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(54) **NON-LINEAR FIN HEAT SINK**

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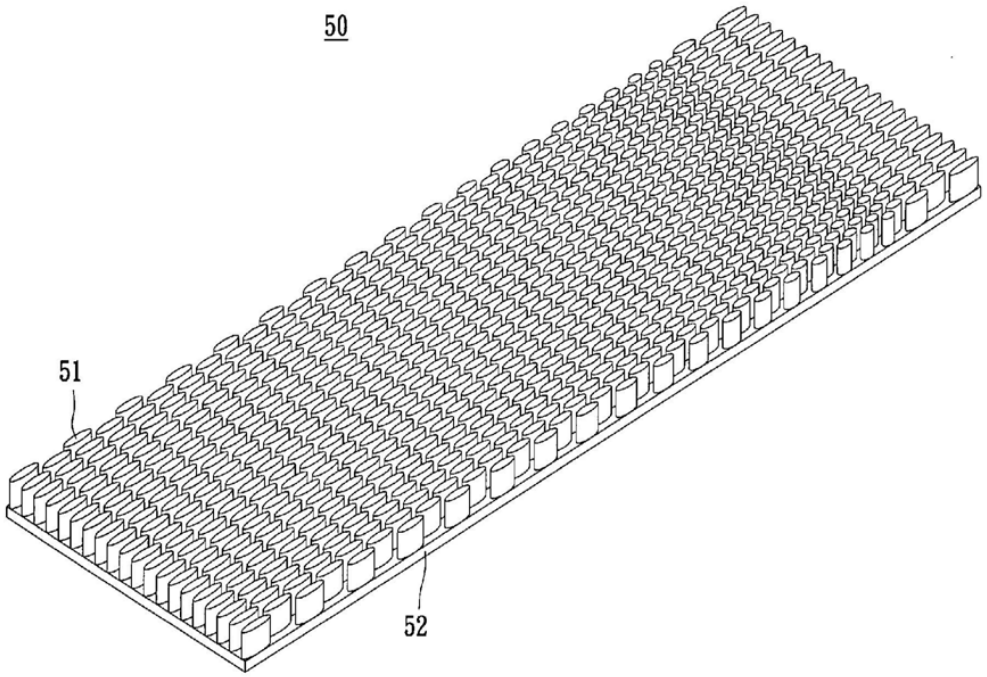
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(52) **U.S. Cl.** **165/80.3**

(57) **ABSTRACT**

A non-linear fin heat sink is provided for dissipating/removing heat uniformly from a device, where the heat generation is non-uniform over that device, while also providing a small and relatively lightweight heat sink. The heat sink has extended surface protrusions that are optimally shaped in recognition of convective heat transfer, conductive heat transfer, and flow resistance allowing the heat sink to offset the temperature rise of a coolant media and provide enhanced cooling for the coolant temperature, deliver optimized cooling efficiency per the local physical properties of the coolant media, be used with a fluid for effectuating heat transfer; either liquid coolant, gas coolant or a combination thereof. Furthermore the heat sink features turbulence enhancement of the coolant stream by a pin array through which coolant stream passes, such fin array featuring a non-linear shape, spacing, and height pattern to provide optimal cooling while simultaneously reducing volume and flow resistance.



SEC et al. v. MRI
SEC Exhibit 1006.0001
IPR2023-00254

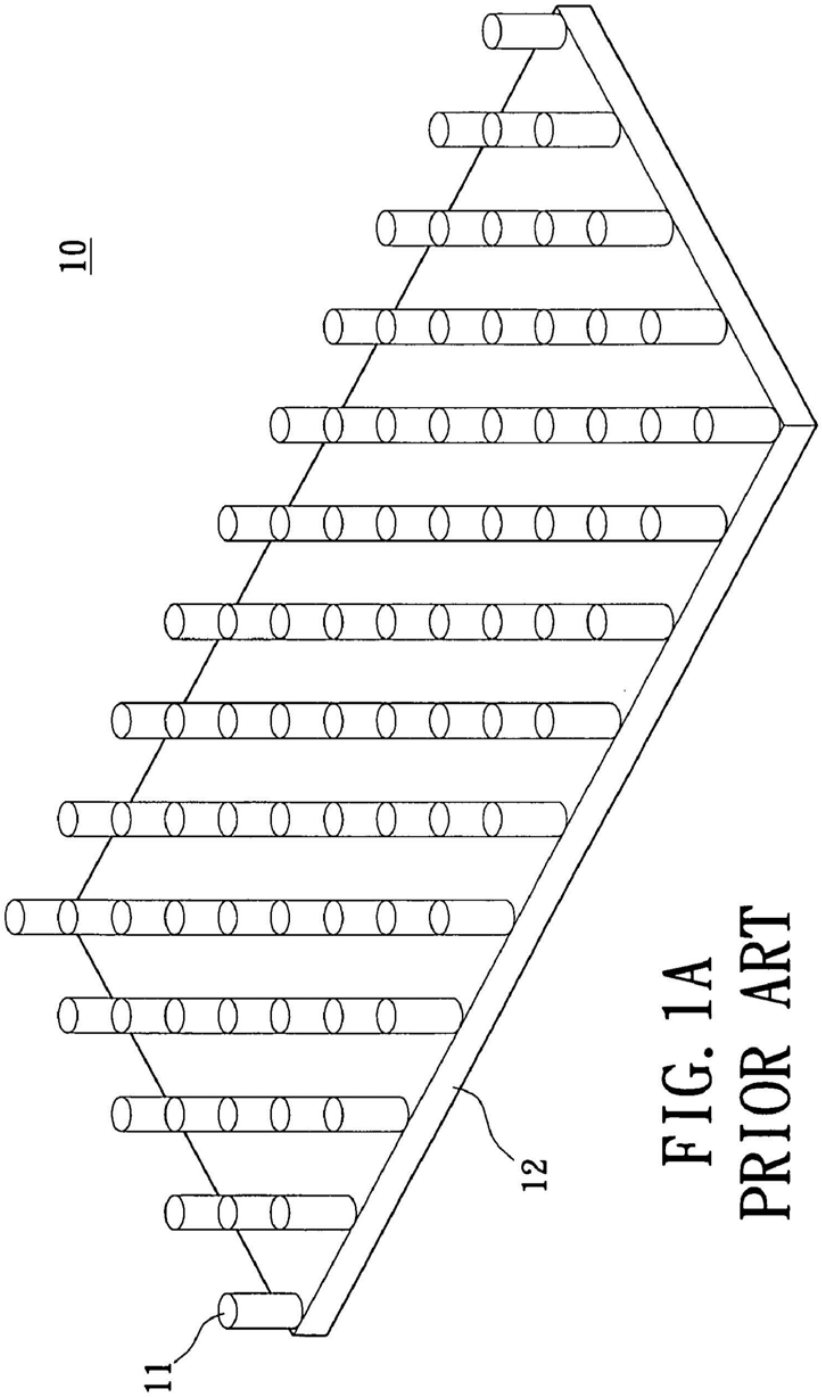


FIG. 1A
PRIOR ART

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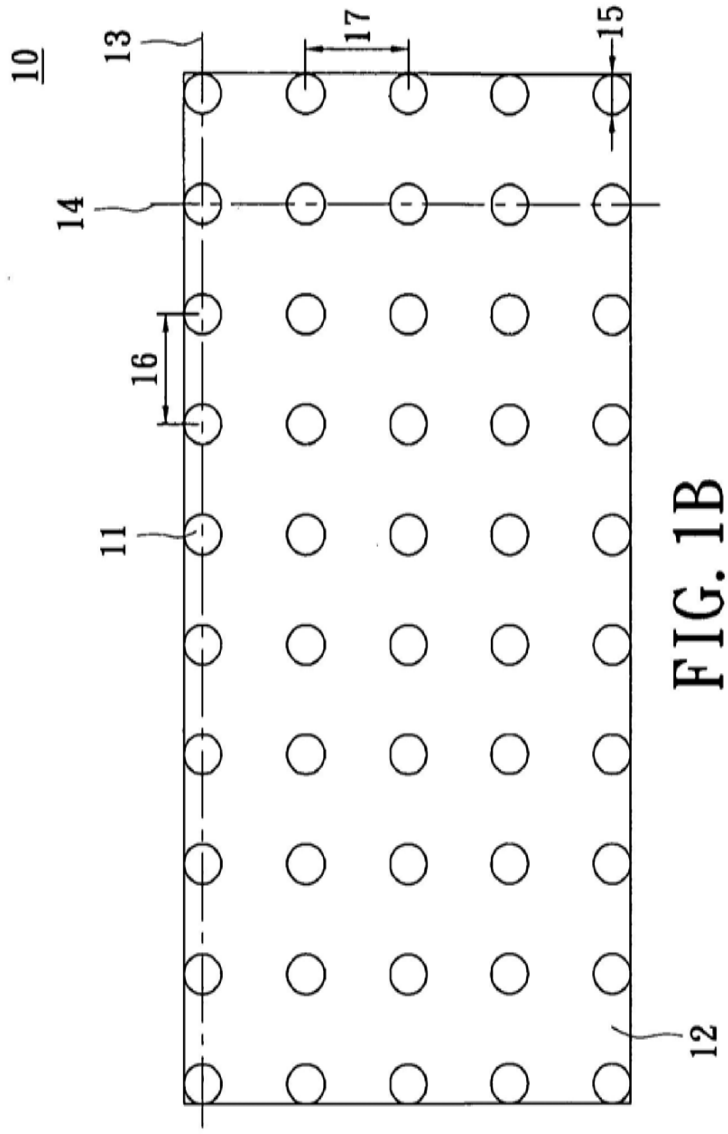


FIG. 1B
PRIOR ART

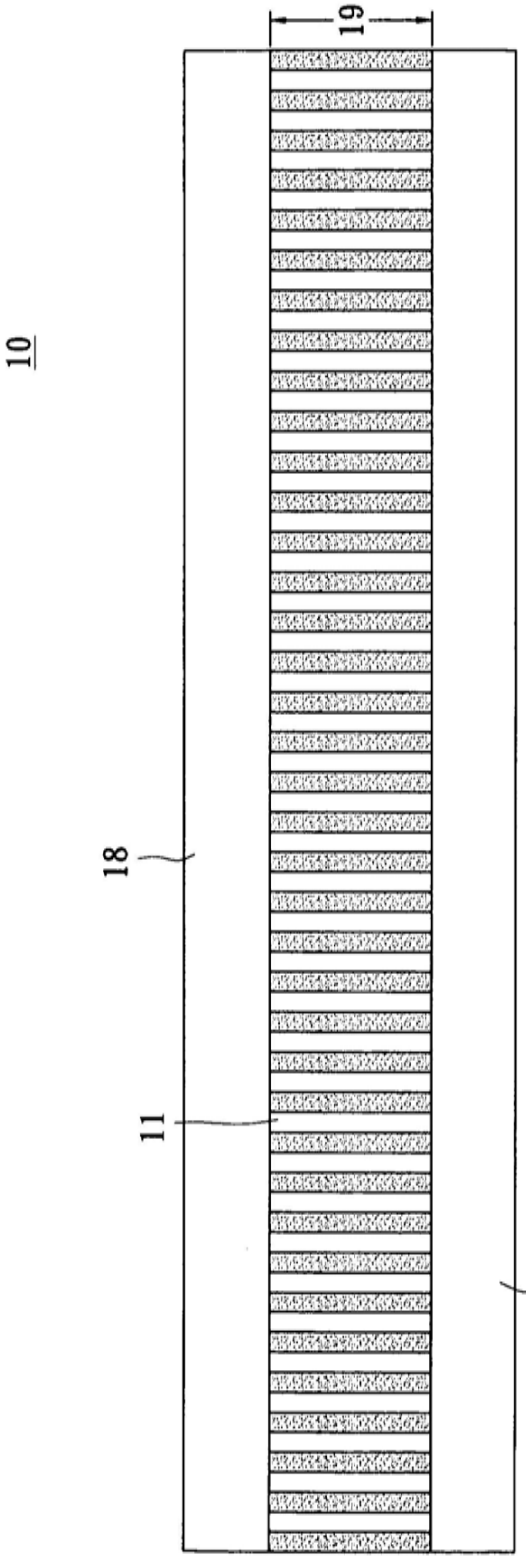


FIG. 1C
PRIOR ART

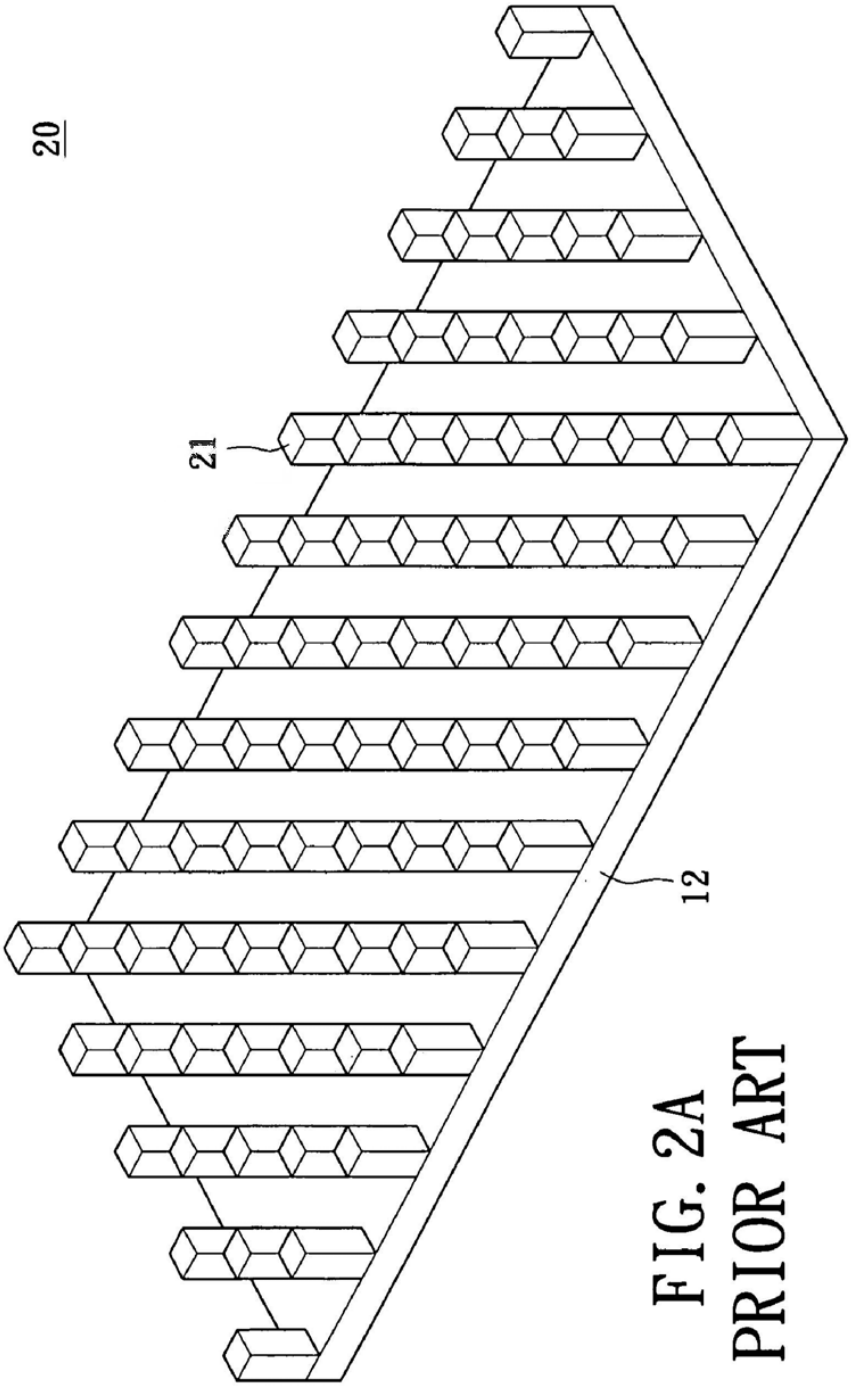


FIG. 2A
PRIOR ART

SEC et al. v. MRI
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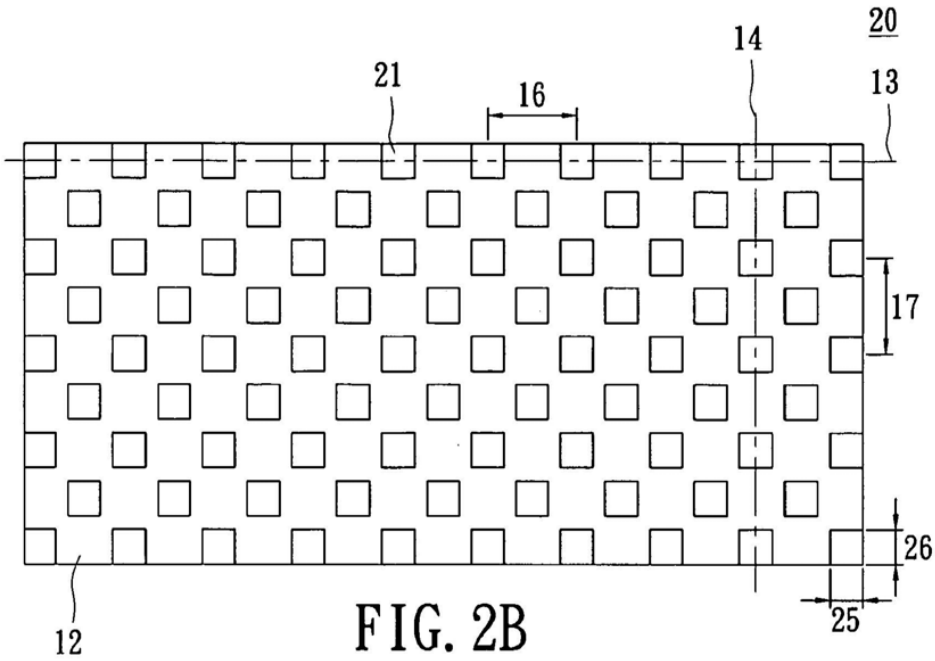


FIG. 2B
PRIOR ART

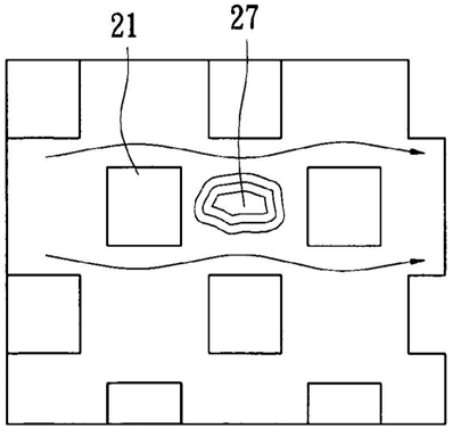


FIG. 2C
PRIOR ART

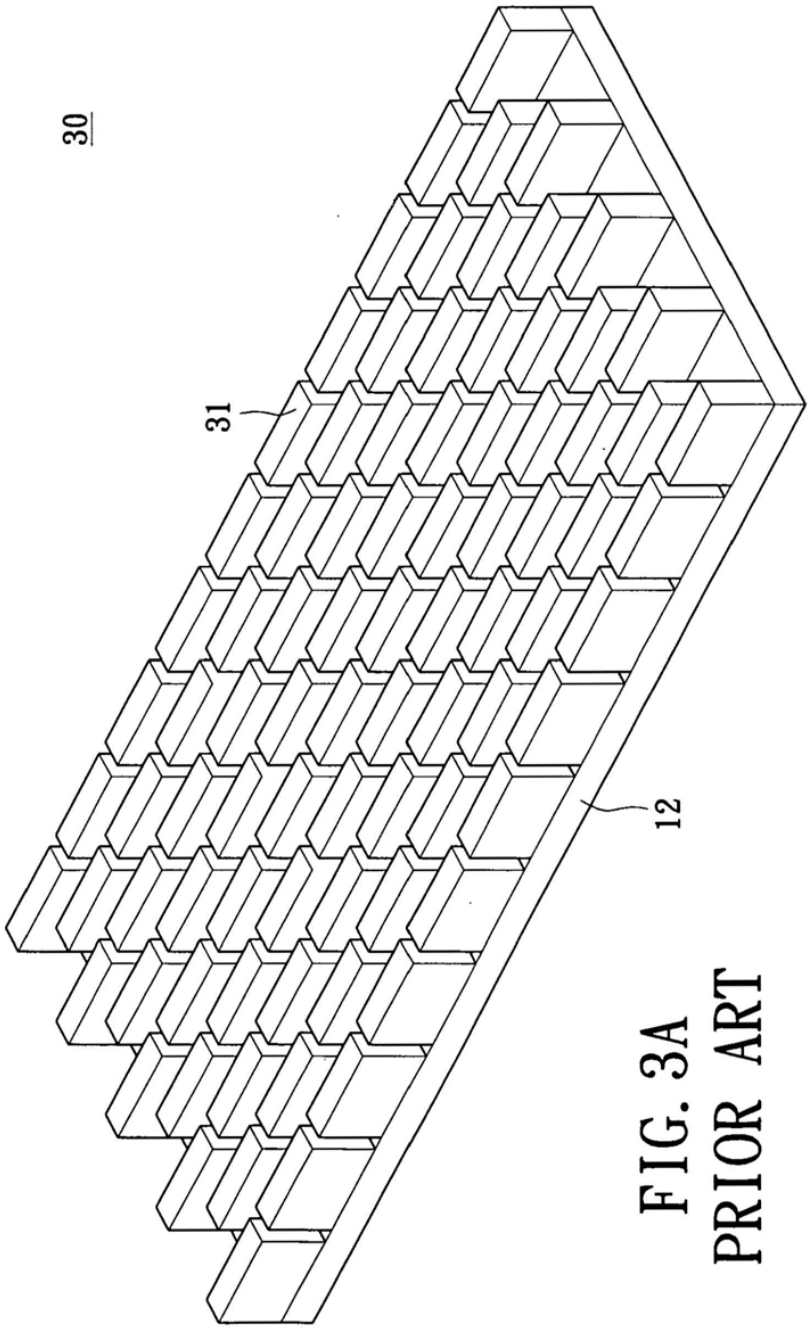


FIG. 3A
PRIOR ART

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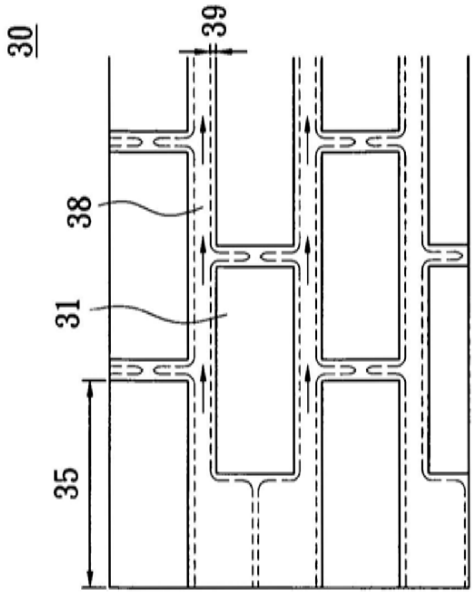


