Loop Heat Pipes, 2001 to 2005

Qpedia continues to present selected patents which were awarded to developers around the world to address cooling challenges. After reading this multi-part series, you will become more aware of both the historic developments and the latest breakthroughs in both product design and applications.

We are focusing on patented technologies to show the breadth of development in thermal management product sectors. Limited by article space, we are presenting a small number to offer a representation of the entire field. You are encouraged to do your own patent investigation. Further, if you have been awarded a patent and would like to have it included in these reviews, please send us your patent number or patent application.

In this issue, our spotlight is on Loop Heat Pipes. There is much discussion about their deployment in the electronics industry, and these patents show some of the salient features that are the focus of different inventors.

MULTIFUNCTIONAL CAPILLARY SYSTEM FOR LOOP HEAT PIPE STATEMENT OF GOVERNMENT INTEREST

6,227,288 B1, Gluck, et al

A Multifunctional Capillary System is located within and between a single compensation chamber (CC) and the evaporator of a loop heat pipe. It provides: vapor-liquid interface control for all gravity states from the micro-gravity condition of space (near 0-g) through the earth's gravitational condition (1-g), with liquid supply to the evaporator via wicking from the CC in micro-gravity, and for all orientations (tilts) of the CC-evaporator assembly in earth gravity. As a single compensation chamber is used, dual compensation chamber penalties of weight and wide- temperature-variation are avoided. The system has combined, parallel wicking structure, paths, and joints for micro-gravity and 1-g liquid acquisition.

PATENT NUMBER	TITLE	INVENTORS	DATE OF AWARD
6,227,288 B1	MULTIFUNCTIONAL CAPILLARY SYSTEM FOR LOOP HEAT PIPE STATEMENT OF GOVERNMENT INTEREST	Gluck, et al	May 8, 2001
6,564,860 B1	EVAPORATOR EMPLOYING A LIQUID SUPERHEAT TOLERANT WICK	Kroliczek, et al	May 20, 2003
6,945,319 B1	SYMMETRICAL HEAT SINK MODULE WITH A HEAT PIPE FOR SPREADING OF HEAT	Li, et al	Sep. 20, 2005

The wick system is comprised of an axial-groove, evaporator-core secondary wick-concentric, contiguous, and in intimate contact with the primary evaporator wick. This secondary wick mates to a porous vane assembly in the CC. The design provides wicking continuity at this and at other joints within the system. In both the micro-gravity environment and under worst case 1-g orientation (CC below evaporator) the design can supply liquid to the primary wick under a wide range of temperature and power for steady state, startup, and transient conditions.

Background of the Invention

The invention is in the field of heat transmission and transport using loop heat pipes.

The loop heat pipe (LHP) is a thermal control and heat transport device initially developed in Russia. Its original purpose was to provide passive (no moving parts) cooling for a missile. It was later used by the Russians for spacecraft cooling. It has since been fabricated and tested by companies in the U. S. It has been space flight tested in Space Shuttle Hitchhiker Canisters and will be used in a number of spacecraft missions. The LHP can transport large quantities of heat over long distances with moderate temperature difference, and can be designed to be mechanically flexible.

FIG. 1 shows a LHP a schematic of a typical LHP. It consists of an evaporator with a porous wick, a contiguous compensation chamber, condenser, and vapor and liquid transport lines. A two-phase (liquid and vapor) working fluid, such as ammonia, is used. Heat applied at the evaporator wall causes vaporization of the liquid at the outer surface of the wick. This vaporization and fluid surface tension causes a curved meniscus to form in the wick. The pressure rise due to this curved meniscus drives fluid to circulate about the loop. The smaller the pore size of the wick, the greater the pressure rise that can be generated. Heat removal causes the liquid to condense, and sets up a steady fluid motion.

