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Gray

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[54] **METHODS AND APPARATUS FOR HEAT TRANSFER IN ROTATING BODIES**

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[52] **U.S. Cl.** **60/39.66,** 60/39.71, 60/267, 165/86, 165/105, 416/96

[51] **Int. Cl.** **F02c 7/12**

[58] **Field of Search** .... 165/86, 105; 416/96; 60/39.66, 39.51 R, 39.71, 267

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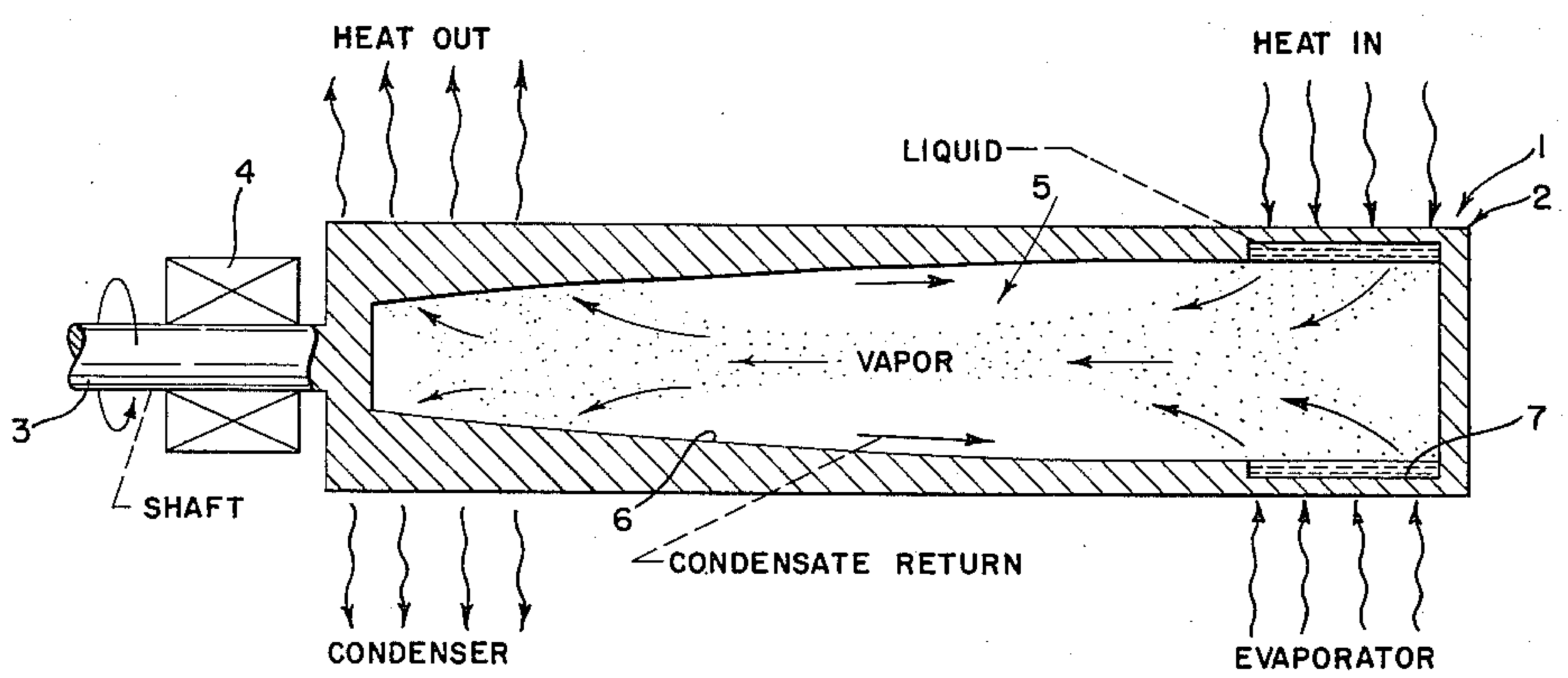
[57] **ABSTRACT**

Methods and apparatus for transferring heat axially in an axially rotating body, the body having a sealed co-axial cavity holding a small inventory of liquid, the cavity having generally convergently tapered axial walls, the body rotating at high speeds to form a substantially uniform liquid annulus at the wide end of the cavity, subjecting the body to external heat at the wide end of the cavity to vaporize the liquid, cooling the opposite end of the body to condense the vapor at the narrow end of the cavity, the rotation of such body being sufficient to produce centrifugal acceleration in the condensate large enough to have a centrifugal acceleration component parallel to the wall at any given point on the wall in excess of 1G at all points on the wall not covered by the liquid annulus; drill assemblies, motors, turbines and air conditioning devices employing the methods and apparatus of the invention.

6 Claims, 7 Drawing Figures



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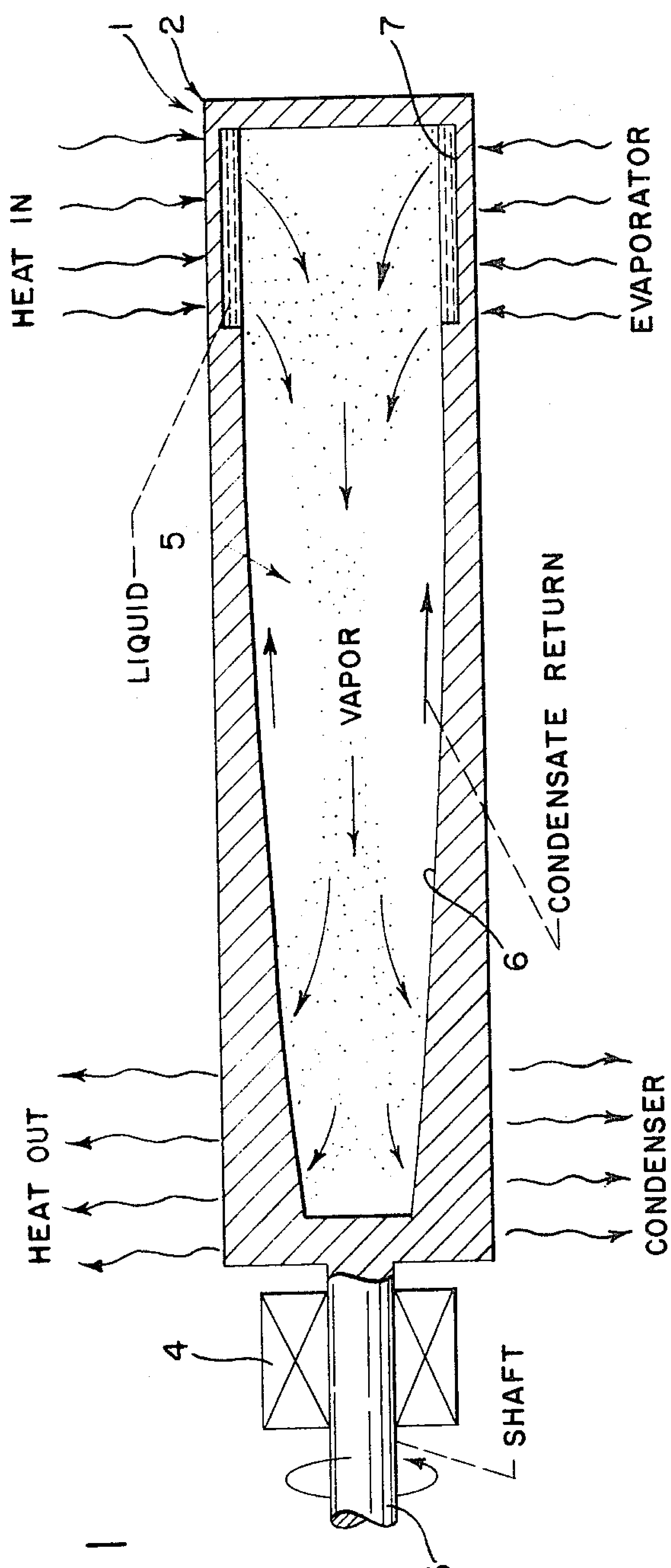

