United States Patent

[11] 3,537,514

[72] Inventor William J. Levedahl Baltimore, Maryland
[21] Appl. No. 823,227
[22] Filed March 12, 1969 Division of Ser. No. 592,362, Nov. 7, 1966, now pending
[45] Patented Nov. 3, 1970
[73] Assignee By mesne assignment, to Teledyne, Inc., Los Angeles, California a corporation of Delaware. by mesne assignment

[54] **HEAT PIPE FOR LOW THERMAL CONDUCTIVITY WORKING FLUIDS**
2 Claims, 10 Drawing Figs.

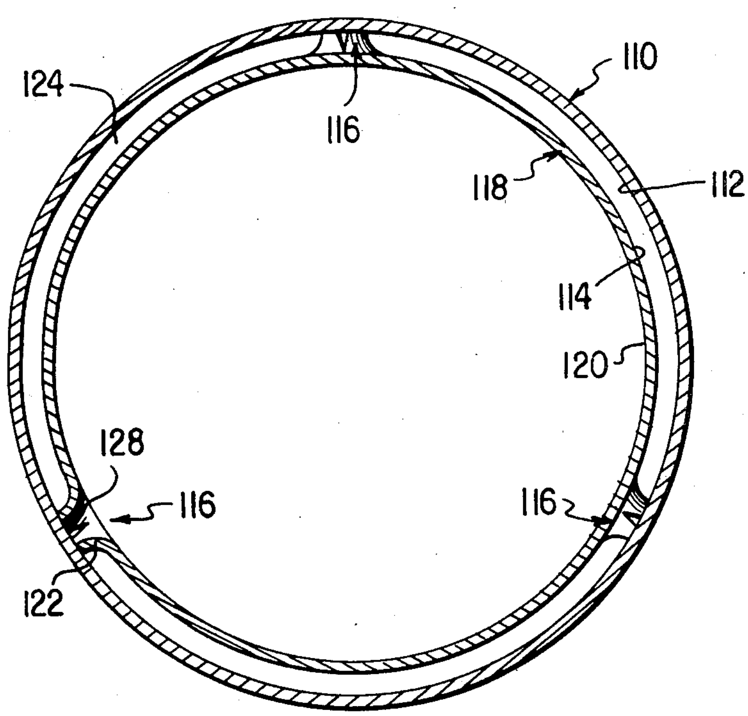
[52] U.S. Cl. 165/105, *137/13, *137/81.5; *138/178; *141/31 (*By disclosure only)
[51] Int. Cl. F28d 15/00
[50] Field of Search 165/105; 62/514

[56] **References Cited**
UNITED STATES PATENTS

3,305,005	2/1967	Grover et al.	165/105
3,402,767	9/1968	Bohdansky et al.	165/105
3,403,075	9/1968	Fiebelmann	165/105X
3,414,475	12/1968	Fiebelmann	165/105X

Primary Examiner—Albert W. Davis, Jr.
Attorney—Fleit, Gipple and Jacobson

ABSTRACT: An improved heat pipe fluid transport section, or capillary, is formed by a wall parallel to the heat pipe wall and spaced so as to provide a continuous capillary passage therebetween. The wall is perforated to allow for the passage of vapor into and out of the passage.