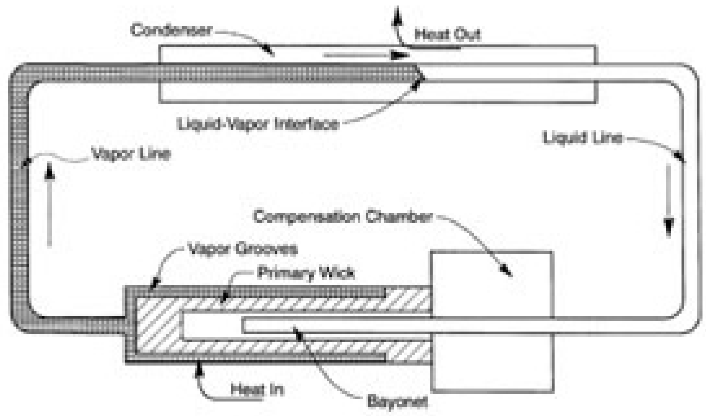


Fig. 1

FIG. 2 is a scanned image of a photograph of the evaporator-compensation chamber assembly (including heater plate) of a Russian LHP. The compensation chamber is a separate element with a larger diameter than the evaporator.

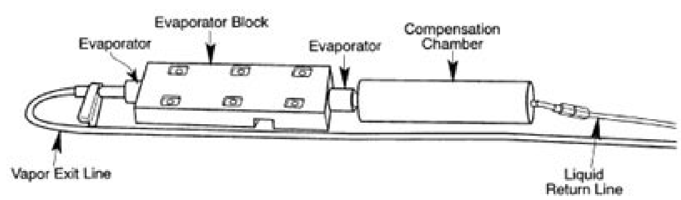


Fig. 2

FIG. 3 shows a possible adverse vapor-liquid configuration in this assembly in the micro-gravity (near 0-g) condition of space. This configuration is adverse in that the liquid in the compensation chamber is separate from and does not wet the evaporator wick. Of course, other vapor- liquid configurations in micro-gravity, many of which wet the wick, are possible. However, spacecraft components must always be designed to operate under the worst possible condition.

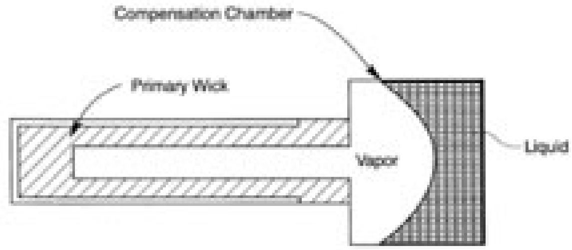


Fig. 3

Similarly, FIG. 4 shows an adverse vapor-liquid configuration in 1-g (earth gravity); this is caused by the orientation (tilt) of the assembly with respect to the earth's gravity vector. Other orientations in earth gravity, as shown for example by the horizontal orientation of FIG. 5, can result in an acceptable vapor-liquid location. Because of evaporator non-wetting illustrated by FIG. 4, LHP usage in 1-g conditions has been constrained to orientations that are near horizontal or where the compensation chamber is above the evaporator.