FIG. 1

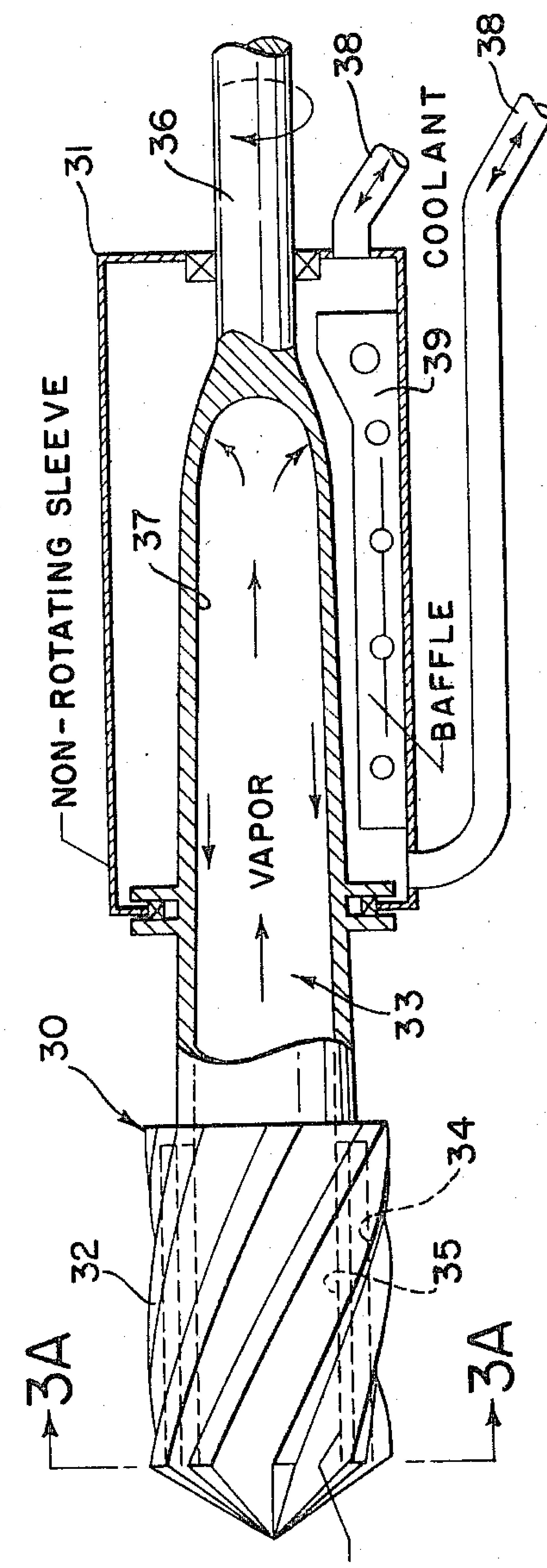

FIG. 3

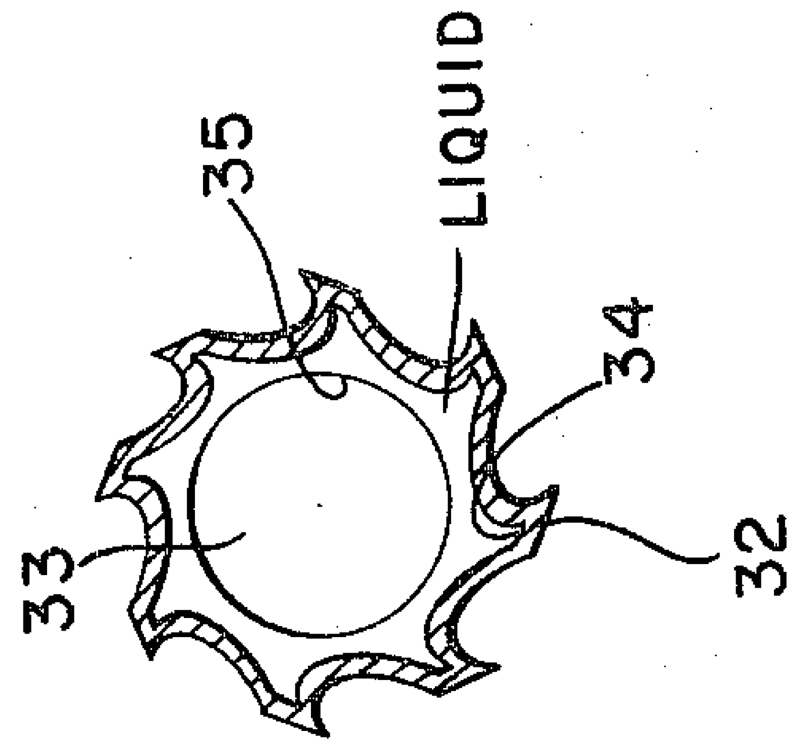

FIG. 3A

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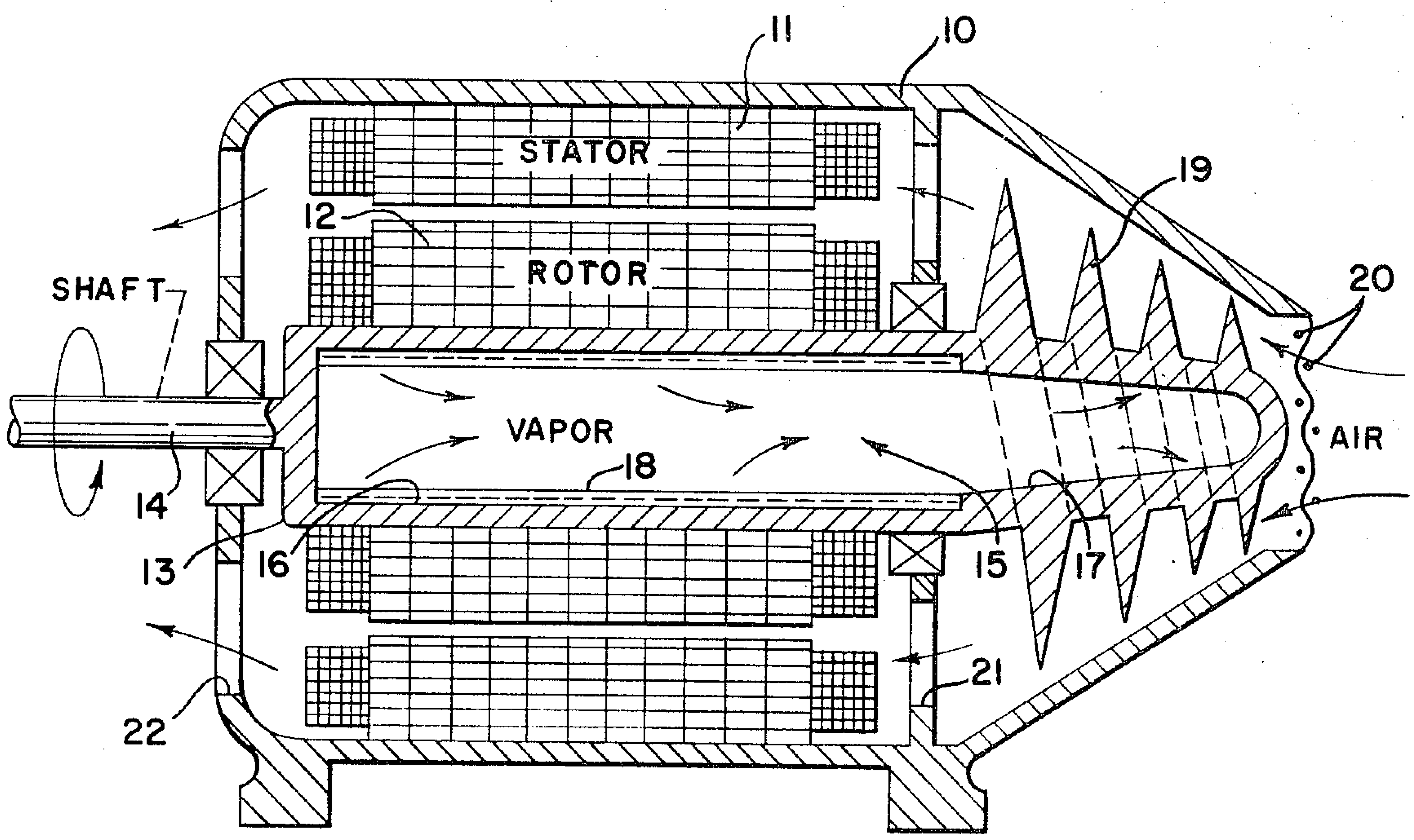

