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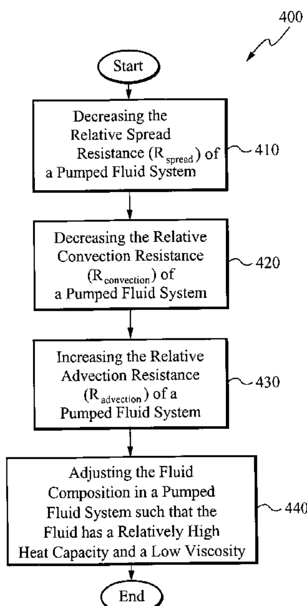
(54) **Title:** PUMPED FLUID COOLING SYSTEM AND METHOD



(57) **Abstract:** The present invention is a pumped fluid cooling system and method. The pumped fluid cooling system and method includes new relative magnitudes of advection, convection and spreading components of the resistance for a pumped fluid system. The pumped fluid cooling system and method also includes adjusting the chemical composition of the working fluid, specifically adjusting the composition and viscosity as the sensitivity to the fluid heat capacity per unit mas increases.

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PUMPED FLUID COOLING SYSTEM AND METHOD

FIELD OF THE INVENTION:

The present invention relates generally to the field of cooling systems. More
5 specifically, the present invention relates to the field of pumped fluid cooling systems.

BACKGROUND OF THE INVENTION:

In current pumped fluid cooling systems as depicted in Figure 1, the total "temperature budget," or the difference between the peak device temperature ($T_{Device, peak}$) and the
10 temperature of the cold fluid inlet ($T_{fluid\ inlet}$) is consumed by the total heat power (q(W)) flowing through four separate resistances.

Figure 1 illustrates such a resistance model for an exemplary pumped fluid cooling system. The device/attach resistances ($R_{Device\ and\ attach}$) dissipate a significant amount of q(W). However, the device/attach resistances are not related to the present invention and need no
15 further explanation. The spreading resistance (R_{spread}) accounts for spreading the heat from a small device into a larger heat exchanger (hx). The R_{spread} increases with the ratio of the hx to device area. The convection resistance ($R_{convection}$) accounts for conducting the heat into the fluid from the hx walls. It is equal to 1/hA, where h is the convection coefficient and A is the total wetted surface area within the hx. This resistance increases strongly with increasing
20 values of the minimum feature size of the hydraulic diameter (d).

Still referring to Figure 1, the advection resistance ($R_{advection}$) accounts for the heating of the fluid as it transverses the hx, and is approximately equal to C/mc, where m is the mass flowrate and c is the specific heat capacity per unit mass and C is a constant near 0.5. Traditional heat exchangers use relatively large dimensions ranging in size from two times to
25 four times the size of the area of the device being cooled. These dimensions result in relatively large values of R_{spread}. Traditional heat exchangers also have large internal features, usually 0.3mm or larger. These dimensions result in relatively large values of $R_{convection}$. These relatively large values of R_{spread} and $R_{convection}$ result in an inefficient pumped fluid system.

Referring now to Figure 2, a resistance model of a current pumped fluid system 20 of
30 the prior art is illustrated. As stated earlier, current pumped fluid systems 20 utilize heat exchangers that are two to four times the size of the device being cooled. This current design therefore includes a large spreading resistance 22, which continues to increase as the surface

area ratio of (hx/device being cooled) increases. Furthermore, current pumped fluid systems 20 have large hydraulic diameters (d). Referring back to the $R_{convection}$ formula 1/hA, as the hx d increases the total wetted surface area A decreases, thus according to 1/hA, causing a relatively large convection resistance 24.

5 Because the current pumped fluid systems 20 have large values of d (and very small values of A), a great deal of the temperature budget is used in this part of the resistance chain. To stay within the total temperature budget at this point requires the current pumped fluid system 20 to have a very small advection resistance 26. Therefore, referring back to the $R_{advection}$ formula C/cm, the $R_{advection}$ may be reduced significantly by creating very large mass 10 flow rates m. Of course, this puts large demands on the pump requirements for a pumped fluid system 20.

It should also be noted that pumped fluid cooling systems of the prior art require specific fluids to operate effectively with the system, e.g., to avoid freezing at low temperatures. Such fluids include those with high concentrations of ethylene glycol or 15 propylene glycol, or similar substances. The characteristics of such fluids include a low heat capacity and a high viscosity and do not function well in a system having a reduced flowrate.

<u>SUMMARY OF THE INVENTION:</u>

The present invention is a pumped fluid cooling system and method. The pumped 20 fluid cooling system and method includes new relative magnitudes of advection, convection and spreading components of the resistance for a pumped fluid system. The pumped fluid cooling system and method also includes adjusting the chemical composition of the working fluid, specifically adjusting the composition and viscosity as the sensitivity to the fluid heat capacity per unit mass increases.

25 In one aspect of the present invention, a pumped fluid cooling system for cooling a device comprises a heat exchanger, the heat exchanger including an interface layer coupled to the device for cooling the device and a fluid pumped through the interface layer of the heat exchanger, the fluid having an inlet temperature and an outlet temperature, wherein the pumped fluid cooling system is configured such that the difference between the fluid outlet 30 temperature and the fluid inlet temperature is at least 30% of the difference between a hottest temperature of the fluid in the heat exchanger and the fluid inlet temperature.

The pumped fluid cooling system further comprises a plurality of microchannels configured in a predetermined pattern along the interface layer wherein the plurality of microchannels have an internal feature size in the range of 15-300 microns. The plurality of microchannels have a surface to volume ratio greater than $1000m^{-1}$. The pumped fluid cooling system further comprises a plurality of pillars configured in a predetermined pattern along the interface layer wherein the plurality of pillars have an internal feature size in the range of 15-300 microns. The plurality of pillars have a surface to volume ratio greater than $1000m^{-1}$.

The pumped fluid cooling system further comprises a microporous structure disposed on the interface layer wherein a plurality of pores in the microporous structure have an internal feature size in the range of 15-300 microns. The plurality of pores of the microporous structure have a surface to volume ratio greater than $1000m^{-1}$. A first surface area of the interface layer that is coupled to the device is less than or equal to 150% of a second surface area of the device that is coupled to the interface layer. The viscosity of the fluid at its average temperature in the heat exchanger is less than 150% of the viscosity of water. The heat capacity per unit mass of the fluid at its average temperature in the heat exchanger is greater than 80% of the heat capacity per unit mass of water. The fluid consists of at least 90% water by mass.

In another aspect of the present invention, a method of efficiently cooling a device in a pumped fluid cooling system comprises decreasing a spread resistance between an interface layer of a heat exchanger and the device, decreasing a convection resistance between a fluid and the interface layer of the heat exchanger, wherein the fluid is pumped through the interface layer, and further wherein the fluid has an inlet temperature and an outlet temperature, increasing an advection resistance and adjusting the composition of the fluid to increase the heat capacity per unit mass and decrease the viscosity, wherein the difference between the fluid outlet temperature and the fluid inlet temperature is at least 30% of the difference between a hottest temperature of the fluid in the heat exchanger and the fluid inlet temperature.

The step of decreasing the convention resistance includes configuring a plurality of microchannels in a predetermined pattern along the interface layer wherein the plurality of microchannels have an internal feature size in the range of 15-300 microns. The plurality of

microchannels have a surface to volume ratio greater than $1000m^{-1}$. The step of decreasing the convection resistance includes configuring a plurality of pillars in a predetermined pattern along the interface layer wherein the plurality of pillars have an internal feature size in the range of 15-300 microns. The plurality of pillars have a surface to volume ratio greater than $1000m^{-1}$.

