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# (54) PUMPED FLUID COOLING SYSTEM AND METHOD

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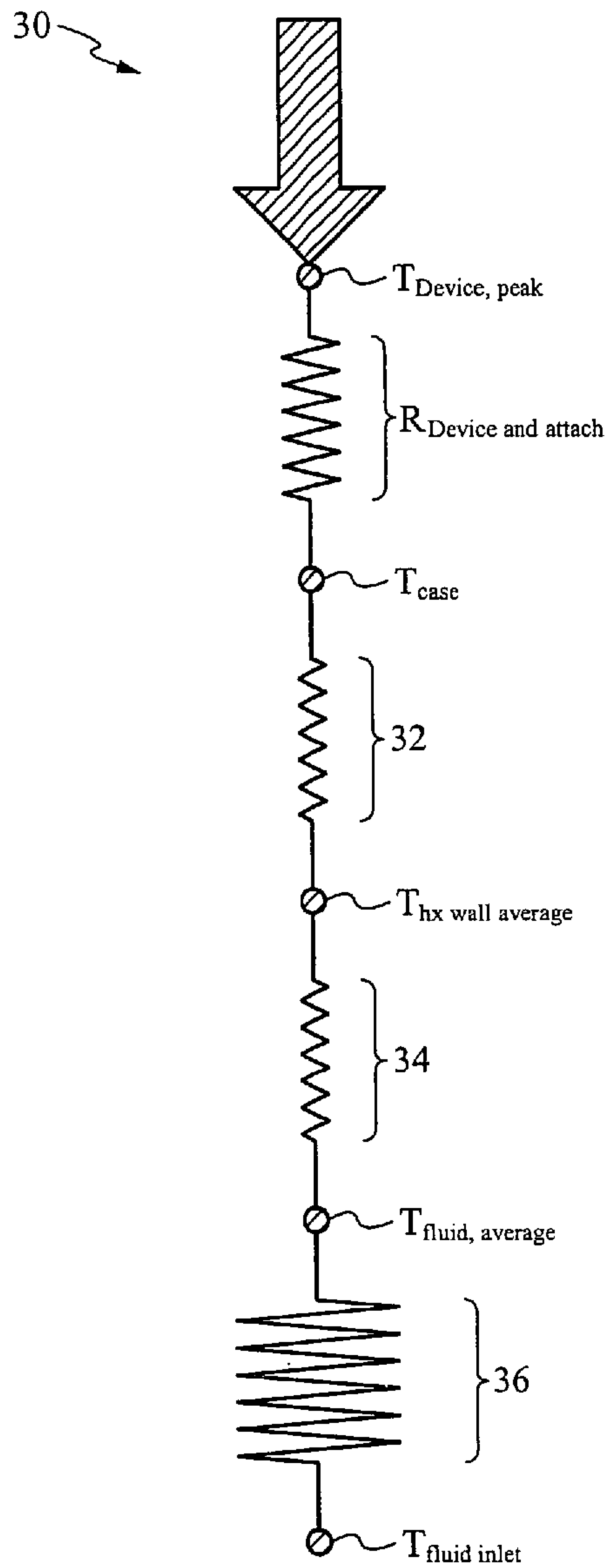
(57) **ABSTRACT**

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.