FIG. 2

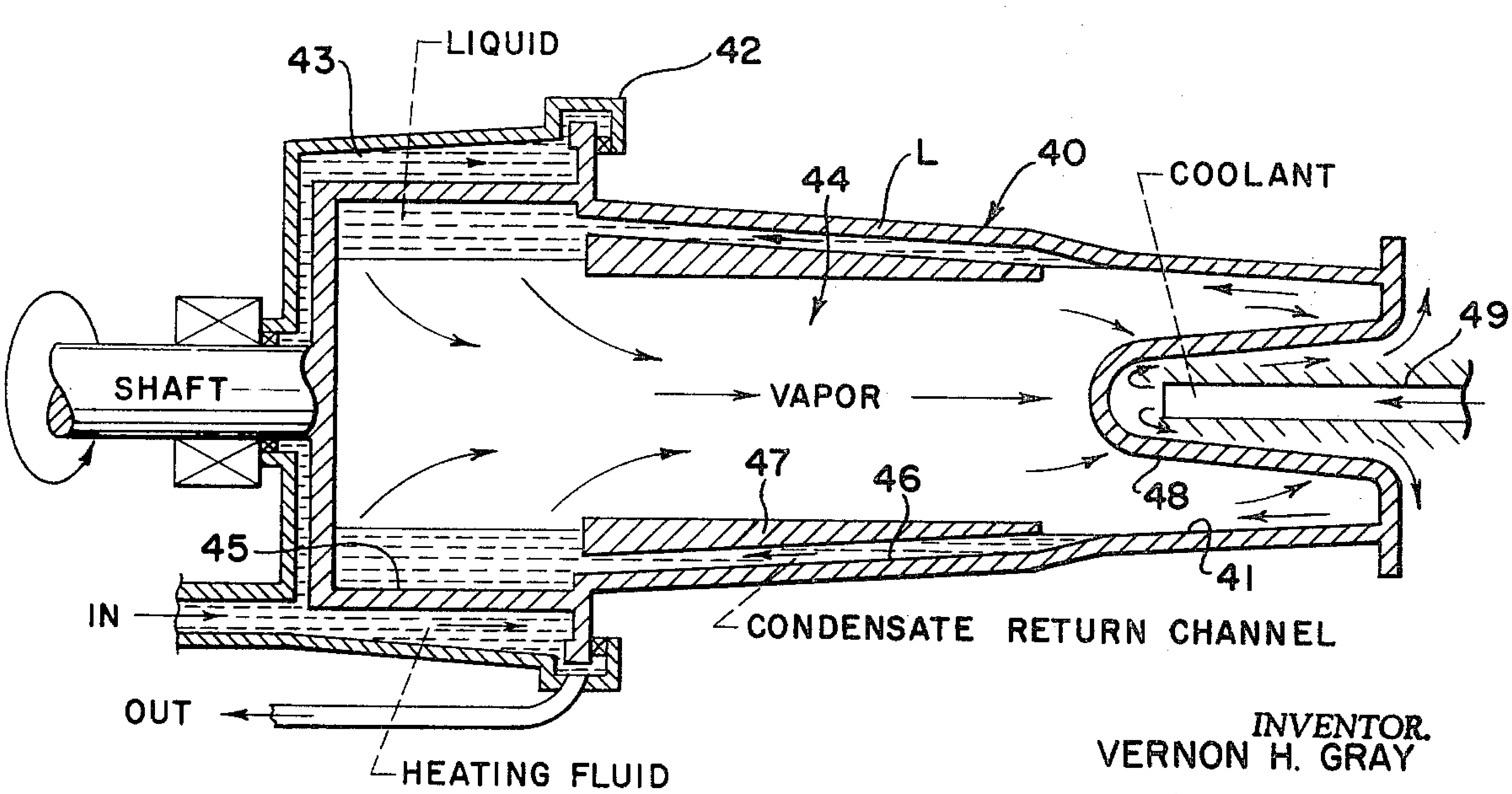

FIG. 4

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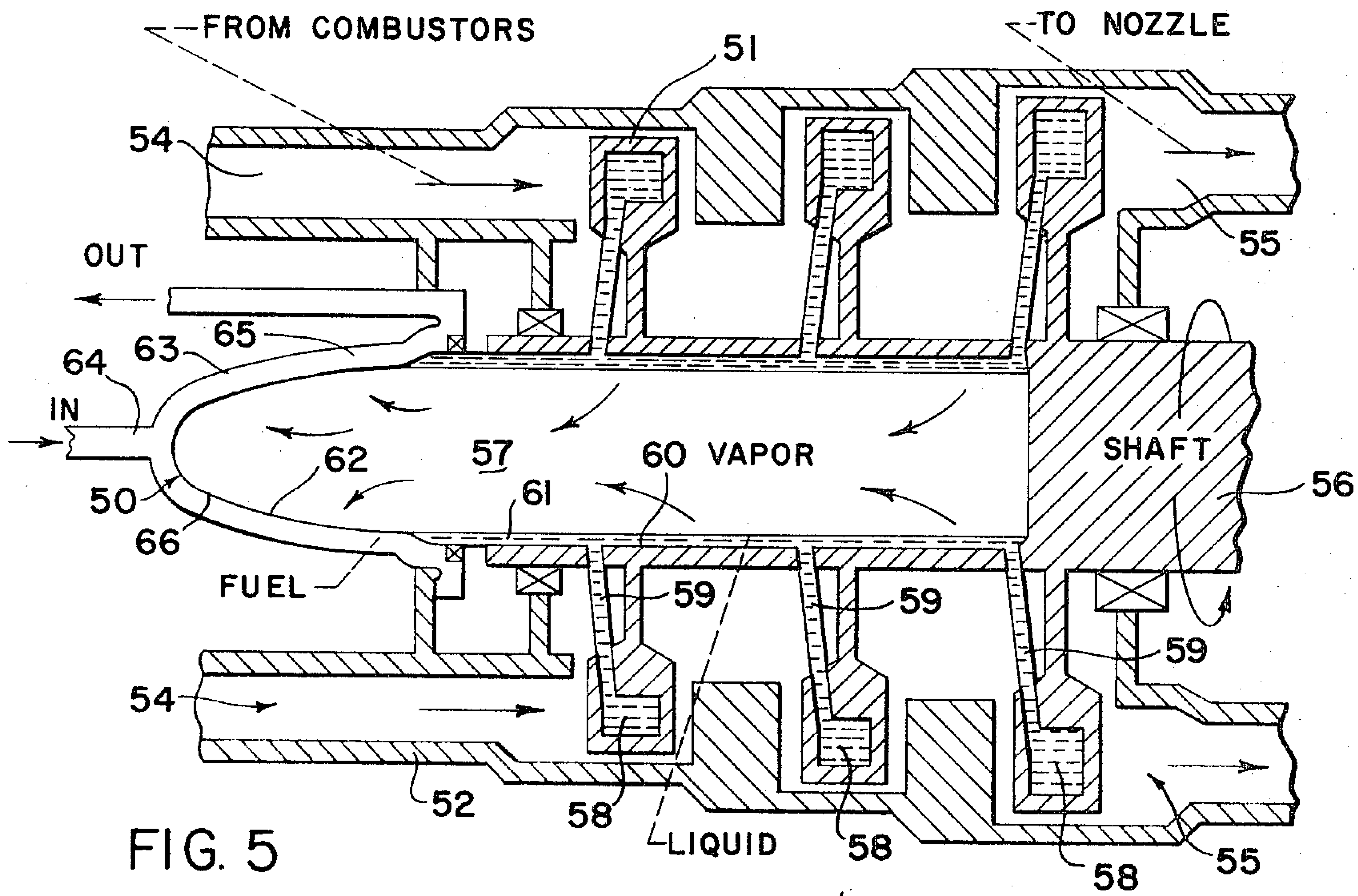