The step of decreasing the convection resistance includes disposing a microporous structure on the interface layer wherein a plurality of pores in the microporous structure have an internal feature size in the range of 15-300 microns. The plurality of pores of the microporous structure have a surface to volume ratio greater than $1000m^{-1}$. The step of decreasing the spread resistance includes reducing a first surface area of the interface layer that is coupled to the device such that the first surface area is less than or equal to 150% of a second surface area of the device that is coupled to the interface layer. The step of adjusting the composition of the fluid includes decreasing the viscosity of the fluid at its average temperature in the heat exchanger, such that the viscosity is less than 150% of the viscosity of water. The step of adjusting the composition of the fluid includes increasing the heat capacity per unit mass of the fluid at its average temperature in the heat exchanger, such that the heat capacity per unit mass is greater than 80% of the heat capacity per unit mass of water. The fluid consists of at least 90% water by mass.

In yet another aspect of the present invention, a pumped fluid cooling system for cooling a device comprises means for decreasing a spread resistance between an interface layer of a heat exchanger and the device, means for decreasing a convection resistance between a fluid and the interface layer of the heat exchanger, wherein the fluid is pumped through the interface layer, and further wherein the fluid has an inlet temperature and an outlet temperature, means for increasing an advection resistance and means for adjusting the composition of the fluid to increase the heat capacity per unit mass and decrease the viscosity, wherein the difference between the fluid outlet temperature and the fluid inlet temperature is at least 30% of the difference between a hottest temperature of the fluid in the heat exchanger and the fluid inlet temperature.

The means for decreasing the convention resistance includes means for configuring a plurality of microchannels in a predetermined pattern along the interface layer wherein the plurality of microchannels have an internal feature size in the range of 15-300 microns. The plurality of microchannels have a surface to volume ratio greater than $1000m^{-1}$. The means for

decreasing the convection resistance includes means for configuring a plurality of pillars in a predetermined pattern along the interface layer wherein the plurality of pillars have an internal feature size in the range of 15-300 microns. The plurality of pillars have a surface to volume ratio greater than $1000m^{-1}$.

5 The means for decreasing the convection resistance includes means for disposing a microporous structure on the interface layer wherein a plurality of pores in the microporous structure have an internal feature size in the range of 15-300 microns. The plurality of pores of the microporous structure have a surface to volume ratio greater than $1000m^{-1}$. The means for decreasing the spread resistance includes means for reducing a first surface area of the 10 interface layer that is coupled to the device such that the first surface area is less than or equal to 150% of a second surface area of the device that is coupled to the interface layer.

The means for adjusting the composition of the fluid includes means for decreasing the viscosity of the fluid at its average temperature in the heat exchanger, such that the viscosity is less than 150% of the viscosity of water. The means for adjusting the composition of the fluid 15 includes means for increasing the heat capacity per unit mass of the fluid at its average temperature in the heat exchanger, such that the heat capacity per unit mass is greater than 80% of the heat capacity per unit mass of water. The fluid consists of at least 90% water by mass.

In yet another aspect of the present invention, an apparatus for cooling an integrated 20 circuit comprises a heat exchanger including an interface layer coupled to the integrated circuit, wherein a first surface area of the interface layer that is coupled to the integrated circuit is less than or equal to 150% of a second surface area of the integrated circuit that is coupled to the interface layer, such that a spread resistance between the interface layer and the integrated circuit is decreased, a plurality of microchannels configured in a predetermined 25 pattern along the interface layer wherein the plurality of microchannels have an internal feature size in the range of 15-300 microns and a surface to volume ration greater than $1000m^{-1}$, such that a convection resistance is decreased and a fluid pumped through the heat exchanger, such that a flowrate of the fluid increases an advection resistance, wherein the fluid consists of at least 90% water by mass. The viscosity of the fluid at its average temperature in 30 the heat exchanger is less than 150% of the viscosity of water. The heat capacity per unit mass of the fluid at its average temperature in the heat exchanger is greater than 80% of the heat capacity per unit mass of water.

In yet another aspect of the present invention, a pumped fluid cooling system for cooling a device comprises a spread resistance, wherein the spread resistance is decrease when a heat exchanger including an interface layer is coupled to the device, further wherein a first surface area of the interface layer that is coupled to the device is less than or equal to 150% of a second surface area of the device that is coupled to the interface layer, a convection resistance, wherein the convection resistance is decreased when a plurality of microchannels is configured in a predetermined pattern along the interface layer, and further wherein the plurality of microchannels have an internal feature size in the range of 15-300 microns and a surface to volume ration greater than $1000m^{-1}$ and an advection resistance, wherein the advection resistance is increased when a fluid is pumped through the heat exchanger, such that a flowrate of the fluid decreases, wherein the fluid consists of at least 90% water by mass. The pumped fluid cooling system as claimed in claim 37 wherein the fluid is water.

BRIEF DESCRIPTION OF THE DRAWINGS:

Figure 1 is a graphical representation illustrating an exemplary temperature budget resistance model.

Figure 2 is a graphical representation illustrating a temperature budget resistance model according to the prior art.

Figure 3 is a graphical representation illustrating a temperature budget resistance model according to an embodiment of the present invention.

Figure 4A is a graphical representation illustrating a top view of a manifold layer of a heat exchanger in accordance with the present invention.

Figure 4B is a graphical representation illustrating an exploded view of a heat exchanger with a manifold layer in accordance with the present invention.

Figure 5 is a graphical representation illustrating a perspective view of an interface layer having a micro-pin layer and a foam layer in accordance with the present invention.

Figure 6 is a flowchart illustrating a method of efficiently cooling a device in a pumped fluid cooling system in accordance with the present invention.

DETAILED DESCRIPTION OF THE INVENTION:

Figure 3 is a graphical representation of the preferred embodiment of the present invention. The preferred embodiment of the present invention includes new relative magnitudes of the advection 36, convection 34, and spreading 32 components of the resistance for a pumped fluidic system (PFS) 30, which enable lower pump flowrates and, consequently, pumps that are smaller and consume less power. The new relative magnitudes of these resistances are enabled by a micro hx as described below with feature sizes in the range of 15-300 microns. Still referring to Figure 3, this micro hx of the PFS 30 of the preferred embodiment of the present invention allows for a smaller spread resistance 32 and smaller convection resistance 34, thereby conserving the temperature budget. This conservation allows for a higher advection 36 component.

Referring back to the advection formula once again, C/mc where m is the flowrate, reducing the flowrate m will cause the advection 36 component to increase. This increase in the advection 36 component may continue until the total temperature budget is spent. Therefore, in effect, the decrease spreading 32 and convection 34 components allow for a micro hx having a smaller flowrate m and higher advection 36 component, thereby resulting in less work for the pump, and thus a more efficient PFS 30.

The micro hx of the present invention decreases the spreading 32 component by reducing the size of the cooling surface of the micro hx such that it is less than or equal to 150% of the size of the surface of the device that is being cooled by the micro hx. The convection 34 component is again equal to 1/hA, where h is the convection coefficient and A is the total wetted surface area of the micro hx. This convection 34 component is decreased as the wetted surface area in the micro hx is greatly increased relative to current pumped fluidic systems. The wetted surface area of the micro hx is increased by incorporating pillars, foam and/or channels having internal feature sizes in the range of 15-300 microns and surface to volume ratios greater than $1000m^{-1}$. The structure of the micro hx is explained in greater detail below.

In order to better understand the description of the preferred embodiment of the present invention described above, it is necessary to also understand the structure and operation of a micro hx according to an embodiment of the present invention. However, it should be understood that the description of the heat exchanger below represents but one design applicable to the present invention, and it has been contemplated that the system and

method of the present invention may be applied to any heat exchanger having the requisite dimensions of the preferred embodiment of the present invention.