FIG. 3C
PRIOR ART

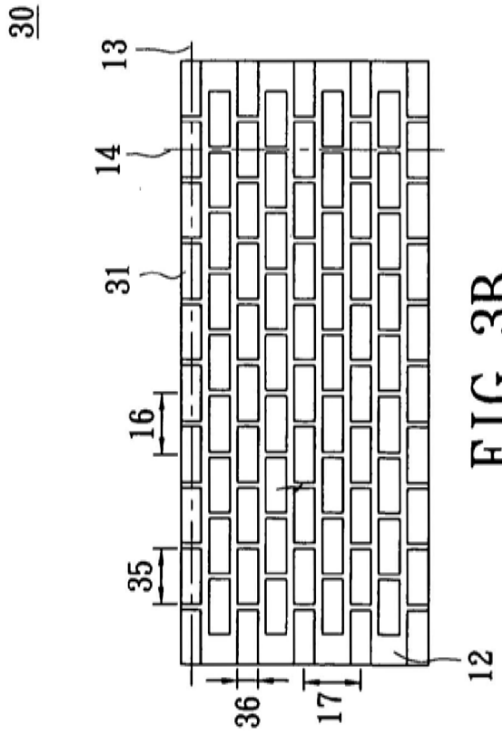


FIG. 3B
PRIOR ART

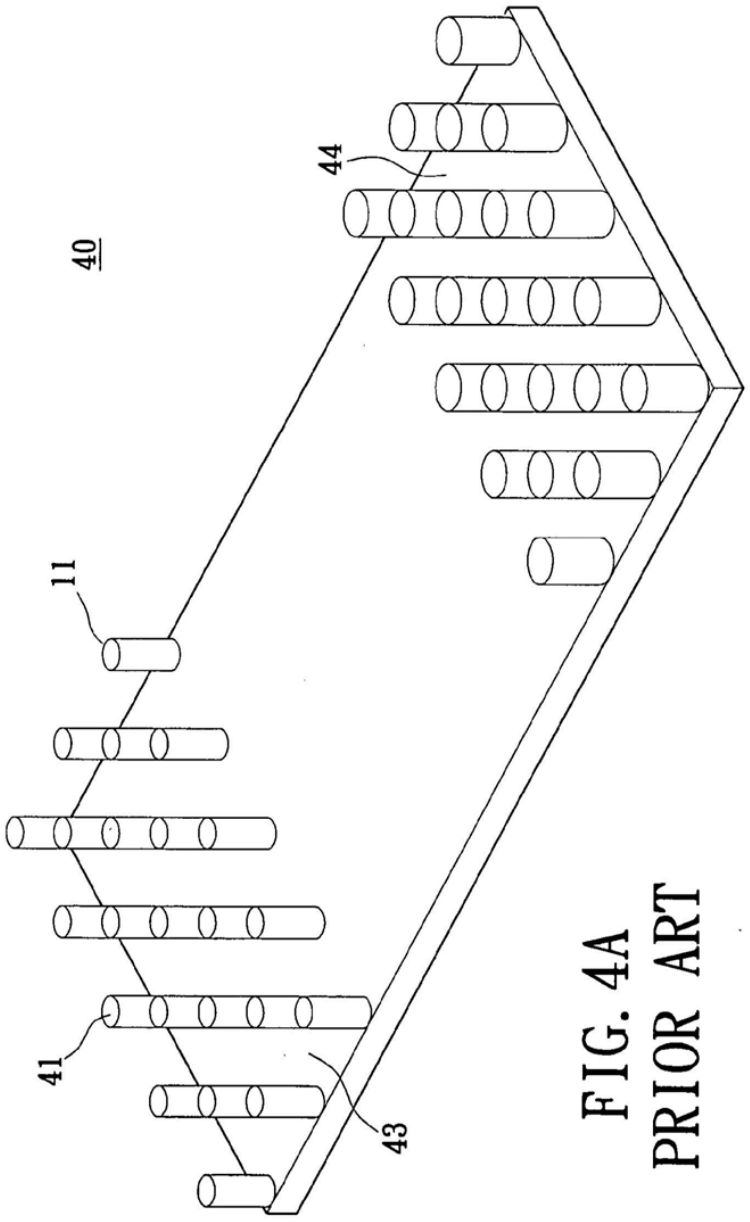


FIG. 4A
PRIOR ART

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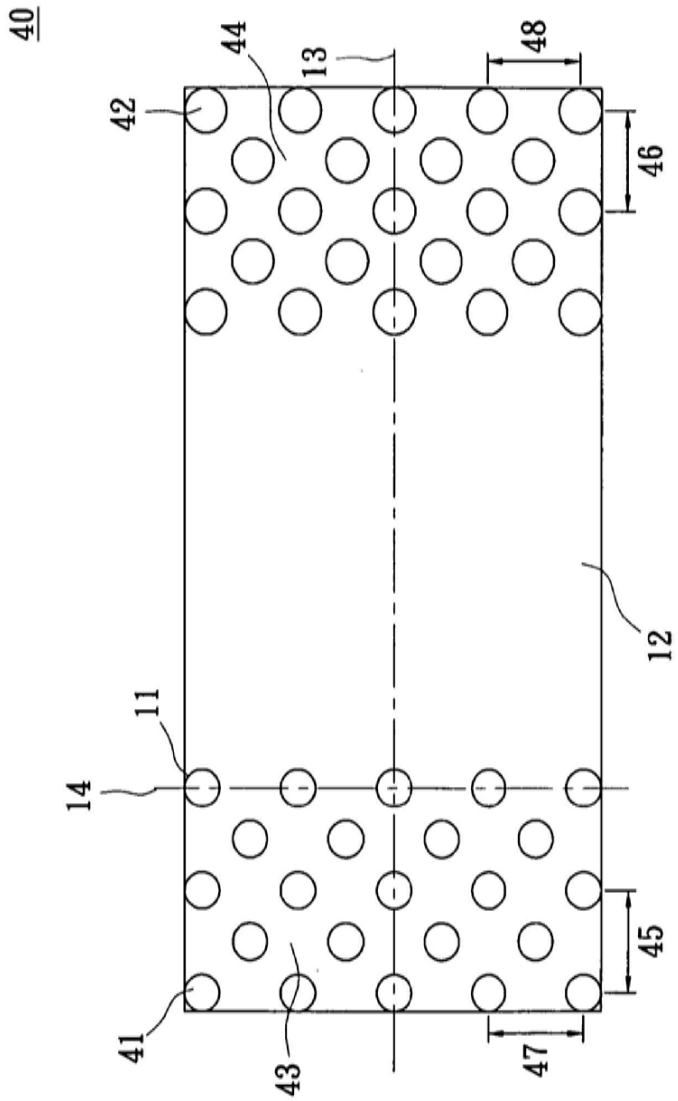


FIG. 4B
PRIOR ART

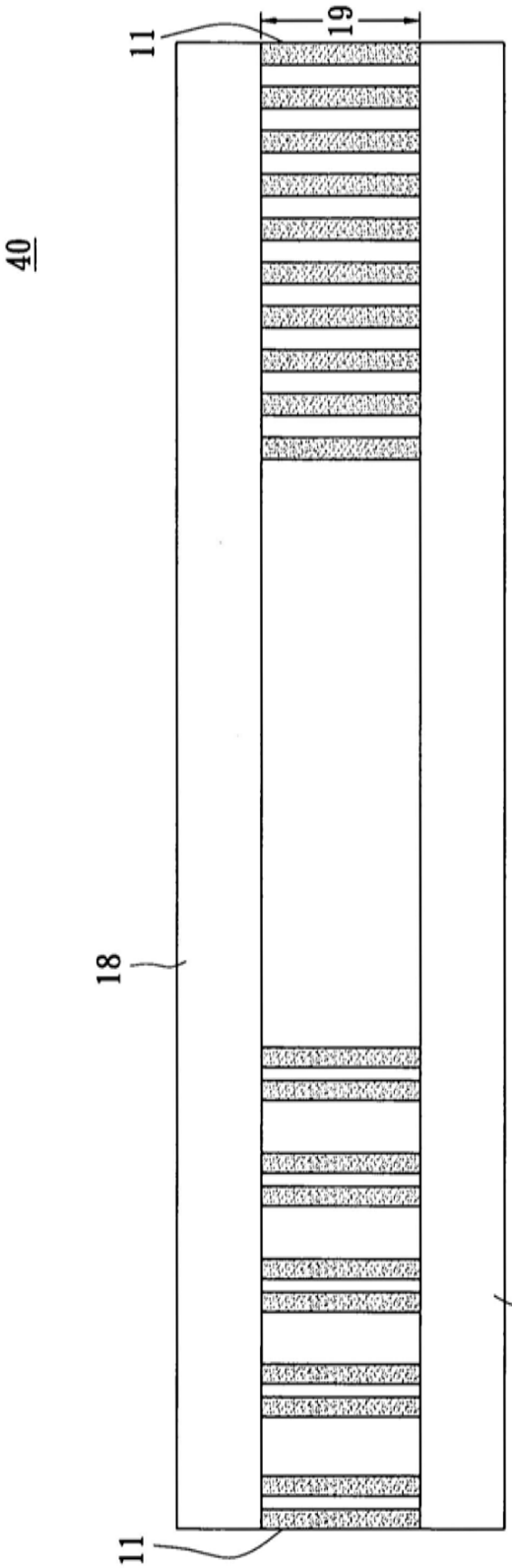


FIG. 4C
PRIOR ART

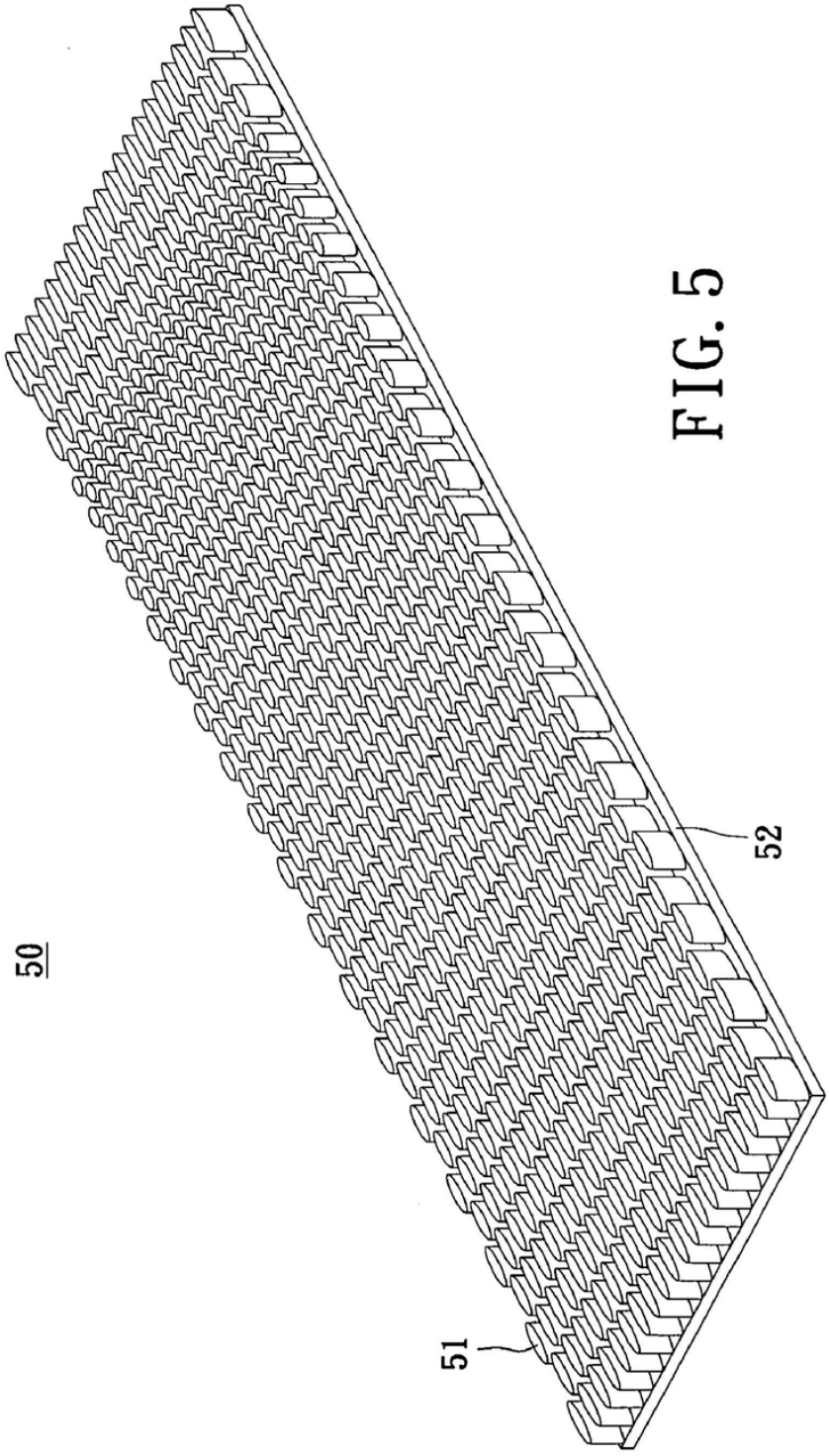


FIG. 5

SEC et al. v. MRI
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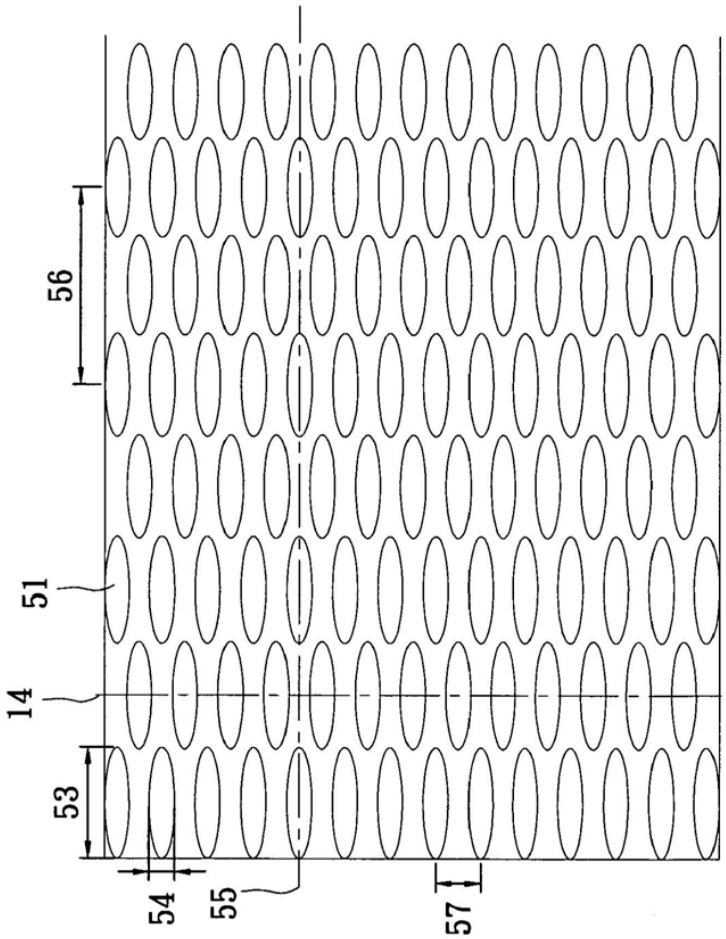


FIG. 6

SEC et al. v. MRI
SEC Exhibit 1006.0013
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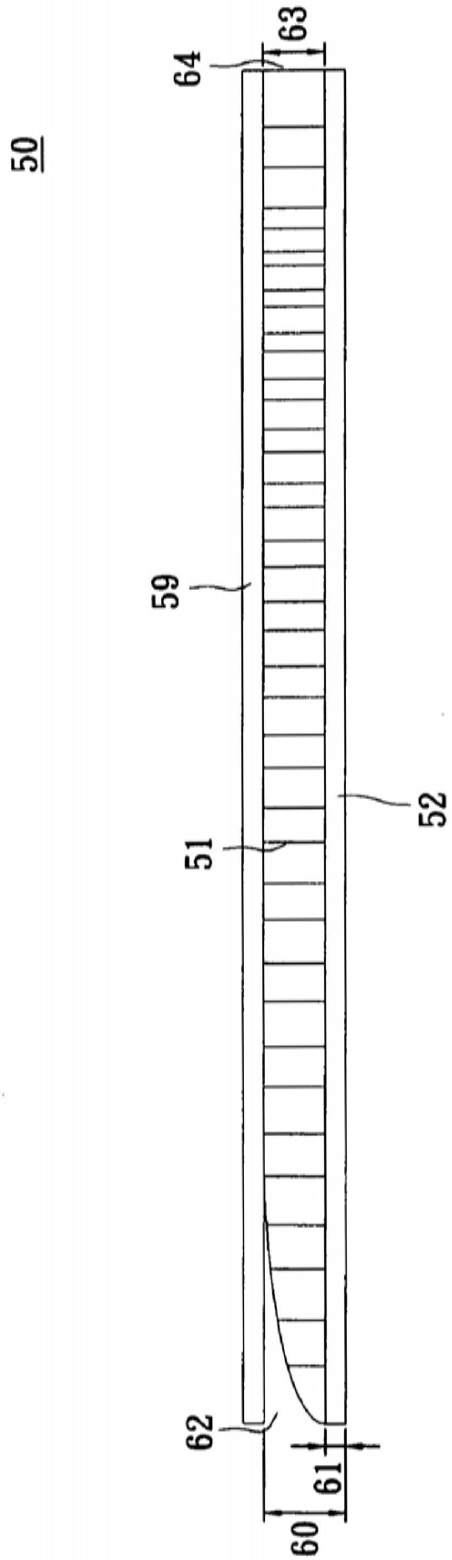