FIG. 5

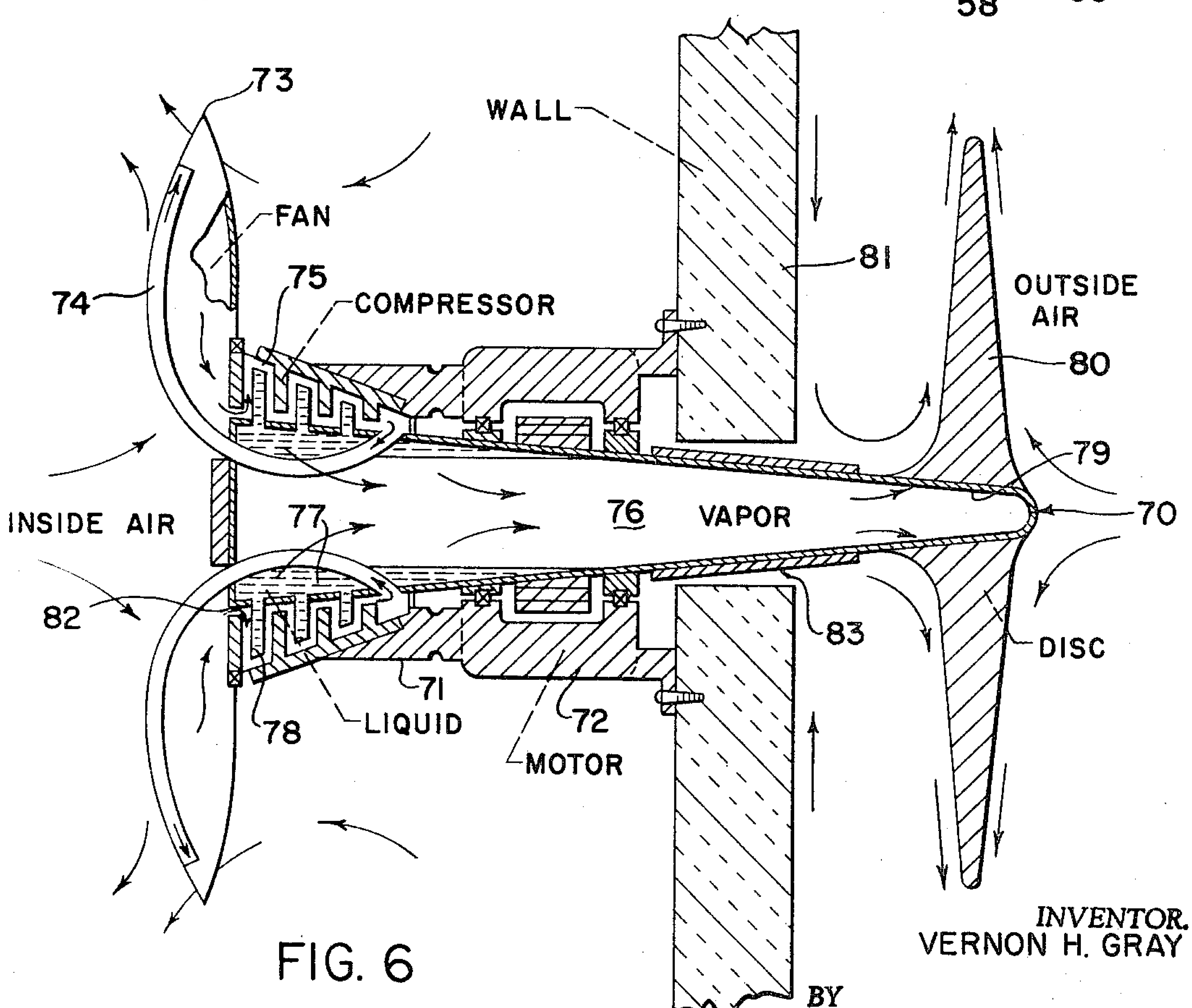

FIG. 6

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METHODS AND APPARATUS FOR HEAT TRANSFER IN ROTATING BODIES

BACKGROUND AND SUMMARY OF INVENTION

This invention relates generally to heat transfer and more particularly to the heating and cooling of rotating bodies and to improvements in "heat pipes."

Where a rotating body is subjected to heat or tends to heat up in operation, such as in the case of the rotor of an electric motor or in the case of the bit of a power drill, it is often beneficial and sometimes essential to hold the temperature of the body below certain limits. In the case of a drill bit, for example, the metal will lose its hardness if it is allowed to reach temperatures above a certain level. In the case of a rotor in a motor or turbine, significantly less expensive materials may be employed if temperatures are minimized.

For these and other reasons, considerable attention has been devoted to means for cooling rotating bodies.

In the normal case, a rotating body undergoes a limited degree of natural cooling by conduction from the hotter regions to the cooler regions, and by surface radiation and convection which is typically enhanced by the velocity with which the body rotates in the ambient medium. Standard means for improving cooling are to lower the temperature of the ambient medium or to increase circulation of the medium (e.g., by flowing oil over the drill bit and drill site as is common in industrial drilling operations).

A further method has been to increase the surface area of exposed portions of the rotating body, such as with fins, vanes or the like.

Still another technique has been to fill the interior of the body with a liquid which circulates to transfer heat from the hot region to a cooler region. Typically, this system has been employed with external pumping means since it is difficult to generate sufficient circulation within a closed system to provide significant cooling. High rates of heat transfer can be obtained in this manner; however, external pumps are undesirable in many applications where heat transfer is desired, and self-contained systems will be preferred if sufficient heat flux can be obtained.

A further means for heat transfer, which is applicable to the cooling of rotating bodies, falls within the general heading of "heat pipes."

The term "heat pipe" as used herein refers to any device that transfers heat by means of evaporation and condensation of a fixed amount of fluid within a sealed cavity of any shape formed in the device. In the operation of a heat pipe, a quantity of liquid locates in the relatively hot region of the cavity (the "evaporator" region) where it absorbs heat from the cavity walls in that region causing it to evaporate. The vapor flows to the cooler region of the cavity where it gives off heat to the walls of the cavity in that region (the "condenser" region) and condenses into a relatively cool liquid. The condensed, cooled liquid is then returned to the hotter zone of the cavity to repeat the cycle.

The principal advantage of heat pipes is that they are self-contained. A disadvantage of existing heat pipes is that they return the condensate to the evaporator region rather slowly, which has been a limiting factor on their heat flux capacity. This is offset to some extent, however, by the fact that they employ a vapor cycle and, therefore, need to circulate only about 1/10th or less fluid mass to obtain heat transport comparable to an all liquid system.

In the past, there have been two common techniques for returning the condensed liquid from the condenser zone to the evaporator zone. In one technique, the evaporator zone is positioned lower than the condenser zone so that the condensate returns under the influence of gravity. A common example of a gravity return system is a domestic steam heating system where the water is vaporized in a furnace, the vapor rising in a pipe to a radiator where it cools, condenses, and runs back down the pipe under gravity to the furnace. A second example of a gravity system is shown in Lincoln U.S. Pat. No. 2,813,698 where the liquid condensate flows down the lower sloped wall of the rotating pipe as in FIG. **3.** A slightly different return pumping technique is shown in FIG. **2** of the Lincoln patent, where the interior surface of the pipe has helical vanes and the pipe is rotated to pump the condensate back to the evaporator zone by screw motion. Being gravity dependent, this type of pipe may be broadly classified as a gravity return. Other examples of gravity return systems are the chemists' reflux capsule, the French Vapotron and the thermosyphon. Return pumping forces in a gravity system can be, however, relatively weak (being limited to a maximum of **1**G acceleration. Gravity return systems are not suitable for many heat transfer applications including where the condenser is necessarily positioned beneath the evaporator (for example, in overhead drilling), or in turbines for aircraft or in vehicles (where the attitude of the axis is variable and random accelerations are present), or in space flight (where the effect of gravity is essentially zero). Gravity return systems are further limited in that fluid flow head losses and evaporator heating fluxes are limited to values obtainable under **1**G conditions.