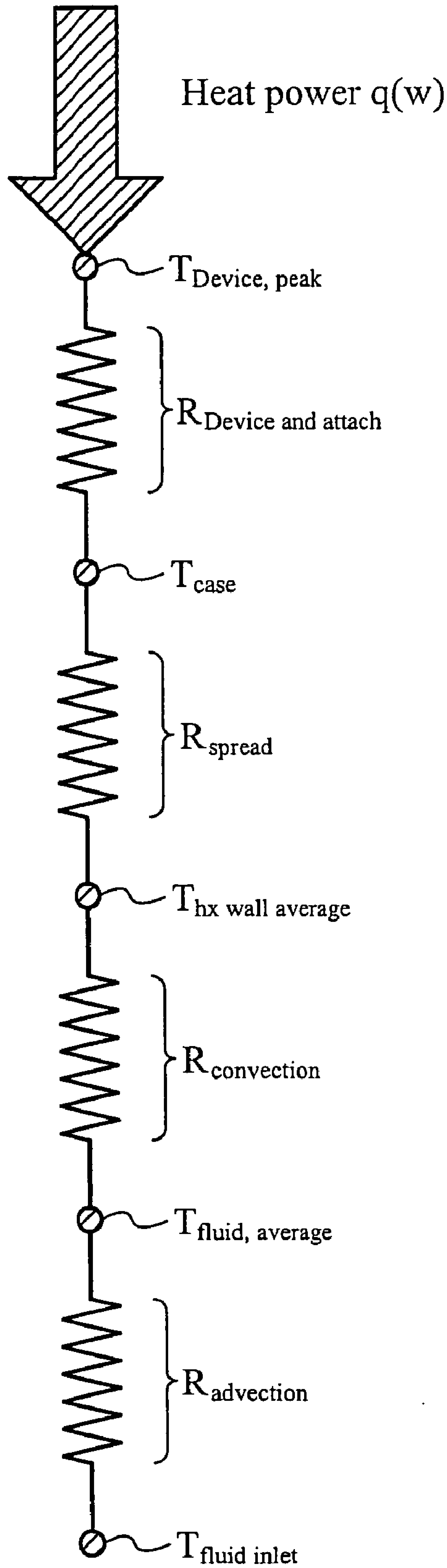

Heat power q(w)
$T_{Device, peak}$
$R_{Device and attach}$
$T_{case}$
$R_{spread}$
$T_{hx wall average}$
$R_{convection}$
$T_{fluid, average}$
$R_{advection}$
$T_{fluid inlet}$

