarchy of vapor passages wherein very numerous, very small passages feed vapor into larger, less numerous passages which may, in turn, feed into a few rather large passages. Capillary arrays of this type simultaneously minimize the pressure differential necessary to cause vapor flow from the array and also minimize the amount of capillary material removed or deleted to form the passages and thus not available for liquid transport. This array also allows a network of very closely spaced passages which form, or are connected to, regional areas of vaporization, to be placed adjacent the heated surface of the capillary material.

Elaborate arrays having a multiple step hierarchy of passages are not necessary in many heat transfer situations. Thus, for example, an inexpensive very high heat flux "-boiling" surface which is in contact with bulk liquid with which vapor may freely mix, for example, simply provides a layer of capillary material perforated by a large number of slits or holes reaching from the bulk liquid surface through or almost through the capillary material to the heat source surface. In such a heat transfer surface structure, the gross heat and liquid flows are in opposed direction, although within the capillary material, heat flow is substantially normal to liquid flow.

FIG. 3 illustrates in perspective a capillary vaporizer surface constructed according to principles of this invention. As illustrated in this embodiment, there is a heat source wall 11, preferably of a relatively high thermal conductivity metal. The heat source wall may, for example, be a wall separating heat transfer fluids in a heat exchanger, or may be a surface portion of a heat producing electronic component or the like.

In intimate thermal contact with the heat source wall 11 is a channeled layer of porous material having a high efficiency $\epsilon_E$. It is to be understood that, as used herein, the term "porous" is meant to include capillary materials whether the capillary liquid conduits are strictly "pores" or not. Thus, for example, a solid surface with grooves or a folded foil might be used instead of a porous matrix in some embodiments. Also, in describing a capillary as having a larger or smaller pore size, what is specifically meant is that the capillary structure has a lower or higher value, respectively, of the effective pore surface to volume ratio $\delta$, rather than any particular dimension of the capillary structure. The channeled layer adjacent the wall 11 can be considered to be a plurality of parallel spaced apart strips 12 of porous material joined to the heat source wall 11. Preferably, the strips 12 are flared in the portion adjacent the heat source wall so that a substantially continuous layer of porous material is provided on the heat source wall with the thickness of the layer increasing from a very small thickness intermediate the strips to an appreciable thickness in the principal portion of the strip. Each of the strips 12 is overlaid by a strip 13 of similar size having a larger pore size (that is, lower $\delta$) than the porous material forming the strips 12. The larger pore size material 13 acts as a temporary reservoir of liquid during operation of the capillary vaporizer surface.

In a typical embodiment, the surface illustrated in FIG. 3 is employed as a so-called boiling heat transfer surface where the surface is in contact with bulk liquid that is free to mix with vapor produced at the surface, that is, the liquid reaching the porous capillary vaporizer is not previously confined within a capillary wick adjacent the vaporizer. It should be noted that this capillary vaporizer surface is not employed exclusively in a situation where the surface is immersed in a body of liquid but may be employed in a situation where the surface is intermittently wetted by a liquid and intermittently exposed only to vapor as may occur in a high flow rate heat exchanger. In this situation, the period of time that the surface is not wetted by liquid is extremely short. Therefore, a relatively small strip 13 of larger pore material can act as a temporary liquid reservoir of sufficient capacity to provide liquid to the vaporizer during the intervals that the local surface is not wetted.

During operation of the heat transfer surface illustrated in FIG. 3 heat flows through the wall 11 and into the porous vaporizer strips 12 at their wide portions. Liquid contacts the vaporizer strips primarily in the larger pore portion 13, and due to surface tension forces flows through the coarser pored material to the finer pored vaporizer strips 12. Heat also flows through the vaporizer strips 12 and vaporization of liquid occurs primarily at the very large number of regional areas formed by the sloping bottom surfaces 14 in the channels between the strips. These surfaces are nearest heat source wall, so that the heat need flow only through a relatively short path of vaporizer, and the surfaces are also adjacent the channels so that the vapor formed can freely escape from the capillary vaporizer surface without passing through the pores.

It should be observed that the width and height of the strips of porous material on the heat transfer surface are quite small, and, for example, may be about 0.002 to about 0.04 inch. The smaller sizes are preferable from the point of view of enhanced heat transfer since a greater total length of strips can be accommodated on a surface of given area. As pointed out hereinabove, a greater heat transfer rate can be obtained since the rate is dependent on a total length of strip and is not affected by the size of the strip, and close packing of small strips gives higher heat flux density than fewer longer strips. The smaller strips are, however, somewhat more difficult to fabricate, and in order to avoid expense, some sacrifice may be made in heat transfer characteristics and large strips may be employed.

A porous capillary vaporizer, as illustrated in FIG. 3, is readily made by bonding a layer of high efficiency porous material to the heat source wall, preferably by diffusion bonding to avoid plugging pores, and in a similar manner a layer of coarser pored material is bonded on the finer material. The surface can then be grooved to produce the structure illustrated in FIG. 3. It will also be apparent that the two-layer structure is not require in all situations, and a less expensive structure can be prepared with a single channeled layer on the heat source wall.

FIGS. 4 and 5 illustrate in two perpendicular cross sections a compound capillary vaporizer surface incorporating principles of this invention. As illustrated in this embodiment there is a heat source wall 21, which, as before, is preferably a high thermal conductivity metal through which heat flows into the capillary vaporizer from some heat source (not shown). Immediately adjacent and in intimate thermal contact with the heat source wall are a plurality of capillary vaporizer strips 22 formed of a porous material having a high efficiency $\epsilon_E$. The strips are placed in close proximity to each other so that there are a plurality of channels therebetween, and the width and height of the strips 22 is preferably in the same order as those strips illustrated in FIG. 3

The strips 22 may have a cross section such as illustrated in FIG. 3, or may merely be rectangular strips either in direct contact with the heat source wall or with a thin layer (a few mils) of additional porous material uniformly covering the heat source wall.

Liquid is delivered to the vaporizer strips 22 in the compound vaporizer surface by somewhat larger strips 23 running approximately perpendicular to the smaller strips 22 on the heat transfer surface. The larger strips 23 are spaced apart to leave channels 24 therebetween that are larger than the channels between the smallest strips 22. Overlying the larger strips 23 are bars 26 running approximately perpendicular to the larger strips 23 and parallel to the heat source wall 21. The bars 26 are mutually spaced apart to leave channels 27 therebetween that are appreciably larger than the channels 24 between the larger strips 23. The larger strips 23 and bars 26 are made of a porous material having a high wicking efficiency $\epsilon_w$ and would normally have larger pores (that is, lower $\delta$) than the smallest strips 22 immediately adjacent the heat source wall.

As an example of the scale involved, the bars 26 may in one example be about 0.016 inch wide and 0.032 inch high in a direction normal to the heat source wall. The size of the bars can, in general, range upwardly from these values, It might also be noted that the strips 23 and bars 26 being made of the same material, can be made in a single operation wherein one ridged die is pressed towards another ridged die with the ridges of the two dies perpendicular to each other. This combination of bars and strips can also be made from a slab of porous material by cutting the channels 24 in one direction and the channels 27 in opposite direction sufficiently deeply to intersect.

Overlying the bars 26 and in intimate contact therewith are still larger bars 28 running parallel to the heat source wall, and approximately perpendicular to the smaller bars 26. The larger bars 28 are mutually spaced apart to form channels 29 therebetween that are larger than the channels 27 between the smaller bars 26. These larger bars are preferably made with an even larger pore diameter (that is, lower $\delta$) than the smaller bars 26. These larger bars act as temporary liquid reservoirs during operation of the capillary vaporizer.

In operation, liquid contacts the larger bars 28 either as bulk liquid in intermittent contact with the exposed faces of the bars or in some embodiments by an additional wick (not shown) bringing liquid thereto. The liquid flows from the larger bars 28 under surface tension forces, as hereinabove described, into the smaller bars 26 and thence into the strips 23 closer to the heat source wall. These bars and strips do not have any substantial heat flow therethrough and behave as wicks for conducting liquid to the smallest strips 22 in contact with the heat source wall 21. Thus, the larger strips and bars service as a liquid distribution system for bringing liquid to the smallest strips 22 where it flows to the interfaces between the strips 22 and the channels between them, from which interfaces it evaporates into the channels. These interfaces are, thus, regional areas of vaporization "formed" by the channels. In some embodiments a thin layer of porous material may be provided over the entire heat source wall 21. Such a layer makes little difference in the maximum heat flux capability of the vaporizer but can appreciably reduce the temperature across it. The regional area of vaporization as used herein includes the interfaces of adjacent strips (in this embodiment) on opposite edges of the channel and the porous layer, if any, adjacent the heat source surface.