Generally, a heat exchanger captures thermal energy generated from a heat source by passing fluid through selective areas of the interface layer which is preferably coupled to the heat source. In particular, the fluid is directed to specific areas in the interface layer to cool the hot spots and areas around the hot spots to generally create temperature uniformity across the heat source while maintaining a small pressure drop within the heat exchanger. As discussed in the different embodiments below, the heat exchanger utilizes a plurality of apertures, channels and/or fingers in the manifold layer as well as conduits in the intermediate layer to direct and circulate fluid to and from selected hot spot areas in the interface layer. Alternatively, the heat exchanger includes several ports which are specifically disposed in predetermined locations to directly deliver fluid to and remove fluid from the hot spots to effectively cool the heat source.

Figure 4A illustrates a top view of an exemplary manifold layer 106 of the present invention. In particular, as shown in Figure 4B, the manifold layer 106 includes four sides as well as a top surface 130 and a bottom surface 132. However, the top surface 130 is removed in Figure 4A to adequately illustrate and describe the workings of the manifold layer 106. As shown in Figure 4A, the manifold layer 106 has a series of channels or passages 116, 118, 120, 122 as well as ports 108, 109 formed therein. The fingers 118, 120 extend completely through the body of the manifold layer 106 in the Z-direction, as shown in Figure 4B. Alternatively, the fingers 118 and 120 extend partially through the manifold layer 106 in the Z-direction and have apertures as shown in Figure 4A. In addition, passages 116 and 122 extend partially through the manifold layer 106. The remaining areas between the inlet and outlet passages 116, 120, designated as 107, extend from the top surface 130 to the bottom surface 132 and form the body of the manifold layer 106.

As shown in Figure 4A, the fluid enters the manifold layer 106 via the inlet port 108 and flows along the inlet channel 116 to several fingers 118 which branch out from the channel 116 in several X and Y directions to apply fluid to selected regions in the interface layer 102. The fingers 118 are preferably arranged in different predetermined directions to deliver fluid to the locations in the interface layer 102 corresponding to the areas at and near the hot spots in the heat source. These locations in the interface layer 102 are hereinafter referred to as interface hot spot regions. The fingers are configured to cool stationary interface

hot spot regions as well as temporally varying interface hot spot regions. As shown in Figure 4A, the channels 116, 122 and fingers 118, 120 are disposed in the X and Y directions in the manifold layer 106 and extend in the Z direction to allow circulation between the manifold layer 106 and the interface layer 102. Thus, the various directions of the channels 116, 122 and fingers 118, 120 allow delivery of fluid to cool hot spots in the heat source 99 and/or minimize pressure drop within the heat exchanger 100.

The arrangement as well as the dimensions of the fingers 118, 120 are determined in light of the hot spots in the heat source 99 that are desired to be cooled. The locations of the hot spots as well as the amount of heat produced near or at each hot spot are used to configure the manifold layer 106 such that the fingers 118, 120 are placed above or proximal to the interface hot spot regions in the interface layer 102. The manifold layer 106 allows one phase and/or two-phase fluid to circulate to the interface layer 102 without allowing a substantial pressure drop from occurring within the heat exchanger 100. The fluid delivery to the interface hot spot regions creates a uniform temperature at the interface hot spot region as well as areas in the heat source adjacent to the interface hot spot regions.

The dimensions as well as the number of channels 116 and fingers 118 depend on a number of factors. In one embodiment, the inlet and outlet fingers 118, 120 have the same width dimensions. Alternatively, the inlet and outlet fingers 118, 120 have different width dimensions. The width dimensions of the fingers 118, 120 are within the range of and including 0.25-1.00 millimeters. In one embodiment, the inlet and outlet fingers 118, 120 have the same length and depth dimensions. Alternatively, the inlet and outlet fingers 118, 120 have different length and depth dimensions. In another embodiment, the inlet and outlet fingers 118, 120 have varying width dimensions along the length of the fingers. The length dimensions of the inlet and outlet fingers 118, 120 are within the range of and including 0.5 millimeters to three times the size of the heat source length. In addition, the fingers 118, 120 have a height or depth dimension within the range and including 0.25-1.00 millimeters. In addition, less than 10 or more than 30 fingers per centimeter are disposed in the manifold layer 106. However, it is apparent to one skilled in the art that between 10 and 30 fingers per centimeter in the manifold layer is also contemplated.

It is contemplated within the present invention to tailor the geometries of the fingers 118, 120 and channels 116, 122 to be in non-periodic arrangement to aid in optimizing hot spot cooling of the heat source. In order to achieve a uniform temperature across the heat

source 99, the spatial distribution of the heat transfer to the fluid is matched with the spatial distribution of the heat generation. As the fluid flows along the interface layer 102, its temperature increases and as it begins to transform to vapor under two-phase conditions. Thus, the fluid undergoes a significant expansion which results in a large increase in velocity.
5 Generally, the efficiency of the heat transfer from the interface layer to the fluid is improved for high velocity flow. Therefore, it is possible to tailor the efficiency of the heat transfer to the fluid by adjusting the cross-sectional dimensions of the fluid delivery and removal fingers 118, 120 and channels 116, 122 in the heat exchanger 100. This effect will also be realized in single phase flow.

10 For example, a particular finger can be designed for a heat source where there is higher heat generation near the inlet. In addition, it may be advantageous to design a larger cross section for the regions of the fingers 118, 120 and channels 116, 122 where a mixture of fluid and vapor is expected. Although not shown, a finger can be designed to start out with a small cross sectional area at the inlet to cause high velocity flow of fluid. The particular finger or
15 channel can also be configured to expand to a larger cross-section at a downstream outlet to cause a lower velocity flow. This design of the finger or channel allows the heat exchanger to minimize pressure drop and optimize hot spot cooling in areas where the fluid increases in volume, acceleration and velocity due to transformation from liquid to vapor in two-phase flow.

20 In addition, the fingers 118, 120 and channels 116, 122 can be designed to widen and then narrow again along their length to increase the velocity of the fluid at different places in the microchannel heat exchanger 100. Alternatively, it may be appropriate to vary the finger and channel dimensions from large to small and back again many times over in order to tailor the heat transfer efficiency to the expected heat dissipation distribution across the heat source
25 99. It should be noted that the above discussion of the varying dimensions of the fingers and channels also apply to the other embodiments discussed and is not limited to this embodiment.

Alternatively, as shown in Figure 4A, the manifold layer 106 includes one or more apertures 119 in the inlet fingers 118. In a three tier heat exchanger 100, the fluid flowing along the fingers 118 flows down the apertures 119 to the intermediate layer 104. In addition,
30 as shown in Figure 4A. the manifold layer 106 includes apertures 121 in the outlet fingers 120. In the three tier heat exchanger 100, the fluid flowing from the intermediate layer 104 flows up the apertures 121 into the outlet fingers 120.

The inlet and outlet fingers 118, 120 are open channels which do not have apertures. The bottom surface 103 of the manifold layer 106 abuts against the top surface of the intermediate layer 104 in the three tier exchanger 100 or abuts against the interface layer 102 in the two tier exchanger. Thus, in the three-tier heat exchanger 100, fluid flows freely to and from the intermediate layer 104 and the manifold layer 106. The fluid is directed to and from the appropriate interface hot spot region by conduits 105 the intermediate layer 104. It is apparent to one skilled in the art that the conduits 105 are directly aligned with the fingers, as described below or positioned elsewhere in the three tier system.