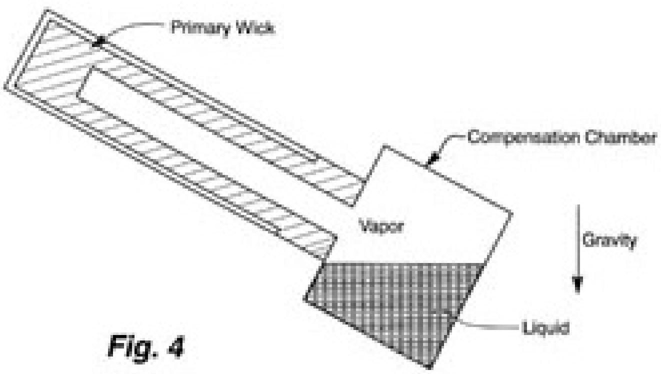


Fig. 4

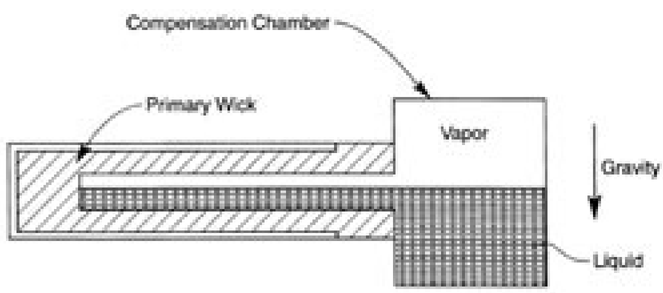


Fig. 5

The above noted deficiencies of LHPs have prompted both Russian and U. S. researchers to seek corrective measures. These have usually consisted of the incorporation of an auxiliary or secondary wick. The principal behind this secondary wick is illustrated by FIG. 6. This shows liquid flowing under capillary pressure from a larger to a smaller pore. The pressure drop going from vapor to liquid in the large and small pores is given by $\Delta P_1 = 2\sigma\cos\theta_1/R$ and by $\Delta P_2 = 2\sigma\cos\theta_2/r$, respectively. Here σ is the surface tension, θ the contact angle, and R and r are the radii of curvature, respectively.

With the vapor pressure the same in the two pores, $\Delta P_1 = P_v - P_{L1}$ and by $\Delta P_2 = P_v - P_{L2}$. Equating P_v in the two equations for the same contact angle, θ, in the two pores, there results $P_{L1} - P_{L2} = 2\sigma \cos\theta(1/r - 1/R)$. Pressure within the liquid is higher in the large pore than in the small one and hence liquid flow ensues in that direction.

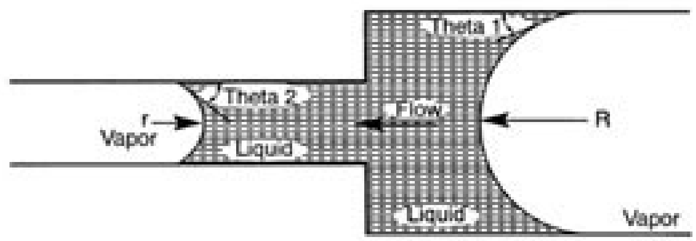


Fig. 6

The Russian version of this wick follows from their powder metal technology. FIG. 7 shows two such wicks, one for each compensation chamber in a dual compensation chamber LHP. The wicks, shown by the coarse crosshatching, occupy the annular region of each compensation chamber, butting against the main or primary wick in the evaporator. Properties of these wicks are: 93% porosity, 600 microns effective pore diameter, and 1.5×10^{-5} m^2 permeability. For comparison the corresponding properties of the primary wick, the driving capillary force in the LHP, are: 72% porosity, 2.3 microns effective pore diameter, and 4×10^{-14} m^2 permeability.

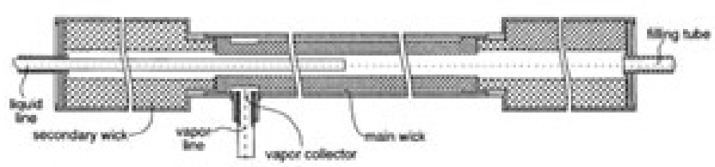


Fig. 7

The secondary wick of FIG. 7, by containing liquid within its pores, does provide interface control within the compensation chamber. However, as regards the liquid supply to the evaporator, its properties are a compromise between micro-gravity and 1-g requirements, and thus do justice to neither. Moreover, the design is deficient in that the secondary wick merely butts, but does not overlap, the primary evaporator wick.

In micro-gravity, capillary-driven flow must overcome only the pressure loss in the medium through which it is flowing, i.e., there is no hydrostatic (gravity) head loss. The capillary pressure difference driving the flow is given for liquids that wet perfectly by $AP=4\sigma/d$, while the laminar flow pressure loss is given by $AP=\mu uL/K$. Here, σ is the surface tension, d is the pore diameter, μ is the liquid viscosity, u is the liquid velocity, L is the length traversed, and K is the permeability. The permeability is inversely proportional to the flow resistance of the medium and is given by $K=\varepsilon d_h^2/32$, where ε is the porosity and d_h the hydraulic diameter of the medium. For randomly packed spheres permeability is given approximately by $K=0.00667d^2\varepsilon^3/(1-\varepsilon)^2$. Solving for the resultant velocity in the medium, it is found that $u= (4)(0.00667)\sigma d\varepsilon^3/\mu L(1-\varepsilon)^2$. Thus it is seen that velocity increases as pore diameter, d, increases.