The vapor that is formed in the channels between the smallest strips 22 flows into the channels 24 between the larger strips 23. The vapor either flows directly into the larger channels or may pass for a short distance parallel to the heat source wall before reaching one of the larger channels 24. The distance that the vapor need flow through the small channels between the smallest strips 22 is quite short, so that there is as little as possible flow resistance in that short passage.

The vapor within the passages 24 between the larger strips then flows into the still larger channels 27 between the bars 2 6. The vapor flows through the larger channels into the still larger channels 29 between the largest bars 28 and thence escapes from the capillary vaporizer. Thus, it will be seen that there is a liquid flow path through the successively smaller and more numerous bars 28 and 26 and the strips 23 to the smallest and most numerous strips 22. The vapor flows countercurrent to the liquid in a parallel path from the smallest and most numerous channels through the successively larger and less numerous channels 24, 27 and 29. The liquid flow path is thus separated from the vapor flow path, and there is no interference between the counterflowing fluids.

The liquid flows through porous materials having successively smaller pore sizes (larger $\delta$) so as to support the increased pressure differentials due to viscous drag and due to the increasing vapor pressure from the increasing temperature. As the cross sections become smaller, the number of flow paths increases so that the total area available for liquid flow remains substantially constant throughout the structure in any plane parallel to the heat source wall and the length of the path over which the liquid flows also decreases so as to minimize the liquid pressure drop. In a similar manner, the vapor flow through successively smaller numbers of successively larger channels to escape from the structure, and the overall area available for vapor flow remains substantially constant in all planes parallel to the heart source wall. It is generally preferred that the cross-sectional area available for vapor flow and liquid flow be about equal since this yields a near optimum overall heat transfer efficiency for the capillary vaporizer over a broad range of operating conditions.

Both of the structures hereinabove described and illustrated in FIG. 3, and in FIGS. 4 and 5 can be considered as vented capillary vaporizers since holes or passages are provided for escape or venting of vapor from the capillary matrix. Determination of the performance characteristics of a vented capillary vaporizer divides into two parts (1) the vaporizing surface structure, including the heated surface and the immediately adjacent capillary material and passages, that is, the region where heat conduction through the matrix and evaporation from the passage walls take place, and (2) the liquid distribution and vapor collection manifolds wherein fluid flow occurs and heat flow can be ignored.

Considering the second part first, the liquid may be distributed directly to the capillary material forming the vaporizing surface structure, that is, the layer of capillary material immediately adjacent the heat source surface, as in the simple "-boiling" surface structure hereinabove described and illustrated in FIG. 3, wherein the vapor is likewise directly collected and discharged into the same region that supplies the liquid so that the liquid and vapor are comingled. In this type of structure, any heat flux limitations lie either in the vaporizing surface structure, or in "mechanical" interference between the incoming liquid and outgoing vapor, which is best determined empirically since the general mathematical solution of the dynamic situation without liquid or vapor manifolds is exceedingly complex.

In a structure as hereinabove described and illustrated in FIGS. 4 and 5, the liquid is distributed to the vaporizing surface structure by capillary material which also contains passages for the vapor to escape. The liquid flow is calculated in the same manner as hereinabove described wherein equations 11 and 15, or 18 and 19 were derived utilizing the fluid parameters: density $\rho$, heat of vaporization per unit volume $h$, surface tension $\sigma$, and viscosity $\eta$; and the matrix parameters: fluid conductivity $\alpha$, pore surface to volume ratio $\delta$, and wick matrix microstructure efficiency $\epsilon_w$; and pressure differences external to the vaporizer, such as $\Delta P_g$ and $\Delta P_R$.

Generally the same calculation procedure is employed. The pressure drop across the matrix required to force the desired fluid flow rate is calculated in terms of the fluid properties, the external pressure differences, and $\alpha$ and $\delta$. Then the equation $\alpha=\epsilon_w/2\delta^2$ is used to eliminate $\alpha$ in favor of the efficiency factor $\epsilon_w$ which is not dependent on scale or size. The external pressure differences are added and the result set equal to $\sigma\delta$ and solved for the fluid flow rate. The expression for the flow rate is then differentiated with respect to $\delta$, set equal to zero, and solved for $\delta$ so as to get the value of $\delta$ giving the maximum flow rate. A capillary matrix structure is then selected having a $\delta$ as close as possible to that calculated and efficiency factor $\epsilon_w$ as high as possible and the actual values of $\delta$ and $\epsilon_w$ entered in the equation for the flow rate, thus giving the solution.

The pressure drop due to vapor flow in the passages is calculated using standard techniques for fluid flow. Care must be taken in these calculations since the vapor flow often shifts from laminar in the smaller passages to turbulent flow in the larger passages. The pressure drop thus calculated is included as one of the "external" pressure drops in the liquid flow calculations. In finding the vaporizing surface characteristics, it is important to note that the use of multiple passages to vent a capillary vaporizer does not alter the basic equations 55 and 56 giving the maximum heat flux *per unit length*, and the maximum values of $\sigma\delta$ of $P_x$ for a conductively heated capillary vaporizer strip. Multiple vapor passages in the vaporizing surface structure, however, permits the use of as large a number of vaporizer strips per unit length as is necessary to obtain the desired heat flux density. Thus, for example, a passage through a porous material adjacent and parallel to the heat source surface creates two vaporizer strips, one on each side thereof, each capable of the heat flux per unit length given in equation 55. Thus, increasing the number of passages adjacent the heat source surface, increases the maximum available heat flux proportionately until a limit is reached wherein the resistance to vapor flow becomes excessive due to the small size of passages.

The increase in heat flux occurs since the flux is not dependent on scale, i.e., size, but only on shape.

In the same manner as for parallel strips, a hole normal to the heat source surface penetrating through or nearly through the capillary matrix produces a circular vaporizer "strip" around the hole where it approaches the heat source surface and such a hole also serves to increase the heat flux capability of the surface, although due to the unusual shape of the "-strip", quantitative determination of the increase is mathematically more difficult.

Calculation of the maximum heat flux per unit heat source area $H_{max}/A_v$ for any selected vaporizer surface structure is straightforward when the evaporator strip shape efficiency factor $G_1$ and the fluid and matrix parameters are known, since

$$\frac{H_{\max}}{A_v}=S\left(\frac{H_{\max}}{x}\right)=SG_1\epsilon_E\sigma\left(\frac{hk_s}{12\eta\frac{dP_v}{dT}}\right)^{1/2}\left(1-\frac{\Delta P_x}{\sigma\delta}\right) \tag{60}$$

where $S$ is the total length of the vaporizer strips per unit heat source area. Thus, for example, if there are $n_p$ passages per unit distance, forming an equal number of regional areas of vaporization, each of which is composed of two vaporizer strips, side by side and parallel and adjacent to the heat source surface, then $S=2n_p$ and

$$\frac{H_{\max}}{A_v}=2n_p\left(\frac{H_{\max}}{x}\right)=2n_pG_1\epsilon_E\sigma\left(\frac{hk_s}{12\eta\frac{dP_v}{dT}}\right)^{1/2}\left(1-\frac{\Delta P_x}{\sigma\delta}\right) \tag{61}$$

Thus, the heat flux density is directly proportional to the density of passages.

The geometrical shape efficiency factors $G_1$ and $G_2$ may pose appreciable difficulty in mathematical determination except for a few very simple shapes, such as a thin rectangular slab or a wedge. For other shapes, either rough estimates can be made based on similarities to the thin slab or wedge shapes or analog or numerical solutions must be resorted to. This, however, does not interpose any substantial difficulties since, as pointed out hereinabove, each shape has a single numerical value for the factors $G_1$ and $G_2$ independent of any other factor. Thus, all scaling and size laws are implicit in equation 55 and do not require a knowledge of the geometrical factors $G_1$ and $G_2$ which are required only when the numerical value of the maximum heat flux must be calculated and not when the effect of various parametric changes is required. Generally, it is preferred to use structures having as high a value of the geometrical shape factor $G_1$ as possible to maximize the heat flux per unit area $H_{max}/A_v$. Therefore, structures described herein are selected to have an estimated value for the geometrical factor in the range of about 0.5 to 1.0.