Although Figure 4B shows the three tier heat exchanger 100 with the manifold layer, the heat exchanger 100 is alternatively a two layer structure which includes the manifold layer 106 and the interface layer 102, whereby fluid passes directly between the manifold layer 106 and interface layer 102 without passing through the interface layer 104. It is apparent to one skilled in the art that the configuration of the manifold, intermediate and interface layers shown are for exemplary purposes and is thereby not limited to the configuration shown.

As shown in Figure 4B, the intermediate layer 104 includes a plurality of conduits 105 which extend therethrough. The inflow conduits 105 direct fluid entering from the manifold layer 106 to the designated interface hot spot regions in the interface layer 102. Similarly, the apertures 105 also channel fluid flow from the interface layer 102 to the exit fluid port(s) 109. Thus, the intermediate layer 104 also provides fluid delivery from the interface layer 102 to the exit fluid port 109 where the exit fluid port 108 is in communication with the manifold layer 106. The conduits 105 are positioned in the interface layer 104 in a predetermined pattern based on a number of factors including, but not limited to, the locations of the interface hot spot regions, the amount of fluid flow needed in the interface hot spot region to adequately cool the heat source 99 and the temperature of the fluid. The conduits have a width dimension of 100 microns, although other width dimensions are contemplated up to several millimeters. In addition, the conduits 105 have other dimensions dependent on at least the above mentioned factors. It is apparent to one skilled in the art that each conduit 105 in the intermediate layer 104 has the same shape and/or dimension, although it is not necessary. For instance, like the fingers described above, the conduits alternatively have a varying length and/or width dimension. Additionally, the conduits 105 may have a constant depth or height dimension through the intermediate layer 104. Alternatively, the conduits 105 have a varying depth dimension, such as a trapezoidal or a nozzle-shape, through the intermediate layer 104.

The intermediate layer 104 is horizontally positioned within the heat exchanger 100 with the conduits 105 positioned vertically. Alternatively, the intermediate layer 104 is positioned in any other direction within the heat exchanger 100 including, but not limited to, diagonal and curved forms. Alternatively, the conduits 105 are positioned within the intermediate layer 104 in a horizontally, diagonally, curved or any other direction. In addition, the intermediate layer 104 preferably extends horizontally along the entire length of the heat exchanger 100, whereby the intermediate layer 104 completely separates the interface layer 102 from the manifold layer 106 to force the fluid to be channeled through the conduits 105. Alternatively, a portion of the heat exchanger 100 does not include the intermediate layer 104 between the manifold layer 106 and the interface layer 102, whereby fluid is free to flow therebetween. Further, the intermediate layer 104 alternatively extends vertically between the manifold layer 106 and the interface layer 102 to form separate, distinct intermediate layer regions. Alternatively, the intermediate layer 104 does not fully extend from the manifold layer 106 to interface layer 102.

Figure 4B illustrates a perspective view of the interface layer 102 in accordance with the present invention. As shown in Figure 4B, the interface layer 102 includes a bottom surface 103 and a plurality of microchannel walls 110, whereby the area in between the microchannel walls 110 channels or directs fluid along a fluid flow path. The bottom surface 103 is flat and has a high thermal conductivity to allow sufficient heat transfer from the heat source 99. Alternatively, the bottom surface 103 includes troughs and/or crests designed to collect or repel fluid from a particular location. The microchannel walls 110 are preferably configured in a parallel configuration, as shown in Figure 4B, whereby fluid flows between the microchannel walls 110 along a fluid path. Alternatively, the microchannel walls 110 have non-parallel configurations.

It is apparent to one skilled in the art that the microchannel walls 110 are alternatively configured in any other appropriate configuration depending on the factors discussed above. In addition, the microchannel walls 110 have dimensions which minimize the pressure drop or differential within the interface layer 102. It is also apparent that any other features, besides microchannel walls 110 are also contemplated, including, but not limited to, pillars 203 (Figure 5), roughed surfaces, and a micro-porous structure, such as sintered metal and silicon foam 213 (Figure 5) or a combination. An alternative interface layer 202 incorporating both pillars 203 and foam microporous 213 inserts is depicted in Figure 5. However, for exemplary

purposes, the parallel microchannel walls 110 shown in Figure 4B is used to describe the interface layer 102 in the present invention.

Referring back to the assembly in Figure 4B, the top surface of the manifold layer 106 is cut away to illustrate the channels 116, 122 and fingers 118, 120 within the body of the manifold layer 106. The locations in the heat source 99 that produce more heat are hereby designated as hot spots, whereby the locations in the heat source 99 which produce less heat are hereby designated as warm spots. As shown in Figure 4B, the heat source 99 is shown to have a hot spot region, namely at location A, and a warm spot region, namely at location B. The areas of the interface layer 102 which abut the hot and warm spots are accordingly designated interface hot spot regions. As shown in Figure 4B, the interface layer 102 includes interface hot spot region A, which is positioned above location A and interface hot spot region B, which is positioned above location B.

As shown in Figures 4A and 4B, fluid initially enters the heat exchanger 100 through one inlet port 108. The fluid then flows to one inlet channel 116. Alternatively, the heat exchanger 100 includes more than one inlet channel 116. As shown in Figures 4A and 4B, fluid flowing along the inlet channel 116 from the inlet port 108 initially branches out to finger 118D. In addition, the fluid which continues along the rest of the inlet channel 116 flows to individual fingers 118B and 118C and so on.

In Figure 4B, fluid is supplied to interface hot spot region A by flowing to the finger 118A, whereby fluid preferably flows down through finger 118A to the intermediate layer 104. The fluid then flows through the inlet conduit 105A, positioned below the finger 118A, to the interface layer 102, whereby the fluid undergoes thermal exchange with the heat source 99. The fluid travels along the microchannels 110 as shown in Figure 4B, although the fluid may travel in any other direction along the interface layer 102. The heated liquid then travels upward through the conduit 105B to the outlet finger 120A. Similarly, fluid flows down in the Z-direction through fingers 118E and 118F to the intermediate layer 104. The fluid then flows through the inlet conduit 105C down in the Z-direction to the interface layer 102. The heated fluid then travels upward in the Z-direction from the interface layer 102 through the outlet conduit 105D to the outlet fingers 120E and 120F. The heat exchanger 100 removes the heated fluid in the manifold layer 106 via the outlet fingers 120, whereby the outlet fingers 120 are in communication with the outlet channel 122. The outlet channel 122 allows fluid to flow out of the heat exchanger through one outlet port 109.

The inflow and outflow conduits 105 are also positioned directly or nearly directly above the appropriate interface hot spot regions to directly apply fluid to hot spots in the heat source 99. In addition, each outlet finger 120 is preferably configured to be positioned closest to a respective inlet finger 119 for a particular interface hot spot region to minimize pressure
5 drop therebetween. Thus, fluid enters the interface layer 102 via the inlet finger 118A and travels the least amount of distance along the bottom surface 103 of the interface layer 102 before it exits the interface layer 102 to the outlet finger 120A. It is apparent that the amount of distance which the fluid travels along the bottom surface 103 adequately removes heat generated from the heat source 99 without generating an unnecessary amount of pressure drop.
10 In addition, as shown in Figures 4A and 4B, the corners in the fingers 118, 120 are curved to reduce pressure drop of the fluid flowing along the fingers 118.

It is apparent to one skilled in the art that the configuration of the manifold layer 106 shown in Figures 4A and 4B is only for exemplary purposes. The configuration of the channels 116 and fingers 118 in the manifold layer 106 depend on a number of factors,
15 including but not limited to, the locations of the interface hot spot regions, amount of flow to and from the interface hot spot regions as well as the amount of heat produced by the heat source in the interface hot spot regions. Any other configuration of channels 116 and fingers 118 is contemplated.