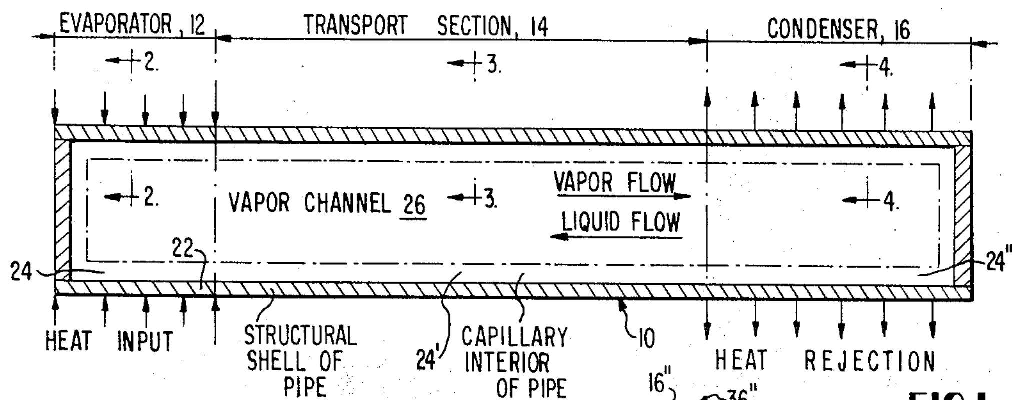


FIG.1

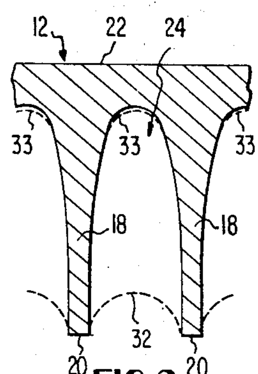


FIG.2

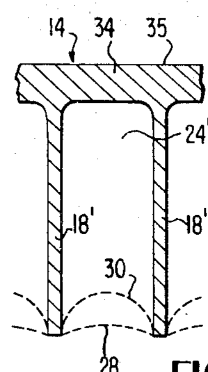


FIG.3

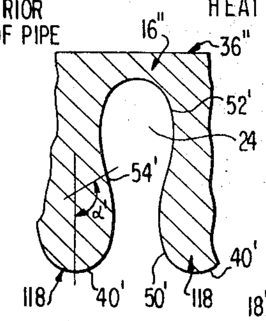


FIG.7

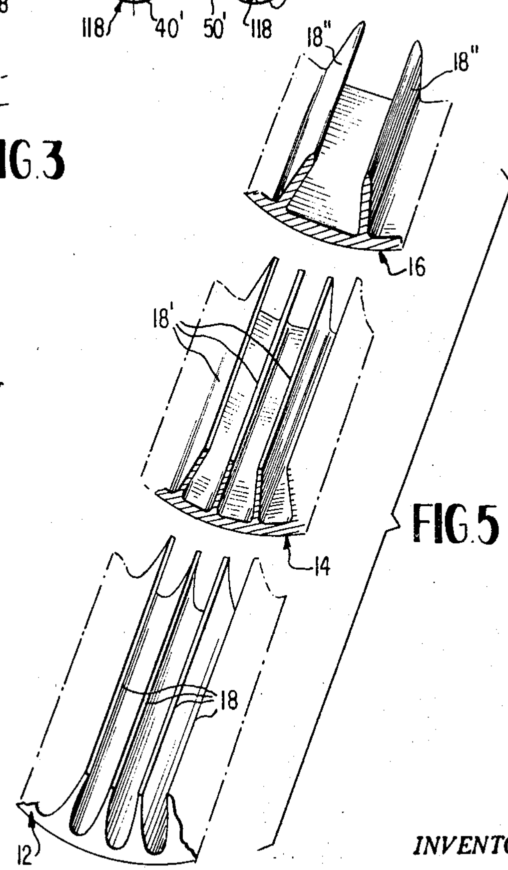


FIG.5

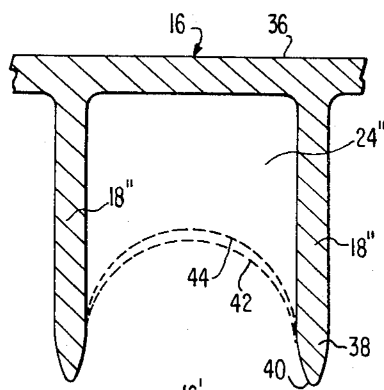


FIG.4

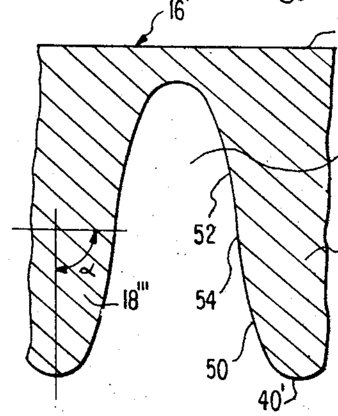


FIG.6

INVENTOR.
WILLIAM J. LEVEDAHL

BY Sughrue, Rothwell, Mion, Zinn & Macpeak
ATTORNEYS

FIG. 8

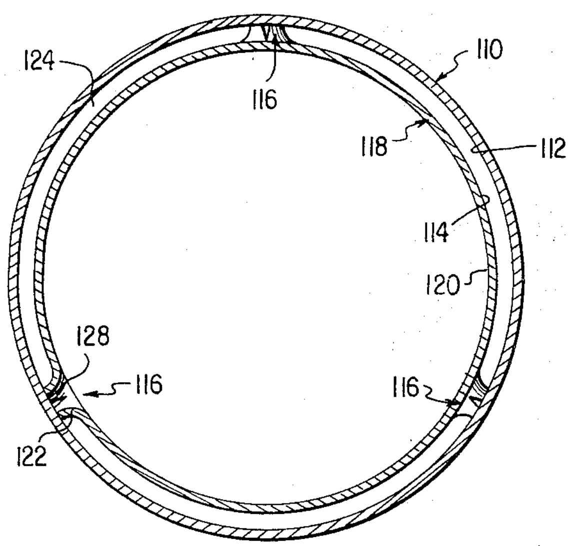


FIG. 9

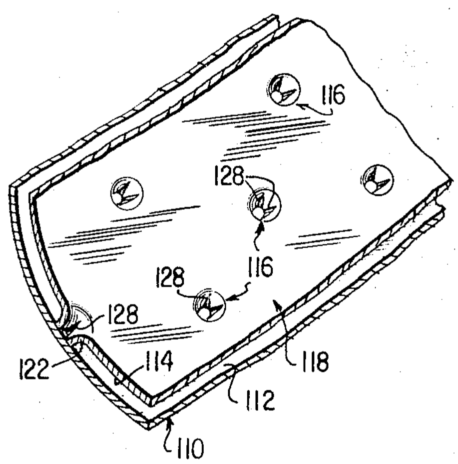
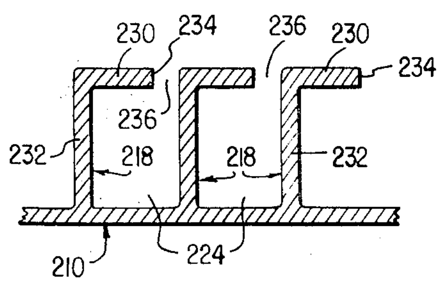


FIG. 10



INVENTOR.
WILLIAM J. LEVEDAHL

BY Sughrue, Rothwell, Mion, Zinn & Macpeak

ATTORNEYS.

HEAT PIPE FOR LOW THERMAL CONDUCTIVITY WORKING FLUIDS

This application is a division of U.S. Pat. application of Ser. No. 592,362 filed Nov. 7, 1966.

This invention relates to heat pipes and more particularly, to optimum heat pipe configurations especially useful with working fluids of low thermal conductivity.

The heat pipe has lately come into vogue and in its simplest form comprises a container, normally metallic, employing on the inner surface a capillary structure which is essentially saturated with a vaporizable fluid. The heat pipe acts to transfer heat, almost isothermally, from one point on the external surface to any other point by a vaporization-condensation cycle. In operation, a heat pipe consists essentially of four regions, each serving one primary function:

1. The evaporator which transfers heat to the inflowing liquid, thereby vaporizing it;
2. The vapor channel which permits vapor to flow from the evaporator to the condenser;
3. The condenser which condenses the vapor by removing heat from it; and
4. The capillary liquid transport section in which the liquid condensate flows back to the evaporator.

Each of these regions can impose a restriction on the usable power level or on the closeness with which the heat input temperature approaches the heat rejection temperature. Thus, each region should be designed in such a way as to maximize its own performance within the overall system's limitations which require compromises in order to optimize the overall heat pipe system. The heat pipe, therefore, functions as a reflux condenser, or evaporating-condensing device which uses a capillary or "wick" section to return condensed liquid from the condenser to the evaporator, replacing the liquid pump or the gravity-induced natural circulation flow in conventional systems. The heat pipe is a totally enclosed, simple, mechanically static device which can transport large quantities of heat over sizable distances essentially isothermally, and is not dependent upon gravity. It is also capable of acting as a thermal control device when used in special configurations and/or with the addition of inert gas.