In order to illustrate the magnitude of the heat flux densities involved in the improved vented capillary vaporizer, constants appropriate to water in a copper matrix at 100° C. are employed in equation 61. First, the product of surface tension $\sigma$ and effective pore surface to volume ratio $\delta$ is found from equation 56

$$\sigma\delta \leq \frac{G_2\epsilon_E k_S}{w\left(\frac{1}{y}\right)k_M} \times 59\left(\frac{2.16\times10^3\times3.68\times10^4}{1.2\times10\times2.8\times10^{-3}\times3.8}\right)$$

$$= 1.47\times10^6\left(\frac{G_2\epsilon_E k_S}{w\left(\frac{1}{y}\right)k_M}\right)\frac{\text{dynes}}{\text{cm.}^2} \qquad (62)$$

Further assuming reasonable values for the geometric constant $G_2=1$, efficiency of the vaporizer matrix $\epsilon_E=0.2$ thermal conductivity ratio $k_S/k_M=4$ and a geometry $w\left\{1/t\right\}=1$ then the product $\sigma\delta \cong 1.18\times10^6$ dynes/cm.$^2$ which is slightly over one atmosphere. It might be noted that for the "ideal" case, where $\epsilon_E=1$, and $k_t/k_M=2$, then the product $\sigma\delta=2.94\times10^6$ dynes/cm.$^2$ Usual operating conditions for a vaporizer have "external" pressure gradients $\Delta P_x$ values ranging from about $10^4$ to about $2\times10^5$ dynes/cm.$^2$, that is about 10 centimeters to 2 meters equivalent water head. Higher values of $\Delta P_x$ can be dealt with as indicated in equations 39 through 42.

Now finding the maximum heat flux per unit area $H_{max}/A_v$ for water in a copper matrix at 100° C.

$$\frac{H_{max}}{A_v} = 2n_pG_1\epsilon_E\times59\left(\frac{2.16\times10^{-3}\times3.8}{12\times2.8\times10^{-2}\times3.68\times10^4}\right)^{1/2}\left(1-\frac{\Delta P_x}{\sigma\delta}\right)$$

$$= 288n_pG_1\epsilon_E\left(1-\frac{\Delta P_x}{\sigma\delta}\right) \qquad (63)$$

Further assuming that $G_1=0.8$, $\epsilon_E=0.2$, and $\Delta P_x=\sigma\delta/4$, then $H_{max}/A_v=34.5n_p$ watts/cm.$^2$ and $\Delta P_x=3.0\times10^5$ dynes/cm.$^2$.

Theoretically, the number of vapor passages, or regional areas of vaporization, per unit length $n_p$ can be made as large as desired, and only limit on $H_{max}/A_v$ lies in the fact that the external pressure $\Delta P_x$ contains a term equal to the pressure drop due to the vapor flow out of the vaporizer and through the rest of the system, if any. This term is usually one to 10 times the maximum vapor velocity pressure $\rho_v v^2/2$ for most vaporizer designs with the simple structures of FIGS. 3, 4 and 5 being close to one. Thus the external pressure drop $\Delta P_x$ increases with increasing heat flux $H_{max}/A_v$, thereby decreasing the term $(\sigma\delta-\Delta P_x)$. As a practical matter there are also other considerations with very large values of $n_p$ since construction of the vented vaporizer becomes increasingly difficult as the passages or channels become smaller, and also higher values of the effective surface to volume ratio mean smaller pores which are clogged more easily by deposits left when the liquid evaporates.

Thus, for example, a vaporizing surface structure suitable for many applications would have a number of channels $n_p$ of about 40 per centimeter, that is, about 0.010 inch between channels. This gives a maximum heat flux per unit area of about 1,380 watts per square centimeter for water in a porous copper matrix at 100° C. A vaporizer surface structure for spot cooling a small area, such as an electronic device, may have as many as 200 channels per centimeter so that the maximum heat flux $H_{max}/A_v$ is in the order of 6,900 watts per square centimeter for water in a copper matrix at 100° C. These heat flux densities are more than an order magnitude above those that are usually available with pool boiling, for example, which has a maximum heat flux of about 100 watts per square centimeter for water at atmospheric pressure. MOst other vaporization or convective cooling systems have even lower heat flux densities than pool boiling.

FIG. 6 illustrates in cross section a single vaporizer strip structure similar to that illustrated in FIG. 3. The structure includes a heat source wall 31, which is impervious to liquid and vapor and through which heat flows to the larger parallel face of a trapezoidal cross section vaporizer strip 32 formed of relatively small pore, high thermal conductivity porous material. On the smaller parallel face of the trapezoidal vaporizer strip 32 is a feeder strip 33 of relatively larger pore material in intimate contact with the smaller pore material of the vaporizer. The liquid (not shown) is at least intermittently in contact with the feeder strip 33, and the liquid flows through the porous matrix to the sloping faces 34 of the trapezoidal strip as indicated by the solid arrows. Vaporization of the liquid occurs principally at the surfaces 34, and vapor escapes from the region between the strip and an adjacent similar strip as indicated by the dashed arrows.

The vaporizer strip structure illustrated in FIG. 6 essentially forms two vaporizer strips in the sense that S is used in equation 60, or it can be considered that the space on either side of the porous material is combined to be equivalent to one passage. Equation 61 may therefore be used to calculate the maximum heat flux density, with $n_p$ being the frequency of vaporizer strip structures on the surface. The vaporizer strip structure illustrated in FIG. 6 can be considered to be formed of two truncated sectors having liquid entering from one face, heat entering from a second face, and vapor escaping from a third face, with the direction of heat flow and liquid flow being substantially perpendicular. Approximate calculations for such a structure indicate a geometrical shape factor $G_1$ of about 0.8 to 0.9 for this geometry.

FIG. 6 also shows by a dashed line 36 an internal liquid-vapor interface within the porous matrix of the vaporizer. This interface 36 arises since the heat source wall is at a higher temperature than the vaporizing surface 34 in order for heat to flow from the wall to the surface. When operating at high heat flux densities there is a region near the heat source, as indicated by the interface line, where the temperature in the matrix is sufficiently high to drive liquid from the pores and replace it with vapor. At low heat flux, the difference in temperature across the vaporizer will be relatively low, and the interface 36 will be relatively nearer the heat source wall or may disappear when the matrix is completely liquid filled. When the heat flux is high, the temperature gradient will be higher, and the interface will be relatively nearer the vaporizer surfaces. Eventually there is a maximum continuous heat flux which cannot be exceeded without the internal liquid-vapor interface moving so near the surface as to so "choke" the liquid flow that insufficient liquid is supplied to replace that vaporized, leading to a "drived-out" condition and possibly damage to the structure. The heat transfer surface structure will accommodate any heat flux less than the maximum available.

FIG. 7 comprises a capillary vaporizer surface structure which can be considered to be a detail of a larger structure such as illustrated in FIGS. 4 and 5. As illustrated in FIG. 7, the heat source wall 21 has a smaller strip 22 in intimate thermal contact therewith, and the strip 22 is also in contact with a larger feeder strip 23 of larger pored material. As seen in this detail, liquid flows from the larger strip 23 through the small vaporizer strip 22, which can for purposes of calculation be considered to be two connected vaporizer strips. The liquid flow is indicated by solid arrows in FIG. 7 and vapor flow from the side faces 38 of the vaporizer strip 22 is indicated by dashed arrows.

As in FIG. 6, a vapor-liquid interface 39 occurs within the vaporizer strip bounding the region where the temperature is high enough that only vapor is stable, and the lower temperature region where liquid is stable within the porous matrix. The three dimensional shape of the interface 39 is complicated by the fact that liquid also flows through the vaporizer strip 22 in a direction along its length, that is, normal to the plane of the paper in FIG. 7, and the interface is further from the heat source wall in the portion of the smaller strip 22 between the larger feeder strip 23 and the wall, then it is beneath a channel 24 (FIG. 5) between a pair of feeder strips 23.

FIGS. 8, 9 and 10 illustrate alternative detailed structures of a capillary vaporizer surface structure incorporating principles of this invention. In each of these figures, A single vaporizer strip structure and one-half of a channel on each side thereof is illustrated, and it will be understood that similar parallel structures occur repetitively on the heat transfer surface.

As illustrated in FIG. 8, a heat source wall 41 is coated with a continuous layer 42 of fine pored, relatively high thermal conductivity vaporizer material, from the opposite surface of which vaporization occurs as indicated by the dashed arrows. A larger pore size feeder strip 43 seen in cross section has a narrowed portion with a small face 44 in contact with the vaporizer strip 42. Liquid flows from the feeder strip 43 to the vaporizer 42 as indicated by the solid arrows. An internal liquid vapor interface 46 occurs within the vaporizer 42 at high heat fluxes and the maximum heat flux is obtained when the interface just contacts the surface from which vaporization occurs at the midpoint of the channel between adjacent feeders 43.