Referring to figure 4B, the preferred embodiment of the present invention includes
20 microchannels 110 in the interface layer 102. In order to achieve the desired decrease in convection resistance as described previously, the internal feature size of the microchannels 110 are in the range of 15-300 microns, and the surface to volume ratios of the microchannels are greater than $1000m^{-1}$. Of course, the present invention contemplates further embodiments contemplating microchannels not entirely within the stated ranges.

25 Referring now to figure 5, further embodiments also contemplate utilizing alternatives to the microchannels 110 (figure 4B) of the preferred embodiment such as pillars 203, roughed surfaces or a micro-porous structure, such as sintered metal and silicon foam 213. Furthermore, any of these alternatives could be used instead of the microchannels 110, or they could be used in combination as an alternative interface layer 202. Furthermore, these
30 alternatives may be used in any conceivable combination with the microchannels 110. Of course, any alternative listed above or combination thereof has internal features sizes and a

surface to volume ration that conforms to those set out in the preferred embodiment of the present invention.

Also critical in achieving the desired relative resistance levels is the fluid composition used in the pumped fluid system on the preferred embodiment of the present invention. Specifically, the heat capacity and viscosity become important when the desired relative resistance levels are achieved. Using micro dimensions as those described in the preferred embodiment can dramatically increase the pumping pressure drop. Using low fluid flowrates makes the performance highly sensitive to the fluid heat capacity per unit mass, which governs its heat absorbing properties.

Therefore, in order for the system to operate properly with the desired relative resistance levels, fluid with very high heat capacity per unit mass (enabling high absorption) and low viscosity (enabling low pressure drop in a micro hx) are required. Preferably, a fluid at its average temperature in the heat exchanger, having a viscosity, greater than 150% of the viscosity of water and a heat capacity greater than 80% of water is required. Also in the preferred embodiment of the present invention, the fluid consists of at least 90% of water by mass.

Figure 6 depicts a method of efficiently cooling a device in a pumped fluid cooling system 400 of the preferred embodiment of the present invention. The method 400 starts in step 410, by decreasing the relative spread resistance in a pumped fluid system. This is achieved by limited the size of the cooling surface of the micro hx relative to the surface of the device being cooled. Preferably, to equal to or less than 150% of the surface of the device being cooled. In step 420, the relative convection resistance of the pumped fluid system is decreased by increasing the total wetted surface area in the micro hx. This is accomplished by reducing the internal feature sizes of the micochannels in the micro hx, preferably to a size in the range of 15-300 microns, with a surface to volume ratio of $1000m^{-1}$.

Still referring to Figure 6, in step 430 the relative advection resistance for the pumped fluid system is increased. This is preferably done by decreasing the flowrate m, where the advection resistance equals C/mc, where C is a constant near 0.5 and c is the specific heat capacity per unit mass. The last step in this method 400 is step 440, by adjusting the fluid composition in the pumped fluid system such that the fluid has a relatively high heat capacity and a low viscosity. Preferably, the viscosity of the fluid at its average temperature in the heat exchanger is less than 150% of the viscosity of water and the heat capacity per unit mass of

the fluid at its average temperature in the heat exchanger is greater than 80% of the heat capacity per unit mass of water. This is preferably achieved by adjusting the fluid such that it consists of at least 90% water by mass.

The present invention has been described in terms of specific embodiments incorporating details to facilitate the understanding of the principles of construction and operation of the invention. Such reference herein to specific embodiments and details thereof is not intended to limit the scope of the claims appended hereto. It will be apparent to those skilled in the art that modifications can be made in the embodiment chosen for illustration without departing from the spirit and scope of the invention. Specifically, it will be apparent to one of ordinary skill in the art that the device of the present invention could be implemented in several different ways and have several different appearances.

CLAIMS

What is claimed is:

1 1. A pumped fluid cooling system for cooling a device, the pumped fluid cooling system
2 comprising:
3 a. a heat exchanger, the heat exchanger including an interface layer coupled to the
4 device for cooling the device; and
5 b. a fluid pumped through the interface layer of the heat exchanger, the fluid
6 having an inlet temperature and an outlet temperature,
7 wherein the pumped fluid cooling system is configured such that the difference
8 between the fluid outlet temperature and the fluid inlet temperature is at least 30% of the
9 difference between a hottest temperature of the fluid in the heat exchanger and the fluid inlet
10 temperature.

1 2. The pumped fluid cooling system as claimed in claim 1 further comprising a plurality
2 of microchannels configured in a predetermined pattern along the interface layer
3 wherein the plurality of microchannels have an internal feature size in the range of 15-
4 300 microns.

1 3. The pumped fluid cooling system as claimed in claim 2 wherein the plurality of
2 microchannels have a surface to volume ratio greater than $1000m^{-1}$.

1 4. The pumped fluid cooling system as claimed in claim 1 further comprising a plurality
2 of pillars configured in a predetermined pattern along the interface layer wherein the
3 plurality of pillars have an internal feature size in the range of 15-300 microns.

1 5. The pumped fluid cooling system as claimed in claim 4 wherein the plurality of pillars
2 have a surface to volume ratio greater than $1000m^{-1}$.

1 6. The pumped fluid cooling system as claimed in claim 1 further comprising a
2 microporous structure disposed on the interface layer wherein a plurality of pores in
3 the microporous structure have an internal feature size in the range of 15-300 microns.

1 7. The pumped fluid cooling system as claimed in claim 6 wherein the plurality of pores
2 of the microporous structure have a surface to volume ratio greater than $1000m^{-1}$.

1 8. The pumped fluid cooling system as claimed in claim 1 wherein a first surface area of
2 the interface layer that is coupled to the device is less than or equal to 150% of a
3 second surface area of the device that is coupled to the interface layer.

1 9. The pumped fluid cooling system as claimed in claim 1 wherein the viscosity of the
2 fluid at its average temperature in the heat exchanger is less than 150% of the viscosity
3 of water.

1 10. The pumped fluid cooling system as claimed in claim 1 wherein the heat capacity per
2 unit mass of the fluid at its average temperature in the heat exchanger is greater than
3 80% of the heat capacity per unit mass of water.

1 11. The pumped fluid cooling system as claimed in claim 1 wherein the fluid consists of at
2 least 90% water by mass.

1 12. A method of efficiently cooling a device in a pumped fluid cooling system, the method
2 comprising:
3 a. decreasing a spread resistance between an interface layer of a heat exchanger
4 and the device;
5 b. decreasing a convection resistance between a fluid and the interface layer of the
6 heat exchanger, wherein the fluid is pumped through the interface layer, and
7 further wherein the fluid has an inlet temperature and an outlet temperature;
8 c. increasing an advection resistance; and
9 d. adjusting the composition of the fluid to increase the heat capacity per unit
10 mass and decrease the viscosity,
11 wherein the difference between the fluid outlet temperature and the fluid inlet
12 temperature is at least 30% of the difference between a hottest temperature of the fluid in the
13 heat exchanger and the fluid inlet temperature.

1 13. The method as claimed in claim 12 wherein the step of decreasing the convention
2 resistance includes configuring a plurality of microchannels in a predetermined pattern
3 along the interface layer wherein the plurality of microchannels have an internal
4 feature size in the range of 15-300 microns.

1 14. The method as claimed in claim 13 wherein the plurality of microchannels have a
2 surface to volume ratio greater than $1000m^{-1}$.

1 15. The method as claimed in claim 12 wherein the step of decreasing the convection
2 resistance includes configuring a plurality of pillars in a predetermined pattern along
3 the interface layer wherein the plurality of pillars have an internal feature size in the
4 range of 15-300 microns.