FIG. 7

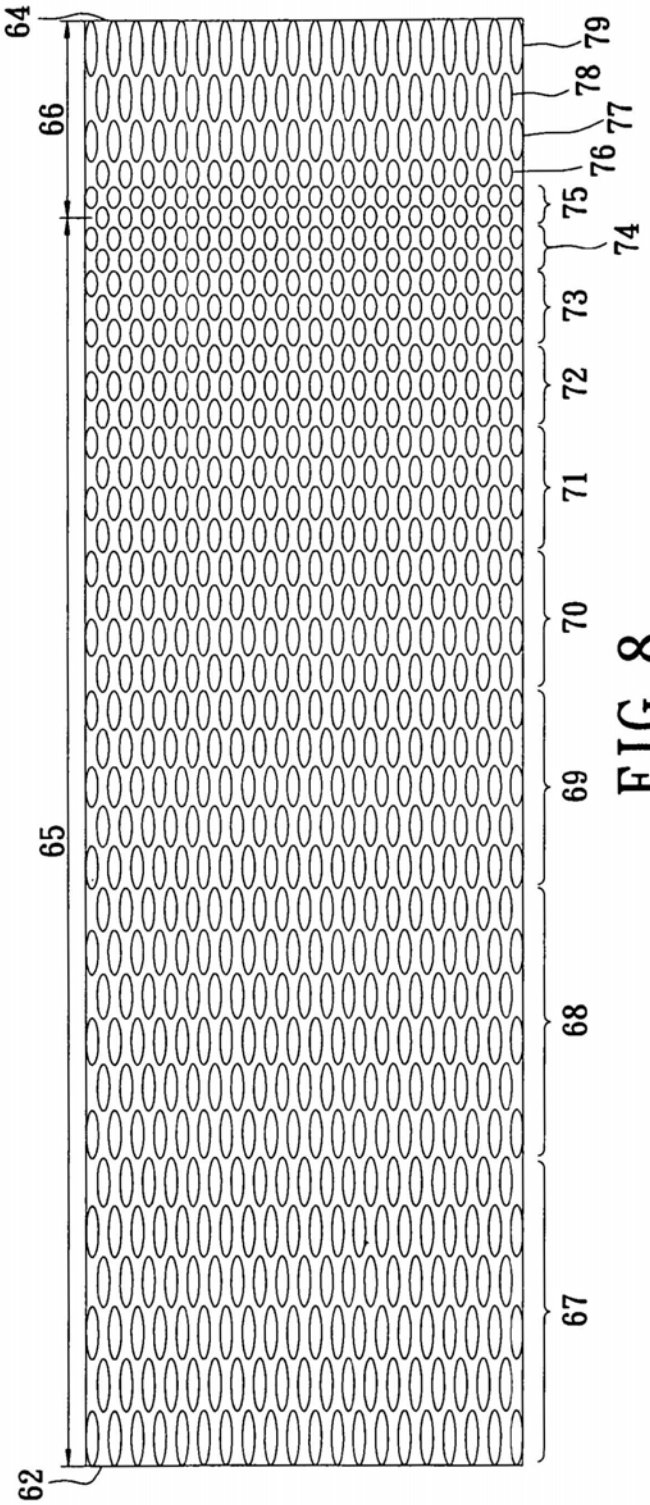


FIG. 8

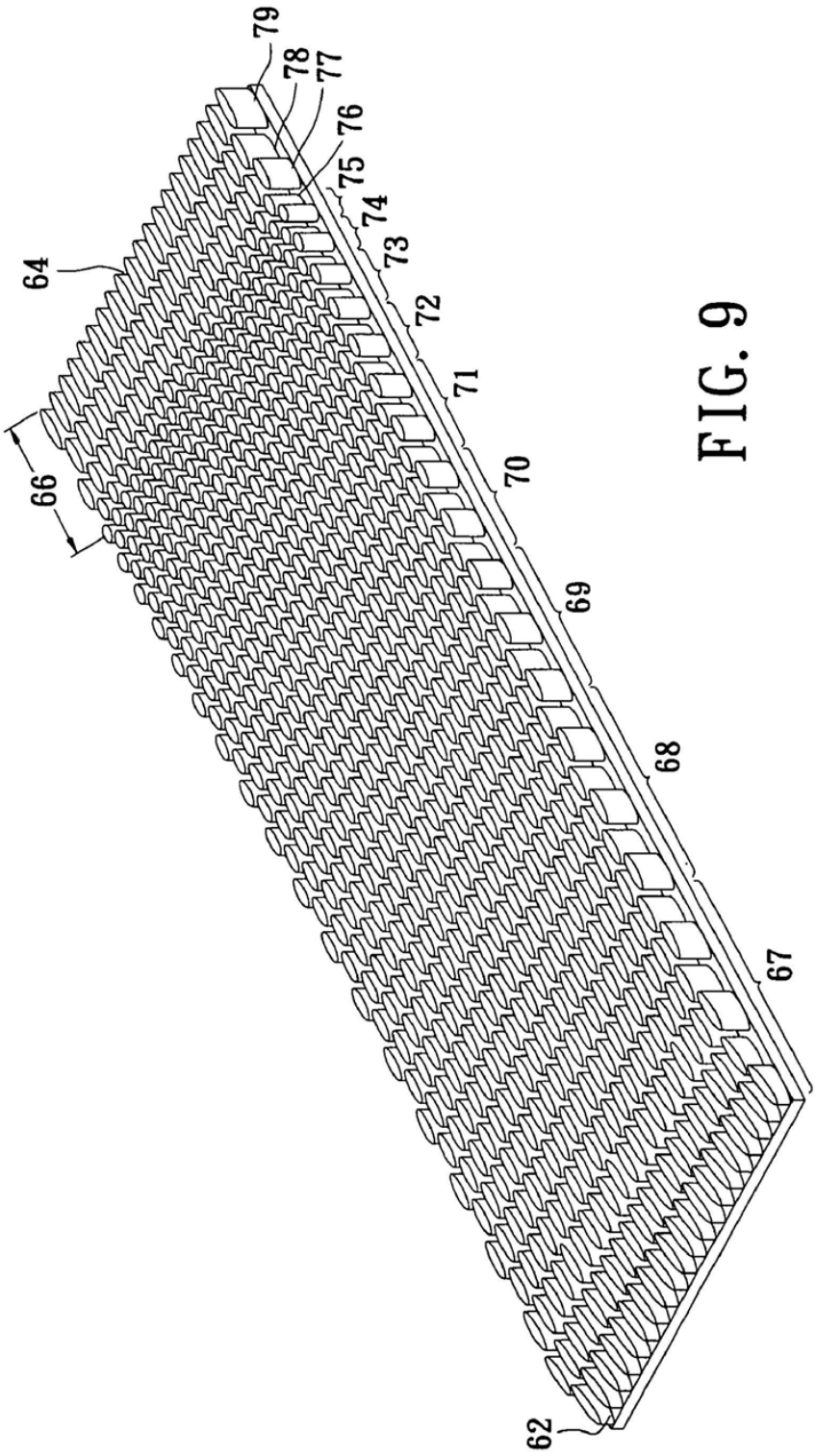


FIG. 9

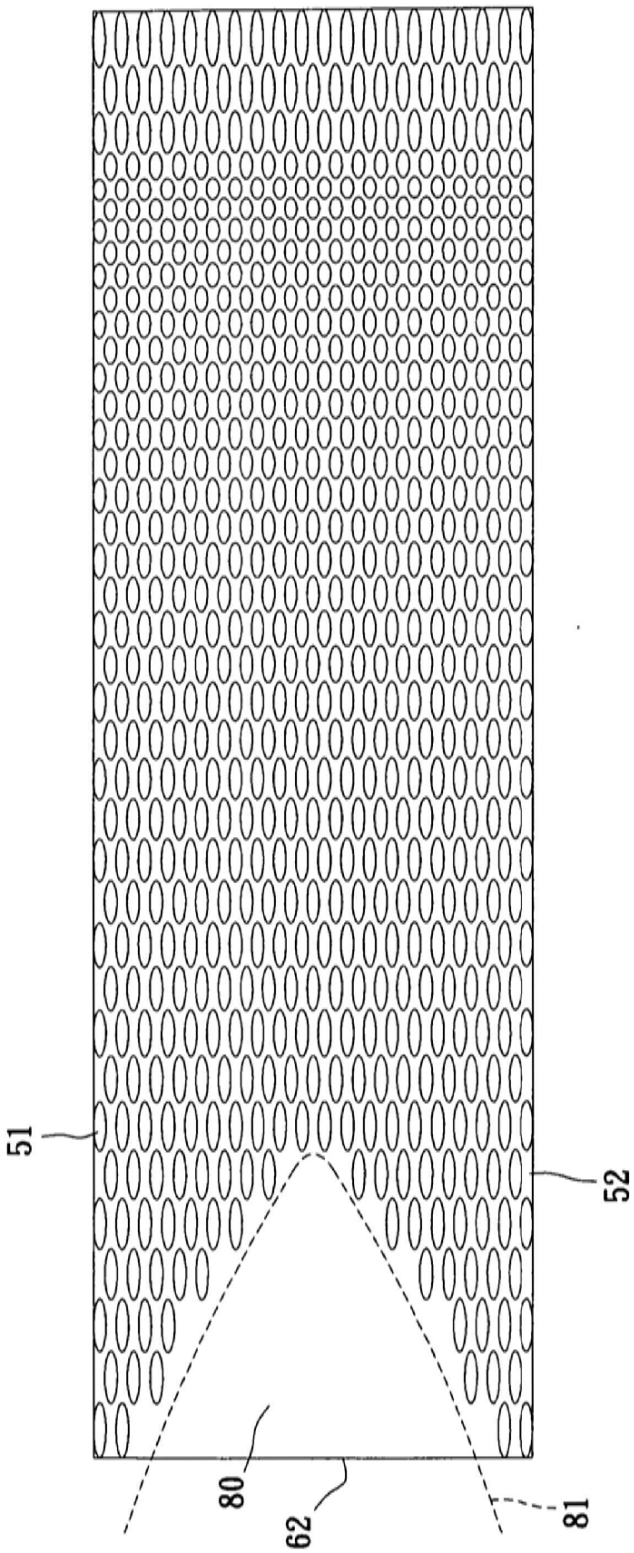


FIG. 10

SEC et al. v. MRI
SEC Exhibit 1006.0017
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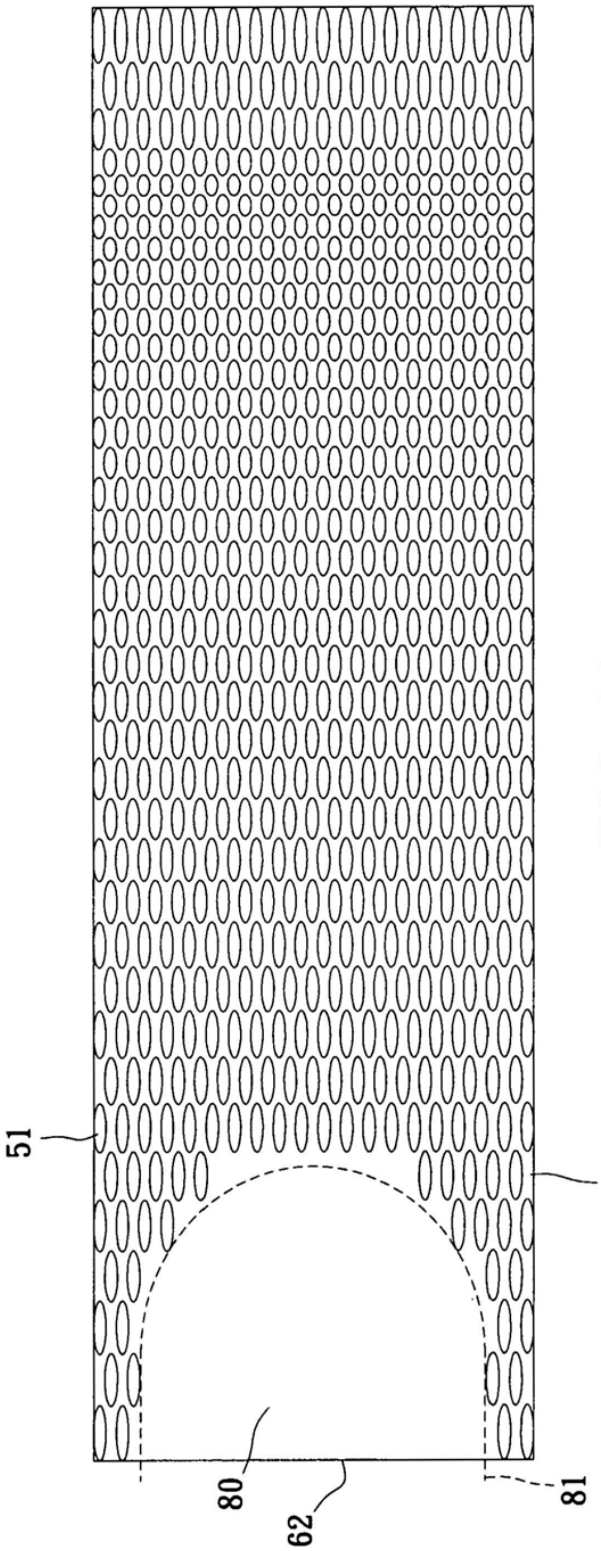


FIG. 11

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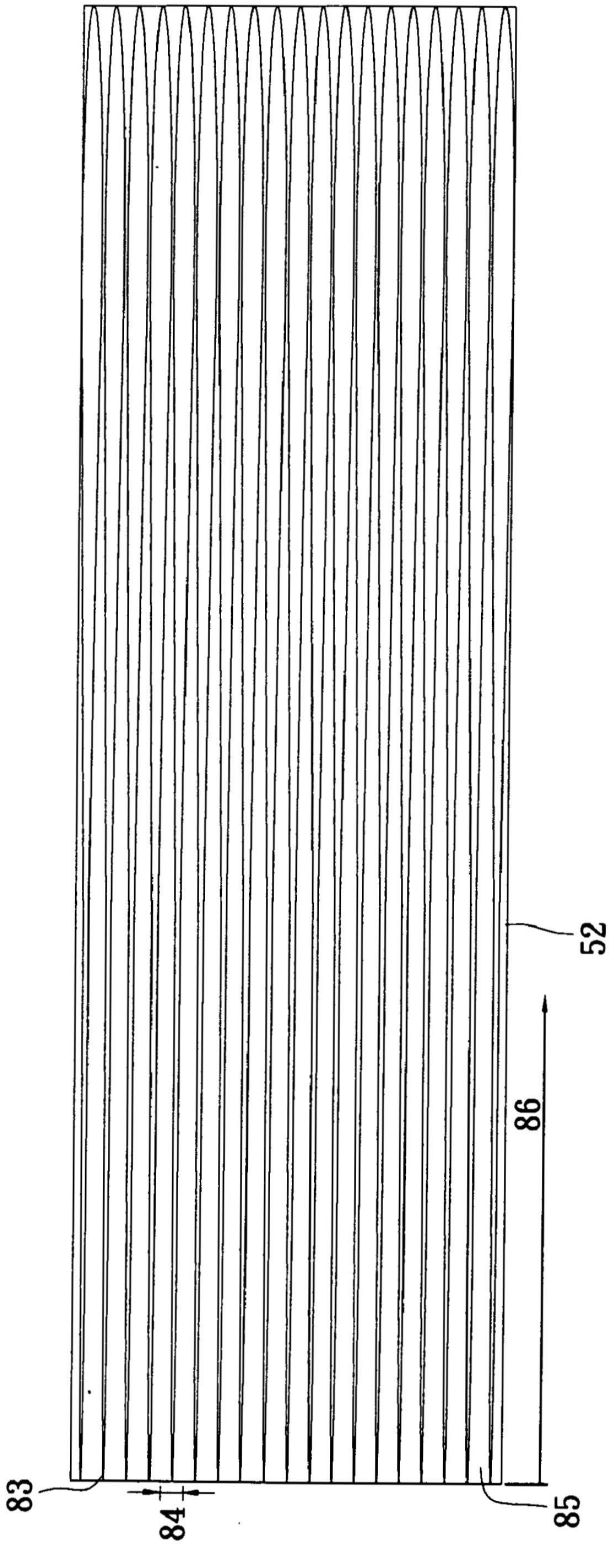


FIG. 12

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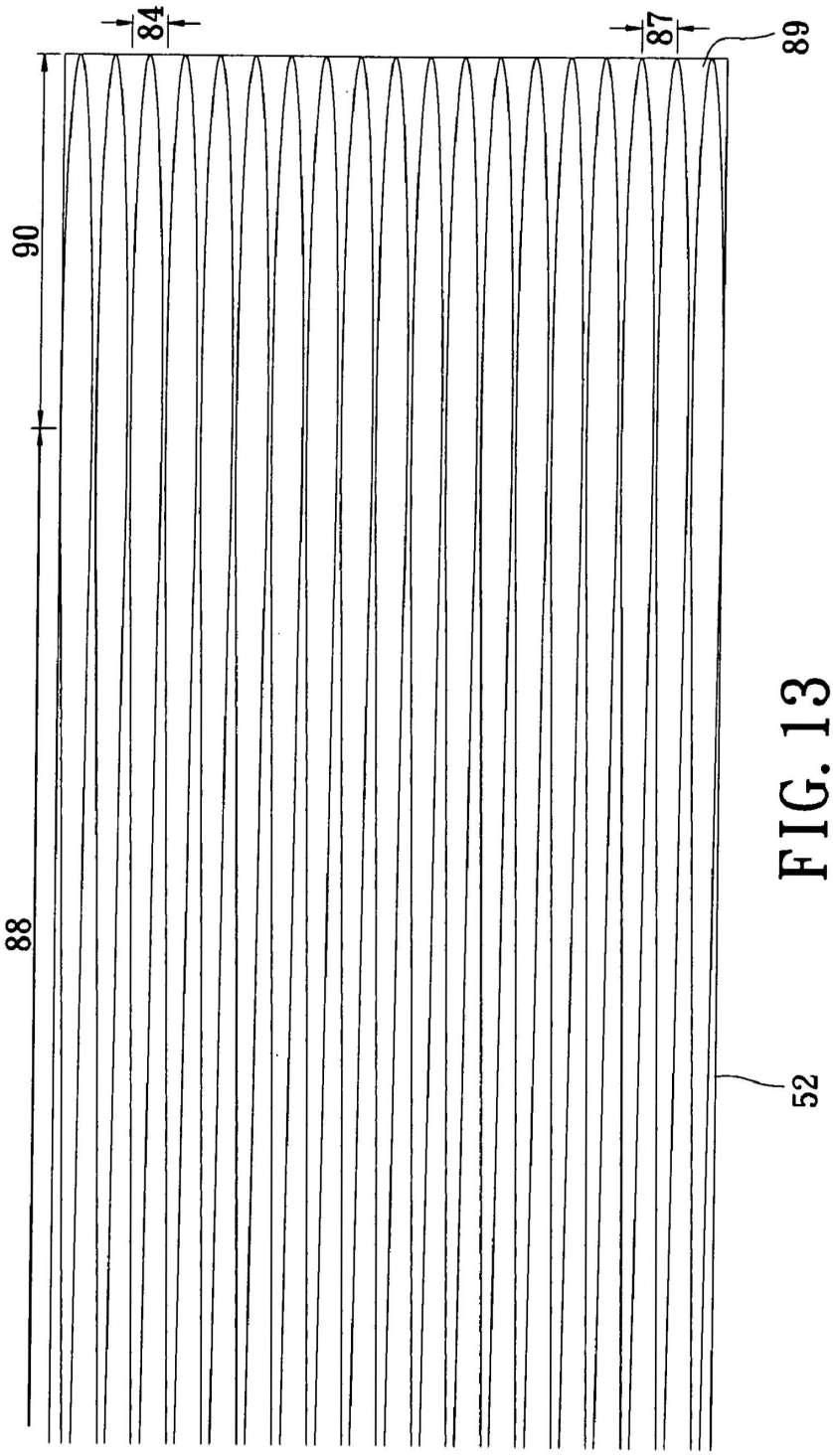


FIG. 13

SEC et al. v. MRI
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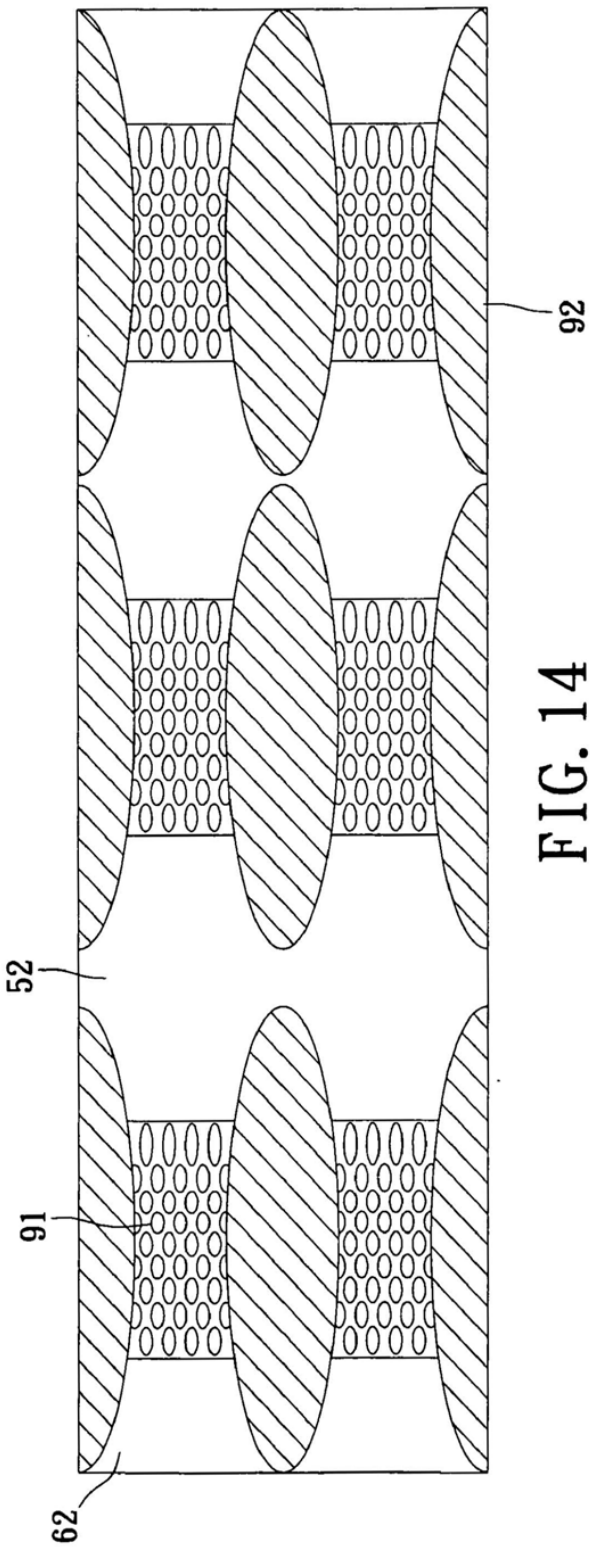


FIG. 14

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SEC Exhibit 1006.0021
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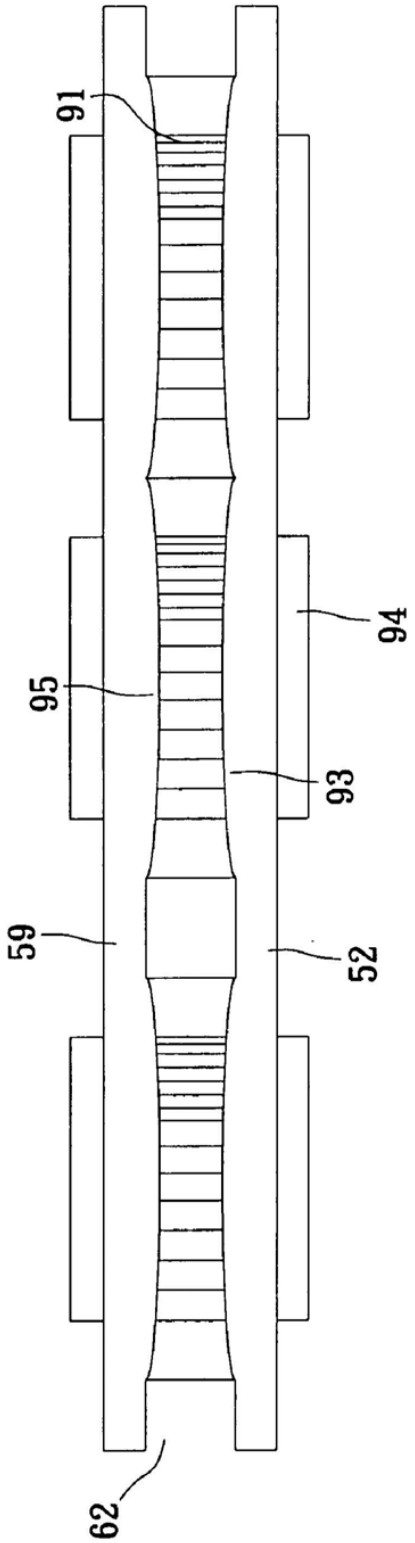


FIG. 15

SEC et al. v. MRI
SEC Exhibit 1006.0022
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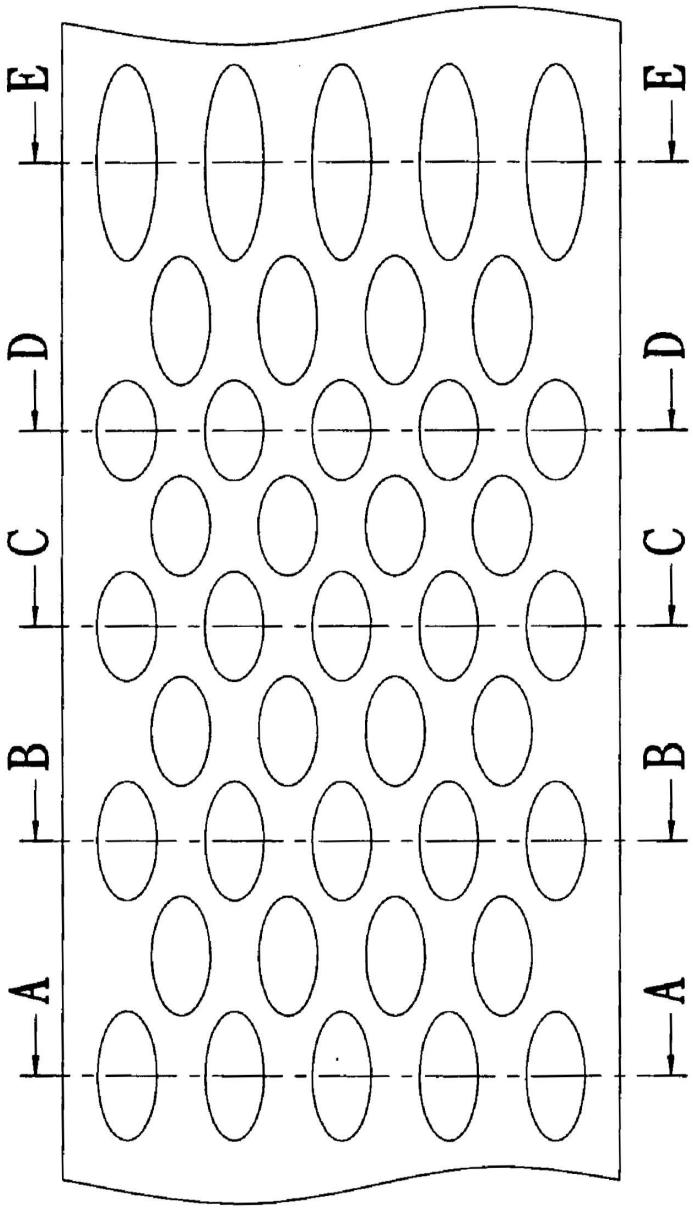


FIG. 16A

SEC et al. v. MRI
SEC Exhibit 1006.0023
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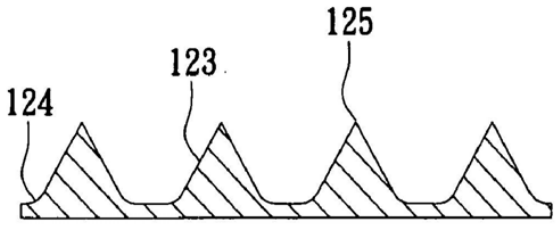