FIG. 9 illustrates a heat transfer surface structure that is mathematically substantially identical to the structure illustrated in FIG. 8. As illustrated in this embodiment, a heat source wall 47 is provided with V-shaped grooves, the sides of which are lined with a fine pored capillary material 48. A portion of the vaporizer at the crest of ridges in the heat source wall 47 is in contact with a relatively coarse pore feeder strip 49 running approximately perpendicular to the ridges. Liquid flows from the feeder 49 through the capillary material 48, as indicated by the solid arrows. Vapor escapes from the surface of the capillary material into the V-shaped channels as shown by the dashed arrows. A vapor liquid interface 51 forms between the heat source wall 47 and the surface of the capillary material from which vaporization occurs, in substantially the same manner as hereinabove illustrated in FIG. 8.

FIG. 10 illustrates a heat transfer surface structure particularly useful when the external pressure drop $\Delta P_x$ against which fluid flow must be provided is relatively high. In this structure a heat source wall 53 is provided with raised rectangular ridges 54 so as to define a plurality of rectangular channels 56 therebetween. The tops of the raised portions 54 of the wall are in intimate thermal contact with a porous matrix 57 that is supplied with a vaporizable liquid. The liquid flows through the porous matrix as indicated by the solid arrows, and vaporization occurs at the face 58 of the matrix adjacent the channels 56 as heat flows from the wall into the matrix. At high heat fluxes a liquid vapor interface 59 forms in the porous matrix as before. The regions of porous matrix forming the vaporizer strips for purposes of calculating the maximum heat flux have unusually large thickness to width ratios $l/w(\bar{l}/t)>1$, and therefore will allow more area for fluid flow, which proves useful when the external pressure $\Delta P_x$ is high so that $\delta$ must be high to support the pressure differential, thus forcing the fluid conductivity $\alpha$ to be low.

The capillary vapor strip designs illustrated in FIGS. 6 to 10 are merely exemplary of good vaporizer designs, and it should be apparent to one skilled in the art that many more shapes and combinations of shapes of porous materials can be used to form the porous matrix and the passages to vent vapor from the surfaces where vaporization occurs.

A "simple" capillary vaporizer surface structure as illustrated in FIG. 3 (as contrasted to a "compound" boiling surface as illustrated in FIG. 4 and 5) permits a particularly large number of different configurations since the vapor passages only have to penetrate through or nearly through the capillary matrix material between the heat source wall and the region occupied by combined liquid and vapor. Also, the passages in any capillary vaporizer may be of almost any shape, for example, grooves, slots or holes of any shape, and even random structure such as a large pore matrix in which the large pores are the vapor passages, and the matrix is composed of material that is itself a fine pore matrix that carries the liquid. Layers of such material, with the size of the larger pores, and possibly the smaller pores, increasing with increasing distance from the heat source surface may also be used as an inexpensive alternate to structures such as illustrated in FIGS. 4 and 5.

Fabrication techniques are equally varied since the passages may be formed by bonding closely spaced strips, cylinders, grids, or even irregular lumps of porous material to the heat source surface and to each other. Another fabrication technique for a capillary vaporizer surface is to first coat the impervious heat source surface with a continuous layer of porous material and then multiply crack the porous matrix, for example, by stretching or bending the composite layers, or by further sintering of the porous material to induce additional shrinkage.

Except for the simplest types of capillary vaporizer surfaces for contact with bulk liquid, the vented capillary vaporizers comprise at least a heat source surface, a vaporizer surface structure and a liquid feed structure. Both of the latter being of capillary material for inducing liquid flow due to surface tension forces. It is important to distinguish these two structures since during operation of the vaporizer the capillary material forming the surface structure transmits both heat and liquid while the capillary material forming the feed structure need transmit only liquid, and in many situations it is desirable that it be a good thermal insulator. These functional differences between the two structures are reflected in the efficiency factors for the two materials; that for the feed structure or wick being $\epsilon_w=2\alpha\delta^2$; while the efficiency factor for the vaporizing surface structure is $\epsilon_E=2\delta(3\alpha(k_M/k_S))^{1/2}$. In some situations, both the surface vaporizer structure and the feed structure may be fabricated of the same porous material for lower cost; however, there may be some sacrifice in performance.

The vented capillary vaporizers provided in the practice of this invention may be categorized in steps of increasing complexity of structure and as to the extent to which the liquid and vapor flow are thereby isolated.

A "simple" heat transfer surface is hereinabove described and illustrated in FIG. 3, and comprises a layer of porous material bonded to a heat source surface and perforated with passages penetrating directly into the porous material to, or almost to, the heat source surface. The vapor produced is thus vented directly into the same region carrying the liquid.

A "compound" heat transfer surface is hereinabove illustrated in FIGS. 4 and 5, and is similar to the simple structure but with passages carrying at least a portion of the vapor a short distance parallel to the heat transfer surface before venting it into the region of mixed liquid and vapor. Thus, in a portion of the region adjacent the surface, the outgoing vapor and incoming liquid have separate flow paths. The compound surface can be considered to be a simple heat transfer surface capped with feeder strips of porous material perpendicular to the vaporizer strips on the heat transfer surface forming liquid and vapor manifolds for separated countercurrent flow of the two fluids.

A third type of vented capillary vaporizer is one where the liquid is fed thereto by a wick such as, for example, in a heat pipe. In this structure, the liquid is always in some capillary structure, either the wick, an intermediate feeder, or the vaporizer, and the flow paths of the liquid and vapor are thereby separated. There is no need, however, to thermally or mechanically isolate the liquid filled wick from the vapor since the capillary forces in the wick support the pressure difference between the liquid and vapor. If, for some reason, the vapor impinging on the wick is superheated, liquid will merely evaporate from the surface of the wick to keep it cool. Thus, the actual vaporizer surface structure can be quite similar to the compound boiling surface, with the feed structure in contact with a wick. Such a wick-fed vaporizer structure is illustrated in FIGS. 11 through 16, hereinafter described in greater detail.

A fourth type of vaporizer can be characterized as a "capillary pump" vaporizer that is employed when the incoming liquid to the vaporizer is in bulk, and it is desired that the outgoing vapor be at a higher pressure than the incoming liquid. By bulk liquid is meant that the liquid is not flowing through a capillary structure sufficiently fine pored to support the desired pressure difference. In a capillary pump vaporizer, the incoming liquid must be both mechanically and thermally isolated from the outgoing vapor until the liquid flows into a sufficiently fine pored matrix within or adjacent the vaporizer to support the required pressure differential. If the incoming liquid is not thermally isolated and, if necessary, cooled, its temperature and thus its vapor pressure will approach that of the outgoing vapor which may cause bubbles and eventual vapor lock in the incoming liquid passage. Generally, at least part of both mechanical and thermal isolation is achieved by separating the bulk liquid input manifold from the complex vaporizer structure by a wicking layer of low thermal conductivity porous material with sufficiently fine pores to support the pressure difference between the liquid and vapor. A capillary pump type of vaporizer is illustrated in FIGS. 17 to 19 and described hereinafter in greater detail.

FIGS. 11 to 14 show in cross-sectional views one of the third type of vented capillary vaporizer wherein liquid is fed thereto by a wick. FIG. 15 shows one element of this rather complex structure in perspective to help clarify the shape of the parts and FIG. 16 magnifies a small portion of FIG. 14 for greater clarity. Such a wick-fed, vented vaporizer design is suitable for use in a multiply segmented heat pipe such as described and illustrated in the aforementioned copending patent application entitled SEGMENTED HEAT PIPE. Such a vaporizer design is also useful in ordinary heat pipes or the like where it is desired to maximize the heat flux density while minimizing the temperature drop across the capillary vaporizer by having a very short heat conduction path.

As illustrated in this embodiment, a cylindrical impervious wall 61 bounds the structure on the sides. A heat source wall or partition 62 separates the vented capillary vaporizer from a heat source 63 which is not illustrated in great detail but which can be a somewhat similar surface structure or some other surface adapted for vapor condensation as in a segmented heat pipe or may be a primary heat source. Immediately adjacent the heat source wall 62 is a thin evaporator surface layer 64 of porous material having a high efficiency $\epsilon_E$ (FIG. 16).

It is convenient in discussing the vented vaporizer illustrated in FIGS. 11 through 16 to define orthogonal $x$ and $y$ coordinates lying in a plane parallel to the heat source wall 22. Thus, the cross sections of FIGS. 11 and 13 are taken in planes parallel to the $xy$ plane. The cross section of FIG. 12 is taken in a plane parallel to the plane containing the $y$-axis and the axis of the tube, and FIGS. 14 and 16 are cross sections taken in a plane parallel to the plane containing the $x$-axis and the axis of the tube.