Heat pipes have been developed primarily for use with high temperature high termal conductivity fluids, such as sodium and lithium. Water or other nonmetallic fluids have received less attention largely because of their low thermal conductivity which results in low overall heat transfer coefficients in the evaporator and condenser sections. Most heat pipes are circular in cross section and use rolled wire screen wicks. When high heat flux devices use working fluids of low thermal conductivity, such as water, phosphorus sesquisulfide, ammonia, hydrocarbons and most other nonmetallic working fluids, the condenser heat transfer coefficient tends to be poor, providing a heat rejection temperature significantly below the saturated vapor temperature. The high temperature difference is proportional to the heat flux, and is explained by the poor thermal conductivity of the relatively thick layer of condensate through which the heat must be transferred in such prior art heat pipes which utilize wicks of high porosity on the inner heat pipe surface.

The liquid transport section usually conducts little heat to or from the walls. Its length is dependent upon the application and may range from zero to being the longest section of the heat pipe. If long, it may provide the major limitation on the quantity of heat to be transported by the device.

The evaporator section may offer a major basic limitation in either of two ways: either by providing too low a heat transfer coefficient and requiring an excessive temperature difference between the heat source and the vapor, or by physically limiting the useful heat flux by "vapor-blanketing" the surface, resulting in "burnout".

In prior art devices, uniform screen wire mesh or wool is normally used to line the entire inside of the heat pipe. This material has poor thermal conductivity in the radial direction, thus the radial heat transfer coefficient through the saturated, relatively thick wick is nearly that of the fluid alone. Longitudinal flow through the wick is impeded by virtue of the many wires perpendicular to the flow path, thus all heat transfer and liquid flow functions are impeded by the wick configuration.