In 1-g, capillary driven flow must overcome both flow pressure loss and hydrostatic head due to gravity. Velocity is now given by $u=[0.00667\sigma d^2\varepsilon^3/\mu L(1-\varepsilon)^2][4\sigma/d-\Delta pgL]$, where Δp is the difference between liquid and vapor density, and g is the acceleration due to earth gravity, 9.8 m/sec^2. The dependence of liquid velocity on pore diameter is now more complex. Indeed, unless the pore diameter is sufficiently small such that $4\sigma/d$ is greater than ΔpgL there is no flow. Where the hydrostatic term, ΔpgL, becomes significant, pore diameter must be small rather than large to cause liquid to flow. This is just the opposite of the result found for the micro-gravity case. Thus, the design approach taken entails the choice of secondary wick pore size that is a compromise between two conflicting requirements.

With an analysis similar to that above for effective pore diameter, it can be shown that it is much preferred that the secondary wick overlap the primary wick, rather than butting it. It was seen above that the permeability of the secondary wick can be orders of magnitude greater than that of the primary wick (1.5×10^{-5} vs. 4×10^{-14} m^2). With overlap, the supply liquid within the secondary wick encounters much less flow resistance in reaching the far end of the primary wick than if it had to traverse the much denser primary wick. The overlapping wick does, however, suffer from the pore diameter compromise discussed above.

The U.S. approach to secondary wick design is closely held and rarely revealed. However, the designs appear to use 100 to 200 mesh screens rolled or formed to create channels or arteries. They appear to extend from the compensation chamber along most of the length of the primary wick, making only partial or sector contact. Designs of this type cannot have much of a static wicking height capability, as pore size is determined by the gap between the screen layers. At best, this gap can be taken to be of the order of the wire diameter, 114 microns for a 100-mesh screen. The resultant static wicking height in ammonia at 25°C is 2.6 cm.

These designs are then primarily for micro-gravity or for near horizontal orientations of the compensation chamber- evaporator assembly in 1-g. They are of little or no utility for compensation chamber-evaporator orientations where the compensation chamber is below the evaporator. Additionally, contact between the secondary and primary wicks within the evaporator appears to be irregular, sector contact.

An alternate approach for liquid supply to the evaporator wick for any orientation of the compensation chamber- evaporator assembly in 1-g is the use of dual compensation chambers. Such an assembly was shown in FIG. 7. (Yuri Maidanik et al, Institute of Thermal Physics, Ural Division of Russian Academy of Sciences, Technical Report for Stage 2 of Project No. 473 for the International Science and Technology Center, Moscow, Russia, 1997). A photograph of the entire

LHP with this assembly is shown in FIG. 8. The premise behind this design is that, for orientations of the assembly away from the horizontal, one of the two compensation chambers is always above the evaporator. Possible orientations of a dual compensation chamber LHP are shown in FIG. 9.