FIG. 16B

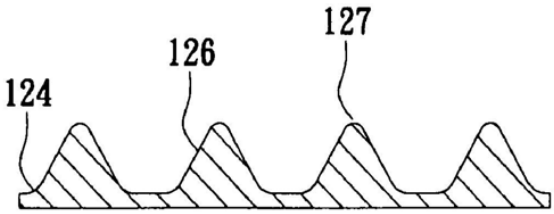


FIG. 16C

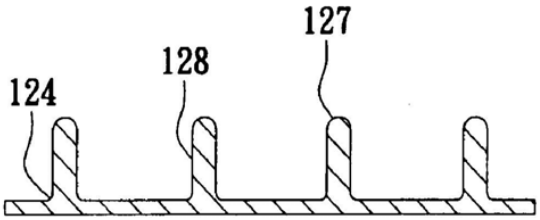


FIG. 16D

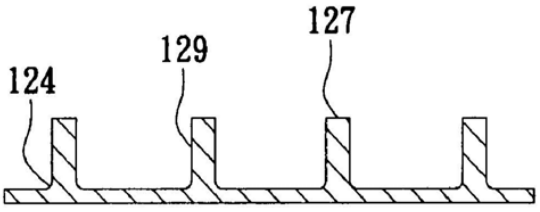


FIG. 16E

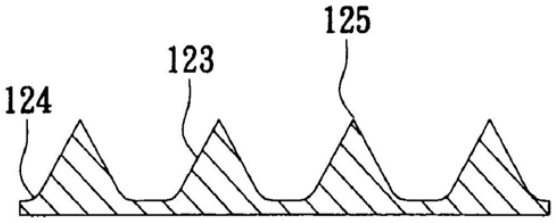


FIG. 16F

SEC et al. v. MRI
SEC Exhibit 1006.0024
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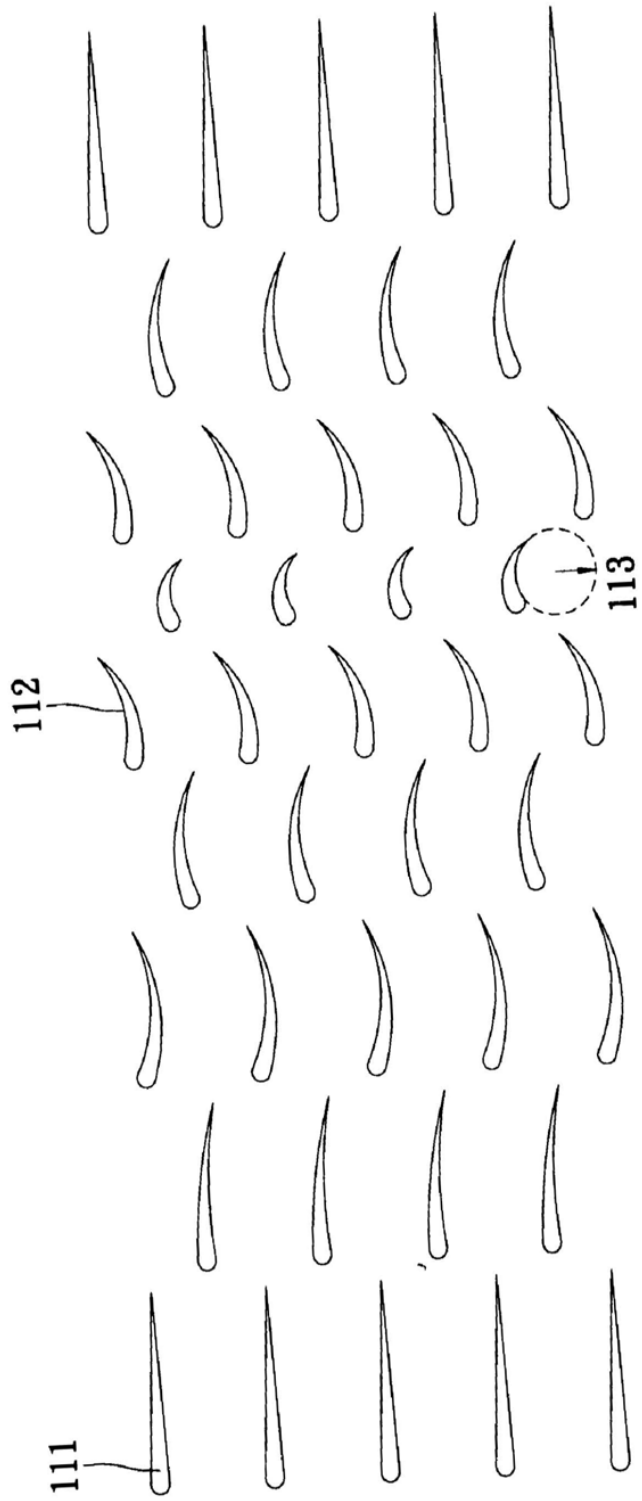


FIG. 17

SEC et al. v. MRI
SEC Exhibit 1006.0025
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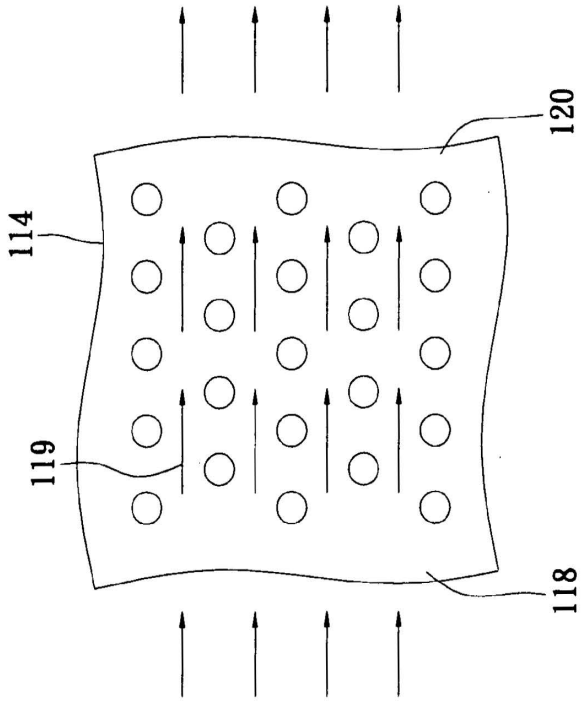


FIG. 18B

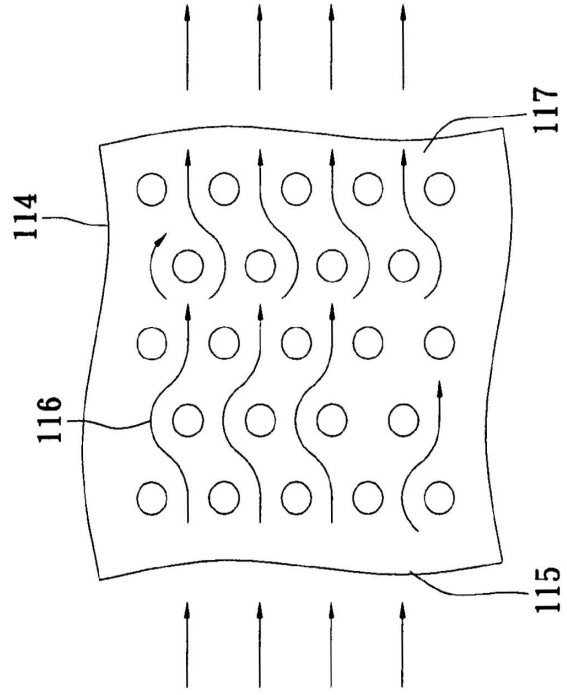


FIG. 18A

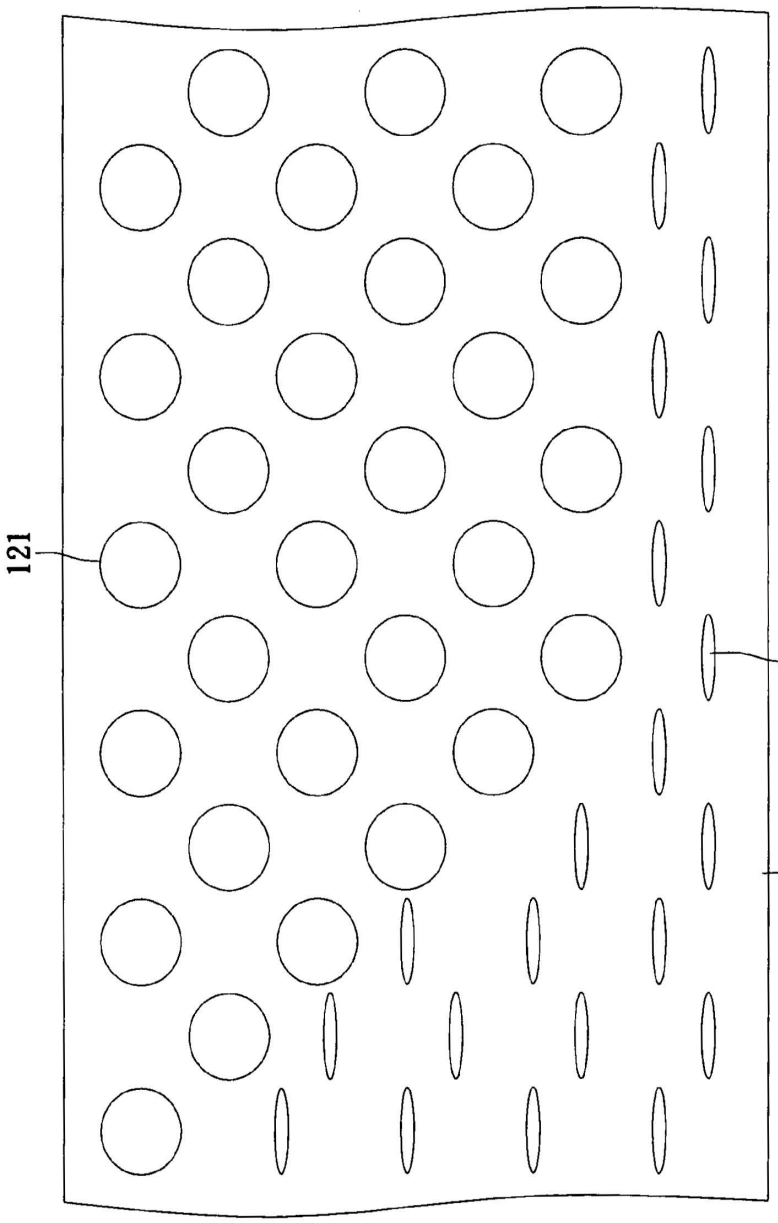


FIG. 19

SEC et al. v. MRI
SEC Exhibit 1006.0027
IPR2023-00254

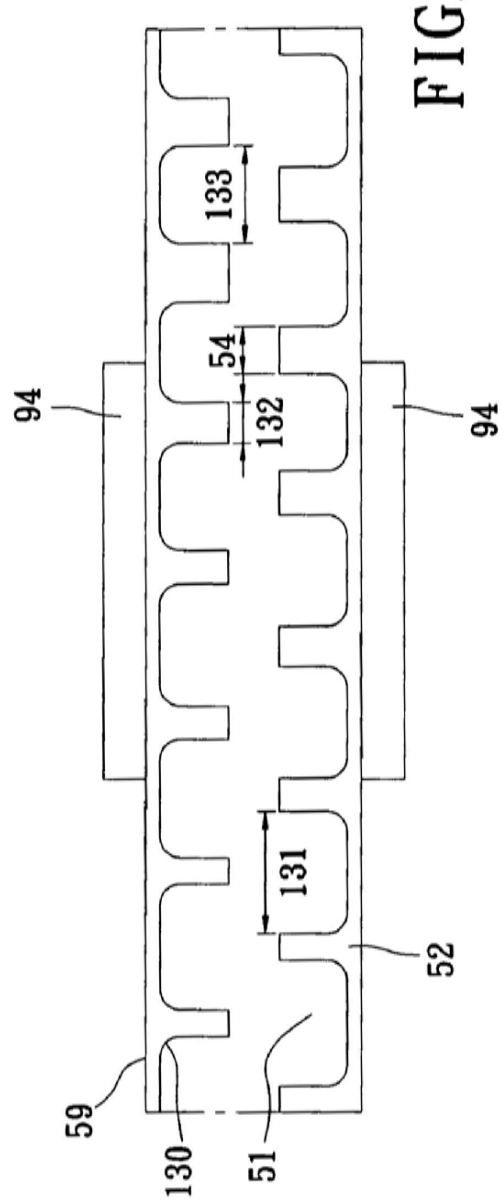


FIG. 20A

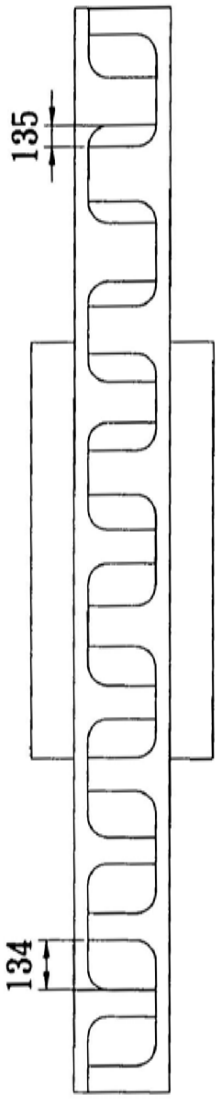


FIG. 20B

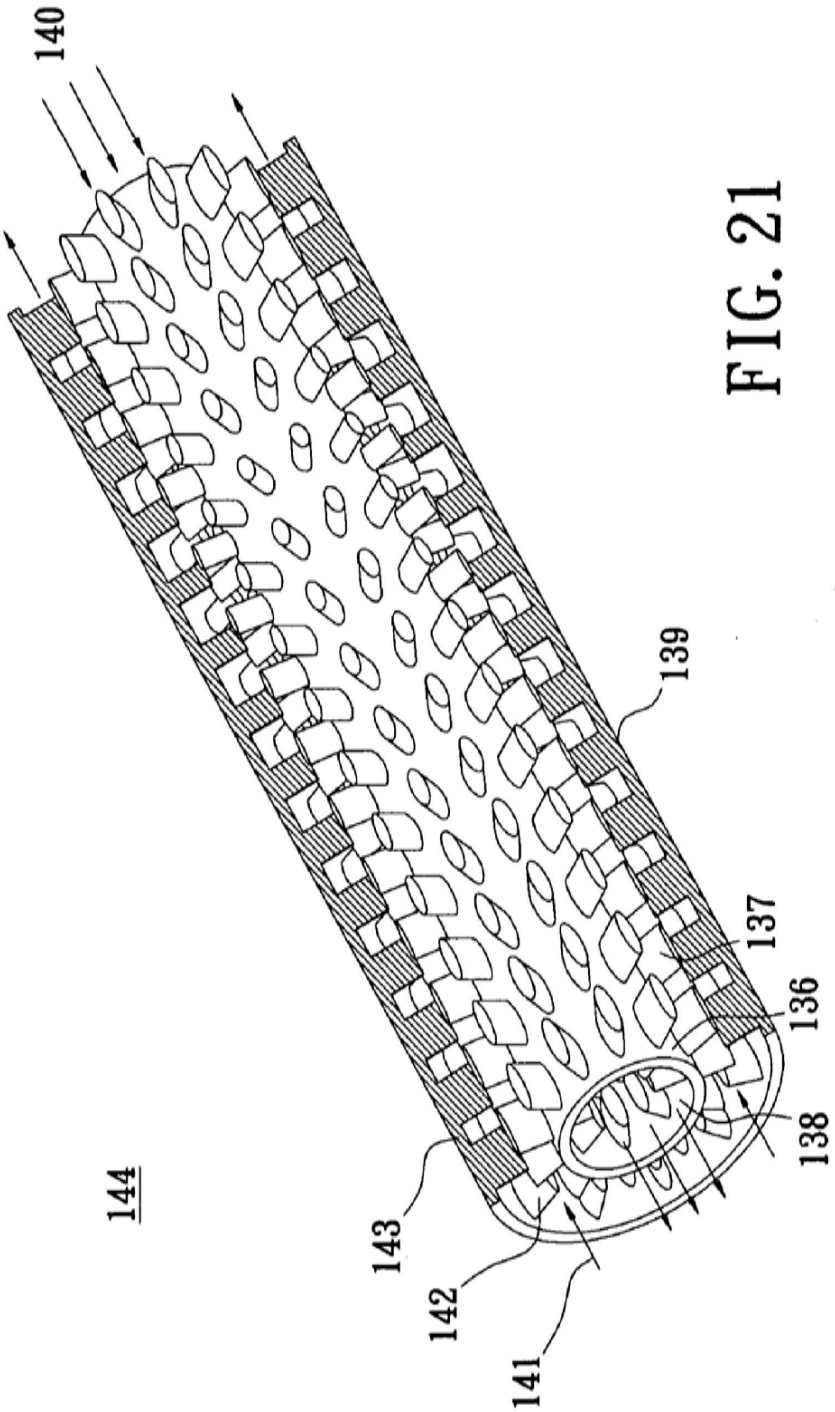
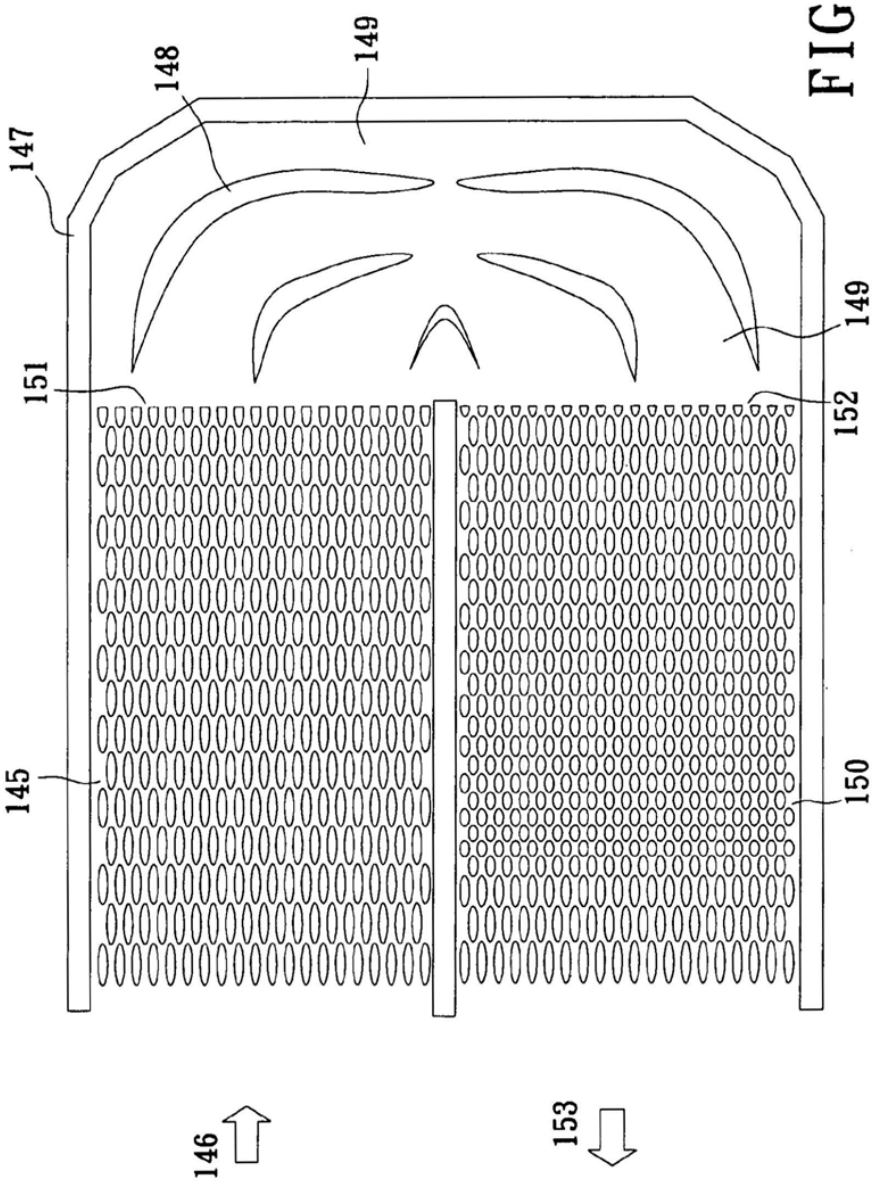


FIG. 21

SEC et al. v. MRI
SEC Exhibit 1006.0029
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FIG. 22



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SEC Exhibit 1006.0030
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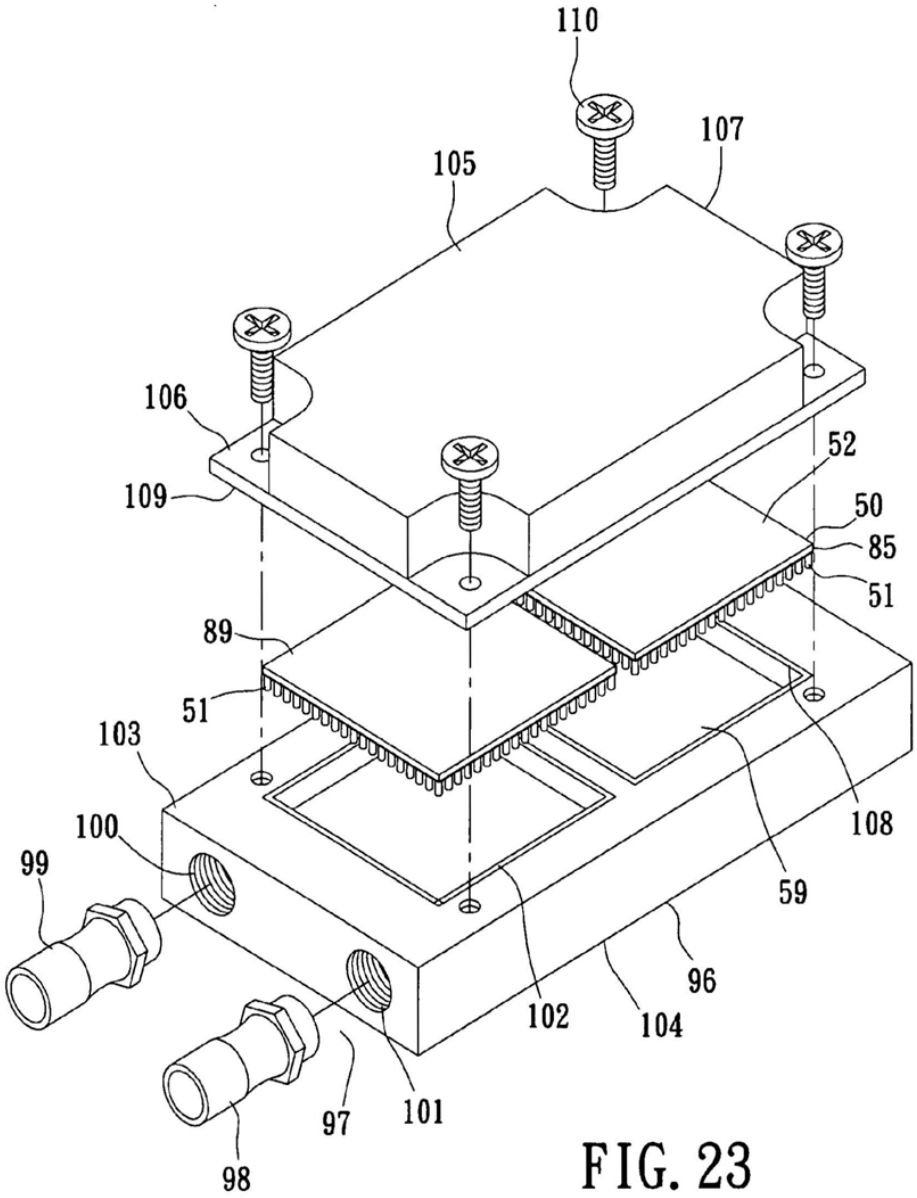