It is, therefore, a primary object of this invention to provide an improved heat pipe configuration wherein the capillary means offers minimum impedance to both the heat transfer and liquid flow.

It is a further object of this invention to provide an improved heat pipe which incorporates "ideal" capillary configurations to achieve for the separate sections, optimum condensation, evaporation and capillary transport of the working fluid.

It is a further object of this invention to provide an optimized heat pipe wherein the improved condenser configuration insures both a minimum fluid layer thickness between the condensing vapor and the surface on which the condensation takes place and a minimum temperature differential between this surface and the heat source.

It is a further object of this invention to provide an optimized heat pipe in which the improved configuration of the condenser section ensures rapid removal of the condensed fluid to maintain a condensation layer of minimum thickness.

It is another object of this invention to provide an optimized heat pipe configuration which achieves vaporization of the liquid within the evaporator section with minimum temperature difference between the vapor, the liquid and the external heat source.

It is a further object of this invention to provide an optimized heat pipe configuration characterized by minimum pressure drop throughout the vapor channel length.

It is a further object of this invention to provide an improved, optimized heat pipe configuration which employs capillary liquid transport means having maximum fluid flow capability per unit cross section with minimum frictional restriction to liquid flow.

It is a further object of this invention to provide an improved optimized heat pipe configuration in which the heat pipe evaporator, condenser and liquid transport sections may be separately formed and readily coupled in abutting end-to-end fashion.

Other objects of this invention will be pointed out in the following detailed description and claims and illustrated in the accompanying drawings which disclose, by way of example, the principle of the invention and the best mode which has been contemplated of applying that principle.

In the drawings:

FIG. 1 is a side elevation, in section, of the "optimized" heat pipe configuration of the present invention;

FIG. 2 is a vertical section of the heat pipe evaporator taken about lines 2–2 of FIG. 1;

FIG. 3 is a vertical section of the heat pipe transport section taken about lines 3–3 of FIG. 1;

FIG. 4 is a vertical section of the heat pipe condenser shown in FIG. 1 about lines 4–4;

FIG. 5 is an exploded, perspective view of a portion of the heat pipe shown in FIG. 1 stressing the abutting relationship of the extruded evaporator, transport and condenser sections in a preferred form;

FIG. 6 is a vertical section of an alternate embodiment of the improved heat pipe condenser configuration of the present invention;

FIG. 7 is a vertical section of yet another embodiment of the improved heat pipe condenser configuration of this invention;

FIG. 8 is a sectional view of an alternate embodiment of the improved fluid transport section of the present invention;

FIG. 9 is a perspective view of a portion of the heat fluid transport section of FIG. 8; and

FIG. 10 is a vertical section of yet another embodiment of the improved heat pipe fluid transport section of the present invention.

In general, the apparatus of the present invention comprises an improved heat pipe of optimized configuration consisting of a casing which defines a closed chamber including in series, condenser, fluid transport and evaporator sections. The closed chamber includes a vapor channel with a vaporizable liquid carried by the chamber. The improvement resides in capillary means formed by a number of spaced, generally parallel surfaces which act as an optimum uninterrupted capillary flow path or channel for condensed liquid moving from the condenser section to the evaporator section. The optimized configuration for the evaporator section is characterized by channels having a cross section of continuously decreasing radius of curvature in the direction toward the heat input surface. The condenser section also has an optimized configuration wherein parallel fins are characterized by a continuously increasing radius of curvature from the tip of the fin to some point at which the radius becomes infinite. Beyond this point the curvature may become concave and its radius progressively smaller.

The heat pipe of the present invention for use with working fluids of low thermal conductivity having an optimized configuration is indicated at **10** in FIG. **1** and includes the three basic sections common to all heat pipes: an evaporator section **12**, a transport section **14** and a condenser section **16**. The ends of the pipe are closed and are provided with capillary fluid transport means on the inner surface thereof. In order to transfer maximum heat flux in a generally isothermal fashion, the heat pipes require different cross-sectional configurations, including capillary transport means, for each section to effect optimized section operation. In the prior art devices employing, in general, screens, wicking, etc., in contact with the inner surface of the cylindrical heat pipe, it is obvious that the configuration for the evaporator, transport and condenser sections remain the same. In the "optimized" heat pipe of the present invention, for each section of the heat pipe, "ideal" capillary configurations are provided to achieve for the specific function of the respective section, optimum condensation, evaporation and capillary transport as required.