The other common liquid return pumping technique for heat pipes employs capillary action. Here, the walls of the pipe are covered with wicks, felts, screens, grooves, porous materials or the like so that the condensate returns to the evaporator under capillary action. Capillary pumping will operate in the absence of gravity and to a limited extent against gravity and is, therefore, suitable for some operations where gravity return pumping cannot be employed. Capillary return pumping is, however, of relatively low magnitude, being limited by the low pressure difference across the menisci in the capillaries. Moreover, in such systems, heat fluxes are limited to values well below normal peak nucleate boiling levels since, when nucleate boiling occurs in the condensate, the vapor formed therein tends to block the capillary passages and severely retards the return pumping effect.

My invention provides a self-contained heat transfer system for rotating bodies that can provide heat flux which is superior to the heat flux capacity of any comparable prior self-contained heat transfer system.

In my invention, I provide the rotating body with an interior sealed cavity which has a certain diameter at the hotter (evaporator) end of the body tapering up to a slightly smaller diameter at the cooler (condenser) end as, for example, is shown in FIG. **1.**

I locate a small inventory of liquid in the cavity and rotate the body at high speed. At high speed, the liquid inventory forms a uniform annulus at the evaporator end. Heat transferred through the evaporator wall vaporizes some of the liquid and the vapor so formed flows toward the axis of rotation and axially toward the condenser end. Here the vapor condenses on the cooled walls and the liquid condensate is pumped back along the tapered cavity walls by the small component of centrifugal acceleration tangential or parallel to the taper of the wall (hereinafter sometimes referred to as "centrifugal pumping acceleration" or CPA). The speed at which I rotate the body is sufficiently great to produce a component of centrifugal acceleration in the condensate parallel to the tapered cavity wall which is in excess of 1G acceleration at essentially all points on the condenser walls, and often preferably many times greater.

Accordingly, I provide a self-contained, vapor cycle heat transfer device which returns the condensate from the condenser region to the evaporator region at a relatively high velocity even against gravity or in the absence of gravity, and which can provide return pumping acceleration many times greater than 1G to provide heat transfer ability which is greatly improved over that of conventional heat pipes and the like.

My invention is particularly well suited to provide substantial heat transfer between two ends of a rotating body having a relatively long axial dimension and a relatively small diameter, since I can generate fairly large return pumping acceleration along a very small angle slope. Similarly, my invention is also well suited for providing heat transfer in a narrow body having a fixed diameter, such as in the bit of a drill where the outer diameter of the drill is necessarily limited by the diameter of the hole to be drilled.

The benefits of my invention are obtainable only in a vapor cycle or two-phase system, and only in such a system where the cool condensing surface is kept relatively free of liquid. The benefits of my invention rapidly decrease as the amount of liquid on the condensing wall increases. Excess liquid occurs on the condensing wall generally in two ways. Either the liquid inventory in the evaporator region is so great that it overflows onto the condenser wall, or return pumping forces are so weak that condensate builds up in the condenser region faster than it can be pumped away. In either case, the effect is the same. Cooling in the condenser region is greatly impaired, return pumping slows down, and the heat flux capacity of the entire system is substantially reduced. When excess liquid occurs in the condenser region, a two-phase system begins to resemble a single phase system. This is undesirable as discussed previously.

It should be noted that at horizontal attitude under the influence of gravity, the liquid in the evaporator forms a non-uniform annulus when centrifugal acceleration of the liquid just exceeds 1G. As centrifugal acceleration in the liquid approaches the relatively high levels required to produce centrifugal pumping acceleration in excess of 1G according to the present invention, the liquid annulus becomes highly uniform and the pressure of the liquid increases substantially. This provides highly efficient boiling and vaporization effects since it greatly increases the convection of vapor bubbles and liquid in the annulus, and also provides a smooth interface at relatively high heat fluxes. By contrast, when boiling occurs at 1G, the interface is distorted and turbulent and tends to disperse relatively large droplets into the vapor which reduces the heat transfer effectiveness of the system. Accordingly, a device constructed in accordance with the principles of my invention may accommodate relatively high levels of heating without causing undue surface turbulence.

Because of the tendency of centrifugal acceleration to increase convection, the denser, cooler liquid flows away from the axis and quickly displaces the less dense heated liquid near the hot evaporator wall which, in turn, flows rapidly toward the axis. This prompt movement from the evaporator wall to the interface enhances evaporation, suppresses boiling and improves the heat transport capabilities of the system as a whole. A more complete discussion of the effects of boiling and liquid vaporization may be found in my U.S. Pat. No. 3,508,402.

These and other advantages of my invention will become apparent from the following description and accompanying drawings.

BRIEF DESCRIPTION OF DRAWINGS

In the accompanying drawings:

FIG. 1 is a longitudinal sectional schematic view of a rotating heat pipe constructed in accordance with the principles of the present invention.

FIG. 2 is a longitudinal sectional schematic view of an electric motor according to the invention.

FIG. 3 is a longitudinal sectional schematic view of a drill bit assembly constructed according to the invention.

FIG. 3A is a transverse section taken along lines 3A — 3A of FIG. 3.

FIG. 4 is a longitudinal sectional schematic view of an alternate rotating heat pipe construction according to the invention.

FIG. 5 is a longitudinal sectional schematic view of a turbine engine constructed according to the invention.

FIG. 6 is a longitudinal sectional schematic view of an air conditioning unit constructed according to the invention.

DETAILED DESCRIPTION

Referring to the drawings in greater detail, FIG. 1 shows a centrifugal pumping heat pipe 1 constructed in accordance with the principles of the present invention. The heat pipe 1 has a generally cylindrical body portion 2 with an axial shaft 3 extending from one end. The shaft 3 is journalled for rotation in a bearing 4. An elongated cavity 5 is formed coaxially in the body portion 2. The cavity is generally frusto-conical with substantially parabolic side walls 6. At the large end of the cavity, there is a cylindrical reservoir 7 which has a slightly larger diameter than the larger diameter of the frustum. A small inventory of liquid is sealed in the cavity. The amount of liquid employed is such that when the pipe is rotated during operation, the liquid forms a uniform annulus which fills the reservoir 7 but does not overflow onto the parabolic walls 6. The amount of liquid may be less than the amount required to fill the reservoir 7 completely, but should be great enough that the reservoir 7 does not boil dry during operation.

In operation, the end of the pipe 1 wherein the reservoir 7 is located (the "evaporator" end) is subjected to heat which causes the liquid in the reservoir to vaporize. The vapor tends to fill the central empty space in the cavity. The opposed end of the pipe (the "condenser" end) is subjected to relatively cooler temperatures so that when the vapor encounters the walls **6** in the condenser end, it condenses. The pipe is rotating at a very high rate so that the condensate which forms on the walls **6** will be subjected to a radial or centrifugal acceleration which is great enough that the component of that acceleration parallel to the wall **6** at any point on the wall will be in excess of 1G.

The CPA or centrifugal pumping acceleration of a body at a point on a sloped wall in a direction parallel to the wall at that point is equivalent to $\omega^2 r \sin\theta$, where ω = the angular velocity of the body, θ = the angle of a tangent to the surface of the wall at that point relative to the axis, and r = the distance from the axis. It will be seen that centrifugal acceleration tends to increase as r increases, so that the slope of the wall may be decreased as r increases without decreasing the tangential component of centrifugal acceleration. For this reason, the walls in the condenser portion of the cavity are preferably curved as shown in FIG. **1**.

In most applications of the present invention, the outer diameter of the rotating body will be limited as, for example, in the case of a drill for drilling a hole of a particular size. For this and other reasons, it will be desirable to determine what cavity wall curvature will produce the optimum pumping acceleration. In the usual case, this will be where the net pumping acceleration is relatively uniform at all points on the slope.