FIG. 23

SEC et al. v. MRI
SEC Exhibit 1006.0031
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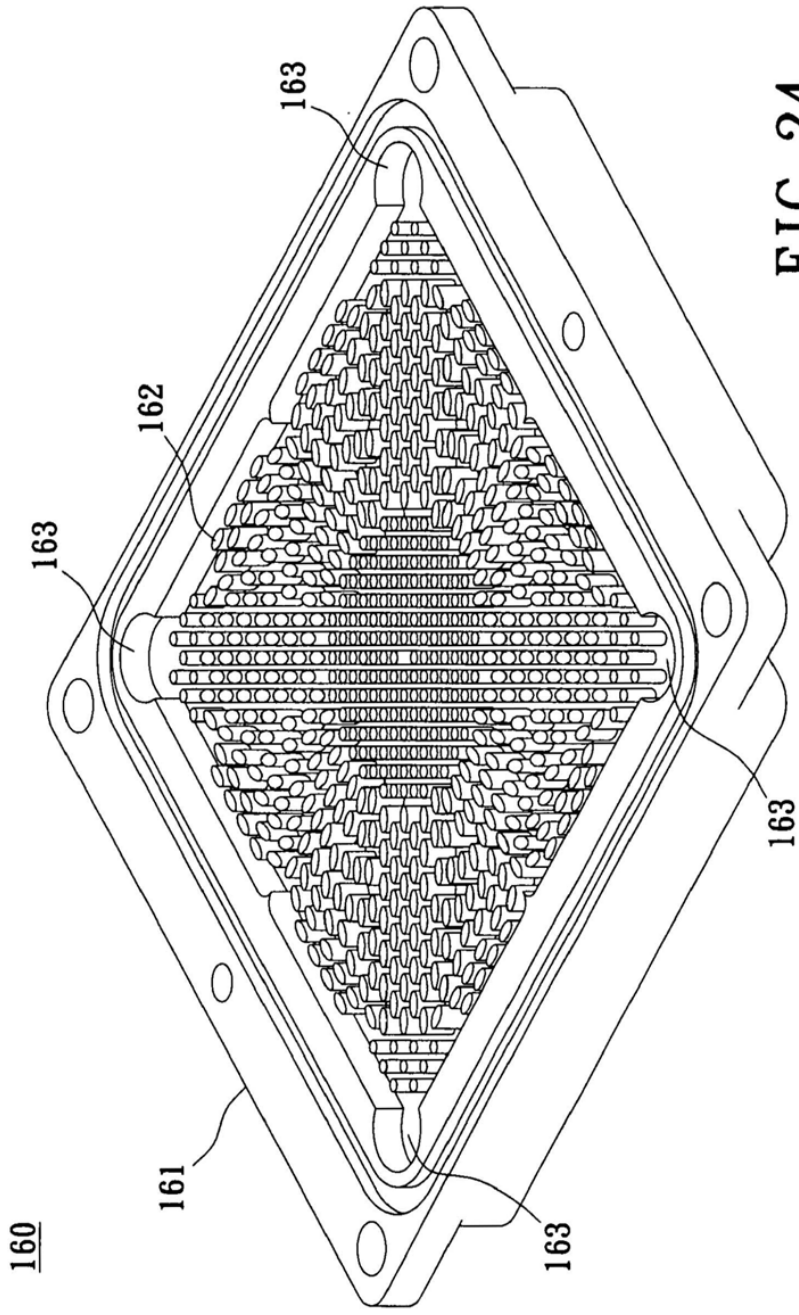
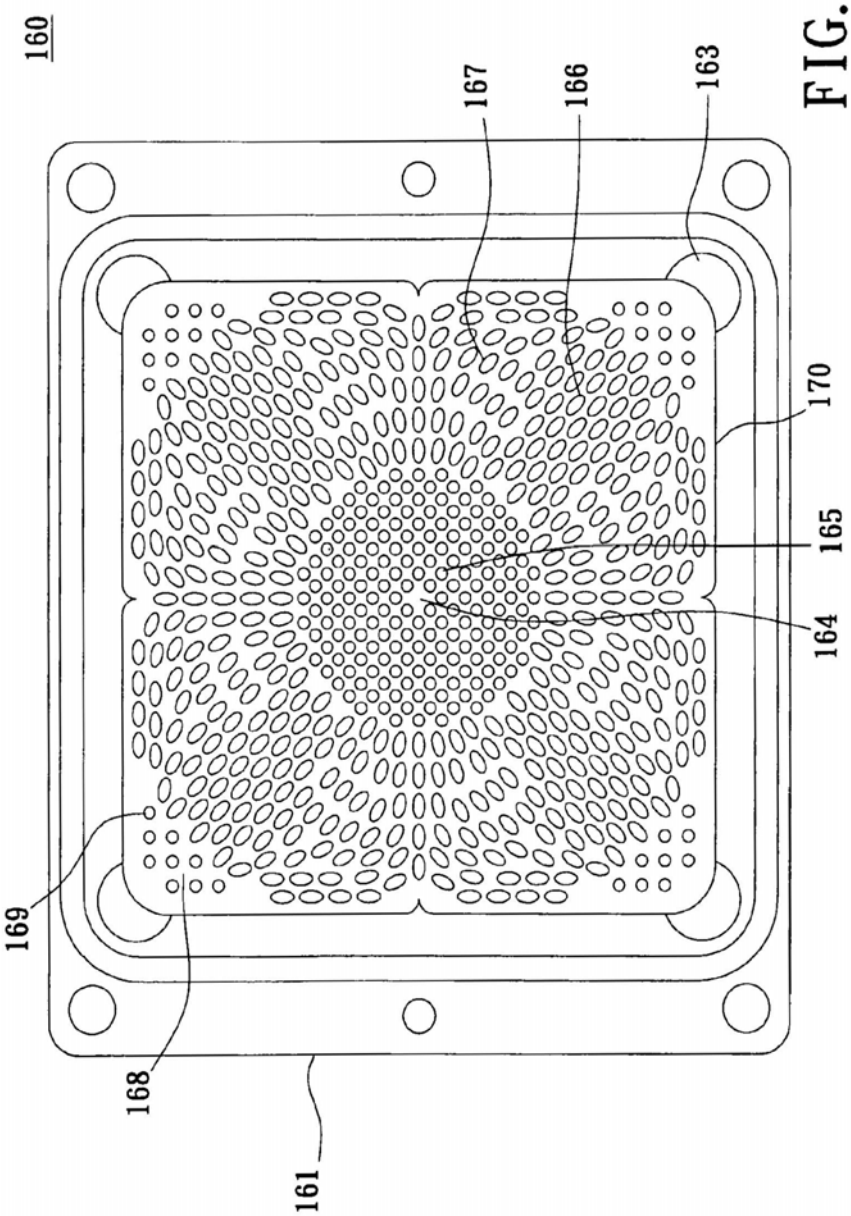


FIG. 24

SEC et al. v. MRI
SEC Exhibit 1006.0032
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FIG. 25



SEC et al. v. MRI
SEC Exhibit 1006.0033
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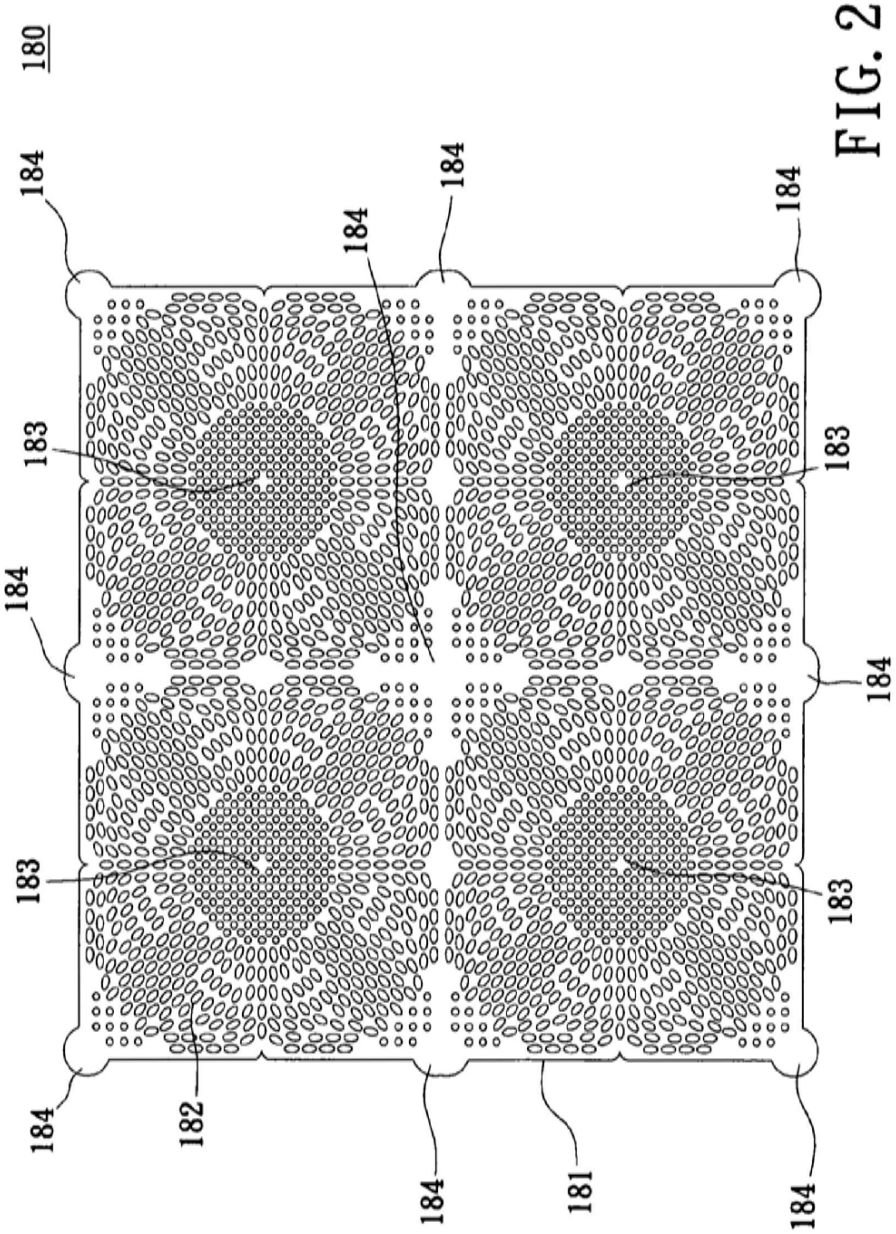


FIG. 26

NON-LINEAR FIN HEAT SINK

BACKGROUND OF THE INVENTION

[0001] 1. Field of the Invention

[0002] The present invention relates to methods and apparatuses for heat transfer. More particularly, the invention relates to optimized extended surfaces used for cooling electronic components and other objects whereas such methods and apparatuses involve heat transfer, such as the removal, absorption and/or dissipation of heat.

[0003] 2. Description of the Related Art

[0004] A "heat sink" (alternatively spelled "heatsink") is a device used for removing, absorbing and/or dissipating heat from a thermal system. Generally speaking, conventional heat sinks are founded on well known physical principles pertaining to heat transference. Heat transference concerns the transfer of heat (thermal energy) via conduction, convection, radiation or some combination thereof. In general, heat transfer involves the movement of heat from one body (solid, liquid, gas or some combination thereof to another body (solid, liquid, gas or some combination thereof). In the present invention the term "heatsink" may also apply to heat exchangers, radiators, air and liquid-cooled coldplates, and other devices through which heat is transferred.

[0005] The term "conduction" (or "heat conduction" or "thermal conduction") refers to the transmission of heat via (through) a medium, without movement of the medium itself, and normally from a region of higher temperature to a region of lower temperature. "Convection" (or "heat convection" or "thermal convection") is distinguishable from conduction and refers to the transport of heat by a moving fluid which is in contact with a heated body. According to convection, heat is transferred, by movement of the fluid itself, from one part of a fluid to another part of the fluid. "Radiation" (or "heat radiation" or "thermal radiation") refers to the emission and propagation of waves or particles of heat. The three heat transference mechanisms (conduction, convection and radiation) can be described by the relationships briefly discussed immediately hereinbelow.

[0006] Conductive heat transfer, which is based upon the ability of a solid material to conduct heat therethrough, is expressed by the equation $q=kA_c\Delta T/L$, wherein: q=rate of heat transfer (typically expressed in Watts) from a higher temperature region to a lower temperature region; k=thermal conductivity (W/m K), which is a characteristic of the material composition; Ac=cross-sectional area (m^2) of the material (perpendicular to the direction of heat flow; ΔT=temperature difference (° C.), which is the amount of temperature drop between the higher temperature region and the lower temperature region; and L=length (m) of the thermal path through which the heat is to flow.

[0007] Convective heat transfer, which is based upon the ability of a fluid to transfer heat energy through intimate contact with a solid surface, is expressed by the equation $q=h_cA_s\Delta T$, wherein: h_c=fluid convection coefficient (W/m^2 K), wherein h_c is determined by factors including the fluid's composition, temperature, velocity, and turbulence; and, A_s=surface area (m^2) which is in contact with the fluid.

[0008] Radiative heat transfer, which is based upon the ability of a solid material to emit or absorb energy waves or particles from a solid surface to fluid molecules or to different temperature solid surfaces, is expressed by the equation $q=A_s\in\sigma(T_s^4-T_a^4)$, wherein: $\in$=dimensionless emissivity coefficient of a solid surface, which is a characteristic of the material surface; σ=Stefan–Boltzmann constant; A_s=surface area (m^2) which radiates heat; T_s=absolute temperature of the surface (K); and, T_a absolute temperature of the surrounding environment (K).

[0009] It is theoretically understood that, regardless of the heat transfer mechanism, heat transfer rate q can be increased by increasing one or more of the numerator factors on the right side of the equation.

[0010] In current practical contexts, heat sinks, coldplates and heat exchangers are generally designed with a view toward furthering the conductive properties of the heat sink by augmenting the thermal conductivity k, for conduction; the surface area, A_s, and heat transfer coefficient hc, for convection. In this regard, according to conventional practice, a heat sink structure is made of a highly thermally conductive solid material, thereby maximizing the conductivity k characteristic of the heat sink; an extended surface comprising a plurality of manufacturable fins or pins, thereby maximizing the surface area A_s; and a geometric shape in contact with the fluid medium, thereby maximizing the heat transfer coefficient h_c.

[0011] Following conventional design practice, the heat sink structure tends to be rendered large (e.g., bulky or voluminous), therefore heat sinks are often rated by a heat transfer efficiency, or thermal resistance θ, found by dividing the ΔT, temperature rise of the heat source by the power input, i.e., ° C./W, whereby a lower value for thermal resistance θ, equates to a more efficient design.

[0012] As surface area and volume is increased, ancillary issues such as flow resistance and mass must be minimized. In order to gauge these ancillary effects on the efficiency of a heat sink, pumping power P_p, (measured in W), and mass M, (in kg) can be weighted and added to the efficiency equation resulting in $\eta=\Delta TP_pM/W$. Flow resistance can be particularly important because this resistance increases at the square power of coolant velocity. High flow resistance may require larger pumps or fans to generate additional pumping power, which may also require additional cooling capability.

[0013] Due to manufacturing costs, optimized heat sinks are usually limited to a linear array of identical fins having fixed spacing, which are intended to increase the surface area available for heat transfer and increase the heat transfer coefficient. These heat sinks are further compromised by containing simple fin shapes such as squares or rectangles, and occasionally round pins.

[0014] Several factors combine to reduce the effectiveness of these conventional heat sinks. One of the most common problems is that the heat absorbed by a coolant media results in a higher temperature media. Due to the temperature rise of the coolant and because a passive heat sink can not cool a heat source below the temperature of the coolant, the temperature of the last device in a row of equally powered components will be hotter than the upstream components. The temperature rise of the coolant is found by $\Delta T=q/\dot{m}c_p$, where $\dot{m}$=mass flow rate (kg/s) and c_p=specific heat (J/kg K) of the coolant media. This effect can also greatly change the coolant properties. Therefore, a linear fin array which is optimized for a specific inlet coolant temperature will not provide optimum heat transfer for the coolant after heat is absorbed. An extreme aspect of coolant media property change is in high heat flux applications whereby a saturated liquid enters a heat exchanger, becomes a two-phase flow through nucleate boiling, and subsequent vapor flow.

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[0015] In addition, heat is not usually spread evenly across the heat input surface of the heat sink. Common practice is to have a plurality of small heat sources share a common heat sink. In such cases, a linear array of fin protrusions will require the same amount of pumping power to flow through the unheated regions as the heated regions.

[0016] Thus, there are potential problems associated with conventional approaches to effectuating heat sink cooling of an entity behaving at a high power density. Firstly, prior art manufacturing approaches result in an array of fin protrusions that are more optimized for cost and not for heat transfer. Secondly, a low-cost prior art fin array, consisting of identical fins with identical spacing, will waste pumping power on unheated regions, usually resulting in the need for larger fans or pumps. Thirdly, prior art fin arrays have no provision to account for the temperature rise of a coolant media or the changes in physical properties of the coolant, resulting in decreased efficiency. Fourthly, prior art heat sinks are often grossly overweight, due to the limitations of the manufacturing approach.

[0017] Of interest in the art are several United States patents, each of which is hereby incorporated herein by reference Klein et al. U.S. Pat. No. 4,151,548 issued Apr. 24, 1979 teaches the use of square or diagonal cross-section pegs in a fluid flow whereby turbulence is created to enhance cooling. Klein also teaches that opposing inlet and outlet ports cause a higher velocity between the ports. Klein does not teach the use of efficient structures or the role of flow resistance.

[0018] Pellant et al. U.S. Pat. No. 4,188,996 issued Feb. 19, 1980 describes a device that contains a plurality of spaced parallel channels. The channels being divided by studs spaced longitudinally in an effort to promote more fluid turbulence. Pellant does not teach the use of efficient structures or the role of flow resistance.

[0019] Iversen U.S. Pat. No. 4,712,609 issued Dec. 18, 1987 discloses a roughened heat exchanger surface with a coolant flow heated to boiling and producing pressure gradients to remove nucleate bubbles. Although Iversen teaches that low flow resistance is important, Iversen does not teach, and makes no provision for the fact that the optimum heat transfer surface for liquid flow is very different than the optimum for two-phase and gaseous flow.

[0020] Steffen et al. U.S. Pat. No. 4,997,034 issued Mar. 5, 1991 teaches a heat transfer surface consisting of diamond-shaped protrusions on a pie-shaped plate and recognition of manufacturing ease and flow resistance. Steffen does not teach that different aspect ratios will produce different heat transfer and flow resistance results, nor does Steffen teach the use of mixed shapes and heights of protrusions.

[0021] Wolgemuth et al. U.S. Pat. No. 5,453,911 issued Sep. 26, 1995 discloses the use of nozzles to cause impingement of a coolant onto the baseplate of an insulated gate bipolar transistor (IGBT) or silicon-controlled rectifier (SCR), and deflectors to cause greater a greater heat transfer coefficient at hot spots. Wolgemuth does not teach the importance of flow resistance, or that gross changes in flow direction and velocity can have a very negative impact on flow resistance, nor does Wolgemuth disclose the use of shaped protrusions to efficiently cool hot spots.

[0022] Romero et al. U.S. Pat. No. 5,915,463 issued Jun. 29, 1999 instructs the use of an optimized fin array to cool discrete components and a method of manufacture. Romero asserts that the fin surfaces perpendicular to coolant flow do not significantly contribute to heat transfer, directly contradicting a large body of published literature.

[0023] Frey et al. U.S. Pat. No. 5,978,220 issued Nov. 2, 1999 discloses the use of fusion bonded heat sinks to cool IGBT modules, whereby the heat sink pins are circular, perpendicular to coolant flow, and in a hexagonal pattern. Although Frey teaches that thermal resistance can be optimized by varying pin diameter and spacing, Frey does not disclose the detrimental effect of flow resistance in that optimization or a method to counter heat absorption by the coolant.

[0024] Becker et al. U.S. Pat. No. 6,039,114 issued Mar. 21, 2000 instructs the use of a cooling body consisting of protruding lugs. Becker teaches that the volume of the lugs is greater than the volume of the flow channels thereby producing homogeneous flow resistance. Becker does not disclose how the shape or pattern of said lugs can be optimized to reduce said flow resistance, or how the geometric shape of the cooling body may be changed to cool local regions of greater heat flux.

[0025] Cannell et al. U.S. Pat. No. 6,729,383 issued May. 4, 2004 teaches a heat sink with non-thermally conductive fins, heat transfer occurring on the heat sink plate. The pins serve to promote fluid turbulence. Cannell teaches that a disparity among pins is acceptable, but that configurational regularity promotes uniformity of heat transfer. Cannell does not teach that flow resistance is an important variable or that certain shapes are more efficient than other shapes. Although Cannell discloses a large number of embodiments there is no rational for using one embodiment over another.

[0026] Rinehart et al. U.S. Pat. No. 7,173,823 issued Feb. 6, 2007 discloses a fluid-cooled assembly wherein lies a heat sink. Rinehart teaches that the cooling pins at the fluid inlet may be of a smaller diameter than at the fluid outlet because of heat absorption by the fluid, but does not disclose that other shapes and spacing of fins are more effective, or that shortening the pins can offer less flow resistance while simultaneously increasing fin efficiency.

[0027] Referring to FIG. **1***a* thru **1***d*, of a prior art configuration, a round pin heat sink **10** is shown. Prior art pins **11** are round and attached to a prior art base **12**. Referring now to FIG. **1***b*, prior art round pins **11** are arranged as such in prior art longitudinal rows **13**, parallel to the flow. Prior art longitudinal rows **13** are in staggered relationship with each other so that pins **11** in alternating longitudinal rows **13** are transversely (columnarly) **14** aligned as show. Referring now to FIG. **1***c*, prior art round pin **11** pattern shown in FIG. **1***b* can also be conceived to reveal a prior art heat sink **10** having prior art round pins **11** arranged in transverse columns **14** situated perpendicular to the flow so that alternating transverse are longitudinally (row-wise) aligned.

[0028] Referring to FIG. **1***b* and FIG. **1***c* of the prior art configuration, it is shown that prior art round pins **11** have equal prior art pin diameter **15**, equal longitudinal pin spacing **16**, and equal prior art transverse pin spacing **17**, whereas the terms "longitudinal" and "transverse" are in relation to the primary flow direction.

[0029] Referring now to FIG. **1***d* of the prior art round pin heat sink **10**, prior art upper flow boundary surface **18** can be added, therefore prior art upper flow boundary surface **18** and prior art base **12** act as boundary layers for flow. Prior art upper flow boundary surface **18** and prior art base **12** are both planar. The prior art round pins **11** uniformly extend an overall prior art pin height **19** from prior art base surface **12**. Every

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prior art round pin **11** has the same prior art pin height **19**. Depending on the specific requirements of prior art round pin heat sink **10**, pins **11** may simply support, touch, or not touch prior art upper flow boundary surface **18**. Since prior art upper flow boundary surface **18** and prior art base surface **19** are both planar, the distance therebetween is constant.

[0030] Referring to FIG. **2***a* thru **2***c*, a prior art square pin heat sink **20** is shown. Prior art square pins **21** have an equal or near-equal prior art square pin longitudinal dimension **25** and prior art square pin transverse dimension **26**. Prior art square pins **21** are attached to prior art base **12**. Referring now to FIG. **2***b*, prior art square pins **21** are arranged as such in longitudinal rows **13**, parallel to the flow. Rows **13** are in staggered relationship with each other so that pins **21** in alternating rows **13** are transversely (columnarly) **14** aligned. Prior art square pins pattern shown in FIG. **2***b* can also be conceived to reveal a prior art heat sink having prior art square pins **21** arranged in transverse columns **14** situated perpendicular to the flow so that alternating columns are longitudinally (row-wise) aligned (not shown).

[0031] Referring to FIG. **2***b* of the prior art configuration, it is shown that prior art square pins **21** have equal prior art longitudinal pin spacing **16**, and equal prior art transverse pin spacing **17**.

[0032] Referring now to FIG. **2***c* of the prior art square pin heat sink **10**, prior art square pin **11** is seen to cause a prior art discontinuity in the flow velocity **27**. Prior art flow discontinuity **27** has elements of recirculation and velocity stagnation resulting in diminished heat transfer efficiency and greater pressure drop. Prior art flow discontinuity **27** is common among most prior art heat sinks having in-line or staggered patterns of shapes. Prior art flow discontinuity **27** is most pronounced when the ratio of transverse pin dimension to longitudinal pin dimension is greater than or equal to unity. Consequently, prior art flow discontinuity **27** diminishes as the ratio of transverse pin dimension to longitudinal pin dimension diminishes, but pressure drop caused by pin surface skin friction increases. The size of prior art flow discontinuity **27** grows with increased flow turbulence, indicated by the Reynolds number, Re. Where $Re=\rho UD/\mu$, and ρ is the fluid density (kg/m^3), U is the fluid velocity (m/s), D is a characteristic dimension (m), and μ is the absolute viscosity ($N\ s/m^2$).