Referring then to evaporator section **12**, the evaporator **12** may have a very high heat flux, and in the prior art designs, are subject to "burnout" or film-blanketing. In order to improve the heat transfer coefficient, it is necessary to pass heat through only a relatively thin layer of liquid. Obviously, such a layer inevitably exists near the conjunction of the liquid meniscus with the essentially parallel walls of a channel. If the heat can be transported to this area, evaporation can take place with a minimum temperature differential and even if boiling occurs, the maximum heat transfer coefficient is achieved here. Thus, relatively thick-walled channels are desirable in the evaporator section. Only flat or concave surfaces are of value since the convex surfaces act to repel liquid rather than conduct the same. Referring to the cross-sectional view shown in FIG. **2**, the optimized configuration is shown to include relatively thick fins **18** which are square-edged providing the fins with the necessary capability of transferring maximum heat all the way from the outer surface **22** without a large temperature differential. The evaporator section, of course, should be made from metal or other good thermal conductor. Aluminum, for instance, may be easily extruded into the configuration shown in FIG. **2** and has a high coefficient of thermal conduction. The spaced fins **18** act to form a capillary channel **24** whose cross section is characterized by the tapering of the sidewalls inwardly, that is, away from the vapor channel indicated best at **26** centrally of the heat pipe. The cross-sectional configuration of the capillary section of the evaporator may be best described by stating that the channels formed by the fins have a continuously decreasing radius of curvature from the square-edged tips **20** toward the heat source, that is, exterior wall surface **22**.

In order to best appreciate the action of the low thermal conductivity liquid within capillary channel **24** in migrating away from the central vapor channel **26**, reference must be had to the capillary action of the same liquid as it moves through the transport section **14**. The liquid is transported through the capillary channels **24'** of the transport section **14** as a result of fluid pressure difference. With liquid constantly being condensed in the right-hand end of the heat pipe, a flat meniscus **28** of the liquid occurs within the capillary transport section adjacent the condenser portion **16** at the right-hand side of the heat pipe. Due to frictional loss, this is changed to a full or semicircular meniscus **30** at the left-hand end of the transport section when the heat pipe is operating at its maximum capacity. This same action continues in evaporator section **12** to continuously replace liquid being vaporized in this section. Since it is preferable to have vaporization occur as close to the heat input source as possible, there is provided a channel configuration characterized by a decreasing radius of curvature.

With the surface tension of the liquid tending to continuously move the liquid toward the small radius bottom of the capillary groove, the liquid at the ends of the semicircular meniscus **32** approximates a tangent position to the surface of fins **18** which at this point are nearly flat. This results in a nearly optimized condition under which maximum thin film surface area is utilized. Depletion of liquid within the channel or groove moves the meniscus toward the small radius bottom of the channel as indicated by dotted line **33** with evaporation closer to the heat source. Near the extreme left-hand end of the heat pipe, evaporation will occur at the point within capillary channel **24** most remote from the square-edged tips **20**. Note, however, regardless of capillary channel length or size, there will always be fluid flow to all points within the channel **24** due to normal capillary transport action whenever the heat input is at or below its maximum design value.

In order to continuously supply the liquid to be evaporated within the capillary channels **24**, it is necessary to transport, by capillarity, the liquid as it condenses within condenser section **16**. The mass rate of flow which can be transported by capillary action is directly proportional to the fluid's FIG. of merit, to the ratio of hydraulic diameter to channel length, and to the difference between the inverse of the meniscus radius at one end and at the other.

Reference to FIG. **3** shows in section the optimized configuration for the capillary channel of transport section **14**. The cross-sectional configuration is characterized by relatively thin channel fins **18'** having a flat surface extending perpendicular from the base section **34**. The most efficient available system is one having a hydraulic diameter equal to twice the capillary radius required at the capillary end, and consisting of thin radial fins extending inwardly from the wall. Note, however, that the smaller the width of channel **24'** the greater the effect of the wall resistance of the fins **18'** to the flow of liquid moving longitudinally of capillary channel **24'**. Thus, there must be some compromise between extremely close spacing of the walls for maximum capillary flow efficiency and the detrimental effect of the frictional restraint by the stationary walls to the moving body of liquid.