Parallel to the heat source wall 62 and extending in the $y$ direction are a plurality of parallel strips 66 of porous material, each having a cross section substantially as illustrated in FIG. 8 with a narrow edge in contact with the porous layer 64 on the heat source wall and dividing the free surface of the porous layer 64 into a number of regional areas of vaporization. The plurality of strips 66 defines a plurality of intermediate channels 67 lying parallel to the heat source wall and extending in the $y$ direction. The parallel strips 66 serve to deliver liquid to the porous material 64 on the heat source wall, and the channels 67 serve to carry vapor away from the porous material as liquid is vaporized in the manner hereinabove described for FIG. 8.

Extending transversely to the strips 66 in contact with the porous layer are a plurality of larger strips 68 of rectangular cross section and extending parallel to the heat source wall in the $x$ direction. The spaces between the larger strips 68 form larger $x$ direction channels 69, each of which is in fluid communication with a plurality of the smaller $y$ direction channels 67 so that vapor from a plurality of the smaller channels feeds into the larger channels 69 during operation of the vaporizer. The larger strips 68 are in contact with the smaller strips 66 and can even be formed integrally therewith of the same material. Both sets of strips are formed of a porous material having a high wicking efficiency $\epsilon_w$. In a typical embodiment for a heat pipe about one-half inch diameter the smaller channels 67 may be present with a frequency in excess of about 100 channels per inch, and the larger channels 69 may be provided in the order of about 36 channels per inch.

In order to feed liquid to the larger strips 68, a capillary wicking structure is provided which can be conveniently divided into a conventional cylindrical fluid source wick 71 of porous material extending along the tubular wall 61 from some cooler region (not shown towards the other end of the heat pipe, and a somewhat more complex wicking structure between the fluid source wick 71 and the larger strips 68. The interior surface of the wick 71 defines a passage denominated herein as a vapor way.

This intermediate wicking structure includes a ring 72 of porous material and having an outside diameter approximately that of the tubular wall 61 of the heat pipe. At the end of the ring in contact with the porous wick 71, the inside surface 74 of the ring 72 is in the form of an elliptical cylinder having the major axis of the ellipse in the $x$ direction and the minor axis in the $y$ direction as seen in the cross section of FIG. 11. In order to mate with this elliptical inside shape, the inside of the wick 71 is provided with a pair of beveled portions 73 so as to provide an elliptical cross section at the end of the wick in contact with the end of the ring 72.

The elliptical internal surface 74 of the ring 72 extends the full length of the ring at the major axis, that is, in a plane in the $x$ direction as seen in FIG. 14. The inside surface of the ring is also provided with a beveled region 76 which can be considered as the surface of an elliptical cone, that is, a cone having an elliptical base and an axis normal to and centered on the elliptical base. The major axis of the elliptical cone is in the $y$ direction, and the minor axis is in the $x$ direction, and in the plane where the ring 72 comes in contact with the larger strips 68 the minor axis of the elliptical cone is equal to the major axis of the elliptical inside surface 74 on the ring. That is, at the median plane parallel to the $x$ axis there is no bevel on the inside of the ring and there is a maximum bevel at the $y$ median plane. The major axis of the elliptical cone in the plane in contact with the larger strips is greater than the diameter of the tubular wall 61 so that the only end surface portion of the ring 72 in contact with the larger strips 68 comprises a pair of opposed crescent-shaped sectors each having a circular outside edge and an elliptical inside edge as seen in the cross section of FIG. 13. The crescent-shaped portions of the ring 72 in the plane of the cross section of FIG. 13 are in contact with a plurality of the larger strips 68 near their ends so that liquid passes from the wick 71 through the ring 72 into the ends of the larger strips.

Delivering liquid solely to the ends of the larger strips 68 by the crescent-shaped portions of the ring 72 in this embodiment is not sufficient for supplying liquid across the full face of the vented capillary vaporizer at maximum heat flux; therefore, in order to supply liquid to the larger strips at a region intermediate their ends, a pair of similar parallel crossbar structures 77 having their greatest extent in the $y$ direction are provided across the ring. A portion of the crossbars 77 is illustrated in perspective in FIG. 15 for the purpose of clarifying the shape of the crossbar and slope of a pair of outer faces 78. Each of the crossbars 77 has a flat face 79 (seen in the cross section of FIG. 13) in contact with the larger strips 68 for transferring liquid thereto. The ends 81 of the bars 77 are in contact with, the ring 72 for accepting liquid therefrom (one of the curved ends 81 is hidden at the far end of the bar 77 illustrated in FIG. 15).

The two sloping outer faces 78 of the bars are opposite the flat face 79 in contact with the larger strips 68. The two faces 78 slope so that the crossbar is relatively thicker on one side 82 nearer the axis of the heat pipe, and relatively thinner on the opposite side 83 more remote from the axis of the heat pipe. The two faces 78 also slope from their intersection with the ring 72 where they are relatively further from the flat face 79 toward a central intersection of the two faces where they are relatively nearer the flat face 79. These two directions of slope of the faces 78 result in their intersecting along a straight line interconnecting a point 84 on the higher side 82, and a second point 86 on the lower side 83. The point 86, more remote from the axis of the heat pipe, is at, or nearly at, the flat face 79 of the crossbar, and the point 84, nearer the axis of the heat pipe, is appreciably more remote from the flat face 79 than is the other point 86.

The sloping structure of the crossbars 77 is provided for maintaining a cross section in the bars proportioned to the quantity of liquid that flows through the cross section as affected by the distance over which the liquid must flow through the bar to reach the desired strips 68 while also keeping the cross-sectional area open for venting the vapor from each region proportioned to the vapor flow from that region. The same principle is responsible for the bevel 73 on the wick 71 and the bevel 76 on the ring 72. The rationale behind the shape of the crossbar can be further recognized by noting the dashed lines 87 on FIG. 13 which represent the bounds of the area of surface structure to which liquid is delivered by one of the crossbars 77. Liquid is delivered to regions outside the bounds of the lines 87 by the other crossbar or the crescent-shaped portions of the ring 72. From these bounds it will be seen that the area to which liquid is delivered nearer the center of the heat pipe is larger than the area more remote from the center. It will also be noted that the distance liquid need flow in the crossbar from its intersection with the ring to the center is greater on the side 82 nearer the center of the heat pipe, and shorter on the side 83 more remote from the center of the heat pipe.

The sloping faces 78 on the crossbars and the bevels 73 and 76 on the wick and ring, respectively, are provided for enlarging the area through which vapor passes as it leaves the region bounded by the channels 69. The slopes and bevels provide the largest possible cross-sectional area and shortest path length for the vapor without significant hindering of liquid flow cross section. It should be recognized that the regions bounded by the two crossbars 77, and between the crossbars and ring form a manifold with and condensing relatively large passages in serial flow connection with the more numerous smaller channels 69 which are further in fluid communication with the smallest most numerous channels 67 adjacent the heat source surface, thus assuring that all the vapor formed at the regional areas of vaporization on the surface of the porous layer 64 flows into the vapor way within the central passage of the wick. In this way, the vented capillary vaporizer illustrated in FIGS. 11 through 16 for a wick-fed heat pipe is analogous to the compound vented capillary vaporizer hereinabove described and illustrating in FIGS. 4 and 5 except that instead of feeding liquid to the vaporizer from a source of bulk liquid, as in FIGS. 4 and 5, the liquid reaches the heat pipe vaporizer by way of a wick 71 and the vapor is delivered to a vapor way instead of being directly returned to mix with the bulk liquid.

FIGS. 17 through 19 illustrate the fourth type of vented capillary vaporizer which can be considered to be a capillary pump wherein the output vapor pressure can be higher than the input bulk liquid pressure. FIGS. 17 and 18 are cross sections through a single region of heat transfer surface structure and it should be understood that a plurality of side-by-side structures such as illustrated in FIG. 17 may be repeated indefinitely to cover a large surface area. A single repetition of the structure such as illustrated in FIG. 17 may, for example, be provided every one-half inch or more along the surface, and such a structure may have an indefinite length in a direction along that of FIG. 18 as may be limited only by the flow capacity of the liquid flow conduits, and cooling means provided as hereinafter described. The structure illustrated is, of course, only part of the heat transfer system and elsewhere means are provided for containing and condensing or releasing the vapor formed.

FIG. 19 is an enlarged detail of the structure adjacent a heat source wall 89. Immediately adjacent the wall, and in thermal contact therewith, are a multiplicity of rectangular strips 90 forming a multiplicity of channels 91 and an equal number of regional areas of vaporization. Running transversely to the fine strips 90 are larger strips 92, having channels 93 therebetween, and these larger strips 92 are in contact with still larger strips 94, having channels 95 therebetween. Thus, the structure nearest the heat source provides a hierarchy of strips 94, 92 and 90 of decreasing size and increasing number for conducting liquid toward the heat source wall, and a hierarchy of channels 91, 93 and 95 of increasing size and for conducting vapor away from the heat source wall and directing it into the gaps between adjacent delivery structures for flow into the vapor way common to several structures such as illustrated in FIG. 17 making up the heat transfer surface structure.