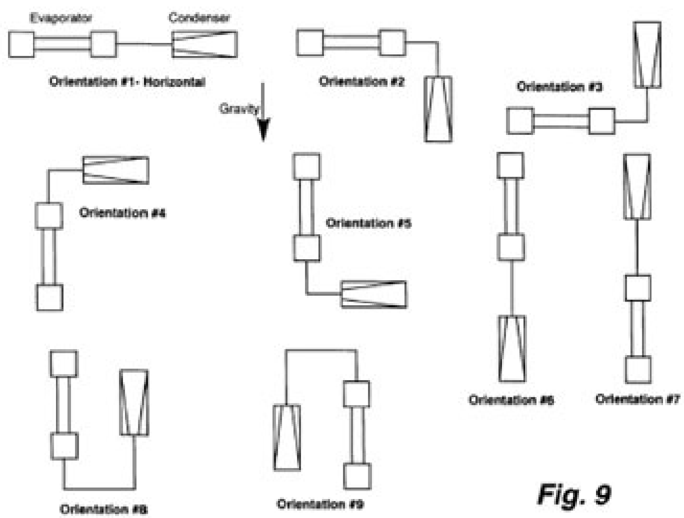


Fig. 9

The obvious penalty of a dual compensation chamber LHP is the weight of the second compensation chamber and the liquid contained therein. Recent performance tests at the Air Force Research Laboratory have revealed an additional, significant penalty. This is shown, for example, for a 40° C. condenser temperature in FIG. 10, where steady-state saturation temperature is plotted against power for the nine orientations of FIG. 9. Saturation temperature is seen to vary widely. For orientations 5, 6 and 8-whose common feature is a vertical evaporator with liquid return from below-this temperature is always hotter than the ambient (18 to 23°C.). For orientations 3, 4, and 7-whose common feature is condenser above evaporator-this temperature can be quite cold, approaching -30°C. at low power. For a number of applications such wide temperature variation is a serious problem or is entirely unacceptable. The overall representation of the current invention is shown in FIG. 11a

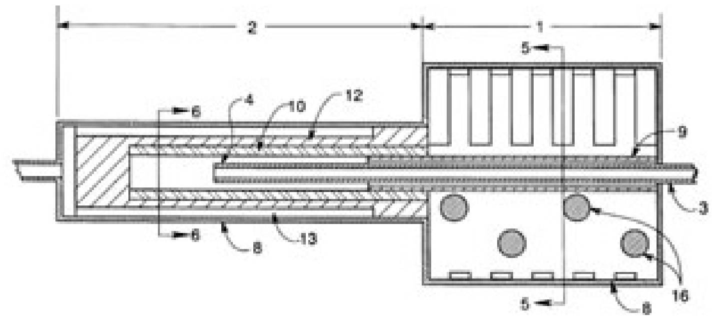


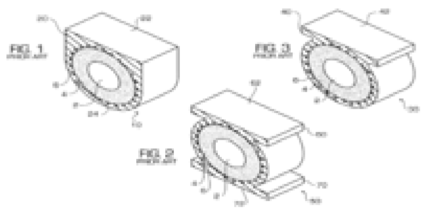
Fig. 11a

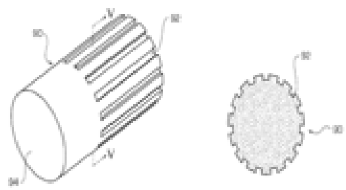
EVAPORATOR EMPLOYING A LIQUID SUPERHEAT TOLERANT WICK

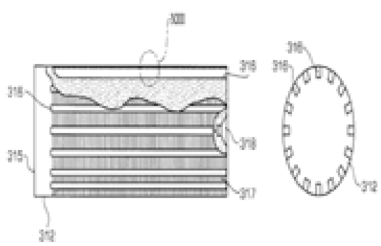
6,564,860 B1, Kroliczek, et al.

A capillary wick for use in capillary evaporators has properties that prevent nucleation inside the body of the wick, resulting in suppression of back-conduction of heat from vapor channels to the liquid reservoir. Use of a central liquid flow channel in the wick is eliminated, and pore size in the wick is chosen to maximize available pressure for fluid pumping, while preventing nucleation in the wick body. The wick is embodied with different geometries, including cylindrical and flat. A flat capillary evaporator has substantially more planar heat input surfaces for convenient mating to planar heat sources. The flat capillary evaporator is capable of being used with working fluids having high vapor pressures (i.e., greater than 10 psi). To contain the pressure of the vaporized working fluid, the opposed planar plates of the evaporator are brazed or sintered to opposing sides of a metal wick. Additionally, a terrestrial loop heat pipe and a loop heat pipe having overall flat geometry are disclosed.

There are numerous instances where it is desirable to transfer heat from a region of excess heat generation to a region where there is too little heat. The object is to keep the region of heat generation from getting too hot, or to keep the cooler region from getting too cold. This is a typical thermal engineering problem encountered in a wide range of applications including building environmental conditioning systems, spacecraft thermal control systems, the human body, and electronics.