Under some conditions the best cross-sectional configuration of the transport section may be that of two concentric tubular members such as cylinders, with the inner cylinder separating the interior vapor flow channel from the exterior liquid flow channel. As shown in FIGS. **8** and **9**, outer casing **110** acts in a similar manner to the casing **10** of the embodiment of FIG. **1**. In this case, the inner concentric tubular member **118** acts in conjunction with outer casing **110** to form therebetween spaced, generally parallel surfaces **114** and **112** respectively, forming a capillary flow path **124** for condensed liquid and a concentric inner vapor channel **126**. Perforations **116**, in this case circular in shape, exist in the inner wall to permit the escape of any vapor which may become entrapped in the liquid channel **124** and at the same time allow liquid condensing on the inner surface **120** of tubular member **118** to move into the liquid flow channel **124**. While the perforations are circular in configuration, they may be rectangular or any other shape. As shown, in forming the circular perforations **116**, the forming tool displaces a portion **122** of wall member **118** radially outward towards the inner surface **112** of the concentric outer tube or casing means **110**. During manufacture the protruding section **122** splits, forming irregular slots **128** which allow the gases to escape from the liquid flow channel toward the central vapor channel **126**. It is noted that the end of deformed section **122** forming the circular opening **116** actually abuts the inner surface **112** of tubular casing **110** and therefore acts to locate and space the inner tubular member **118** with respect to outer tubular member **110**. Obviously, the randomly spaced perforations **116** may be otherwise formed, thus requiring some other means for maintaining the spatial gap between the capillary liquid transport surfaces **112** and **114**. In preferred optimized configurations employing this concept, the radial thickness of the liquid channel **124** would be approximately equal to the circumferential spacing between the fins in the evaporator section. The smallest dimension of any perforations such as **116** formed in the inner wall **118** should be less than this radial thickness. One advantage of the arrangement shown in FIGS. **8** and **9** is that a separation of the oppositely directed flows of liquid and vapor is achieved, thereby virtually eliminating the possibility of any flow instabilities resulting from normal interactions at the interface which exists between the liquid and gaseous phases. Another advantage of this configuration is that the hydraulic diameter is the largest possible for a given meniscus radius.

FIG. **10** shows another configuration for the liquid transport section of the heat pipe which is also designed to minimize the possibility of flow instabilities at the liquid-vapor interface, while eliminating the need for a separate inner, concentric member. The tubular casing **210** of the fluid transport section of the heat pipe has integrally formed fins **218** which are spaced slightly from each other. The fins are L-shaped in configuration having a base section **230** and a right-angle leg section **232**. The L-shaped fins are inverted with respect to the casing member **210** such that the free ends **234** of the base section lie generally in the same plane, but in the embodiment shown, are spaced slightly from the next succeeding adjacent fin. They act to form small circumferential gaps **236** which are fully equivalent to the perforations **116** of the previously described embodiment. Thus, in both these embodiments, the configuration of the fluid transport section tends to minimize the surface per unit volume of transported liquid while at the same time eliminating the shear surface at the interface between the liquid and gaseous phases of the embodiment shown in FIG. **3**. This is especially important in the heat pipe devices of the present invention in which the gas and liquid flow in opposite directions. With respect to the embodiment of FIG. **10**, it may be even preferable to position the fin sections **218** closer together to the point where the tip **234** of base section **230** is in abutting contact with the adjacent fin. This would act to completely enclose the capillary liquid flow channels **224**. The L-shaped fins could be extruded or otherwise formed as straight fins, and subsequently bent over to close or to nearly enclose the liquid flow channels **224**.

For maximum optimization, especially in the areas where the liquid volume is changing due to condensation or evaporation, it may be advisable to taper the channels slightly in an axial direction rather than conform to a strict parallel fin arrangement. In heat pipe configurations involving inner and outer tubular members, such heat pipe constructions are characterized by capillary flow surfaces which are generally equidistant from each other along their perimeters.

The radial depth of the fin or channel is limited only if the channel is to be tested or operated inverted to a gravity field or subjected to an acceleration, in which case the capillary force exerted by the menisci **28** or **30** must be sufficient to support the mass of fluid, such as water, above it. Of course, this limitation also holds for the evaporator and condenser sections. As liquid is condensed within condenser section **16** and flows toward the left-hand end of the condenser section, the meniscus tends to achieve a minimum radius equal to one-half the channel width **44** and retains this radius at the entry to the transport section as indicated at **28**, whereupon flow through the transport section is effected by the pressure differential which exists between this end of the transport section and the evaporator and at which the pressure is lower due to the sharper radius of curvature shown at **30**. While the evaporator section **12** is preferably of a heat conducting material, such as metal, there is no such requirement for the transport section, and as a matter of fact, it may be formed of a material having low thermal conductivity to prevent the dissipation of heat radially of the transport section. If formed of extruded metal, it may require an insulative shield about outer surface **35**.