1 16. The method as claimed in claim 15 wherein the plurality of pillars have a surface to
2 volume ratio greater than $1000m^{-1}$.

1 17. The method as claimed in claim 12 wherein the step of decreasing the convection
2 resistance includes disposing a microporous structure on the interface layer wherein a
3 plurality of pores in the microporous structure have an internal feature size in the range
4 of 15-300 microns.

1 18. The method as claimed in claim 17 wherein the plurality of pores of the microporous
2 structure have a surface to volume ratio greater than $1000m^{-1}$.

1 19. The method as claimed in claim 12 wherein the step of decreasing the spread
2 resistance includes reducing a first surface area of the interface layer that is coupled to
3 the device such that the first surface area is less than or equal to 150% of a second
4 surface area of the device that is coupled to the interface layer.

1 20. The method as claimed in claim 12 wherein the step of adjusting the composition of
2 the fluid includes decreasing the viscosity of the fluid at its average temperature in the
3 heat exchanger, such that the viscosity is less than 150% of the viscosity of water.

1 21. The method as claimed in claim 12 wherein the step of adjusting the composition of
2 the fluid includes increasing the heat capacity per unit mass of the fluid at its average
3 temperature in the heat exchanger, such that the heat capacity per unit mass is greater
4 than 80% of the heat capacity per unit mass of water.

1 22. The method as claimed in claim 12 wherein the fluid consists of at least 90% water by
2 mass.

1 23. A pumped fluid cooling system for cooling a device, the pumped fluid cooling system
2 comprising:
3 a. means for decreasing a spread resistance between an interface layer of a heat
4 exchanger and the device;
5 b. means for decreasing a convection resistance between a fluid and the interface
6 layer of the heat exchanger, wherein the fluid is pumped through the interface
7 layer, and further wherein the fluid has an inlet temperature and an outlet
8 temperature,
9 c. means for increasing an advection resistance; and
10 d. means for adjusting the composition of the fluid to increase the heat capacity
11 per unit mass and decrease the viscosity,
12 wherein the difference between the fluid outlet temperature and the fluid inlet
13 temperature is at least 30% of the difference between a hottest temperature of the fluid in the
14 heat exchanger and the fluid inlet temperature.

1 24. The pumped fluid cooling system as claimed in claim 23 wherein the means for
2 decreasing the convention resistance includes means for configuring a plurality of
3 microchannels in a predetermined pattern along the interface layer wherein the
4 plurality of microchannels have an internal feature size in the range of 15-300 microns.

1 25. The pumped fluid cooling system as claimed in claim 24 wherein the plurality of
2 microchannels have a surface to volume ratio greater than $1000m^{-1}$.

1 26. The pumped fluid cooling system as claimed in claim 23 wherein the means for
2 decreasing the convection resistance includes means for configuring a plurality of
3 pillars in a predetermined pattern along the interface layer wherein the plurality of
4 pillars have an internal feature size in the range of 15-300 microns.

1 27. The pumped fluid cooling system as claimed in claim 26 wherein the plurality of
2 pillars have a surface to volume ratio greater than $1000m^{-1}$.

1 28. The pumped fluid cooling system as claimed in claim 23 wherein the means for
2 decreasing the convection resistance includes means for disposing a microporous
3 structure on the interface layer wherein a plurality of pores in the microporous
4 structure have an internal feature size in the range of 15-300 microns.

1 29. The pumped fluid cooling system as claimed in claim 28 wherein the plurality of pores
2 of the microporous structure have a surface to volume ratio greater than $1000m^{-1}$.

1 30. The pumped fluid cooling system as claimed in claim 23 wherein the means for
2 decreasing the spread resistance includes means for reducing a first surface area of the
3 interface layer that is coupled to the device such that the first surface area is less than
4 or equal to 150% of a second surface area of the device that is coupled to the interface
5 layer.

1 31. The pumped fluid cooling system as claimed in claim 23 wherein the means for
2 adjusting the composition of the fluid includes means for decreasing the viscosity of
3 the fluid at its average temperature in the heat exchanger, such that the viscosity is less
4 than 150% of the viscosity of water.

1 32. The pumped fluid cooling system as claimed in claim 23 wherein the means for
2 adjusting the composition of the fluid includes means for increasing the heat capacity

1 per unit mass of the fluid at its average temperature in the heat exchanger, such that the
2 heat capacity per unit mass is greater than 80% of the heat capacity per unit mass of
3 water.

1 33. The pumped fluid cooling system as claimed in claim 23 wherein the fluid consists of
2 at least 90% water by mass.

1 34. An apparatus for cooling an integrated circuit, the apparatus comprising:
2 a. a heat exchanger including an interface layer coupled to the integrated circuit,
3 wherein a first surface area of the interface layer that is coupled to the
4 integrated circuit is less than or equal to 150% of a second surface area of the
5 integrated circuit that is coupled to the interface layer, such that a spread
6 resistance between the interface layer and the integrated circuit is decreased;
7 b. a plurality of microchannels configured in a predetermined pattern along the
8 interface layer wherein the plurality of microchannels have an internal feature
9 size in the range of 15-300 microns and a surface to volume ration greater than
10 $1000m^{-1}$, such that a convection resistance is decreased; and
11 c. a fluid pumped through the heat exchanger, such that a flowrate of the fluid
12 increases an advection resistance,
13 wherein the fluid consists of at least 90% water by mass.

1 35. The apparatus as claimed in claim 34 wherein the viscosity of the fluid at its average
2 temperature in the heat exchanger is less than 150% of the viscosity of water.

1 36. The apparatus as claimed in claim 34 wherein the heat capacity per unit mass of the
2 fluid at its average temperature in the heat exchanger is greater than 80% of the heat
3 capacity per unit mass of water.

1 37. A pumped fluid cooling system for cooling a device, the pumped fluid cooling system
2 comprising:
3 a. a spread resistance, wherein the spread resistance is decrease when a heat
4 exchanger including an interface layer is coupled to the device, further wherein

1 a first surface area of the interface layer that is coupled to the device is less
2 than or equal to 150% of a second surface area of the device that is coupled to
3 the interface layer;
4 b. a convection resistance, wherein the convection resistance is decreased when a
5 plurality of microchannels is configured in a predetermined pattern along the
6 interface layer, and further wherein the plurality of microchannels have an
7 internal feature size in the range of 15-300 microns and a surface to volume
8 ration greater than $1000m^{-1}$; and
9 c. an advection resistance, wherein the advection resistance is increased when a
10 fluid is pumped through the heat exchanger, such that a flowrate of the fluid
11 decreases,
12 wherein the fluid consists of at least 90% water by mass.

1 38. The pumped fluid cooling system as claimed in claim 37 wherein the fluid is water.