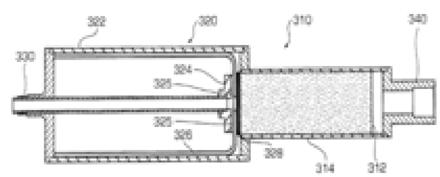


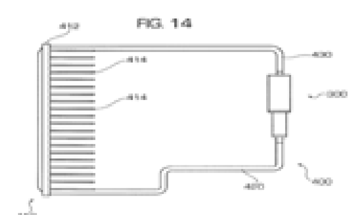


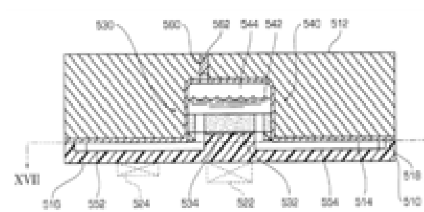
A variety of techniques can be employed to achieve this heat sharing effect. These include heat straps (simple strips of high conductivity material), closed loops of pumped single-phase fluid, heat pipes, mechanically pumped two- phase loops, and capillary pumped two-phase loops.

The most advanced and efficient concept is the capillary pumped two-phase loop and the related loop heat pipe (LHP). LHP technology has recently been developed for spacecraft applications due to its very low weight to heat transferred ratio, high reliability, and inherent simplicity.

An LHP is a two-phase heat transfer system. The LHP is a continuous loop in which both the vapor and the liquid always flow in the same direction. Heat is absorbed by evaporation of a liquid-phase working fluid at the evaporator section, transported via the vaporized fluid in tubing to a condenser section to be removed by condensation at the condenser. This process makes use of a fluid's latent heat of vaporization/condensation, which permits the transfer of relatively large quantities of heat with small amounts of fluid and negligible temperature drops. A variety of fluids including ammonia, water, Freon, liquid metals, and cryogenic fluids have been found to be suitable for LHP systems. The basic LHP consists of an evaporator section with a capillary wick structure, of a pair of tubes (one of the tubes is for supply of fluid in its liquid state, and the other is for vapor transport), and a condenser section. In many applications, the pressure head generated by the capillary wick structure provides sufficient force to circulate the working fluid throughout the loop, even against gravity. In other applications, however, the pressure differential due to fluid frictional losses, static height differentials, or other forces may be too great to allow for proper heat transfer. In these situations it is desirable to include a mechanical pump to assist in fluid movement. Systems employing such pumps are called hybrid capillary pumped loops.







It is an object of the present invention to:

- provide a wick for use in an LHP evaporator that has improved back- conduction performance.
- provide a liquid superheat tolerant wick that will reduce back-conduction in evaporators regardless of evaporator geometry and regardless of whether the vapor pressure of the working fluid used is high or low.
- provide a flat capillary evaporator that has the structural integrity to accommodate high-pressure working fluids, while avoiding the bulky mass of support structures such as ribs or thick walls.
- provide a capillary evaporator having a thin-walled flat geometry with minimal weight.
- provide a capillary evaporator having a thin-walled flat geometry and being suitable for use with both high-pressure and low- pressure working fluids
- provide a capillary evaporator having a thin-walled flat geometry and being suitable for use with low-pressure working fluids.
- provide a capillary evaporator having a geometry with minimal thickness at the heat transfer interface.
- provide a capillary evaporator having a thin-walled flat geometry with minimal temperature difference across the heat transfer interface.
- avoid the need for clamps to hold together the plates of a capillary evaporator having flat geometry.
- avoid the need for a saddle to match the footprint of the heat source to a cylindrical evaporator.
- provide a lightweight, flat capillary evaporator that can be easily integrated, at minimal clearance, with a flat-surface heat source.
- provide the mechanical strength necessary to hold two opposing housing plates of a flat evaporator to a metal wick, and rely on the tensile strength of the wick material, so as to prevent deformation of the plates.
- provide a method for assembling a lightweight flat capillary evaporator.
- provide a capillary evaporator having a liquid superheat tolerant wick.
- provide a capillary evaporator having etched micro-channels as vapor grooves.
- provide an LHP that can reliably operate under terrestrial conditions regardless of the vapor pressure of the working fluid.
- provide an LHP that is physically compact with the various components integrated into a unitary package.