[0033] Referring to FIGS. **3***a* and **3***b*, a prior art plate fin heat sink **30** is shown. Prior art plate fins **31** are usually characterized by having a prior art longitudinal dimension **35** much larger than prior art transverse dimension **36**. Prior art plate fins **31** are attached to prior art base **12**. Referring now to FIG. **3***b*, prior art square pins **31** are arranged as such in longitudinal rows **13**, parallel to the flow. Rows **13** are in staggered relationship with each other so that fins **31** in alternating rows **13** are transversely (columnarly) **14** aligned. Prior art plate fin pattern shown in FIG. **3***b* can also be conceived to reveal a prior art heat sink having prior art plate fins **31** arranged in transverse columns **14** situated perpendicular to the flow so that alternating columns are longitudinally (row-wise) aligned (not shown).

[0034] Referring to FIG. **3***b* of the prior art configuration, it is shown that prior art plate fins **31** have equal prior art longitudinal fin spacing **16**, and equal prior art transverse fin spacing **17**.

[0035] Referring now to FIG. **3***c* of the prior art square pin heat sink **30**, prior art plate fin **31** is seen to cause a prior art discontinuity **38** in the flow velocity. Prior art flow discontinuity **38** has a velocity boundary layer height **39**, measured perpendicular to the primary flow direction, found by the equation $\delta=5x/\sqrt{Re}$, where δ is the velocity boundary layer height **39** at dimension x (m), x is the distance parallel to flow (m) from the initial point of the object, and Re_x is the Reynolds number at dimension x. It is understood that the height of velocity boundary layer **39** represents a near-stagnation zone along the surface of prior art plate fin **31** that causes the heat transfer coefficient to decrease as prior art longitudinal pin dimension **35** increases. The increasing thickness of the velocity boundary layer along the flow path acts to shroud downstream fins or pins in a "shadow" of lower velocity fluid thereby decreasing the heat transfer coefficient as the number of transverse columns **14** increases.

[0036] Referring to FIG. **4***a* thru **4***c*, a prior art two-diameter pin heat sink **40** is shown. Prior art round pins **11** are attached to a prior art base **12**. Prior art round pins **11** are grouped into two distinct regions, an inlet region **43** and an outlet region **44**, that are non-continuous and therefore non-interactive. Prior art inlet pins **41** within inlet region **43** have a smaller diameter than prior art outlet pins **42** within prior art outlet region **44**.

[0037] Referring now to FIG. **4***b*, prior art round pins **11** are arranged as such in longitudinal rows **13**, parallel to the flow. Rows **13** are in staggered relationship with each other so that pins **11** in alternating longitudinal rows **13** are transversely (columnarly) **14** aligned. Prior art inlet pins **41** have equal longitudinal spacing **45** within inlet region **43** and prior art outlet pins **42** have equal longitudinal pin spacing **46** within outlet region **44**. Prior art inlet pins **41** have equal transverse spacing **47** within inlet region **43** and prior art outlet pins **42** have equal transverse pin spacing **48** within outlet region **43**.

[0038] Referring now to FIG. **4***c* of the prior art two-diameter pin heat sink **40**, prior art upper surface **18** and prior art base **12** act as boundary layers for flow. Prior art upper surface **18** and prior art base **12** are both planar. The prior art round pins **11** uniformly extend an overall prior art pin height **19** from prior art base surface **12**. Every prior art round pin **11** has the same prior art pin height **19**. Depending on the specific requirements of prior art round pin heat sink **40**, pins **11** may simply support, touch, or not touch prior art upper surface **18**. Since prior art upper surface **18** and prior art base surface **19** are both planar, the distance therebetween is constant.

SUMMARY OF THE INVENTION

[0039] In view of the foregoing, it is an object of the present invention to provide heat sink method and apparatus which are capable of dissipating/removing heat from a device or other to-be-cooled object which is characterized by a high power density.

[0040] It is another object of this invention is to provide heat sink method and apparatus which provides cooling, for a to-be-cooled object (such as a module) having a baseplate, wherein the cooling is non-uniform over the surface area of the baseplate.

[0041] Another object of the present invention is to provide heat sink method and apparatus which are not large, cumbersome or heavy.

[0042] A further object of this invention is to provide heat sink method and apparatus in which extended surface protrusions are optimally shaped in recognition of convective heat transfer, conductive heat transfer, and flow resistance.

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[0043] Another object of the present invention is to provide heat sink method and apparatus which offsets the temperature rise of a coolant media and provide enhanced cooling for the local coolant temperature.

[0044] A further object of this invention is to provide heat sink method and apparatus which delivers optimized cooling efficiency per the local physical properties of the coolant media.

[0045] The present invention provides a heat sink for cooling an object, and a methodology for accomplishing same. The inventive heat sink is capable of being used in association with a fluid (liquid or gas) for effectuating cooling. Either liquid coolant, gas coolant or a combination two-phase flow can be used in inventive practice.

[0046] The present invention further features turbulence enhancement of the coolant stream by a pin array through which the coolant stream passes. According to many embodiments, this invention additionally features a non-linear shape, spacing, and height pattern to provide optimal cooling while simultaneously reducing volume and flow resistance.

[0047] In accordance with many embodiments of the present invention, a heat sink device for utilization in association with fluid for cooling an object comprises a heat transfer structure which includes a foundation section, plural protrusions, side surfaces, and a lid surface. The foundation section has an upper surface. The protrusions are situated on the upper surface. The foundation bottom surface is adaptable to engagement with a heat source. The fluid streams approximately longitudinally with respect to the upper surface and with respect to the object. According to typical inventive practice, the structure is adaptable to such engagement and association wherein at least one protrusion affects the streaming of the fluid—more typically, wherein plural protrusions, which are some or all of the protrusions, affect the streaming of the fluid.

[0048] In another embodiment of the present invention a heat sink device for utilization in association with fluid for cooling an object comprises a structure which includes a foundation section, plural protrusions, side surfaces, and a lid surface. Wherein protrusions are situated on both the upper surface of the foundation and the lower surface of the lid. The foundation bottom surface is adaptable to engagement with a heat source. The fluid streams approximately longitudinally with respect to the foundation upper surface, the lid lower surface, and the object. Accordingly, the pins on the upper surface and the lower surface may have different shapes spacing, and heights, that when assembled, produce multiple local flowfields within the heat transfer structure.

[0049] The inventive cooling apparatus is for application to any body—for example, an electronic circuitry device or other electronic component.

[0050] The inventive fluid-cooling heat sink apparatus typically comprises fluidity means (e.g., a fluid generation system) and a member. The subject body has a body surface portion. The member has a member surface portion and a plurality of pins projecting therefrom. According to frequent inventive practice the pins are approximately parallel; however, such parallelness is not required in accordance with the present invention. Each pin has a pin end surface portion opposite the member surface portion. The fluidity means includes means emissive of a fluid which is flowable along at least a part of the member surface portion so as to be contiguous at least a part of the body surface portion when at least a part of the body surface portion communicates with at least some of the pin end surface portions. Typically, the pins are arranged and configured in such manner as to be capable of increasing the turbulence of the fluid which passes between the member surface portion and the body surface portion.

[0051] Many inventive embodiments provide a method for cooling an entity such as an electronic component. The inventive method comprises the following steps: (a) providing a device having a device surface area and plural members which jut from the device surface area, the members having corresponding extremities opposite the device surface area; (b) associating the entity with the device, the entity having an entity surface area, the associating including placing the entity surface area in contact with at least some of the extremities; and (c) discharging fluid between the device surface area and the entity surface area so as to be disturbed by at least some of the members.

[0052] This invention meets most military and commercial requirements for dissipating/removing heat. The inventive heat sink: is capable of dissipating heat from a single or multiple high power density devices; can provide uniform or localized cooling over a baseplate surface area; is highly efficient in terms of mass, total volume, pumping power, and thermal resistance; and, carries relatively low manufacture and assembly costs.

[0053] The terms "pin" and "fin," in relation to the present invention, are used somewhat interchangeably. The term "pin" is usually applied to an extended surface protrusion of any height having roughly equivalent dimensions parallel and perpendicular to the general coolant flow direction. The term "fin" usually refers to an extended surface protrusion of any height having a greater dimension parallel to the general flow than perpendicular to the general flow. Hereinafter, "fin" is used to present the structure inside the heat sink.

[0054] In accordance with many embodiments of the present invention, the protrusions may be made of a thermally conductive material such as metal, thereby adding surface area and complementing heat convection by the working fluid with heat conduction by the fins.

[0055] According to inventive embodiments which thus implement thermally nonconductive fins, there is no significant or appreciable thermal conductivity; all or practically all of the heat which is removed from the heat source is removed via convection, wherein the cooling fluid comes into direct contact with a surface or surface portion of the heat source object. A thermally nonconductive material will generally be a nonmetallic material.

[0056] For instance, in inventive applications involving a module having a dielectric (e.g., ceramic) baseplate, the entirety of the heat is removed through the baseplate by the working fluid (e.g., water or air). The invention's fins serve as mechanical support for the ceramic baseplate and to enhance the turbulent flow of the working fluid; the turbulent flow increases the heat-removal effectiveness of the working fluid. The present invention not only provides support for the baseplate to prevent breakage, but also cools the baseplate.

[0057] It should be understood that, according to this invention, the fins do not necessarily project from the heat sink's base section. An inventive feature is that the fins may be interposed between the heat source and the heat sink surface. The heat sink surface bounds the working fluid flow on one side, and a heat source object surface bounds the working fluid flow on the opposite side. In inventive practice, the fins can project from either (i) a base which is part of a module for

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holding an electronic component, or (ii) a base section which is part of the heat sink device, this base section itself representing a sort of "base plate."

[0058] In accordance with many embodiments of the present invention, the terms "heat source" and "coolant media" can be replaced with the terms "cold source" and heated media". The invention can thus operate in either direction of heat flow, i.e. heat source to coolant media or heated media to cold source.

[0059] It should be understood that, according to this invention, the protrusions may be made of a thermally conductive material such as metal, thereby adding surface area and complementing heat convection by the working fluid with heat conduction by the fins.

[0060] The invention can thus operate regardless of which of two opposing substrates the fins project from, viz., an object surface (e.g., a "modular baseplate surface") or a heat sink surface (e.g., a "heatsink baseplate surface"). Therefore, according to many embodiments, a cooling assembly may comprise a modular baseplate, a heatsink base, plural fins and a fluid. The fins located between the two surfaces. The fluid is disposed between the modular baseplate and the heatsink base so as to be disrupted by at least some of the fins. Such inventive arrangements can prove especially propitious for applications involving high heat fluxes, wherein the modular baseplate (and perhaps the rest of the module, as well) is made of a dielectric material, e.g., a nonmetallic material such as ceramic, and thereby affords electrical isolation to the electronic component which is housed by the module.

[0061] Further is should be understood that the invention structure applies to different physical geometries. For example, a multi-sided heat transfer structure wherein some sides transfer heat into the structure and other sides transfer heat out of the structure.

[0062] Other objects, advantages and features of this invention will become apparent from the following detailed description of the invention when considered in conjunction with the accompanying drawings. It is to be understood that both the foregoing general description and the following detailed description are exemplary, and are intended to provide further explanation of the invention as claimed. Other advantages and features of the invention will be apparent from the following description, drawings and claims.

BRIEF DESCRIPTION OF THE DRAWINGS

[0063] The various objects and advantages of the present invention will be more readily understood from the following detailed description when read in conjunction with the appended drawings, in which:

[0064] FIG. **1***a* is an isometric view of a prior art heat sink device, wherein the extended surface is in the form of a linear array of identical round fins, having identical spacing, in a staggered pattern.

[0065] FIG. **1***b* is a top plan view of the prior art configuration shown in FIG. **1***a*, wherein the extended surface is in the form of a linear array of identical round fins, having identical spacing, in a staggered pattern.

[0066] FIG. **1***c* is a top plan view of a prior art configuration wherein the extended surface is in the form of a linear array of identical round fins, having identical spacing, in an in-line pattern.

[0067] FIG. **1***d* is a side elevation view of the prior art configuration shown in FIG. **1***c*.

[0068] FIG. **2***a* is an isometric view of a prior art heat sink device, wherein the extended surface is in the form of a linear array of identical square fins, having identical spacing, in a staggered pattern.

[0069] FIG. **2***b* is a top plan view of the prior art configuration shown in FIG. **2***a*, wherein the extended surface is in the form of a linear array of identical square fins, having identical spacing, in a staggered pattern.

[0070] FIG. **2***c* is a top plan close-up view of the prior art configuration shown in FIG. **2***b*, wherein the extended surface is in the form of a linear array of identical square fins, having identical spacing, in a staggered pattern. Vortex flow discontinuities that are characteristic of this fin pattern are shown.

[0071] FIG. **3***a* is an isometric view of a prior art heat sink device, wherein the extended surface is in the form of a linear array of plate-shaped fins, having identical spacing.

[0072] FIG. **3***b* is a top plan view of the prior art configuration shown in FIG. **3***a*, wherein the extended surface is in the form of a linear array of plate-shaped fins, having identical spacing.

[0073] FIG. **3***c* is a top plan close-up view of the prior art configuration shown in FIG. **3***b*, wherein the extended surface is in the form of a linear array of plate-shaped fins, having identical spacing. Boundary layer flow discontinuities that are characteristic of this fin pattern are shown

[0074] FIG. **4***a* is an isometric view of a prior art heat sink device, wherein the extended surface is in the form of a linear array of thin round fins at the coolant inlet, and thick round fins at the coolant outlet, all having identical spacing.

[0075] FIG. **4***b* is a top plan view of the prior art configuration shown in FIG. **4***a*, wherein the extended surface is in the form of a linear array of thin round fins at the coolant inlet, and thick round fins at the coolant outlet, all having identical spacing.

[0076] FIG. **4***c* is a side elevation view of the prior art configuration shown in FIG. **4***a*.

[0077] FIG. **5** is an isometric view of the preferred embodiment of the present heat sink invention, illustrating the novel non-linear fin shapes, fin heights, and fin spacings.

[0078] FIG. **6** is a top plan close-up view of the preferred embodiment of the present heat sink invention shown in FIG. **5**.

[0079] FIG. **7** is a side elevation view of the heat sink preferred embodiment shown in FIG. **6**.

[0080] FIG. **8** is a top plan view of the present invention showing the present heat sink invention.

[0081] FIG. **9** is an isometric view of the fins of the heat sink of the present invention.

[0082] FIG. **10** is a top plan close-up view of an alternate embodiment of the leading edge of the present invention showing an entrance channel optimized for laminar flow.

[0083] FIG. **11** is a top plan close-up view of an alternate embodiment of the trailing edge of the present invention showing an entrance channel optimized for turbulent flow.

[0084] FIG. **12** is a top plan view of an alternate embodiment of the present invention showing a nonlinear fin structure extending the length of the heat sink.

[0085] FIG. **13** is a top plan close-up view of an alternate embodiment of the present invention showing a nonlinear fin structure extending at the trailing edge of the heat sink.

[0086] FIG. **14** is a top plan view of an alternate embodiment of the present invention showing flow diverters and a plurality of heat sources and discrete fin arrays.

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[0087] FIG. **15** is a side view of the alternate embodiment of FIG. **15** showing placement of heat sources.

[0088] FIG. **16***a* is a top view of the present invention showing locations of cross sectional views a-a, b-b, c-c, and d-d.

[0089] FIG. **16***b* is a cross sectional front view of the present invention along plane a-a of FIG. **18***a* at a distance of roughly ⅛ along the flow length showing fins having a conical profile.

[0090] FIG. **16***c* is a cross sectional front view of the present invention along plane b-b of FIG. **18***a* at a distance of roughly ½ along the flow length showing fins having a truncated concave hyperbolic profile.

[0091] FIG. **16***d* is a cross sectional front view of the present invention along plane c-c of FIG. **18***a* at a distance of roughly ¾ along the flow length showing fins having a truncated concave parabolic profile.

[0092] FIG. **16***e* is a cross sectional front view of the present invention along plane d-d of FIG. **18***a* at a distance of roughly ⅞ along the flow length showing fins having a cylindrical profile.

[0093] FIG. **16***f* is a cross sectional front view of the present invention along plane e-e of FIG. **18***a* at a distance of roughly 15/16 along the flow length showing fins having a conical profile.

[0094] FIG. **17** is a top plan view showing a varying configuration of curved airfoil-shaped fins.

[0095] FIG. **18***a* is a top plan view of a semi-staggered fin array showing an undulating flow path.

[0096] FIG. **18***b* is a top plan view of the semi-staggered fin array of FIG. **20***a* rotated 90o showing an orderly flow path.

[0097] FIG. **19** is a top plan view of a fin configuration having a pattern that varies along the longitudinal and transverse directions of the fin array from large round fins to highly elliptical fins.

[0098] FIG. **20***a* shows a front view of two surfaces with varying fin thickness and varying spaces between adjacent fins.

[0099] FIG. **20***b* shows a front view of the two surfaces of FIG. **22***a* attached to form a flow channel depicting the resulting interspaced fins producing varying spaces between fins.

[0100] FIG. **21** is an isometric cutaway view of an embodiment of the present invention configured as a shell-tube heat exchanger.

[0101] FIG. **22** is a top plan view of an embodiment of the present invention showing fins used to simultaneously channel fluid and affect heat transfer.

[0102] FIG. **23** is an isometric exploded view of an embodiment of the current invention used in a power conversion module assembly.

[0103] FIG. **24** is an isometric view of an embodiment of the present invention using a central imfinging jet and four outlet ports.

[0104] FIG. **25** is a top plan view of the embodiment of the invention shown in FIG. **26** having a central imfinging jet and four outlet ports.

[0105] FIG. **26** is a top plan view of an embodiment of the present invention showing multiple inlet and outlet ports.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

[0106] Referring to FIG. **5**, a non-linear fin heat sink **50** is shown. The fins **51** are cross-sectionally shaped as ellipses and attached to a heat sink base **52**.

[0107] Referring now to FIG. **6**, each elliptical fin **51** has a cross-sectional fin longitudinal dimension **53** and a cross-sectional fin transverse dimension **54**. Longitudinal fin spacing (distance between two consecutive fins in the longitudinal direction, i.e., within a given row **55**) is represented as **56**, and the transverse fin spacing (distance between two consecutive rows **55** or fins **51** in the transverse direction) is represented as **57**. Longitudinal rows **55** are in staggered relationship with each other so that fins **51** in alternating longitudinal rows **55** are transversely (columnarly) **58** aligned.

[0108] Referring now to FIG. **7** of the present invention **50**, upper lid **59** and base **52** act as boundary layers for flow. Upper lid **59** and base **52** are both planar. Elliptical fins **51** extend an overall fin height **60** from upper surface of the base **52**. Depending on the specific requirements of the present invention **50**, fins **51** may simply support, touch, or not touch or be fixedly joined to foundation surface of the upper lid **59**. Since upper lid **59** and base **52** are both planar, the distance therebetween is constant.

[0109] A novel feature of the invention is that for each fin **51**, longitudinal fin dimension **53**, transverse fin dimension **54**, and fin height **60**, are optimized for the local flow field. FIG. **7** shows that the entrance fin height **61** at the leading edge **62** of the non-linear heat sink is less than at the exit fin height **63** at the trailing edge **64** of the heat sink, even though the distance between upper lid surface **59** and base surface **52** is constant. The initial height of fins **61** at leading edge **62** is intended to cause a displacement trip, whereby the transition from laminar flow to turbulent flow is hastened, causing a higher heat transfer coefficient than would otherwise occur. Initial height of fins **61** must meet two requirements to initiate a displacement trip: $\in > 820\ \nu/U$ and $\in > 0.308\delta_d$. Where $\in$ is the height of the fin (m), ν is the kinematical viscosity (m^2/s), U is fluid velocity (m/s) and δ_d is the height of the displacement boundary layer (m) where $1.7208\ Re_x^{-0.5}$. After the initial fin height causes the displacement trip, the height of the downstream fins is based on the increase in the velocity boundary layer thickness when turbulent.

[0110] Referring now to FIG. **8**, a novel feature of the present invention is that fin transverse dimension **54** increases in relation to δ, the thickness of the boundary layer. In this manner, each subsequent fin will protrude through the velocity boundary layer, thereby achieving a greater exposure to high turbulence flow, resulting in higher heat transfer from each fin **51**.

[0111] Another novel feature is that fin aspect ratio, described as fin longitudinal dimension **53** divided by fin transverse dimension **54**, progresses along a log curve from high aspect ratio ellipses at the leading edge of the heat sink **62** to low aspect ratio ellipses terminating at a distance approximately ⅞ of the total heat sink length **65**. For the remaining roughly ⅛ of the heat sink length **66**, fins **51** progress linearly back to high aspect ratio ellipses.

[0112] It is know in the art that for fins having a specific volume and height, greater cross-sectional ellipse aspect ratios will yield greater surface area, a lower coefficient of drag, and a lower heat transfer coefficient. Therefore, since one of the objects of the invention is to minimize flow resistance and volume while simultaneously increasing the heat transfer coefficient, fins **51** change aspect ratio as the distance from the leading edge of the heat sink increases. Fins **51** toward the leading edge have higher aspect ratios because at this longitudinal dimension the fluid has not absorbed enough heat to require a high heat transfer coefficient at the expense

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of volume and flow resistance. As shown in FIG. **8** and FIG. **9**, the first six transverse rows of fins **67** have an aspect ratio of 6.0:1; the next six transverse columns of fins **68** have an aspect ratio of 5.5:1; the next five transverse columns of fins **69** have an aspect ratio of 5.0:1; the next four transverse columns of fins **70** have an aspect ratio of 4.5:1; the next four transverse columns of fins **71** have an aspect ratio of 4.0:1; the next three transverse columns of fins **72** have an aspect ratio of 3.5:1; the next three transverse columns of fins **73** have an aspect ratio of 3.0:1; the next two transverse columns of fins **74** have an aspect ratio of 2.5:1; the remaining two transverse columns of fins **75** in the first ⅞ of flow length **65** have an aspect ratio of 2.0:1.

[0113] Referring now to FIG. **8** and FIG. **9**, as the fluid absorbs heat and the temperature of the fluid rises, lower aspect ratio fins increase the heat transfer coefficient to achieve uniform cooling. Although a log progression to lower aspect ratios is beneficial, the last roughly ⅛ of the heat sink **66** may contribute a majority of the flow resistance. For this reason, at about ⅞ distance along the flow axis, the fins progress linearly to high aspect ratios again. Again referring to FIG. **8**, starting at the leading edge of the last ⅛ of heat sink length **66**, the first transverse rows of fins **76** has an aspect ratio of 3.0:1; the next transverse columns of fins **77** has an aspect ratio of 4.0:1; the next transverse columns of fins **78** has an aspect ratio of 5.0:1; and the last transverse columns of fins **79** has an aspect ratio of 6.0:1. Alternately, fins at the trailing edge may have a reduced height to achieve the same effect.

[0114] FIG. **10** and FIG. **11** depict another embodiment of the present invention which makes greater use of fin configuration variations. For this embodiment, the number of fins along the leading edge is diminished, lending a "U" shape to the heat sink entrance **80**. The exact distance along the flow axis that the diminished fin zone extends is a function of the flow profile. Flow in an enclosed channel in the laminar flow region, wherein Reynolds number is roughly <2,000 will benefit from a parabolic entrance region **81**. Flow in an enclosed channel in the turbulent flow region, wherein Reynolds number is roughly >2,000 will benefit from a hyperbolic entrance region **82**. It appears that the benefit and effect of entrance regions is more applicable to fluids having a greater ratio of kinematical viscosity to thermal diffusivity. This ratio, the Prandtl number is described as $Pr=c_p\mu/k$.