Accordingly, I have calculated the relationship of the radius, surface taper, and angular velocity to one another when it is desired to obtain a uniform centrifugal pumping acceleration at all points on the slope. In these calculations, it is necessary to take into account both the presence (or absence) of actual gravity and the attitude of the axis of the rotating body to the direction of actual gravity. The general equation is as follows:

$r = 0.816(n + m \cos\theta\cos\beta)/(\text{rev./sec.})^2(\sin\theta)$ where r = the distance of the wall from the axis, expressed in feet;

- n = the no. of G's desired for net pumping (including the effect of actual gravity);
- m = the acceleration of actual gravity in G's, ($m = 0$ in outer space, $m = 1$ on earth);
- θ = the angle formed by the axis of rotation and a straight line drawn parallel to the surface of the wall at a particular point on the wall; and
- β = the angle of the axis of rotation relative to the direction of actual gravity, where $\beta = 180°$ when the axis of rotation is vertical with the evaporator end down. $0° \leq \beta \leq 180°$.

The general equation assumes that the body is rotating at a rate sufficient to produce a centrifugal acceleration normal to the axis which is much greater than the actual ambient gravity.

I have also calculated the following simplified equations for special cases:

Case I

Axis of rotation vertical, condenser end down, actual gravity = 1G.

$r = 0.816(n + \cos\theta)/(\text{rev./sec.})^2(\sin\theta)$, ft.

Case II

Axis of rotation vertical, condenser end up, actual gravity = 1G.

$r = 0.816(n - \cos\theta)/(\text{rev./sec.})^2(\sin\theta)$, ft.

This equation is meaningless where $n<1$.

Case III

Axis of rotation horizontal, actual gravity between 0 and 1G, and axis at any angle with actual gravity = 0.

$r = 0.816(n)/(\text{rev./sec.})^2(\sin\theta)$, ft.

From the above four equations, it is possible to generate ideal curves for the wall of the heat pipe. Very good approximate curves may be drawn by beginning at either end of the condenser wall and plotting values for r and θ at small regular increments along axis of the pipe (ΔL). This generates a stepped slope which can be smoothed out by drawing a curve through the intersection of the taper and the radius for each value of ΔL. If desired, more accurate slopes can be generated from the above formulae by iteration.

In generating approximate curves for small taper portions of the wall (i.e., where $\theta < 6°$), one may assume that $\sin\theta = \tan\theta$. Accordingly, the following equations may be employed.

In Case I

$\tan\theta = 0.816(n+1)/r(\text{rev./sec.})^2$

In Case III, $n = 1$

$\tan\theta = 0.816/r(\text{rev./sec.})^2$

Since $\tan\theta = \Delta r/\Delta L$, one may generate the curve by determining the value of θ and r for the wall at one point on the axis and then plotting subsequent points on the wall by inserting values for ΔL and solving for the corresponding values of Δr.

In the evaporator end, it is important to maintain heat flux below levels where critical nucleate boiling ("burn-out") occurs. This critical level tends to increase at high accelerations. For example, water boiled at 400G's and 815,000 BTU/hour, ft^2 is below the burn-out level, yet this heat flux is approximately double the normal critical value at 1G. Generally speaking, burn-out heat flux varies with the one-fourth power of acceleration. In order to produce centrifugal pumping acceleration in excess of 1G, and at multiple G levels, it is necessary to produce high levels of radial centrifugal acceleration on the wall and in the evaporator (since centrifugal pumping acceleration at any point on the taper is equal to centrifugal acceleration at that point times the sine of the slope angle at that point, and since the slope angle is typically quite small). Accordingly, the high levels of acceleration produced in the evaporator tend to raise the heat flux capacity of the evaporator.

Example A

For example, in a two-inch diameter evaporator cylinder turning at about 6,000 revolutions per minute, the centrifugal acceleration of the liquid is about 1,000G's. The heat flux capacity of this evaporator with water is about 1,800,000 BTU/hour, ft^2. This is about 10 times greater than the highest capillary heat pipe heat flux reported prior to this invention.

At the condenser end, it is important to pump the condensate off the walls and back to the evaporator since any build-up of condensate reduces the condensing effectiveness of the walls. For this reason, it is preferable to produce relatively high levels of centrifugal pumping acceleration in the practice of my invention. Since, typically, centrifugal pumping acceleration equivalent to dozens of G's can be produced in devices constructed according to my invention, such devices can operate with much less thermal resistance in the condensate layer than devices with equivalent vertical condensing surfaces at 1G.

The thermal resistance of the condensate layer can be reduced still further by plating the condenser walls with noble metals.

Turning to FIG. 2 of the drawings, there is shown a schematic drawing of an electric motor constructed in accordance with the principle of the present invention. The motor has a housing **10,** a stator **11,** and a rotor **12.** The rotor core **13** has an axial drive shaft **14** extending from one end, and an auger-like portion **19** extending from the other end. The rotor core is further provided with a coaxial sealed cavity **15,** which has cylindrical walls **16** adjacent the rotor windings (defining the evaporator region) and tapered walls **17** adjacent the auger portion **19** (defining the condenser region). The cylindrical walls **16** in the condenser region have a slightly larger diameter than the largest diameter of the condenser walls **17** so that the evaporator walls are recessed to form a well-defined reservoir for the liquid **18.** The rotor is shown rotating at a speed sufficient to form a substantially uniform liquid annulus in the reservoir. The liquid inventory is small enough that it does not overflow the reservoir to cover any appreciable portion of the condenser walls **17** during operation.

The importance of minimizing the amount of liquid on the condenser walls of devices constructed according to my invention has been discussed and, for this reason, a recessed, well-defined reservoir is preferred in this and most other embodiments of my invention as will be readily apparent to persons of ordinary skill in the art.

In the operation of the electric motor shown in FIG. **2,** the rotor **12** develops localized heat in the rotor windings and at the bearings upon which the rotor is journalled in the housing **10.** This heat is conducted through the cylindrical walls **16** of the core **13,** where it is transferred to the liquid **18** in the reservoir. The liquid **18** is vaporized and the vapor flows radially toward the axis and axially to the condenser end where it condenses on the tapered walls **17.** This condensate is subjected to centrifugal pumping acceleration to return it to the reservoir. When the vapor condenses on the walls **17,** it gives off heat which is conducted through the walls **17** and into the auger blades **19.** The motion of the auger blades **19** in the ambient air (entering the housing through an air inlet screen **20**) enhances cooling of the condenser walls **17.** Moreover, the auger **19** acts as a blower, forcing air through apertures **21,** across the rotor and stator windings, and out the opposite side of the housing **10** via vents **22.** This assists cooling of both the rotor **12** and stator **11.**

FIGS. **3** and **3**A provide schematic illustration of a drill bit assembly constructed in accordance with the principles of the present invention. The assembly consists essentially of a drill bit **30** and a non-rotating sleeve **31.** The bit **30** has conventional helical cutting blades **32** formed at one end and is provided with a coaxial sealed cavity **33.** The cavity has an enlarged reservoir portion adjacent the blades **32.** The reservoir is defined by the cavity walls **34** which extend into the cutting blades **32.** Liquid inventory **35** locates in the reservoir during rotation to form a substantially uniform annulus during operation, as shown. The cavity **33** tapers gradually from the reservoir to a smaller diameter at the opposite end. In operation, localized heat build-up in the evaporator region at the blades **32** is transferred away according to principles already discussed, as will be apparent. In addition, the sleeve **31,** enhances heat transfer by flowing coolant (from the pipes **38**) over the outer surface of the shank of the bit **36** and adjacent the condenser walls **37.** Rotational flow in the coolant is minimized by locating one or more apertured baffles **39** on the inner surface of the sleeve **31.**

FIG. **4** of the drawings shows a schematic representation of an alternate form of heat pipe **40** embodying features which may be employed together or singly in specific applications as desired. As shown, the pipe **40** is being employed to transfer heat away from a fluid **43** supplied to the evaporator end of the pipe in a non-rotating jacket **42.** The pipe **40** is provided with coaxial sealed cavity **44** having a cylindrical evaporator wall **45** at the evaporator end, a conically tapered condenser wall **41** at the condenser end, and circumferentially spaced, generally axial submerged channels **46** extending between the evaporator wall **45** and the condenser wall **41** beneath a conically tapered wall **47** mounted in the pipe on radially outwardly extending posts or ribs (not shown).