The arrangement provided in FIGS. 17 to 19 differs from the related structure hereinabove illustrated in FIGS. 4 and 5 in that liquid is delivered to the larger strips 94 by a thermal isolation matrix 96 in contact therewith. The thermal isolation matrix is a porous material having a high wicking efficiency $\epsilon_w$ and as low a thermal conductivity as possible. The structure illustrated in FIGS. 17 and 18 also differs from that hereinabove illustrated in FIGS. 4 and 5 in that the vapor generated adjacent the heat source surface is collected in a vapor way external to the liquid delivery structures while remaining well isolated from the incoming bulk liquid.

The thermal isolation matrix 96 is in intimate thermal contact with a matrix cooler plate 97 formed of a high thermal conductivity metal. Transverse grooves 98 in the cooler plate 97 transmit liquid from approximately triangular conduits 99 to the cooler surface of the thermal isolation matrix 96. One wall of each of the liquid conduits 99 is formed by the wall 101 of a conventional heat pipe. The heat pipe also comprises a conventional wick 102 and an open vapor passage 103 for extracting heat from liquid in the conduits 99 and from the matrix cooler plate 97 for delivery to a cooler heat sink (not shown). A layer of thermal insulation 104 surrounds the heat pipe and liquid conduits and a metal sheath 105 and channel wall 106 prevent liquid in the conduits and vapor surrounding the structure for contacting the insulation. The liquid channel wall 106 also conducts heat to the heat pipe to help keep the liquid in the channel cool.

In operation, the capillary pump-type of vented vaporizer, illustrated in FIGS. 17 and 18, takes bulk liquid that is not in a capillary structure into the conduits 99 for delivery to the porous thermal isolation matrix 96 by way of the transverse grooves 98.

The bulk liquid entering the conduits 99 is at a relatively lower pressure than the vapor escaping from the channels 95 nearer the heat source surface, and the liquid pressure continues to drop as it flows through the conduits 99, grooves 98, and the various capillary matrix structures. In order to prevent the occurrence of vapor in the liquid flow path, the temperature of the liquid must be sufficiently low at any point that its vapor pressure at that temperature is less than the sum of the liquid pressure and the bubble pressure at that point. As previously pointed out, the bubble pressure is $\sigma\delta$ or $\sigma(1/R_1+1/R_2)$ where $R_1$ and $R_2$ are the major radii of curvature of the liquid vapor interface of the largest bubble that can form in the pore or channel. When the liquid wets the pore or channel walls, then $R_1$ and $R_2$ are approximately half the width and half the thickness of the pore or channel, respectively. Thus, for a circular pore of radius $R_p$ the bubble pressure is $2\sigma/R_p$. In the microscopic structures, such as the conduits 99 and grooves 98, $R_1$ and $R_2$ are generally so large that the bubble pressure is negligible, and the liquid in them must remain cool enough that the vapor pressure of the bulk liquid is less than the liquid pressure. In the very small pores of the capillary matrix, however, the radii $R_1$ and $R_2$ are also very small or, which is equivalent, the effective pore surface to volume ratio $\delta$ is large, so that the bubble pressure can be quite high, and the vapor pressure may be considerably greater than the liquid pressure without any bubbles forming. Thus, the temperature within such a capillary matrix may be slightly greater than the temperature of the vapor formed at a free surface of the capillary matrix without forming vapor bubbles in the matrix, while the temperature of the bulk liquid must be kept enough cooler than the temperature of the vapor formed at the free surface of the capillary matrix to balance the liquid-vapor pressure difference.

In order to keep the bulk liquid sufficiently cool to prevent its vaporization, the thermal isolation matrix is selected with as low a thermal conductivity as possible for minimizing the quantity of heat transferred from the vapor within the channels 95 through the matrix to the face in contact with the bulk liquid. Such thermal isolation by means of a low thermal conductivity material is sufficient in some circumstances such as, for example, as in an apparatus as described and illustrated in the aforementioned copending patent application entitled "-Heat Transfer Device With Isolated Fluid Flow Paths." If the thermal isolation is not sufficient, a heat pipe such as illustrated in FIGS. 17 and 18 is provided in thermal contact with the bulk liquid for extracting surplus heat therefrom for preventing vaporization. A heat pipe is preferred in such an application because of the high rate of heat transfer available with relatively small temperature difference; however, it is to be understood that other active or passive cooling means than the heat pipe can be employed as desired, such as, for example, a fluid flow cooling tube. or a heat radiating or conducting rod or fin, or a thermoelectric cooling device.

In each of the above-described vented capillary vaporizers, it is preferred that the vapor passages in the portion immediately adjacent the heat transfer surface be spaced apart by less than about 0.1 inch. If the passages are spaced apart by appreciably more than about 0.1 inch, the additional expense of preparing the structure is not justified by the increase obtained in heat flux density The vented capillary vaporizer structure is more expensive to fabricate than most simple heat transfer structures. Its major advantage is that it is capable of handling much higher heat flux densities than any but the most expensive and complex systems, which must use high pressure pumps and under-cooled liquids in order to obtain comparable heat flux densities. This is true of those devices that exploit in full the highly advantageous characteristics of the vented capillary vaporizer.

As shown in equation 60 and 61 hereinabove, the maximum heat flux density $H_{max}/A_v$ is proportional to the total length of vaporizer strip per unit heat source area, which, in turn, is proportional to the number of passages or regional areas per unit length $n_p$ for passages parallel to the heat source wall. That is, $H_{max}/A_r \sim n_p$. This can also be expressed in terms of the nearest neighbor distance $d$ between passages or regional areas adjacent the heat source surface. $H_{max}/A_r \sim 1/d$, or $H_{max} \sim A_r/d$. In this form, the proportionality holds, not only for passages parallel to the heat source wall, but also for any shape of vented capillary vaporizer so long as the shape remains unchanged while all dimensions are scaled proportionately. A particularly interesting example is a square array of round passages of diameter D perpendicular to the heat source surface. Here, the maximum heat flux density is approximately the same as for the passages parallel to the heat flux surface so long as $D/d=2/\pi$ since the total effective length of "vaporizer strip" per unit heat source area is the same for both examples. In the terminology used herein a passage close and parallel to the heat source surface creates a single regional area of vaporization along it, which for purposes of calculation is divided into two vaporizer strips.

In order to evaluate how small the passage spacing $d$ must be made for the vented capillary vaporizer to be economically competitive, one can best compare its maximum heat flux density with the maximum obtainable with the liquid vaporizing situation of pool boiling. Since the heat flux densities obtained are a function of the liquid employed, water is specified in both examples. Such an example has been set forth hereinabove for the vented capillary vaporizer as equation 63 wherein it was pointed out that $H_{max}/A_v=34.5n_p$ watts/cm.$^2$, or substituting $d=1/n_p$, the maximum heat flux density is $H_{max}/A_v=34.5/$ watts/cm.$^2$.

The maximum heat flux density available for water in pool boiling is about 100 watts/cm.$^2$. Thus, in order to obtain comparable performance $34.5/d=100$ watts/cm.$^2$, or $d=0.345$ cm., or 0.136 inch. In order to be economically practical, the spacing $d$ should afford some heat flux advantage over a simple and inexpensive pool boiling situation. Thus, a spacing $d$ between channels less than about 0.1 inch is preferred. Usually, the spacing between passages or regional areas will be much smaller than 0.1 inch since manufacturing costs of the heat transfer surface structure go up less rapidly than the number of channels produced until very high values of $n_p$ are reached, that is, the channels become a very short distance apart. With a heat transfer surface structure having channels closer than about 0.1 inch spacing, it is usually less expensive to fabricate a smaller area of higher maximum heat flux density than a lower area of lower maximum heat flux density having the same total heat flux capacity.

Several embodiments of heat transfer surface structures incorporating principles of this invention have been described and illustrated herein. It will be apparent, however, that many modifications can be made in these structures. Thus, for example, such structures can be employed in heat transfer environments other than in a heat pipe or in a boiler or refrigeration evaporating tubes as discussed hereinabove. The structures have been described for vaporization of liquid, however, many of the structures described and illustrated are also suitable for condensing vapor, such as, for example, at the cooler end of a heat pipe, when the condensed liquid is removed with a wick or as bulk liquid at sufficient temperature and pressure to prevent bubble formation. The structure is in most cases identical, that is the channel pattern is the same. The material may be different for providing different thermal conductivity and effective pore surface to volume ratio. Otherwise, the only difference is that the direction of flow of liquid and vapor is reversed within and near the structure. A number of geometrical shapes have been identified for structures providing liquid to a heat transfer surface in one area and permitting vapor to escape in another area. Other structures performing this function will be apparent to one skilled in the art.