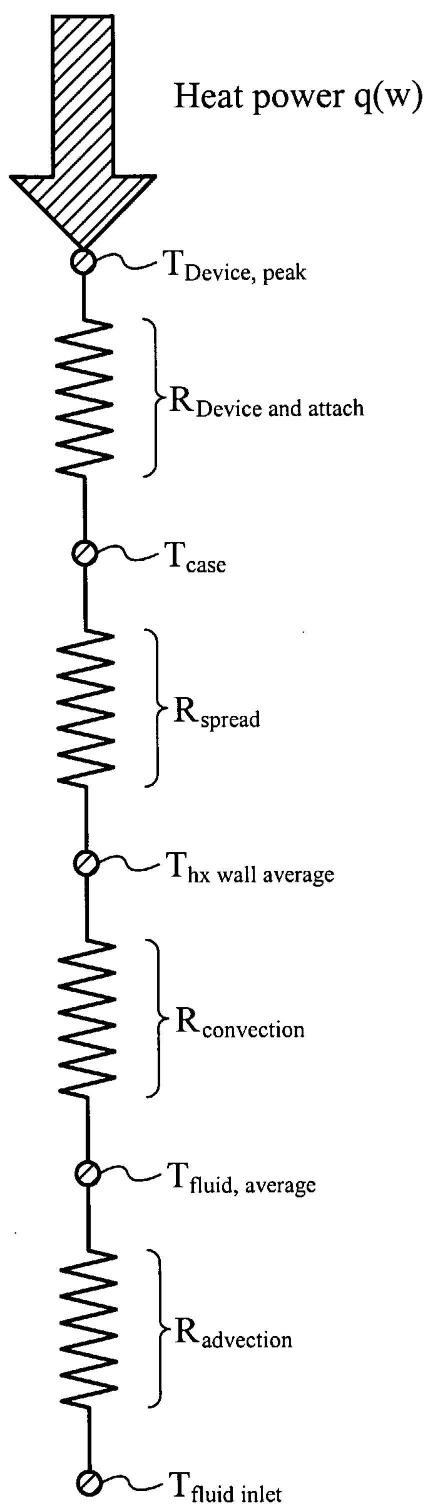

Fig. 1

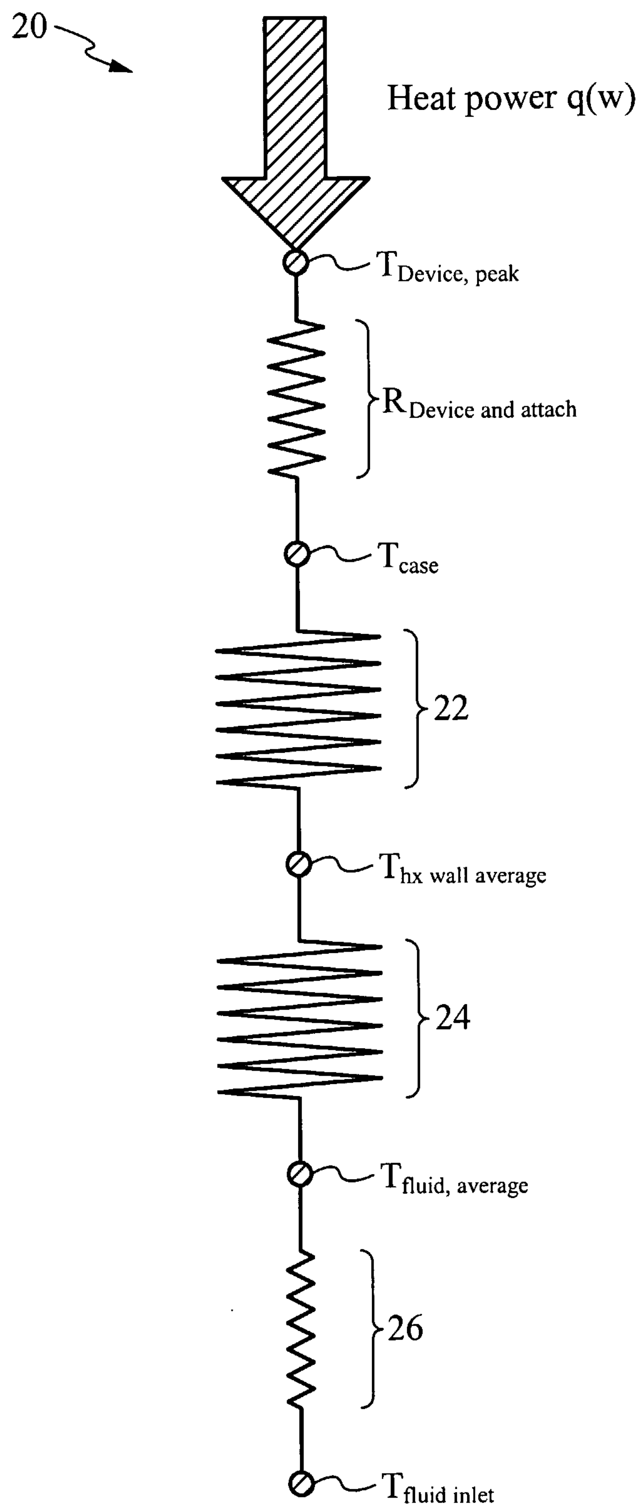

Fig. 2 (PRIOR ART)

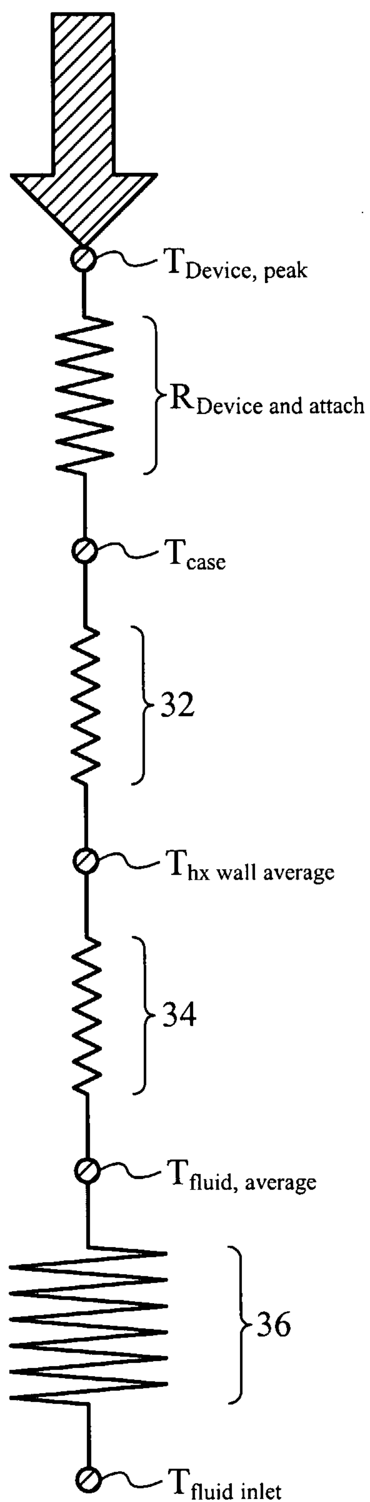

Fig. 3

109
107
108
118D
119A
106
122
119C 119B
116
120
118
118B
118C
121 120 121 118A 119D
107
y
z
x