The submerged passages **46** and conical wall **47** are provided between the condenser and evaporator regions because in the central regions of a heat pipe, there is usually an adiabatic zone in which the vapor and liquid flows are transferred counter-currently. This is a zone of annular flow with the vapor at the center moving at much higher velocities than the liquid along the wall which creates the possibility that the vapor will blow the returning condensate film into waves or mist.

To solve this problem, if it occurs, one may employ submerged condensate passages as shown at **46** in FIG. **4.**

A further feature illustrated in FIG. **4** of the drawings is that the transverse end of the condenser region is folded inwardly to accomplish one or more of several objectives, as follows: (*a*) to increase the area available for cooling when overall length is limited; (*b*) to increase the condensing heat-transfer coefficient by causing the condensate, as soon as it is formed, to be centrifuged off the convex inner surface **48** of the infolded wall and collect on the larger diameter tapered surface **41** for centrifugal pumping back to the evaporator region; and (*c*) to provide a concave outer surface opposite convex inner condensing surface **48,** in which coolant is directed as a jet from the pipe **49** against the concave surface which, because of its shape, causes the coolant to flow back along the wall to conduct heat through the wall away from the inner condenser surface **48.** It will be noted that in (*b*) above, this feature permits one to increase the condensing surface area without increasing the return surface area.

FIG. **5** is a schematic drawing of a turbine-type engine according to the present invention. Here, the turbine rotor **50** has a plurality of radially outwardly projecting turbine blades **51** which are hollow. A housing **52** encloses the rotor **50** and hot combustion gases blow into the housing via passages **54,** past the blades **51,** and out of the housing via outlet passages **55.** The flow of hot combustion gases against the blades **51** imparts angular velocity or acceleration to the rotor **50** and, at the same time, heats the hollow blades **51.** The rotor **50** is provided with a sealed partially liquid-filled coaxial cavity **57** communicating with the hollow blade cavities **58** via small tubes **59.** Heat in the liquid in the blade is transferred either by small vapor bubbles or very strong liquid natural convection currents in the connecting tubes **59** to the interface in the central cavity **57.** Although the colder liquid entering the hollow blades flows counter-currently to the hotter liquid and/or vapor leaving the blades, it is felt that a single relatively small connecting tube per blade is adequate, based on my experience with rotating boilers.

It will be noted that heat transferred from the vapor to the tapered concave condensing surface **62** is conducted away from the opposed convex outer surface **66** by the flow of fuel over that surface **66** through a jacket **63** which surrounds the exterior of the rotor **50** at the condenser end. Fuel enters the jacket from the fuel tanks via inlet pipe **64** and leaves the jacket, pre-heated for combustion, on its way to the combustors via outlet pipe **65.**

FIG. **6** is a schematic drawing of a novel air conditioning unit constructed according to my invention. This unit essentially comprises a rotor generally indicated by the numeral **70,** and a housing generally indicated by the numeral **71.** A motor **72** drives the rotor **70** in rotation in the housing **71,** the motor's rotor windings wound on the rotor **70,** and the motor's stator windings fixed in the housing **71.** Hollow fan blades **73** are fixedly mounted at one end of the rotor **70** and partially filled with a conventional liquid refrigerant, such as freon. A hollow tube **74** communicates between each hollow blade tip and one end of the compressor passage **75** in the compressor. The other end of the compressor passage **75** communicates with each hollow blade cavity near the hub of the blade at compressor passage inlet **82.** When the motor drives the rotor **70** in rotation, the refrigerant is compressed in the compressor passages **75** where it gives off heat of compression to liquid **77** in the evaporator region of the heat pipe cavity **76.** The compressed, liquefied refrigerant flows from the compressor into the tubes **74** toward the tips of the hollow blades **73** where, upon leaving the tubes **74** through orfices near the blade tips, it expands to fill the blades with cold vapor. Room air, induced to circulate past the exterior surfaces of the blades **73,** gives off heat, cooling the room air and heating the cold vaporized refrigerant. The warmed vaporized refrigerant flows through the hollow blades **73** toward the axis where it re-enters the compressor at compressor passage inlet **82,** where it is compressed, liquefied, and the cycle repeated.

When the compressed refrigerant gives off heat of compression to the liquid **77** in the evaporator end of the rotor cavity **76** (note the hollow interior surface **78** of the rotor compressor vanes forming a part of the total evaporator surface in the evaporator region), the vapor flows to the condenser end of the cavity **76,** where it condenses giving off heat to the outside air through the rotating tapered conductive disc **80** (constructed of aluminum or other good heat conductor). As is apparent, the condensate is returned to the evaporator region under centrifugal pumping acceleration according to the principles of the present invention discussed previously.

It will be noted that an insulating sleeve **83** is applied to the outer surface of the rotor **70** at the point where the rotor passes through the wall **81.** This sleeve substantially prevents condensation on the conical interior rotor wall in that region so that heat is not given off into the inside air, but only into the outside air.

It will be apparent that relatively low temperature levels will be encountered at both ends of the rotor **70** (on exterior surfaces), and that the rotor liquid inventory must either have a relatively low boiling point at normal pressure, or the pressure in the rotor cavity must be relatively low.

The air conditioning unit shown in FIG. **6** has several advantages, including (*a*) that it can provide superior cooling in a compact unit; and (*b*) that it requires a very small hole in the wall (e.g., 3 or 4 inches) by comparison to conventional units which require large vents.

In addition, cooling can be further improved by providing a stationary shroud ring to collect condensation of room air moisture on the blades as it sprays off the blades, and this moisture can be ducted to the outside rotating disc **80** near its hub so that the moisture can be centrifuged radially outwardly over the rotating disc surfaces to help cool them.

The above are but a few specific examples of the many devices that can be constructed according to the principles of my invention. While my invention is limited to heat transfer in rotating bodies and has been shown in connection with the heating and cooling of rotating parts, it is not necessary that application of the invention be made where rotation already exists. Rotation may be added to obtain the desirable features (such as in abstractting heat from reactors, furnaces, combustion chambers and the like).

Other applications where shafts have localized heating are in heavily loaded bearings and gears, rollers for conveyor tables handling hot metal, etc. Also, there are reverse applications where shafts are locally cooled, such as hot stirring devices, immersion heaters, constant temperature rollers, etc.

Furthermore, there are numerous alternative designs for the heat pipe. For example, by constructing a pipe with a cavity which is narrow at both ends, tapering down from a larger diameter in the middle, the pipe can be heated in the middle and cooled at both ends. Similarly, constructing a pipe with a cavity which is wide at both ends and narrow in the middle, the pipe can be heated at both ends and cooled in the middle, although it will be preferable and sometimes essential to provide a middle partition to prevent the liquid from concentrating at one end only.

Also, there will be other adjustments in the pipe design to overcome problems in specific applications. For example, when relatively high vapor velocities are encountered, the inside taper will be increased to overcome the higher pressure drops at the cavity wall (or at the liquid-vapor interface). Also, if rapid start-up transients are important, the inside tapered surface can be provided with longitudinal ribs for faster pumping action.

It will be appreciated that several of the devices illustrated herein, for example in FIGS. **2, 4** and **6,** show conical return walls that are not curved to produce uniform net pumping acceleration at all points along the walls. These illustrations were not intended to show optimum wall configuration for the devices shown. However, while properly curved return walls are preferred, the invention may be practiced less effectively with straight conical return walls.