An important function of the optimum configuration heat pipe for use with low thermal conductivity working fluids involves the employment of a condenser or condensation section wherein condensation can take place on a metal surface which is at a temperature as close as possible to the heat rejection temperature and further where the condensate is so rapidly removed that only a thin film remains at any point. Obviously, the condensation surface may be maintained close to the heat sink temperature if the condenser section employs sufficiently thick metallic fins whereupon there is no significant temperature gradient between the outer wall of the heat pipe and the inner tip of the fin. Reference to FIG. **4** shows one form of fin configuration for the capillary channel **24''**. The fins **18''**, preferably extruded of metal, extend at right angles, radially inward of the heat rejection surface **36**. In like manner to the fins **18** of the evaporator section **12**, the fins **18''** of the condenser section **16** are relatively thick, but unlike the evaporator fins, they do not terminate in square tips. The fins **18''** are characterized by tip sections **38** of continuously increasing radius of curvature from the apex **40** of the tip section **38** toward the heat rejecting surface **36** exterior of the pipe. The thickness of the condensing film may be reduced to a very low level by making the condensing surface convex with a relatively sharp radius at the apex **40** of the fin and with a continuously increasing radius of curvature inwardly. The cross section appears to be similar to half of an ellipse and such a configuration would operate satisfactorily. However, the optimum configuration follows a slightly different equation, approximated as follows:

$$\frac{r}{r_0}=\frac{1}{1-A\times 1} \tag{1}$$

where r is the radius of curvature at any point, r_o is the radius of curvature of the apex, A is a constant, and x is the distance along the surface from the apex of the fin. This curvature tends to give an essentially constant film thickness, with correspondingly constant heat flux and film surface temperature.

With fins having parallel surfaces below the tip, Equation (1) gives a distance along the constant-heat-flux, constant temperature curved surface on each side of the centerline $x=\frac{3}{4}\pi r_o$ or 3/2 the distance of a quarter circle of the radius r_0. The half thickness of the fin is approximately 1.16 r_0. If all condensation occurs on the curved surface, then the following equations would closely approximate the temperatures at the tip and base of the fin, as functions of the several parameters:

$$\theta=2.03\,\frac{r_0(Q/A)^{4/3}}{k\left(\frac{\sigma\lambda}{y}\right)^{1/3}} \tag{2}$$

where θ is the temperature difference across the film, (Q/A) is the condensation heat flux incident on the surface, k is thermal conductivity, σ is surface tension, λ is latent heat of vaporization, and γ is kinematic viscosity of the liquid.

The mean heat flux down the constant thickness portion of the fin is about 2.08 times the incident heat flux, and the temperature drop along this portion of the fin is approximately

$$\Delta T=\frac{2.08(Q/A)h}{k_f} \tag{3}$$

where h is the height of the fin, k_f is its thermal conductivity and ΔT is the difference between the tip and base temperatures.

With a continuously increasing radius of curvature away from the apex point **40**, the surface tension of the condensed fluid will cause the fluid to move rapidly and continuously toward the capillary channel 24'' between the lower essentially parallel surfaces 40 of the fins.

This configuration has several remarkable characteristics. The film temperature drop can be made as small as desired, within practical manufacturing limits, by decreasing the radius of curvature r_0. Correspondingly the theoretical heat fluxes are unlimited, but at some point the temperature drop in the fin corresponding to the required flow channel depth will become limiting. Theoretical heat fluxes of tens of watts per cm^2 are theoretically achievable with water using practical configurations at film temperature drops of 1°C.

The lower part of the channel beneath the curved tip sections 38 of the fins may have essentially parallel walls as indicated by walls 40. The thickness of the walls should be the minimum compatible with low temperature drop and with reliable fabricability. Obviously, the walls must be of sufficient thickness so that heat may be readily transferred from the tip section 38 as the heated vapor condenses to the heat rejection surface 36. The channel width required between fins in the condenser section must always be larger than the channel width in the transport and evaporator sections unless the fluid is to be permitted to cover part or all of the curved fin tips. Obviously, it is most desirable that the condensed liquid move rapidly inward toward the base of the condenser section, as indicated by dotted meniscus line 44, since if the liquid is allowed to build up sufficiently, as indicated by dotted meniscus line 42, some of the curved condenser tip section 38 will be covered by a liquid film preventing condensation under optimum conditions. The difference between the evaporator and the condenser channel thicknesses provides the capillary driving head for fluid transport while with the customary low performance condensers, the effective radius at the condenser is nearly infinite. Thus, this concept for a high performance condenser is most attractive when the liquid transport distance is short, and it may lose part of its advantage for extremely long pipes.

Reference to FIG. 5 shows the manner in which the extruded evaporator, transport and condenser sections are moved into abutting position and coupled to produce the heat pipe 10 of FIG. 1. Of course, the abutting relationship of individually extruded nonvarying sections of FIG. 5 may be modified to achieve maximum optimization for low thermal conductivity working fluids. For instance, it may be desirable to continuously extrude all three sections as a unitary element in which modification of the extrusion means achieves the necessary configuration changes depending upon the section momentarily being extruded. In changing from transport section 14 to condenser section 16, if a single material were used, it would be necessary to increase the capillary channel width, as well as to change from a constant thickness fin to one having tip portions approximating a half of an ellipse. Further, since the ends of the evaporator and condenser sections closest to the transport section must act to an increasingly greater extent in a partial transport function, the radial height of the fins could be varied. The radial height of the fins in the evaporator and condenser sections closest to the extreme outer ends may thus be reduced, further tending to maintain the liquid in these sections as close as possible to the heat input surfaces and heat rejection surfaces, respectively.