What I claim is:

1. A heat transfer surface structure comprising:
   - a heat source surface;
   - a quantity of vaporizable liquid and its vapor;
   - a capillary matrix wet by the liquid and having at least a first surface portion in thermal contact with the heat source surface, a second surface portion in contact with the vaporizable liquid, and a third surface portion from which the principal vaporization of liquid from the capillary matrix takes place, said third surface portion being arranged as a multiplicity of regional areas of vaporization spaced sufficiently closely to each other and to the first surface portion that the volume of the capillary matrix through which the liquid must pass between the second surface portion and the third surface portion remains liquid filled;
   - a multiplicity of vapor passages sufficiently larger than the capillary size of the capillary matrix to be vapor filled, said vapor passages being in vapor communication between the regional areas of vaporization of the third surface portion and a region external to the capillary matrix away from which vapor can flow, and wherein a

multiplicity of said vapor passages are embedded substantially into the capillary matrix material.

2. A structure as defined in claim 1 wherein said regional areas of vaporization are spaced apart by less than 0.1 inch.

3. A structure as defined in claim 1 wherein the first surface portion is substantially uniformly separated throughout its extent from the second surface portion.

4. A heat transfer surface structure comprising:

a heat source surface;

a quantity of vaporizable liquid and its vapor;

a capillary matrix wet by the liquid and having at least a first surface portion in thermal contact with the heat source surface, a second surface portion in contact with the vaporizable liquid, and a third surface portion from which the principal vaporization of liquid from the capillary matrix takes place, said third surface portion being arranged as a multiplicity of regional areas of vaporization spaced sufficiently closely to each other and to the first surface portion that the volume of the capillary matrix through which the liquid must pass between the second surface portion and the third surface portion remains liquid filled;

a multiplicity of vapor passages sufficiently larger than the capillary size of the capillary matrix to be vapor filled, said vapor passages being in vapor communication from the regional areas of vaporization of the third surface portion through the capillary matrix to a region external to the capillary matrix away from which vapor can flow.

5. A structure as defined in claim 4 wherein said regional areas of vaporization are spaced apart by less than 0.1 inch.

6. A structure as defined in claim 4 wherein the capillary matrix comprises a plurality of bodies of capillary material separated at least in part by intervening vapor passages.

7. A structure as defined in claim 4 wherein the first surface portion is substantially uniformly separated throughout its extent from the second surface portion.

8. A heat transfer surface structure comprising:

a heat source surface;

a quantity of vaporizable liquid and its vapor;

a capillary matrix wet by the liquid and having at least a first surface portion is thermal contact with the heat source surface, a second surface portion in contact with the vaporizable liquid, and a third surface portion from which the principal vaporization of liquid from the capillary matrix takes place, said third surface portion being arranged as a multiplicity of regional areas of vaporization spaced apart less than 0.1 inch and sufficiently close to the first surface portion that the volume of the capillary matrix through which the liquid must pass between the second surface portion and the third surface portion remain liquid filled; and

a multiplicity of vapor passages sufficiently larger than the capillary size of the capillary matrix to be vapor filled, said vapor passages being in vapor communication between the regional areas of the third surface portion and a region external to the capillary matrix away from which vapor can flow.

9. A structure as defined in claim 8 wherein said capillary matrix comprises:

a first volumetric portion relatively nearer the first surface portion and having a relatively higher thermal conductivity; and

a second volumetric portion relatively further from the first surface portion and having a relatively lower thermal conductivity.

10. A heat transfer surface structure comprising:

a heat source surface;

a quantity of vaporizable liquid and its vapor;

a capillary matrix wet by the liquid and having at least a first surface portion in thermal contact with the heat source surface, a second surface portion in contact wit the vaporizable liquid, and a third surface portion from which the principal vaporization of liquid from the capillary matrix takes place, said third surface portion being arranged as a multiplicity of regional areas of vaporization spaced sufficiently closely to each other and to the first surface portion that the volume of the capillary matrix through which the liquid must pass between the second surface portion and the third surface portion remains liquid filled;

a multiplicity of vapor passages sufficiently larger than the capillary size of the capillary matrix to be vapor filled, said vapor passages further comprising:

a first multiplicity of passages spaced relatively more closely together and situated for receiving vapor from the regional areas of vaporization; and

a second multiplicity of passages spaced relatively less closely together and in vapor communication between the first multiplicity of passages and a region external to the capillary matrix away from which the vapor can flow.

11. A structure as defined in claim 10 wherein said regional areas of vaporization are spaced apart by less than 0.1 inch.

12. A structure as defined in claim 10 wherein the capillary matrix further comprises:

a first volumetric portion relatively nearer the first surface portion and having a relatively larger effective capillary surface to volume ratio $\delta$, and

a second volumetric portion relatively further from the first surface portion and having a relatively smaller effective capillary surface to volume ratio $\delta$.

13. A heat transfer surface structure comprising:

a heat source surface;

a quantity of vaporizable liquid and its vapor;

a capillary matrix wet by the liquid and having at least a first surface portion in thermal contact with the heat source surface, a second surface portion in contact with the vaporizable liquid, nd a third surface portion from which the principal vaporization of liquid from the capillary matrix takes place, said third surface portion being arranged as a multiplicity of regional areas of vaporization spaced sufficiently closely to each other and to the first surface portion that the volume of the capillary matrix through which the liquid must pass between the second surface portion and the third surface portion remains liquid filled, and further comprising a first volumetric portion relatively nearer the first surface portion and having a relatively larger effective capillary surface to volume ratio $\delta$ and a second volumetric portion relatively further from the first surface portion and having a relatively smaller effective capillary surface to volume ratio $\delta$, and

a multiplicity of vapor passages sufficiently larger than the capillary size of the capillary matrix to be vapor filled, said vapor passages being in vapor communication between the regional areas of vaporization of the third surface portion and a region external to the capillary matrix away from which vapor can flow.

14. A structure as defined in claim 13 wherein said capillary matrix further comprises:

a first volumetric portion relatively nearer the first surface portion and having a relatively higher thermal conductivity; and

a second volumetric portion relatively further from the first surface portion and having a relatively lower thermal conductivity.

15. A heat transfer surface structure comprising:

a heat source surface;

a quantity of vaporizable liquid and its vapor;

a capillary matrix wet by the liquid and having at least a first surface portion in thermal contact with the heat source surface, a second surface portion in contact with the vaporizable liquid, and a third surface portion from which the principal vaporization of liquid from the capillary matrix takes place, said third surface portion being arranged as a multiplicity of regional areas of vaporization spaced sufficiently closely to each other and to the first surface portion that the volume of the capillary matrix through which the liquid must pass between the second surface portion and the third surface portion remains liquid filled, and further comprising a first volumetric portion relatively nearer the first surface portion having a relatively higher thermal conductivity and a second volumetric portion relatively further from the first surface portion and having a relatively lower thermal conductivity; and

a multiplicity of vapor passages sufficiently larger than the capillary size of the capillary matrix to be vapor filled, said vapor passages being in vapor communication between the regional areas of vaporization of the third surface portion and a region external to the capillary matrix away from which vapor can flow.

16. A structure as defined in claim 15 wherein the first surface portion is substantially uniformly separated throughout its extent from the second surface portion.

17. A heat transfer surface structure comprising:

a heat source surface;

a quantity of vaporizable liquid and its vapor;

a capillary matrix wet by the liquid and having at least a first surface portion in thermal contact with the heat source surface, a second surface portion in contact with the vaporizable liquid, and a third surface portion from which the principal vaporization of liquid from the capillary matrix takes place, said third surface portion being arranged as a multiplicity of regional areas of vaporization spaced sufficiently closely to each other and to the first surface portion that the volume of the capillary matrix through which the liquid must pass between the second surface portion and the third surface portion remains liquid filled; a multiplicity of vapor passages sufficiently larger than the capillary size of the capillary matrix to be vapor filled, said vapor passages being in vapor communication between the regional areas of vaporization of the third surface portion and a region external to the capillary matrix away from which vapor can flow;

means for thermally insulating the fluid adjacent the second surface portion of the capillary matrix from portions of the external ambient environment having a higher temperature than said fluid.