Many different liquids are suitable for use in devices constructed according to the present invention. Preferred liquids for specific applications will be readily apparent to persons of ordinary skill in the art. For many applications, it will be preferable, or even essential, to utilize a liquid metal (such as liquid sodium, for example). One benefit of liquid metals is that they conduct heat readily (about 100 times faster than water) so that the condensate that forms on the condenser walls does not slow subsequent condensation while it is being pumped back to the evaporator region.

Similarly, the pipe may be constructed of many different materials as will be apparent to persons of ordinary skill, although it is usually preferably formed of a highly conductive, noncorrosive metal such as stainless steel, molybdenum, nickel, or their alloys. It is essential, however, that the pipe be constructed to withstand the substantial internal pressures that may be developed during operation.

It will be understood that further changes and modifications may be made without departing from the spirit and scope of the invention as set forth in the claims appended hereto.

Having described my invention, I claim:

1. A combustion turbine engine comprising a housing;

an elongated body mounted for rotation about its longitudinal axis in said housing, said body being provided with an interior sealed elongated cavity coaxial with said longitudinal axis of said body, said cavity having a first portion locating at one end of said body, and said cavity having a second portion locating at the opposed end of said body, said second portion having generally conical walls with its larger diameter adjacent said first portion, said first portion having a diameter not less than the larger diameter of said second portion, every point on the conical walls defining said second cavity portion having a slope and radial distance from the axis which, when the body rotates at operational velocities, produces a centrifugal pumping acceleration in excess of 1G in a thin film of liquid located at such point;

a liquid inventory sealed in said cavity, said inventory being sufficiently large to form a substantially uniform annulus in said elongated body when said body rotates at said operational velocities, said liquid inventory being sufficiently small to be contained substantially within said first cavity when said annulus is formed;

a plurality of hollow vanes projecting radially from said body radially outwardly of said first cavity portion;

means for producing hot combustion gases to blow past said vanes in said housing and drive said body in rotation at operational levels of angular velocity; and ducts communicating between the hollow in each of said vanes and said first cavity portion, a portion of said liquid inventory in said body cavity flowing outwardly into said hollow vanes via said ducts when said liquid annulus is formed, whereby said vanes are cooled by such liquid; and

a jacket surrounding the body exterior radially outwardly of said second cavity portion, said jacket provided with a fuel inlet leading from a source of fluid fuel, said jacket further provided with a fuel outlet leading to said hot combustion gas producing means, so that fuel from said fuel source may flow through said jacket across the adjacent exterior surface of said body to absorb heat therefrom prior to entering said combustion gas producing means.

2. A process for cooling a gas turbine engine having a combustor, a housing provided with ducts leading from said combustor to exhaust parts, a hollow hermetic rotor mounted for rotation about its central axis in said housing and provided with radial turbine blades extending outwardly into said ducts so that hot combustion gases flowing through said ducts impinge upon said blades to drive said rotor, said rotor being constructed to receive a sealed-in inventory of liquid, said rotor having an evaporator region and means forming a reservoir for said liquid with portions located at said blades, said rotor having a condenser region axially spaced from said evaporator region and from said turbine blades and located adjacent a heat sink, said rotor having an internal condensing surface in said condenser region coaxial with said rotor and having a taper whereby said condensing surface decreases in diameter in a direction away from said reservoir, wherein the steps comprise

providing said rotor with an inventory of liquid in said reservoir having a generally cylindrical internal liquid surface within the rotor and spaced inwardly from said blades, continually transferring heat directly from said turbine blades

to said liquid in said reservoir whereupon said liquid vaporizes and the resulting vapors flow to said condenser regions; continually transferring heat from said condenser region of said rotor to said heat sink so that said condensing surface is cooled and said vapor condensed by contact therewith to effect continual removal of heat from the turbine rotor;

rapidly forcing said condensate off of said condensing surface as soon as it forms to return said condensate to said reservoir and to keep said condensing surface relatively free of liquid at any attitude by rotating said rotor at a velocity providing a high centrifugal acceleration having a component greater than 1G in an axial direction parallel to said tapered condensing surface in essentially each particle of condensate located on said condensing surface; and passing fuel over the outer surface of said rotor in said condenser region before said fuel is transported to the combustor, said fuel absorbing heat from said rotor to cool said condensing surface and to heat said fuel prior to combustion.

3. A process for cooling a gas turbine engine having a combustor, a housing provided with ducts leading from said combustor to exhaust parts, a hollow hermetic rotor mounted for rotation about its central axis in said housing and provided with radial turbine blades extending outwardly into said ducts so that hot combustion gases flowing through said ducts impinges upon said blades to drive said rotor, said rotor being constructed to receive a sealed-in inventory of liquid, said rotor having an evaporator region and means forming a reservoir for said liquid with portions located at said blades, said rotor having a condenser region axially spaced from said evaporator region and from said turbine blades and located adjacent a non-rotating heat sink, said rotor having an internal condensing surface in said condenser region coaxial with said rotor and having a taper whereby said condensing surface decreases in diameter in a direction away from said reservoir, wherein the steps comprise providing said rotor with an inventory of liquid in said reservoir having a generally cylindrical internal liquid surface within the rotor and spaced inwardly from said blades, continually transferring heat directly from said turbine blades to said liquid in said reservoir whereupon said liquid vaporizes and the resulting vapors flow to said condenser region;

passing a cooling liquid through said non-rotating heat sink during rotation of said rotor to cover the outer surface of said rotor in said condenser region with an annular body of liquid so that said condensing surface is cooled and said vapor condensed by contact therewith to effect continual removal of heat from the turbine rotor; and

rapidly forcing said condensate off of said condensing surface as soon as it forms to return said condensate to said reservoir and to keep said condensing surface relatively free of liquid at any attitude by rotating said rotor at a velocity providing a high centrifugal acceleration having a component greater than 1G in an axial direction parallel to said tapered condensing surface in essentially each particle of condensate located on said condensing surface.

4. A process for cooling a gas turbine engine according to claim **3** wherein said condensing surface is curved in axial cross section in such a manner as to provide means for causing generally uniform pumping acceleration along the length of said surface and wherein said last-named component is many times greater than 1G.

5. A multiple-stage fluid-pressure energy-transferring device of the character described having a stator, a rotor mounted to rotate within said stator and having a body shaped to provide an axially elongated hermetic heat pipe coaxial with said rotor and rotatable therewith, said rotor providing a series of axially spaced stages, each stage comprising a multiplicity of circumferentially spaced hollow vanes around the rotor projecting outwardly from said heat pipe to said stator and liquid-filled cooling passages extending outwardly from said tube to said vanes to transfer heat from said vanes by convective circulation of the liquid in said passages, said heat pipe containing a sealed-in inventory of liquid with an axially elongated internal surface of uniform radius located within the pipe in an evaporator region extendinaxially to extending axially cooling passages of all of said stages, said heat pipe having a condenser region at one end of the pipe providing an external heat sink, said evaporator region containing a reservoir for said liquid spaced from said condenser region, means for continually transferring heat from an external source to the liquid in said reservoir to evaporate the same whereupon the resulting vapors move through said pipe to said condenser region, said pipe having a coaxial interior condensing surface in said condenser region with a taper, whereby said condensing surface gradually decreases in diameter in a direction away from said reservoir, said condensing surface being curved in axial cross section to provide means for causing generally uniform pumping acceleration along the length of said surface, external cooling means for continually removing heat from said condenser region to condense the vapors contacting said condensing surface and to cause heat transfer to the heat sink, and means for rotating said heat pipe about said axis at a velocity such that the component of centrifugal acceleration in an axial direction parallel to the tapered condensing surface at essentially any location thereupon is greater than 1G and the condensate formed on said condensing surface is rapidly forced off of said condensing surface to keep it relatively free of liquid at any attitude.

6. A multiple-stage fluid-pressure energy-transferring device according to claim **5** wherein said one end of said heat pipe has a tapered external surface of revolution concentric to said tapered interior condensing surface, said external cooling means comprises a non-rotatable jacket enclosing said condenser region and containing an annular body of cooling liquid surrounding and contacting said tapered external surface, and means are provided for passing a cooling liquid through said jacket.

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