[0115] The methodology of the present invention is not limited to fin heat sinks. As shown in FIG. **12**, a plate fin heat sink has a number of fins **83** parallel to the flow axis and mounted to base **52**. The fin transverse dimension **84** increases in relation to δ, the thickness of the boundary layer as the flow distance **86** from the leading edge **85** increases. In this manner, as flow distance **86** from leading edge **85** increases, the fin surface will protrude through the velocity boundary layer, thereby achieving a greater exposure to high turbulence flow, resulting in higher heat transfer from each fin **83**.

[0116] Again, as shown in FIG. **13** the greatest transverse fin dimension **87** of fin **83** is achieved at roughly ⅞ the length **88** of the heat sink. As longitudinal dimension **88** increases past this point, during the last roughly ⅛ of flow length **90** transverse fin dimension **84** again decreases to the trailing edge **89** of the heat sink. Fins having this profile will achieve greater heat transfer coefficients than simple fixed-thickness plate fins.

[0117] FIG. **14** depicts base **52** showing a plurality of discrete fin patterns **91**. Using the teachings herein it is apparent that the invention can be scaled up or down, or replicated to produce equal improvement at discrete heat sources. Flow enters at leading edge **62** and is diverted by flow diverters **92** toward fin patterns **91**. Flow diverters **92** are shaped to provide minimal flow resistance, while allowing high velocity at fin patterns **91**. By utilizing optimized fin patterns **91** each of a plurality of heat sources, having a multitude of discrete power levels could be cooled equally if so required by the design application.

[0118] FIG. **15** shows a side view of the object of FIG. **14** through section A-A. Protruding from base **52** are fin patterns **91**. Fluid flow is contained by upper lid **59**. As fluid enters at leading edge **62** the fluid velocity is increased by base flow diverters **93** protruding from base **52** and lid flow diverters **95** protruding from upper lid **59**. Attached directly to base **52** and upper lid **59** are a plurality of heat sources **94** which are aligned with areas of higher velocity present within fin patterns **91**.

[0119] Upper lid **59** and base **52** are depicted to be non-planar, as shown in FIG. **16**. Upper lid **59** and base **52** can be inventively practiced so as to have any of a diversity of "topographies." The geometric configuration of an upper lid surface **59** can be entirely planar, entirely non-planar, or some combination thereof. Assuming a planar (flat) base **52**: If upper lid **59** is planar, then fin height **60** is constant; if the upper lid **59** is non-planar, and the base **52** thickness is variable, fin height **60** is not constant.

[0120] The geometry of upper lid **59** and base **52** can be characterized entirely by rectilinearity, entirely by curvilinearity, or by some combination thereof. FIG. **14** and FIG. **15** show both upper lid **59** and base **52** to be displaced from planar by lid flow diverter **95** and base flow diverter **93** and by flow diverters **92**. In inventive practice, any random or rigid geometry may be used for the foundation surface of upper lid **59** and the upper surface of the base **52**, such as, but not limited to, triangular, oval and sinusoidal.

[0121] Table. 1 shows the results of applying the teachings of the present invention when compared to a heat sink using the prior art teachings. Table. 1 lists the physical characteristics of a prior art fin heat sink and a heat sink of the current invention. Most notable are the decrease in thermal resistance, decrease in pumping power required, and decrease in mass and volume, embodied by the invention.

TABLE 1

Heat Sink Material	Copper	
Fluid Volumetric Flow Rate, q (l/min)	11	
Fluid Density, ρ (kg/m^3)	840	
Fluid Absolute Viscosity, μ (Ns/m^2)	0.01	
Fluid Thermal Conductivity, k (W/mK)	0.13	
Fluid Specific Heat, c_P (J/kgK)	2450	
Fluid Inlet Temperature, T_F (° C.)	75	
Ambient Air Environment, T_A (° C.)	75	
Baseplate L × W × D (cm)	17.6 × 6.5 × 0.3	
Power, Q (W)	3,600	
	Prior Art	Invention
Maximum Temperature, T_{MAX} (° C.)	161	140
Mean Baseplate Temperature, T_{MEAN} (° C.)	154	135
Pressure Drop, ΔP (kPa)	37	14
Heat Sink Volume, V (cm^3)	119	108

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TABLE 1-continued

Heat Sink Mass, M (kg)	0.54	0.45
Thermal Resistance, θ (° C./W)	0.024	0.018
Pumping Power, P_P (W)	6.69	2.6
Efficiency, η (1/(θP_PM))	11.5	47.5

[0122] It is emphasized that inventive practice is not limited to the specific geometric ratios represented herein in the drawings. The geometric modalities shown in these figures are intended herein to be "generic" in nature because dissimilar geometric motifs can manifest similar principles and concepts; in particular, different aspect ratio geometries and forms of the fins and/or the base section can be used to generate attributes of thermal performance. It will be apparent to the ordinarily skilled artisan who reads this disclosure that there are thematic commonalities among the geometric modalities specifically disclosed herein, and that many geometric modalities and ratios not specifically disclosed herein can be inventively practiced in accordance with such thematic commonalities and in accordance with other inventive principles disclosed herein.

[0123] The present invention admits of a diversity of embodiments. As elaborated upon hereinbelow, the inventive practitioner can vary one or more dimensional, configurational and/or geometric parameters, including but not limited to the following: (i) fin length and/or height; (ii) fin cross-sectional shape (e.g., elliptical versus circular versus square, etc.); (iii) fin distribution (e.g., non-staggered rows versus staggered rows, angularity of row staggering, even or parallel row orientations versus offset row orientations, angularity of row orientation offset, etc.); (iv) fin spacing (e.g., distances between various fins in various directions); (v) passage depth (e.g., distance between baseplate and upper lid); (vi) passage shape (e.g., relative dispositions of baseplate and upper lid surface, contour (three-dimensional shape) of baseplate, contour (three-dimensional shape) of heat sink base section, etc.); (vii) heat sink base section's outline (two-dimensional) shape; (viii) heat sink base section's transverse dimensions; (ix) fluid inlet configuration; and, (x) fluid outlet configuration.

[0124] Although elliptically-shaped fin **51** cross-sections (having longitudinal fin dimension **53** and transverse fin dimension **54**) are portrayed in the embodiment, it is readily appreciated by the ordinarily skilled reader of this disclosure that fin **51** cross-sections of any shape can be disposed either in angularly offset fashion (e.g., oblique with respect to a selected longitudinal line, such as an edge of the upper lid surface **59**) or in angularly non-offset fashion (e.g., parallel with respect to a selected longitudinal line, such as an edge of the upper lid surface **59**). The figures disclosed herein are merely exemplary insofar as generically demonstrating that inventive practice can use any combination of geometrical cross-sectional shapes, geometrical arrangements and angularities with respect to a longitude (e.g., angle theta can be any value greater than or equal to zero). In fact, the present invention encompasses a potentially infinite number of variations of cross-sectional shapes and locations of fins **51**.

[0125] It is again emphasized that any number or geometric arrangement of fins **51** can be used in inventive practice. With regard to the properties of staggeredness and uniformity (homogeneity), an inventive fin **51** array can be characterized by staggered uniformity, non-staggered uniformity, staggered nonuniformity or non-staggered non-uniformity. Further, any combination of two or more geometric fin **51** shapes can be used for a given fin **51** array.

[0126] Furthermore, the surface roughness of the flow cavity can be varied in accordance with the present invention. Irrespective of the essential geometry defined by upper lid **59** and base **52**, the detailed geometry defined by the surfaces can vary in terms of smoothness versus roughness. Not only the essential geometry of fins **51**, but also the detailed geometry of the boundary surfaces, can be selected so as to affect the flow of fluid in a desired fashion.

[0127] Surface roughness and porosity of fins **51** is an important factor in specific flows. In low viscosity fluids with a relatively high thermal conductivity, fin porosity can add greater surface area and serve to reduce pressure drop. In fluids having a boiling point near the expected operating conditions within non-linear heat sink **50** surface roughness will have a great effect on boiling incipience and wall superheat. In fact, it is a further object of the present invention to affect means for allowing a continuous control of boiling incipience and affectivity as a function of flow length.

[0128] It should be now obvious to those skilled in the art than in addition to local longitudinal and transverse dimensional and geometric alterations; fins may be similarly optimized along an axis measured parallel to the height of the fin. The effect of physical changes in fin cross section vs. distance measured from the fin base are well know in the art, but these characteristics can also be altered on an individual fin or fin region basis to affect local optimized flowfields and solid/fluid interaction. Referring now to FIG. **16***a* a top plan view of the present invention is shown. FIG. **16***b* shows a frontal cross section view of the fins at a distance of roughly ⅛ of the longitudinal distance of the heat sink base **52**. At this location along the longitudinal axis, the fins are of a conical shape **123**. The side walls of the fin progress from the large fin base **124** attached to base **52** to the fin tip **125**. This shape allows a low flow resistance and does not transfer heat very well, that is, the temperature of fin tip **125** is about the same temperature as the local flowable media. FIG. **16***c* shows a frontal cross section view of the fins at a distance of roughly ½ of the longitudinal distance of the heat sink base **52**. At this location along the longitudinal axis, the fins are of a concave hyperbolic shape **126**. The side walls of the fin progress along a concave hyperbolic curve from the fin base **124** attached to base **52** to the truncated fin tip **127**. This shape allows a low flow resistance but transfer heat better than the conical profile **123** of FIG. **16***b*. Truncated fin tip **127** allows a savings of material since the fin tip does not transfer much heat, and also allows the user the option of attaching truncated fin tip **127** to an upper lid wall **59** (not shown) and transferring some heat therethrough. FIG. **16***d* shows a frontal cross section view of the fins at a distance of roughly ¾ of the longitudinal distance of the heat sink base **52**. At this location along the longitudinal axis, the fins are of a concave parabolic shape **128**. The side walls of the fin progress along a concave parabolic curve from the fin base **124** attached to baseplate **52** to the truncated fin tip **127**. This shape allows a maximum heat transfer at a minimum of material usage and shows a high fin efficiency when 100% fin efficiency is described as a fin of the same surface area having infinite thermal conductance. The truncated fin tip **127** allows a savings of material since the fin tip does not transfer much heat, and also allows the user the option of attaching truncated fin tip **127** to an upper lid wall **59** (not shown) and transferring heat therethrough. FIG. **16***e* shows a frontal cross section

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view of the fins at a distance of roughly 7/8 of the longitudinal distance of the heat sink base **52**. At this location along the longitudinal axis, the fins are of a cylindrical profile **129**. The side walls of the fin progress along a straight line perpendicular from base **52**. This shape allows a maximum heat transfer. While fin efficiency is not as high as the concave parabolic profile of FIG. **16***d* heat transfer to the fin tip is maximized which allows the use of the fin tip to transfer heat to the coolant or to transfer heat to an upper lid wall **59** (not shown). FIG. **16***f* shows a frontal cross section view of the fins at a distance of roughly 15/16 of the longitudinal distance of the heat sink base **52**. At this location along the longitudinal axis, the fins are again of conical profile **123** which allows some heat transfer but is used primarily to transition the fluid coolant to the slower flow velocity downstream of the fin array and to minimize the turbulent velocity wake.

[0129] Referring now to FIG. **17**, airfoil-shaped fins **111** are know in the art to have exceptionally low skin flow resistance because of the termination of the laminar boundary layer, but these shapes are usually hindered by a low heat transfer coefficient. As taught herein and shown in FIG. **18**, these shapes can be formed into sections of varying curvature whereby flow is directed to the downstream fin in such a manner as to increase imfingement on one side of the fin surface **112** thereby increasing the heat transfer coefficient. The radius of curvature of the fins **113** can be reduced using an exponential function, just as the aspect ratio of the elliptical fins of the preferred embodiment are adjusted, as a function of the distance from the leading edge of the heat sink. Such improvements are particularly advantageous at very low Reynolds numbers. For high values of Reynolds number, the optimized airfoil shape will share geometric characteristics with a Sears-Haack body.

[0130] Again referring to FIG. **7**, FIG. **8**, and FIG. **17**, it will be seen to those knowledgeable in the art that the curvature of the fins and other characteristics of the geometries of individual fins can be altered to increase or reduce the local turbulence of the flowable media. Specifically, fin configurations having high longitudinal dimension to transverse dimension aspect ratios will have lower turbulence. Also, fin patterns that produce greater cross sectional flow area between fins will have higher turbulence. By varying the placement of these laminator and turbulator fins, a desired level of local heat transfer and local fluid mixing can be achieved.

[0131] In another embodiment of the invention, utilizing the fin pattern of the preferred embodiment, individual fins can be manufactured from different materials and combined with the geometric effects on coolant flow to control heat transfer or conversely individual fins may be made from more than one material. For example, in order to provide a more uniform base temperature, fins near the leading edge may be constructed of a material having a lower thermal conductivity such as aluminum while fins closer to the trailing edge may be constructed of a material having a higher thermal conductivity such as copper. Fins having different material characteristics may also be combined to produce other thermal effects. For example, the fins closer to a heat source may be manufactured from different materials to more closely match the coefficient of thermal expansion of the heat source. In one region of the heat sink, fins may be constructed of platinum while in another region of the heat sink, fins are constructed of beryllium. Although both materials have similar thermal conductivity, beryllium has almost 15 times the heat capacity, which may be useful in high power transient applications.

[0132] In FIGS. **18***a* and **18***b*, another embodiment of the invention is shown that has a specific arrangement of fins, learned from the teachings herein, that when rotated produce a different effect than the original orientation. A fluid flow enters semi-staggered fin array **114** on side A **115** and travels an undulating path **116** to side C **117**. When rotated 90 degrees, flow enters side D **118** of semi-staggered fin array **114** and the flow travels a more orderly path **119** toward side B **120**. Such an arrangement can yield a high heat transfer coefficient at higher flow resistance in the original orientation but will offer low flow resistance and low heat transfer when rotated. By way of example, the present invention may be rotated 180 degrees relative to fluid flow and impart a different set of characteristic flow patterns and effects. In the present invention, the specific novel fin pattern would produce an initial area of low flow resistance and heat transfer coefficient, followed by an area of higher flow resistance and higher heat transfer, followed by a very gradual transition to the original flowfield. Whereas the original orientation of the preferred embodiment produced a uniform base temperature, when rotated 180 degrees, the fin pattern will produce a much higher temperature at the trailing edge of the base than at the leading edge.

[0133] The effect of having a much higher temperature at the trailing edge can be used to benefit cooling applications that rely on a change in the coolant phase from liquid to gas. In this manner the base can be liquid cooled along the leading edge, and gas cooled at the trailing edge. One reason to impart this effect is for process control and assist chemical reactions. For example, the fins may be constructed of a porous material infused with a chemical, or simply coated with a chemical depending on the application. The application of liquid flow may produce an initial chemical reaction having a desired effect downstream of the leading edge. As the distance from the leading edge increases and the heat of the liquid increases exponentially (depending on the non-linear fin pattern), a secondary thermo-chemical reaction may occur. As the fluid changes phase, the incipience of gaseous nucleation may cause tertiary reactions to occur. One embodiment of the present invention may be used to facilitate vapor compression distillation of urine in a disposable canister. In addition to the specific thermo-chemical reactions caused by the example described, specific regions of a fin pattern may be coated with different chemicals which when combined produce an effect only in the presence of the flowable catalyst.

[0134] Depending on the chemical composition of a fin, the fin pattern may be varied to achieve a specific rate of chemical release into the flowable media which depends on the amount of turbulence and/or flow velocity. As shown in FIG. **19**, a base **52** has a region of large round fins that contain a higher core temperature and can be arranged to produce higher fluid velocity, which can be used to increase the rate of release of a chemical. Base **52** also transitions along the transverse axis to a region of highly elliptical fins **122** that produce lower core temperatures and lower velocities and therefore a lower heat transfer coefficient. Within a single disposable canister a number of chemical reactions can occur with or without the use of heat. Using the teachings of the present invention those knowledgeable in the art will therefore see a variety of applications in the fields of medicine, chemical processing, and military weaponry.

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[0135] In another embodiment, shown as FIG. **20***a*, fins **51** protrude not only from base **52** but also from upper lid **59**. As show, upper lid **59** has a plurality of protruding upper lid fins **130** as does base **52** have a plurality of base fins **51**. Base **52** and upper lid **59** both have a heat source **94** attached to the surface opposite base fins **51** and upper lid fins **130** respectively. As shown, upper lid fins **130** have a variable upper lid fin space **133** between adjacent upper lid fins **130** while base fins **51** have a variable base fin space **131** between adjacent base fins **51**. Also, base fin transverse dimension **54** and upper lid fin transverse dimension **132** may be varied to produce a desired effect. When base **52** and upper lid **59** are attached to form a fluid channel as shown in FIG. **20***b* base fins **51** fit into upper lid space between fins **133** and upper lid fins **130** fit into base space between fins **131**. Because space **133** between upper lid fins **130**, space **131** between base fins **51**, upper lid fin transverse dimension **132**, and base transverse fin dimension **54** are varied, accordingly the effect is to provide a number of large spaces between adjacent fins **134** and a number of small spaces between adjacent fins **135**. It is important to note that although heat transfer can be increased by affecting higher fluid velocity, manufacturing considerations often do not allow base fin space **131** or upper lid fin space **133** to be reduced to a value that would cause higher fluid velocities. The technique disclosed herein and in FIG. **20***a* and FIG. **20***b* allows larger fin space for easier manufacturing, yet can still be used to produce higher fluid velocity leading to higher heat transfer coefficients. This technique can also be used whether or not all fins are used for heat transfer. For example, if only base **52** has a heat source **94** properly designed upper lid fins **130** will still produce the effect of higher velocity when used to produce smaller space between adjacent fins **134** and **135**. It can be seen to those knowledgeable in the art that the description of varying transverse dimensions **54**, **131**, **132**, and **133** are by way of example and that other dimension such as longitudinal fin length, thickness, spacing, angle, etc., can also be varied to produce similar effects. It is also seen that although the example of FIG. **20***a* and FIG. **20***b* show base fins **51** and upper lid fins **130** touching upper lid **59** and base **52**, respectively, the fins may be attached, not attached, have a variable space, or may even be designed so that base fin **51** and upper lid fin **130** are aligned. Some number of fins may not be attached while other fins are attached to impart structural strength, or provide a desired clearance. Furthermore, it is seen that although FIGS. **20***a* and **20***b* depict opposing faces containing fins, fins may be contained on other walls or objects at angles to the fluid flow.

[0136] FIG. **21** shows another embodiment of the present invention configured as a shell-tube heat exchanger **144**. Shell tube **139** forms a fluid conduit through which cooling fluid **141** flows. The interior surface of shell tube **139** contains a plurality of internal shell tube fins **142** having geometric shapes locations and sizes using the teachings of the present invention. Shell tube **139** contains a heat exchanger tube **136** which has a plurality of exterior fins **137** and interior fins **138** having geometric shapes locations and sizes using the teachings of the present invention. Hot fluid **140** flows through heat exchanger tube **136**. In the embodiment shown in FIG. **21** internal shell tube fins **142** and exterior fins **137** are interspaced according to the descriptive teachings of FIG. **20***a* and FIG. **20***b*. In use, heat contained in hot fluid **140** is convected and conducted by interior fins **138** to heat exchanger tube **136**. Heat is conducted through heat exchanger tube **136** to exterior fins **137**. Heat is then convected and conducted from the external surface of heat exchanger tube **136** and exterior fins **137** to cooling fluid **141**. In the embodiment shown, shell-tube heat exchanger **144** is in an environment having a higher temperature than cooling fluid **141**. Therefore, both shell tube **139** and internal shell tube fins **142** are constructed of a material having low thermal conductivity. In an alternate embodiment having an environment temperature lower than cooling fluid **141** shell tube **139** and internal shell tube fins **142** would be constructed of a material having high thermal conductivity and shell tube **139** would also contain a plurality of external fins (not shown) located on shell tube external surface **143** to enhance heat transfer to the environment. It is understood that FIG. **21** and the related descriptions are only one of a number of possible variations on the basic scheme of heat exchange.

[0137] Referring now to FIG. **22**, another embodiment of the present invention is shown. Fluid enters the first fin array **145** at entrance location **146**. Fluid flows through first fin array **145** and enters a plenum/manifold **147**. Plenum/manifold **147** contains a number of manifold fins **148** and fluid passages **149**. Plenum/manifold fins **148** and fluid passages **149** are so configured as to direct fluid flow from first fin array **145** to a second fin array **150**. Using the teachings herein, plenum/manifold fins **148** simultaneously route fluid through the 180 degrees turn of plenum/manifold **147** to enter second fin array **150** at an optimum angle, maximize heat transfer and fluid mixing, and minimize flow resistance. It is a novel aspect of this embodiment to affect a bulk fluid directional change within plenum/manifold **147** while applying the teachings herein. It is important to note that the bulk fluid properties may be quite different at fluid entrance **146**, first fin array exit **151**, second fin array entrance **152**, second fin array exit location **153**, and even within plenum/manifold **147**. Using the teachings herein therefore, the specific geometric shape of fins located in different regions of first fin array **145**, plenum/manifold **147**, and second fin array **150** are different. It is noted that although FIG. **22** shows fluid flowing in series through a first and a second fin array, fin arrays may be configured as series/parallel paths with fins using the teachings herein the fluid entrance, fluid exit, and at points between.

[0138] It is reemphasized that the present invention can be practiced in association with any among a multiplicity of geometries. Any of the fin **51** array patterns illustrated in the drawings (and many others not specifically shown) can be inventively practiced regardless of the geometric nature (e.g., planar or non-planar) of the bounding surfaces.

[0139] In the previous figures, fins **51** are shown to be made part of base **52**, protruding from the upper surface of base **52**, toward and contacting the bottom surface of upper lid **59**. Base **52** is part of heat source **94**. However, inventive practice can provide for the fabrication of fins **51** as part of another separate heat sink baseplate which is then attached to heat source base **52**. Again however, those leaned in the art will realize that fins **51** can extend from surface of upper lid **59** and attach to base **52** or any other part of the structure as described for the embodiment of FIG. **20***a* and FIG. **20***b*.

[0140] Referring now to FIG. **23**, of one possible assembly containing the present invention, manifold **96** is a housing fairly representative of that used in commercial practice. Manifold **96** has at least one receiving space to house the non-linear fin heat sink **50**. Non-linear fin heat sink **50** includes a rectangular plate-like foundation base **52** and a plurality of turbulence-enhancing and heat transferring fins

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51 which project therefrom. Manifold 96 has upper lid 59. Each fin 51 is based at its fin root in base 52.

[0141] Manifold 96 further serves to channel the cooling fluid (liquid or gas) 97 through fins 51, thereby enhancing turbulent flow. Manifold 96 provides an incoming coolant port 101 and an outgoing coolant port 100 to which incoming coolant barb 98 and outgoing coolant barb 99 are respectively attached. Manifold 96 has an upper mounting surface 103 and a lower mounting surface 104 and at least one opening 102 sized for attachment of non-linear fin heat sink 50. Those skilled in the art will see that although FIG. 16 shows only upper mounting surface 103 being used for active mounting, lower mounting surface 104 can be used for the same function alone, or in addition to use of upper mounting surface 103. Upper mounting surface 103 has provisions for mounting the object (e.g., device) to be cooled, such as power conversion module 105 which holds one or more heat sources 94, shown in FIG. 15.

[0142] Power conversion module 105 includes module housing 107, which houses at least one heat source 94. Power conversion module 105 has a module baseplate 106 to which heat sources 94 are thermally attached.