18. A heat transfer surface structure comprising:

a heat source surface;

a quantity of vaporizable liquid and its vapor;

a capillary matrix wet by the liquid and having at least a first surface portion in thermal contact with the heat source surface, a second surface portion in contact with the vaporizable liquid, and a third surface portion from which the principal vaporization of liquid from the capillary matrix takes place, said third surface portion being arranged as a multiplicity of regional areas of vaporization spaced sufficiently closely to each other and to the first surface portion that the volume of the capillary matrix through which the liquid must pass between the second surface portion and the third surface portion remains liquid filled;

a multiplicity of vapor passages sufficiently larger than the capillary size of the capillary matrix to be vapor filled, said vapor passages being in vapor communication between the regional areas of vaporization of the third surface portion and a region external to the capillary matrix away from which vapor can flow;

means for removing heat from the fluid adjacent the second surface portion of the capillary matrix, said means being capable of removing said heat even when, without said means, the region surrounding said fluid would have at least as high a temperature as the fluid.

19. A heat transfer surface structure comprising:

a heat source surface;

a quantity of vaporizable liquid and its vapor;

a capillary matrix wet by the liquid and having at least a first surface portion in thermal contact with the heat source surface, a second surface portion in contact with the vaporizable liquid, and a third surface portion from which the principal vaporization of liquid from the capillary matrix takes place, said third surface portion being arranged as a multiplicity of regional areas of vaporization spaced sufficiently closely to each other and to the first surface portion that the volume of the capillary matrix through which the liquid must pass between the second surface portion and the third surface portion remains liquid filled;

a multiplicity of vapor passages sufficiently larger than the capillary size of the capillary matrix to be vapor filled, said vapor passages being in vapor communication between the regional areas of vaporization of the third surface portion and a region external to the capillary matrix away from which vapor can flow;

active means for removing heat from the fluid adjacent the second surface portion of the capillary matrix.

20. A structure as defined in claim 19 wherein the active means for removing heat comprises a heat pipe.

21. A structure as defined in claim 19 wherein the active means for removing heat comprises a thermoelectric cooling device.

22. A heat transfer surface structure comprising:

a heat sink surface;

a quantity of fluid consisting of a vaporizable liquid and its vapor;

a capillary matrix wet by the liquid and having at least a first surface portion in thermal contact with the heat sink surface, a second surface portion through which the fluid is withdrawn from the capillary matrix, and a third surface portion at which the principal condensation of vapor takes place; said third surface portion being arranged as a multiplicity of regional areas of condensation;

a multiplicity of vapor passages sufficiently larger than the capillary size of the capillary matrix to be vapor filled; said vapor passages being in vapor communication between the regional areas of condensation of the third surface portion and a region external to the capillary matrix from which the vapor passages receive vapor.

23. A structure as defined in claim 22 wherein a multiplicity of said vapor passages are embedded substantially into the capillary matrix material.

24. A structure as defined in claim 22 wherein a multiplicity of said vapor passages are in vapor communication from the regional areas of condensation through the capillary matrix to the region external to the capillary matrix from which the vapor passages receive vapor.

25. A structure as defined in claim 22 wherein said regional areas of condensation are spaced apart by less than 0.1 inch.

26. A structure as defined in claim 22 wherein said vapor passages further comprise:

a first multiplicity of passages spaced relatively more closely together and so as to deliver vapor to the regional areas of condensation; and

a second multiplicity of passages spaced relatively less closely together and in vapor communication between the first multiplicity of passages and the region external to the capillary matrix from which the vapor passages receive vapor.

27. A structure as defined in claim 22 further comprising means for maintaining the pressure of the liquid in the capillaries of the capillary matrix lower than the pressure of the vapor adjacent the third surface portion of the capillary matrix.

28. A structure as defined in claim 22 wherein said capillary matrix comprises:

a first volumetric portion relatively nearer the first surface portion and having a relatively higher thermal conductivity; and

a second volumetric portion relatively further from the first surface portion and having a relatively lower thermal conductivity.

PO-1050
(5/69)

# UNITED STATES PATENT OFFICE
# CERTIFICATE OF CORRECTION

Patent No. 3,598,180 Dated August 10, 1971

Inventor(s) Robert David Moore, Jr.

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

Address of Inventor, "Road" should be --Real--.

Drawings, Sheet 1 of 4, FIG. 2, direction of arrow under "y" should be downward instead of upward.

Column 1, line 33, "Thus" should be --This--;

Column 4, line 35, "cn" should be --cm--;

line 76, "more" should be --most--.

Column 5, line 31, "dp/dx" should be --dP/dx--;

Formula 2, should be --$P_B = \sigma\delta$ --.

Column 6, line 29, "dt/dx" should be --dT/dx--.

Column 7, line 13, "equal" should be --equals-- (both occurrences)

line 56, insert --then-- between "H," and "since".

Column 9, line 6, "Thus" should be --This--;

line 14, insert --a-- between "not" and"practical";

line 37, insert a comma between "and y" and "where";

line 51, "x" should be --z--;

line 58, "AT" should be --At--.

Column 10, line 30, "$\Delta P \ _v$" should be -- $\Delta P_v$ --.

Column 11, Formula 36A should be --$\left(\frac{1}{t}\right) = \frac{1}{w}\int_0^w \overline{\left(\frac{1}{t}\right)}\, dz$ --;

... cont'd

PO-1050
(5/69)

# UNITED STATES PATENT OFFICE
# CERTIFICATE OF CORRECTION



Patent No. 3,598,180 Dated August 10, 1971

Inventor(s) Robert David Moore, Jr.

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

... cont'd

Column 11, line 16, "$1/_t$" should be --1/t--;

line 24, "dx" should be --dz--;

in formula 37A, "$\left(\frac{1}{y}\right)$" should be --$\left(\frac{\overline{1}}{t}\right)$--;

line 31, "1/t" should be --$\overline{1/t}$--;

in formula 38, "$^{1/3}$" should be --$^{1/2}$--

Column 12, line 5, "Thus" should be --This--;

line 16, "(1/t)" should be --$(\overline{1/t})$--;

in formula 44, "$\left(\frac{1}{t}\right)$" should be --$\left(\frac{\overline{1}}{t}\right)$;

formula 45, "β" should be --b--;

formula 48, no equal sign after "δ".

Column 13, formula 57, line 40, should be -- $2\frac{\sigma}{\Delta P_x}$ --;

formula 59, line 61, "B max" should be --H max--.

Column 15, line 30, insert --the-- between "nearest" and "heat";

line 48, "large" should be --larger--;

line 57, "require" should be --required--.

.. cont'd

PO-1050
(5/69)

# UNITED STATES PATENT OFFICE
# CERTIFICATE OF CORRECTION



Patent No. 3,598,180 Dated August 10, 1971

Inventor(s) Robert David Moore, Jr.

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

...cont'd

Column 16, line 50, "service" should be --serve--;

line 73, "2 6 " should be --26--.

Column 17, line 24, "heart" should be --heat--.

Column 18, line 3, insert --an-- between "and" and "efficiency";

line 4, insert a comma between "possible" and "and";

line 16, "of" (second occurrence) should be --or--.

Column 19, line 51, insert --the-- between "and" and "only";

line 55, should be --$\rho_V v^2/2$ --

Column 20, line 2, insert --of-- between "order" and "magnitude";

line 5, insert a period after "pressure" and "MOst" should be --Most--;

line 57, "drived-out" should be --dried-out--.

Column 21, line 13, "A" should be --a--.

Column 23, line 55, "22" should be --62--.

Column 24, line 21, "(not shown" should be --(not shown)--

Column 25, line 1, insert --or integral with-- between "with," and "the";

.. cont'd

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# UNITED STATES PATENT OFFICE
# CERTIFICATE OF CORRECTION



Patent No. 3,598,180 Dated August 10, 1971

Inventor(s) Robert David Moore, Jr.

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

cont'd

Column 25, line 51, delete "and condensing" between "with" and "relatively" and insert --three-- ;

line 62, "illustrating" should be --illustrated--.

Column 26, line 20, between "and" and "for" insert --decreasing number forming a vapor manifold--;

line 51, "for" should be --from--.

Column 27, line 1, "microscopic" should be --macroscopic--

line 59, should be --$H_{max}/A_v \smallsetminus n_p$--

Column 28, line 12, should be --34.5/d watts/$cm^2$--;

line 29, "lower" should be --larger--.

Column 29, line 41, "is" should be --in--;

line 52, "remain" should be --remains--;

line 73, "wit" should be --with--.

Column 30, line 36, "nd" should be --and--.

Signed and sealed this 21st day of November 1972.

(SEAL)
Attest:

EDWARD M.FLETCHER,JR.
Attesting Officer

ROBERT GOTTSCHALK
Commissioner of Patents