Fig. 4A

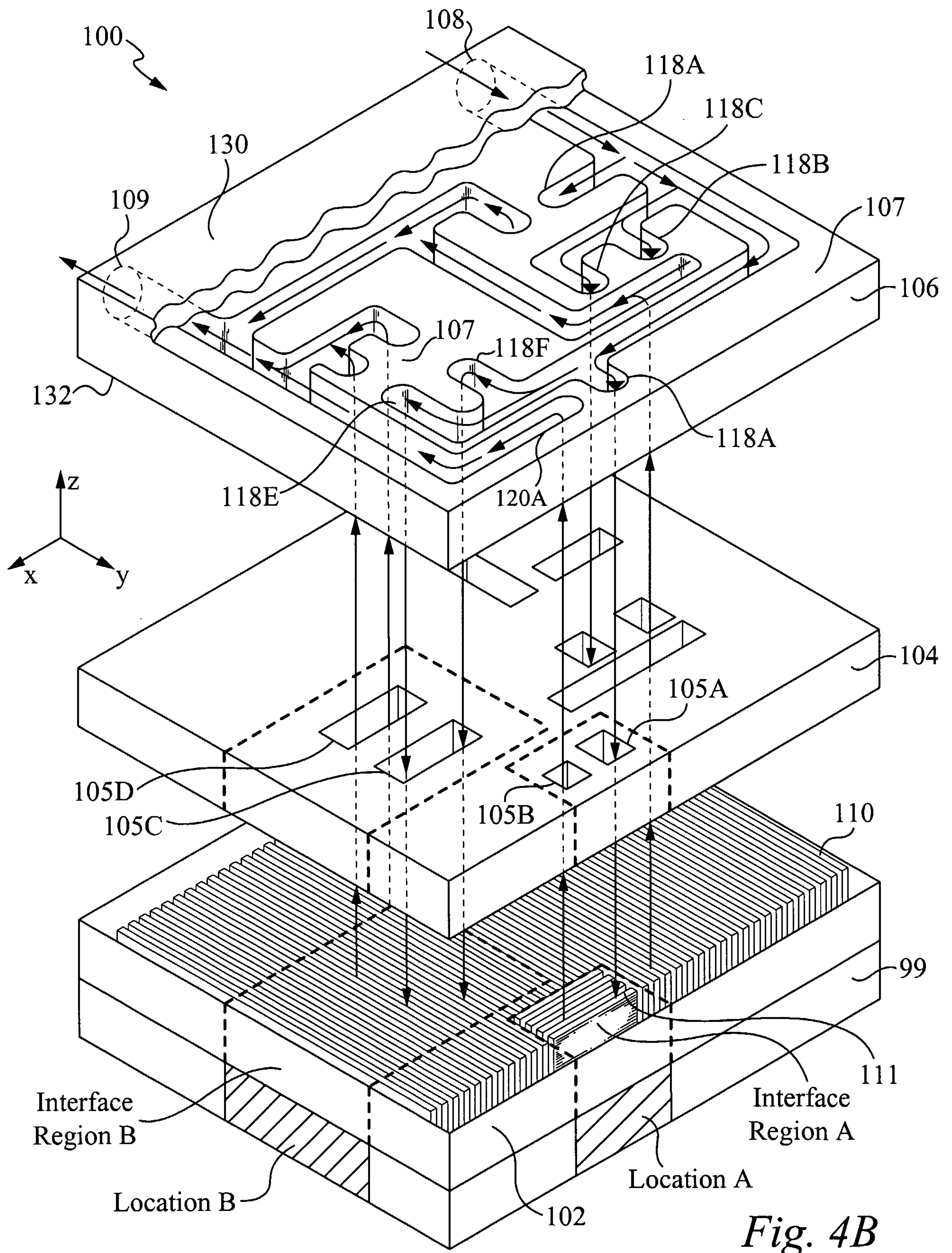

Fig. 4B

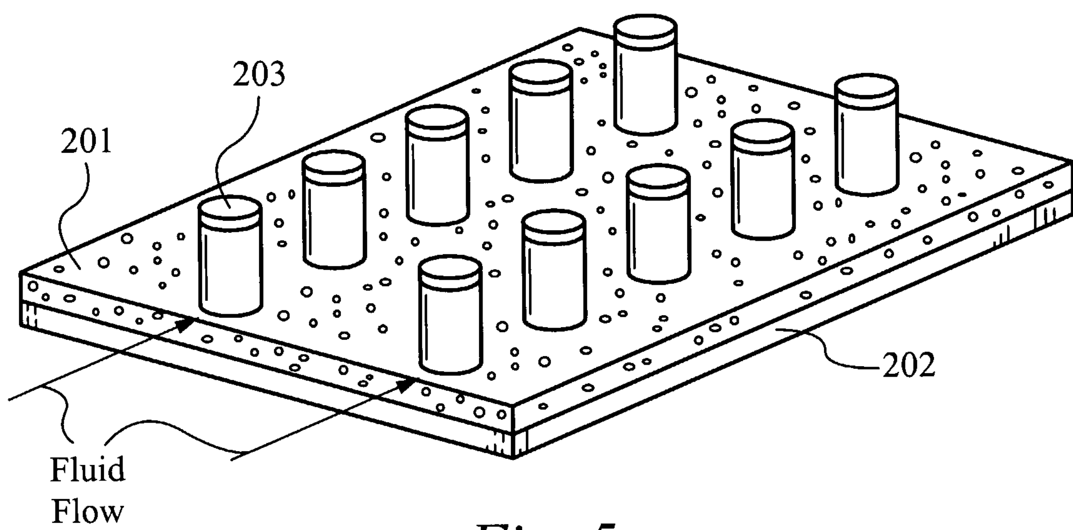

Fig. 5

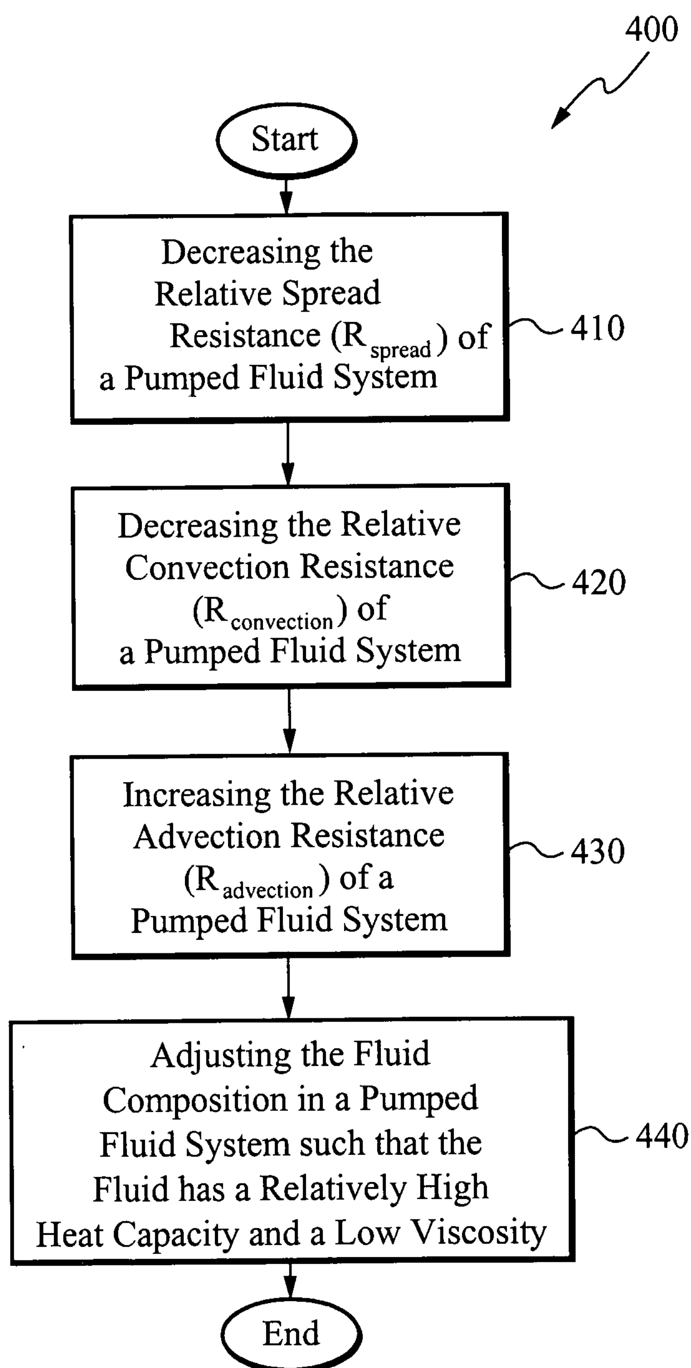

Fig. 6