[0143] As illustrated in FIG. 23, the edges of base 52 are attached to manifold 96 and power conversion module 105 is coupled with the manifold-heatsink assembly. A sealing element (e.g., gasket or O-ring, braze) 108 is provided to prevent coolant leakage. A layer of thermal interface material is provided between module baseplate 106 and heat sink base 52 to ensure effective thermal conductivity of heat from heat source 94 to heat sink base 52.

[0144] Although shown as flat surfaces, module baseplate 106, heat sink base 52, and any other heat transfer interfaces may have specific geometric surface patterns to aid the conduction of heat. By way of example such patterns as hierarchical nested channels (T. Brunschwiler, U. Kloter, H. Rothuizen, and B. Michel, "Hierarchically nested channels for fast squeezing interfaces with reduced thermal resistance", 21st IEEE SEMI-THERM Symposium, San Jose, Calif. 2000) can provide a marked reduction in interface resistance when used with flowable thermal interface materials.

[0145] When assembled, manifold 96 is contiguous with respect to non-linear fin heat sink 50, whereby base 52 and the tips of fins 51 are mounted in a highly thermally conductive leak-proof manner. Likewise heat sources 94 within module housing 107 of power conversion module 105 are attached in a highly thermally conductive manner to module baseplate 106, which is then mounted in a highly thermally conductive manner through thermal interface material 109 to upper mounting surface 103 of manifold 96 by attachment bolts 110. Once power conversion module 105 and the manifold-heatsink unit are joined, fins 51 protrude into the path of coolant fluid 97.

[0146] The fluid flow system will typically include fluid flow means (e.g., including fluid pumping means), fluid inlet means and fluid outlet means. In typical inventive practice, heat sink base 52, upper lid inside the manifold 59 and sidewalls of manifold 96 will define an outline shape (e.g., a rectangular shape) which provides a flow cavity.

[0147] As shown in FIG. 23, cooling fluid 97 is generated pursuant to a fluid system and is conveyed via incoming coolant fitting to incoming coolant port 101 to non-linear fin heat sink 50 to outgoing coolant port 100 to outgoing coolant fitting 99.

[0148] Energy appearing as waste heat emanates from heat sources 94 and passes through several layers of material having various degrees of electrical and thermal conductivity to module baseplate 106, through thermal interface material 109, through heat sink base 52, to fins 51 and convected and conducted to coolant fluid 97.

[0149] In inventive practice, components can be made from a wide variety of materials. In reference to the preferred embodiment, nonlinear fin heat sink 50 and module baseplate 106 are made from copper.

[0150] Referring now to FIG. 24, a non-linear impingement heat sink 160 is shown. Non-linear impingement heat sink 160 comprises a heat transferring baseplate 161, a plurality of fins 162, and four outlet ports 163. Among the many novel features of this embodiment is the varying height of the fins based on the importance of heat transfer versus flow resistance within the immediate local flowfield. For clarity and because means for introducing a jet of impinging fluid and means to convey the outlet fluid are quite varied, they are not shown.

[0151] Referring now to FIG. 25, an isometric view of a non-linear impingement heat sink 160 is shown. In the embodiment depicted, fluid enters at substantially a perpendicular angle to the surface 169 of baseplate 161. Fluid enters as a high velocity, turbulent jet and contacts baseplate surface 169 at a central impingement point (also called a stagnation zone) 164. The high velocity jet vector appearing perpendicular to baseplate surface 169 is converted to a laminar flow with a vector substantially parallel to baseplate surface 169. The rapid change in vector and momentum causes a large reduction in the thickness of the velocity and boundary layers at impingement point 164 resulting in a local area of high heat transfer. Fluid travels in a radial direction away from impingement point 164 and towards four outlet ports 163. Non-linear fins 162 of baseplate 161 are physically grouped into four local regions having specific fluidic and heat transfer function. Immediately surrounding impingement point 164 appears a region of inlet cylindrical fins 165. The function of inlet cylindrical fins 165 is to capture the turbulent chaotic radial flow from the impingement point and provide a region of even laminar flow in all directions parallel to surface 169. While doing this, inlet cylindrical fins 165 transfer heat on all surfaces while providing low pressure drop.

[0152] When the laminarized flow reaches the edge of inlet cylindrical fin region 165 a portion of the flow will progress through a region of elliptical fins 166 that are aligned parallel to the bulk flow and a region of elliptical fins 167 that are at a slight angle to the bulk flow. Moving radially from the impingement point, elliptical fins 166 aligned parallel to the flow are characterized by having high aspect ratios of longitudinal length to transverse width (measured according to the local flow direction. Past this entrance area at a distance roughly halfway between impingement point 164 and outlet port 163 elliptical fins 166 progress to lower aspect ratios having larger transverse dimensions. This change in aspect ratio and transverse size helps to maintain a constant velocity, disrupt the formation of a boundary layer, and allows elliptical fins 166 to have a higher core temperature providing a higher heat transfer coefficient. Flow resistance in this area is still low because of the elliptical shape of fins 166.

[0153] The portion of fluid that did not travel through the region of fins 166 aligned parallel to the bulk flow, moves through a region of elliptical fins 167 that are at a slight angle to the bulk flow. Angled elliptical fins 167 have higher angles

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relative to the vector of the bulk flow as the distance from impingement point **164** increases. The increasing angle of the downstream fins helps to gradually change the fluid direction without a corresponding increase in flow resistance, prevents the occurrence of a thick boundary layer, and again results in a higher heat transfer coefficient than would normally occur. As the flow approaches the fluid boundary wall **170** of baseplate **161** the fluid is separated into two paths that each lead to an opposing outlet port **163**. Separation of the fluid and subsequent impingement at fluid boundary wall **170** provide a slight enhancement in heat transfer while effectively completing the change in fluid direction toward outlet ports **163**.

[0154] As the fluid leaving the region of elliptical fins **166** aligned parallel to the bulk flow and the region of elliptical fins **167** that are at an angle to the bulk flow combines from different directions, local mass flow, velocity and turbulence are increased and the fluid enters a region of outlet cylindrical fins **168** that are designed to transfer heat to the turbulent flow and allow fluid movement in a substantially perpendicular vector away from baseplate surface **169** and out through outlet ports **163**. The fins at this location are necessarily thin to avoid obstructing outlet fluid flow, and because at this radial distance from the central heat source, little heat is left to transfer to the fluid.

[0155] Although most of the embodiments of the present invention show a single inlet location and one or several fluid exit locations, the embodiment depicted in FIG. **26** shows the effect of an optimized non-linear fin structure, using the teachings herein, surrounding multiple fluid inlet and outlet locations to comprise a multiple inlet/outlet non-linear coldplate **180**. As shown a common baseplate **181** has a plurality of non-linear fins **182** used to affect heat transfer. Fins **182** are non-linear by way of having varying aspect ratios (longitudinal length to transverse width to vertical height), varying degrees of parallelness or perpendicularity to bulk fluid flow, varying material compositions and coatings, all based on the desired effect on fluid flow and heat transfer in the area of fluid flow directly adjacent to the individual fin. The present embodiment has four impingement points **183** and nine areas to manage outlet flow **184**. FIG. **26** shows inlet areas **183** and outlet areas **184** spaced evenly, but design practice may dictate more varied spacing based on the location of heat sources. FIG. **26** also shows identical groups of non-linear fins **182** surrounding each inlet location **183**. According to the application, these groups of fins, outlet port sizes, impingement jet diameter and flow rate may change significantly based on the desired amount of heat transfer in relation to the other heat sources.

[0156] There are numerous fluids (gaseous or liquid) which are conventionally used for cooling purposes in heat sink applications, any of which can be used in practicing the present invention with appropriate changes to fin geometries. Air is commonly used to dissipate low heat fluxes, such as in desktop computers.

[0157] Depending on the specific application, utilization of liquids for the cooling of electronic equipment is generally governed by certain requirements, principles and considerations. Among the many such requirements, principles and considerations which would possibly be applicable in inventive practice are the following: (i) a high thermal conductivity will yield a high heat transfer rate. (ii) High specific heat of the fluid will require a smaller mass flow rate of the fluid. (iii) Low viscosity fluids will cause a smaller pressure drop, and thus require a smaller pump. (iv) Fluids with a high surface tension will be less likely to cause leakage problems. (v) A fluid (e.g., liquid) with a high dielectric strength is not required in direct fluid (e.g., liquid) cooling. (vi) Chemical compatibility of the fluid and the heat sink material is required to avoid problems insofar as the fluid reacting to the material with which it comes in contact. (vii) Chemical stability of the fluid is required to assure that the fluid does not decompose under prolonged use. (viii) Nontoxic fluids are safe for personnel to handle and use. (ix) Fluids with a low freezing point and a high boiling point will extend the useful temperature ranges of the fluid; however, for most practical applications, a fluid should be selected to meet the operating conditions of the component to be cooled. (x) Low cost is desirable to maintain affordable systems.

[0158] Fluid-cooled heat sinks used in electronic enclosures and such contexts are usually water-cooled. The heat sink is cooled by the water which is passed therethrough. In many electronic applications, distilled or demineralized water is used to increase the dielectric strength of the water, thereby avoiding electrically coupling components. High heat removal rates can be achieved by circulating water systems. Anhydrous refrigerants are used in place of water or in mixtures with water to keep temperatures of heat sinks at subzero temperatures, thereby increasing the performance of the electronic components. Examples of refrigerants other than water include ammonia, carbon dioxide, CFC-based refrigerants such as R-12 (dichlorodifluoromethane or "freon"), HCFC-based refrigerants such as R134A, and non-CFC substitutes (e.g., for freon) such as R-406A.

[0159] Other embodiments of this invention will be apparent to those skilled in the art from a consideration of this specification or practice of the invention disclosed herein. Various omissions, modifications and changes to the principles described may be made by one skilled in the art without departing from the true scope and spirit of the invention which is indicated by the following claims.

What is claimed is:

1. A non-linear fin heat sink, comprising:

a base;

a plurality of fins disposed on an upper surface of the base, wherein each fin has a cross-sectional fin longitudinal dimension and a cross-sectional fin transverse dimension, and the fins are arranged in a plurality of longitudinal rows and a plurality of transverse rows; and

an upper lid disposed on the top of the fins;

wherein the base and the upper lid are formed a boundary for flowing inside, one side of the heat sink is a leading edge for flowing in and a corresponding side of the heat sink is a trailing edge for flowing out.

2. The non-linear fin heat sink as claimed in claim **1**, wherein the longitudinal rows are in staggered relationship with each other.

3. The non-linear fin heat sink as claimed in claim **1**, wherein the upper lid and the base are both planar, and a distance therebetween is constant.

4. The non-linear fin heat sink as claimed in claim **3**, wherein each height of at least one entrance fins at the leading edge is less than a height of a exit fin at the trailing edge, and the height of the exit fin at the trailing edge is equal to the distance between the upper lid and base.

5. The non-linear fin heat sink as claimed in claim **4**, wherein a height of an initial fin in the entrance fins has to meet two requirements: $\in > 820\ \nu/U$ and $\in > 0.308\delta_d$, wherein $\in$ is the height of the initial fin (m), ν is the kinematic vis-

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cosity (m^2/s), U is fluid velocity (m/s) and δ_d is the height of the displacement boundary layer (m) where 1.7208 $Re_x^{-0.5}$.

6. The non-linear fin heat sink as claimed in claim **5**, wherein the height of the downstream fins in the entrance fins is based on the increase in the velocity boundary layer thickness.

7. The non-linear fin heat sink as claimed in claim **4**, wherein the fin transverse dimension increases in relation to the thickness of the boundary layer.

8. The non-linear fin heat sink as claimed in claim **7**, wherein the fins are arranged to form a first group of fins and a second group of fins.

9. The non-linear fin heat sink as claimed in claim **8**, wherein the first group of pins is from the leading edge to a first predetermined distance of a total heat sink length, and the second group of fins is from the first predetermined distance of the total heat sink length to the trailing edge.

10. The non-linear fin heat sink as claimed in claim **9**, wherein a fin aspect ratio described as fin longitudinal dimension divided by fin transverse dimension progresses from high aspect ratio ellipses at the leading edge to low aspect ratio ellipses at the first distance of the total heat sink length in the first group of fins.

11. The non-linear fin heat sink as claimed in claim **10**, wherein the first transverse row of the first group of fins has an aspect ratio of 6.0:1 and the last transverse row of the first group of fins has an aspect ratio of 2.0:1, with the aspect ratio of the fins in intervening transverse rows decreasing linearly from the first row to the last row.

12. The non-linear fin heat sink as claimed in claim **11**, wherein a first portion of transverse rows of the first group of fins has an aspect ratio of 6.0:1; a next portion of transverse rows of the first group of fins has an aspect ratio of 5.5:1; a next portion of transverse rows of the first group of fins has an aspect ratio of 5.0:1; a next portion of transverse rows of the first group of fins has an aspect ratio of 4.5:1; a next portion of transverse rows of the first group of fins has an aspect ratio of 4.0:1; a next portion of transverse rows of the first group of fins has an aspect ratio of 3.5:1; a next portion of transverse rows of the first group of fins has an aspect ratio of 3.0:1; a next portion of transverse rows of the first group of fins has an aspect ratio of 2.5:1; a remaining portion of transverse rows of the first group of fins has an aspect ratio of 2.0:1.

13. The non-linear fin heat sink as claimed in claim **9**, wherein the fin aspect ratio described as fin longitudinal dimension divided by fin transverse dimension progresses from low aspect ratio ellipses at the first distance of the total heat sink length back to high aspect ratio ellipses at the trailing edge.

14. The non-linear fin heat sink as claimed in claim **13**, wherein the transverse row of the second group of fins has an aspect ratio of 3.0:1; the last transverse row of the second group of fins has an aspect ratio of 6.0:1; and the aspect ratio of the intervening transverse rows increases linearly from the first row to the last row.

15. The non-linear fin heat sink as claimed in claim **14**, wherein a first portion of transverse rows of the second group of fins has an aspect ratio of 3.0:1; a next portion of transverse rows of the second group of fins has an aspect ratio of 4.0:1; a next portion of transverse rows of the second group of fins has an aspect ratio of 5.0:1; and a last portion of transverse rows of the second group of fins has an aspect ratio of 6.0:1.

16. The non-linear fin heat sink as claimed in claim **4**, wherein a number of fins along the leading edge is diminished for lending a heat sink entrance with a "U" shape.

17. The non-linear fin heat sink as claimed in claim **16**, wherein the heat sink entrance is a parabolic entrance region.

18. The non-linear fin heat sink as claimed in claim **16**, wherein the heat sink entrance is a hyperbolic entrance region.

19. The non-linear fin heat sink as claimed in claim **16**, wherein a shape of each fin is changed depending on the distance between each fin and the leading edge.

20. The non-linear fin heat sink as claimed in claim **19**, wherein each fin at a first predetermined distance of the longitudinal distance of the heat sink base is of a conical shape.

21. The non-linear fin heat sink as claimed in claim **20**, wherein each fin comprises a fin base and a fin tip, and each fin progresses from the large fin base attached to the base to the fin tip.

22. The non-linear fin heat sink as claimed in claim **19**, wherein each fin at a second predetermined distance of the longitudinal distance of the heat sink base is of a concave hyperbolic shape.

23. The non-linear fin heat sink as claimed in claim **22**, wherein each fin comprises a fin base and a fin tip, and each fin progresses along a concave hyperbolic curve from the fin base attached to base to the truncated fin tip.

24. The non-linear fin heat sink as claimed in claim **19**, wherein each fin at a third predetermined distance of the longitudinal distance of the heat sink base is of a concave parabolic shape.

25. The non-linear fin heat sink as claimed in claim **24**, wherein each fin comprises a fin base and a fin tip, and each fin progresses along a concave parabolic curve from the fin base attached to base to the truncated fin tip.

26. The non-linear fin heat sink as claimed in claim **19**, wherein each fin at a fourth predetermined distance of the longitudinal distance of the heat sink base is of a cylindrical profile.

27. The non-linear fin heat sink as claimed in claim **26**, wherein the side walls of each fin progress along a straight line perpendicular from the base.

28. The non-linear fin heat sink as claimed in claim **19**, wherein each fin at a fifth predetermined distance of the longitudinal distance of the heat sink base is of a conical shape.

29. The non-linear fin heat sink as claimed in claim **28**, wherein each fin comprises a fin base and a fin tip, and each fin progresses from the large fin base attached to the base to the fin tip.

30. The non-linear fin heat sink as claimed in claim **1**, wherein each fin is formed of an airfoil-shape.

31. The non-linear fin heat sink as claimed in claim **30**, wherein a radius of curvature of the fins are varying depending on the distance from the leading edge.

32. The non-linear fin heat sink as claimed in claim **30**, wherein the radius of curvature of the fins is reduced using an exponential function.

33. The non-linear fin heat sink as claimed in claim **1**, wherein the fins are made from at least one material.

34. The non-linear fin heat sink as claimed in claim **32**, wherein the fins near the leading edge are constructed of materials having a lower thermal conductivity than fins farther from the leading edge.

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35. The non-linear fin heat sink as claimed in claim 33, wherein the fins near the trailing edge are constructed of materials having a higher thermal conductivity than fins farther from the trailing edge.

36. The non-linear fin heat sink as claimed in claim 1, wherein a surface of each fin is a porous surface.

37. The non-linear fin heat sink as claimed in claim 36, wherein each fin is constructed of a porous material infused with a chemical.

38. The non-linear fin heat sink as claimed in claim 36, wherein the surface of each fin is coated with a chemical.

39. The non-linear fin heat sink as claimed in claim 36, wherein the fins are coated with at least one chemical.

40. The non-linear fin heat sink as claimed in claim 33, wherein the fins are arranged with a varying aspect ratio of the fins depending on the distance from the leading edge.

41. A method for producing a non-linear fin heat sink, comprising:

providing a base with a plurality of base fins thereon;

providing an upper lid with a plurality of upper lid fins thereon.

facing a surface of the upper lid with the upper lid fins downwardly corresponding to a surface of the base with the base fins;

combining the upper lid and the base; and

forming a fluid channel.

42. The method as claimed in claim 41, wherein an upper lid fin space is formed between each two adjacent upper lid fins.

43. The method as claimed in claim 42, wherein the base fins respectively fit into the upper lid fin spaces.

44. The method as claimed in claim 41, wherein a base fin space is formed between each two adjacent base fins.

45. The method as claimed in claim 44, wherein the upper lid fins respectively fit into the base fin spaces.

46. A heat exchanger, comprising:

a shell tube having a leading edge and a trail edge, wherein a plurality of internal shell tube fins are disposed on an interior surface thereof;

a heat exchanger tube having a leading edge and a trail edge disposed inside the shell tube; wherein a plurality of exterior fins and interior fins are respectively disposed on an exterior surface and an interior surface of the heat exchanger tube.

47. The heat exchanger as claimed in claim 46, wherein a plurality of external fins are located on an external surface of the shell tube.

48. The heat exchanger as claimed in claim 46, wherein each height of at least one entrance fins at the leading edge is less than a height of an exit fin at the trailing edge.

49. The heat exchanger as claimed in claim 46, wherein each fin has a cross-sectional fin longitudinal dimension and a cross-sectional fin transverse dimension, and the fins are arranged in a plurality of longitudinal rows and a plurality of transverse rows.

50. The heat exchanger as claimed in claim 49, wherein a fin aspect ratio described as fin longitudinal dimension divided by fin transverse dimension progresses from high aspect ratio ellipses at the leading edge to low aspect ratio ellipses at a first distance of the total heat sink length.

51. The heat exchanger as claimed in claim 50, wherein the fin aspect ratio described as fin longitudinal dimension divided by fin transverse dimension progresses linearly from low aspect ratio ellipses at the first distance of the total heat sink length back to high aspect ratio ellipses at the trailing edge.

52. A heat sink, comprising:

a first sink having an entrance and an exit, wherein a first fin array is disposed inside the first sink;

a second sink having an entrance and an exit, wherein a second fin array is disposed inside the second sink; and

a manifold connected between the exit of the first sink and the entrance of the second sink, wherein a plurality of manifold fins are dispose inside the manifold;

wherein a fluid flows through the first fin sink and enters the manifold to turn to enter the second fin sink.

53. The heat sink as claimed in claim 52, wherein each manifold fin has predetermined configuration.

54. The heat sink as claimed in claim 53, wherein a fin aspect ratio described as a fin longitudinal dimension divided by a fin transverse dimension progresses from high aspect ratio ellipses at the entrance to low aspect ratio ellipses at a first distance of the total heat sink length in the second fin array.

55. The heat sink as claimed in claim 54, wherein the fin aspect ratio described as the fin longitudinal dimension divided by the fin transverse dimension progresses linearly from low aspect ratio ellipses at the first distance of the total heat sink length back to high aspect ratio ellipses at the exit in the second fin array.

56. A manifold-heatsink assembly, comprising:

a manifold housing having at least one receiving space and an upper lid formed thereinside;

at least one non-linear fin heat sink attached inside the receiving space, wherein the non-linear fin heat sink comprises a base and a plurality of fins projected from the base the base to contact the upper lid;

a module baseplate covered on the manifold housing;

an incoming coolant port defined on a sidewall of the manifold housing to communicate with the receiving space; and

an outgoing coolant port defined on the sidewall of the manifold housing to communicate with the receiving space;

wherein base upper lid and sidewalls of manifold housing define an outline shape to provide a flow cavity.

57. The manifold-heatsink assembly as claimed in claim 56, wherein a sealing element is disposed on edges of the receiving space to prevent fluid leakage.

58. The manifold-heatsink assembly as claimed in claim 56, wherein a layer of thermal interface material is provided between module baseplate and base.

59. A non-linear impingement fin heat sink, comprising:

a heat transferring baseplate;

a plurality of fins disposed on the heat transferring baseplate; and

at least one outlet ports defined on sides of the heat transferring baseplate;

wherein the fins are arranged to form an impingement point at a portion of the baseplate, an inlet cylindrical fins region surrounding the impingement point, an elliptical fins region surrounding the inlet cylindrical fins region.

60. The non-linear impingement fin heat sink as claimed in claim 59, wherein the elliptical fins region has a first portion of fins and a second portion of fins.

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61. The non-linear impingement fin heat sink as claimed in claim **60**, wherein the first portion of fins is arranged parallel to the bulk flow and the second of fins is at a slight angle to the bulk flow.

62. A multiple inlet/outlet non-linear coldplate, comprising:

a baseplate; and

a plurality of fins disposed on the baseplate;

wherein the fins are arranged to form at least one impingement point and at least one outlet flow portion.

63. A plate fin heat sink, comprising:

a base; and

a plurality of fins mounted on the base, wherein each fin has a fin transverse dimension;

wherein the plate fin heat sink has a leading edge and a trailing edge for a fluid entering and draining the plate fin heat sink.

64. The plate fin heat sink as claimed in claim **63**, wherein the fins are parallel to the flow axis of the fluid.

65. The plate fin heat sink as claimed in claim **63**, wherein the fin transverse dimension of each fin increases from the leading edge to a predetermined length of the heat sink.

66. The plate fin heat sink as claimed in claim **64**, wherein the fin transverse dimension of each fin increases in relation to a, the thickness of the boundary layer.

67. The non-linear fin heat sink as claimed in claim **65**, wherein the fin transverse dimension of each fin decreases from the predetermined length of the heat sink to the trailing edge.

68. A heat sink, comprising

a base;

a plurality of discrete fin patterns wherein each fin pattern is formed of a plurality of fins;

a upper lid covered on the fin patterns; and

a plurality of flow diverters disposed between each two fin patterns.

69. The heat sink as claimed in claim **68**, wherein a plurality of base flow diverters are protruding from the base.

70. The heat sink as claimed in claim **68**, wherein a plurality of lid flow diverters are protruding from the upper lid.

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