Turning to FIG. 6, there is shown an alternative embodiment of the invention insofar as the condenser section of the heat pipe is concerned. The condenser section 16' includes relatively thick fins 18''' which extend radially away from the heat rejection surface 36', the fins being spaced to form a liquid capillary channel 24'''. Again, the extreme tips 40' of the fins 18''' are rounded but in this case, all of the surfaces of the fins are characterized by a continuously increasing radius of curvature with the exception of the point of inflection 54, surface sections 50 and 52 on either side of the point of inflection 54 being curved. The surface 50 is of a continuously increasing radius of curvature in a positive sense, while surface 52 is of decreasing radius of curvature in a negative sense. With the continuously increasing radius of curvature from the tip of the condenser section, the surface tension of the fluid varies through a maximum at the tip to a minimum at the flattened area so that there is a natural tendency for the fluid to flow toward the infinite radius or flattened section at point 54. In actuality, fluid flow continues past point 54 due to inertia of the moving liquid body, reaching surface area 52 which has a continuously increasing radius of curvature in the negative sense to sustain liquid flow towards the heat rejecting surface 36' of this condenser section since surface tension of the thin film liquid continues to decrease.

Reference to FIG. 7 shows vertically, the second alternate configuration for the condenser section of the heat pipe. Condenser section 16'' includes relatively thick fins 118 which extend normally from the heat rejection surface 36''. The spaced fins 118 form a liquid capillary channel 124. The tips 40'' of the fins are rounded in like manner to the previous condenser embodiments, and the surfaces of the fins are also characterized by a continuously increasing radius of curvature from the tip 40'' inwardly to the point of inflection 54'. At this point, the fin surface of continuously increasing radius of curvature changes from one in a positive sense to a surface of decreasing radius of curvature in a negative sense. Thus, there is produced surface sections 50 and 52 on either side of the point of inflection, the surface 50' being of continuously increasing radius of curvature in a positive sense, while surface 52' is of decreasing radius of curvature in a negative sense. The primary difference between the embodiment shown in FIG. 7 and that of the previous embodiment of FIG. 6 is the fact that the point of inflection does not occur after the fin surface from the tip inwardly has covered an angle of curvature of exactly 90° but an angle of curvature in excess thereof. As indicated, the angle α' appears, in this embodiment, to be roughly 110° to 120° as contrasted to FIG. 6 α is exactly 90°.

While the evaporator and condenser sections of the heat pipe for the present invention are particularly adaptable for low thermal conductivity working fluids, these optimized sections have general application to working fluids having high or intermediate thermal conductivity, provided only that the fin material has significantly higher thermal conductivity coefficients than the fluid itself. The optimized capillary liquid transport sections of the present invention have equal application to both high or low thermal conductivity working fluids.

In all the embodiments, showing the different condenser sections, the maximum heat flux per degree temperature drop through the film is only approached when the film thickness from the condensed liquid is constant throughout the condenser section. Under the principles of the present invention, this is achieved by providing surface configurations wherein the flow velocity of the condensed liquid on the condenser surface is proportional to the volume of condensed liquid to thus maintain constant film thickness.

While the apparatus of the present invention is particularly applicable to heat pipe configurations which are cylindrical in form such as that provided by the abutting extruded cylindrical evaporator, transport and condenser sections as best seen in the exploded view of FIG. 5, the heat pipe may comprise tubular members other than cylindrical, having flat planar surfaces with fins normal thereto. It is to be understood therefore, that the work "tubular" is intended to cover heat pipes of rectangular or circular cross section, extruded or otherwise, as well as unobvious configurations such as ovoids, triangles, etc.

While the invention has been particularly shown and described with reference to preferred embodiments thereof, it will be understood by those skilled in the art that the foregoing and other changes in form and details may be made therein without departing from the spirit and scope of the invention.

I claim:

1. An improved heat pipe fluid transport section of optimized configuration comprising casing means of tubular configuration and a second tubular member positioned internally of said casing means with spaced generally parallel surfaces thereto forming continuous capillary means for moving condensed liquid therebetween in a direction axial to the heat

pipe, said second tubular member being perforated to allow liquid to pass into said capillary means and vapor to flow out of said capillary means.

2. The improved fluid transport section as claimed in claim 1 wherein said perforations are apertures randomly formed within said second tubular member to allow liquid condensing on the inner tubular wall surface to pass into the capillary flow channel and gases captured between said tubular members within said capillary flow channel to move into the centrally located vapor channel.