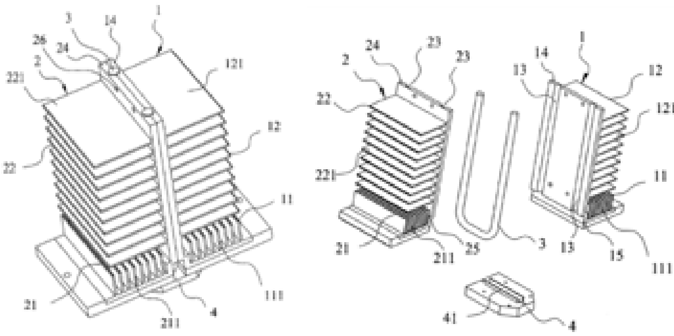
The above objects are obtained by a capillary wick that has a structure resistant to back-conduction. The wick has a configuration that is liquid superheat tolerant.

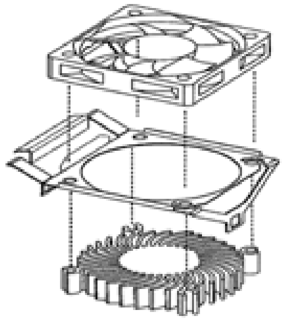
SYMMETRICAL HEAT SINK MODULE WITH A HEAT PIPE FOR SPREADING OF HEAT

6,945,319 B1, Li, et al

Although this invention does not fit into the exact category of LHP, the use of heat pipe in this configuration where the orientation can play significant role in its performance, is worthy of review. The present invention is a symmetrical heat sink module with a heat pipe for spreading of heat, comprising a first and a second sets of fins each with a first and a second heat dissipation areas where the first and the second heat dissipation areas each comprises corresponding concave parts; a curved heat pipe between the concave parts with the bottom being convex out of the bottom of the first and the second sets of fins; and a base at the convex bottom of the curved heat pipe. Accordingly, the first and the second sets of fins are corresponding to each other so that the heat sink module obtains larger heat dissipation area and the function of two-side dissipation to achieve better heat dissipation efficiency.

The prior art of "Heat sink apparatus" is disclosed in U.S. Pat. No. 6,525,939 comprising: a fan comprising at least one attachment hole; a heat sink module mounted on the CPU, comprising a metal heat conduction column, a plurality of arc-shaped cooling fins radiating from the edge of the metal heat conduction column and a plurality of mounting holes positioned at the ends of the arc-shaped cooling fins, corresponding to the attachment holes; and a latch comprising a latch





arm, a plurality of openings and at least one latch hole. Therein, the latch arm is secured to the socket of the CPU; each of the openings correspond to the mounting holes and the attachment holes; the fan and the heat sink module are mounted onto the CPU with the latch; and the metal heat conduction column is a cylinder in direct contact with the CPU and is hollowed out as being filled with a metal having a heat conductibility that is better than the heat conductibility of the metal of the column. Although the prior art mentioned above is able to dissipate heat by a heat sink module, the heat sink module is formed as a unity with limited dissipation area so that the heat dissipation efficiency is limited. So, the prior art cannot fulfill the requirements from the user on actual use.

The main purpose of the present invention is to form a first set of fins and a second set of fins to be symmetrical in the module to get larger heat dissipation area so that a better heat dissipation efficiency is achieved. Another purpose of the present invention is to form the first set of fins and the second set of fins to be symmetrical in the module to dissipate heat at two sides so that a better heat dissipation efficiency is achieved. To achieve the above purposes, the present invention is a symmetrical heat sink module with a heat pipe for spreading of heat, comprising a first and a second sets of fins symmetrically formed; a curved (U-type) heat pipe between the two sets of fins; and a base at the bottom of the curved heat pipe.

The two sets of fins comprise a first heat dissipation area at the bottom of each set of fins; a second heat dissipation area on a surface of each set of fins near the first heat dissipation area; two vicinal concave parts on the opposite surface of each set of fins where the two sets of fins are combined by the two surfaces having the concave parts. The curved heat pipe is deposited in the concave parts of the two sets of fins; and the bottom of the curved heat pipe is convex out of the bottom of the two sets of fins. The base is deposited at the convex bottom of the curved heat pipe, and comprises a connection part corresponding to the bottom of the curved heat pipe. So, by the first and second sets of fins formed symmetrically, the heat sink module gains twice the heat dissipation area and the ability of heat dissipation at two sides to achieve better heat dissipation.