*Fig. 1*

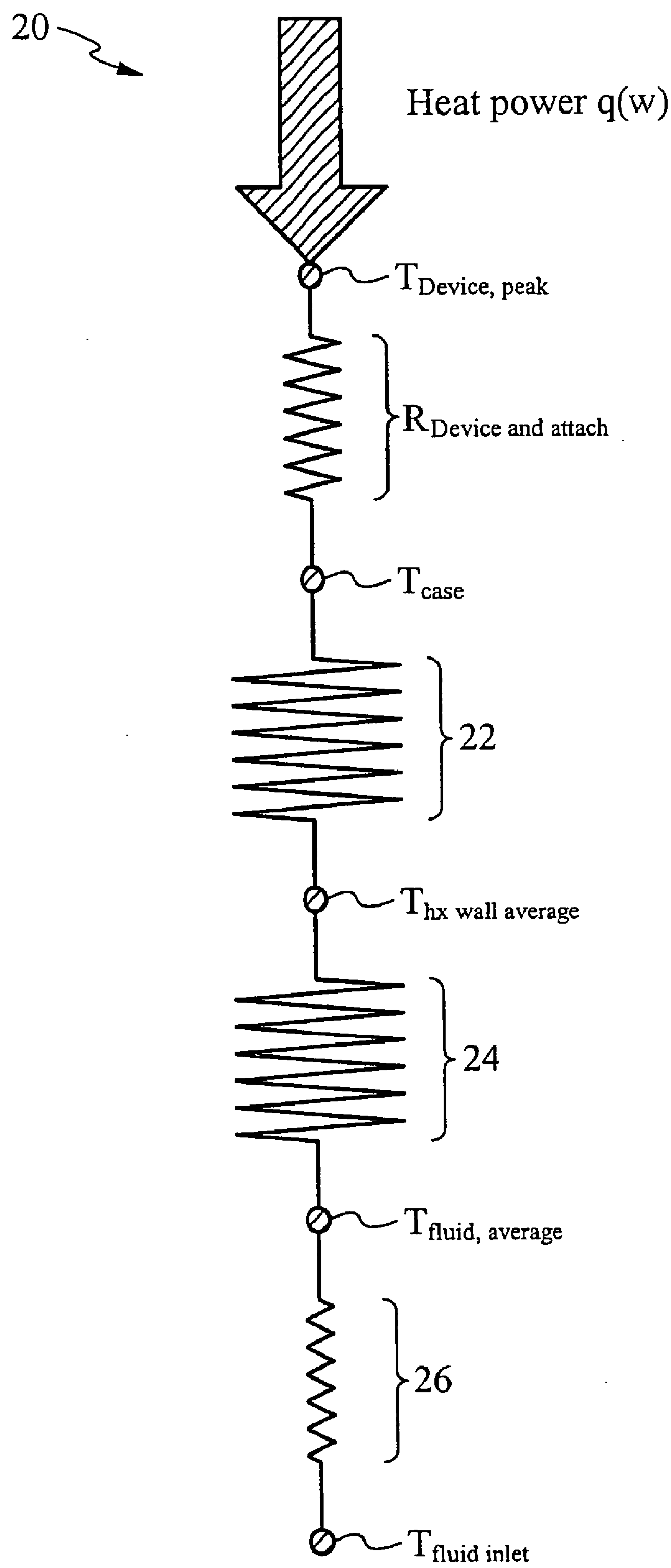



*Fig. 2* (PRIOR ART)

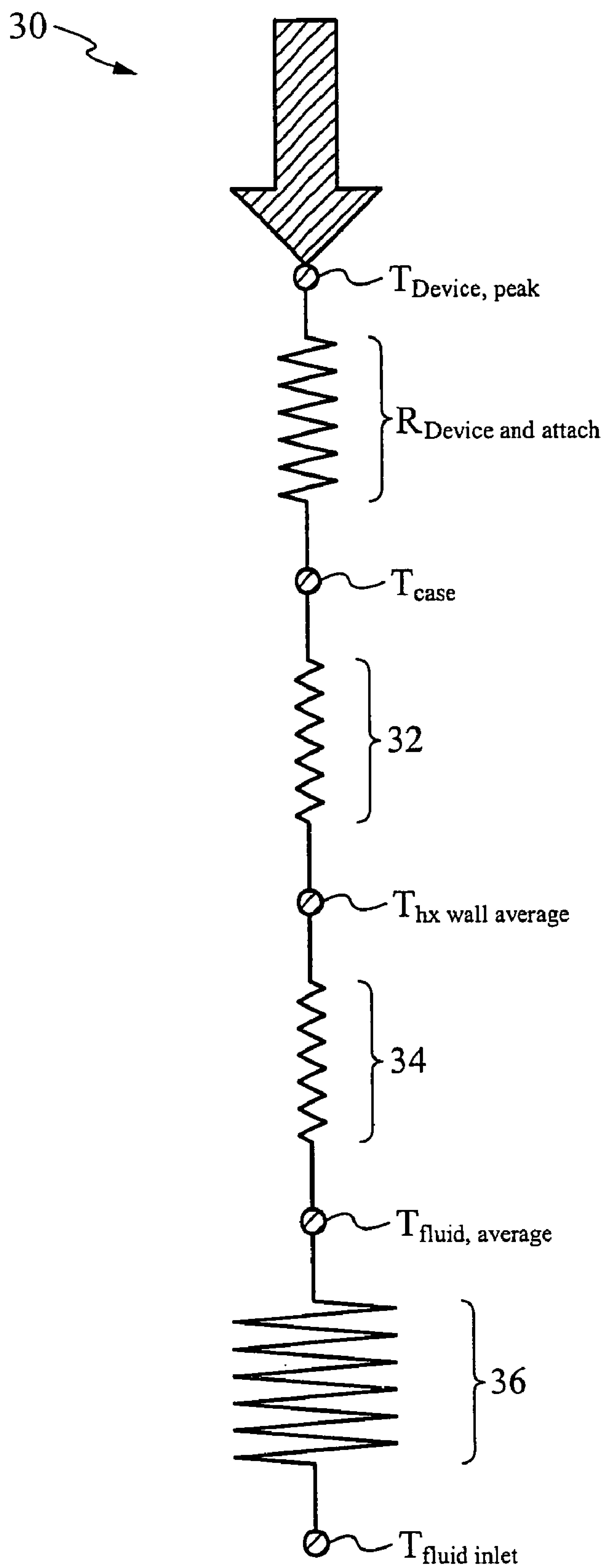

30
$T_{Device, peak}$
$R_{Device and attach}$
$T_{case}$
32
$T_{hx wall average}$
34
$T_{fluid, average}$
36
$T_{fluid inlet}$

*Fig. 3*

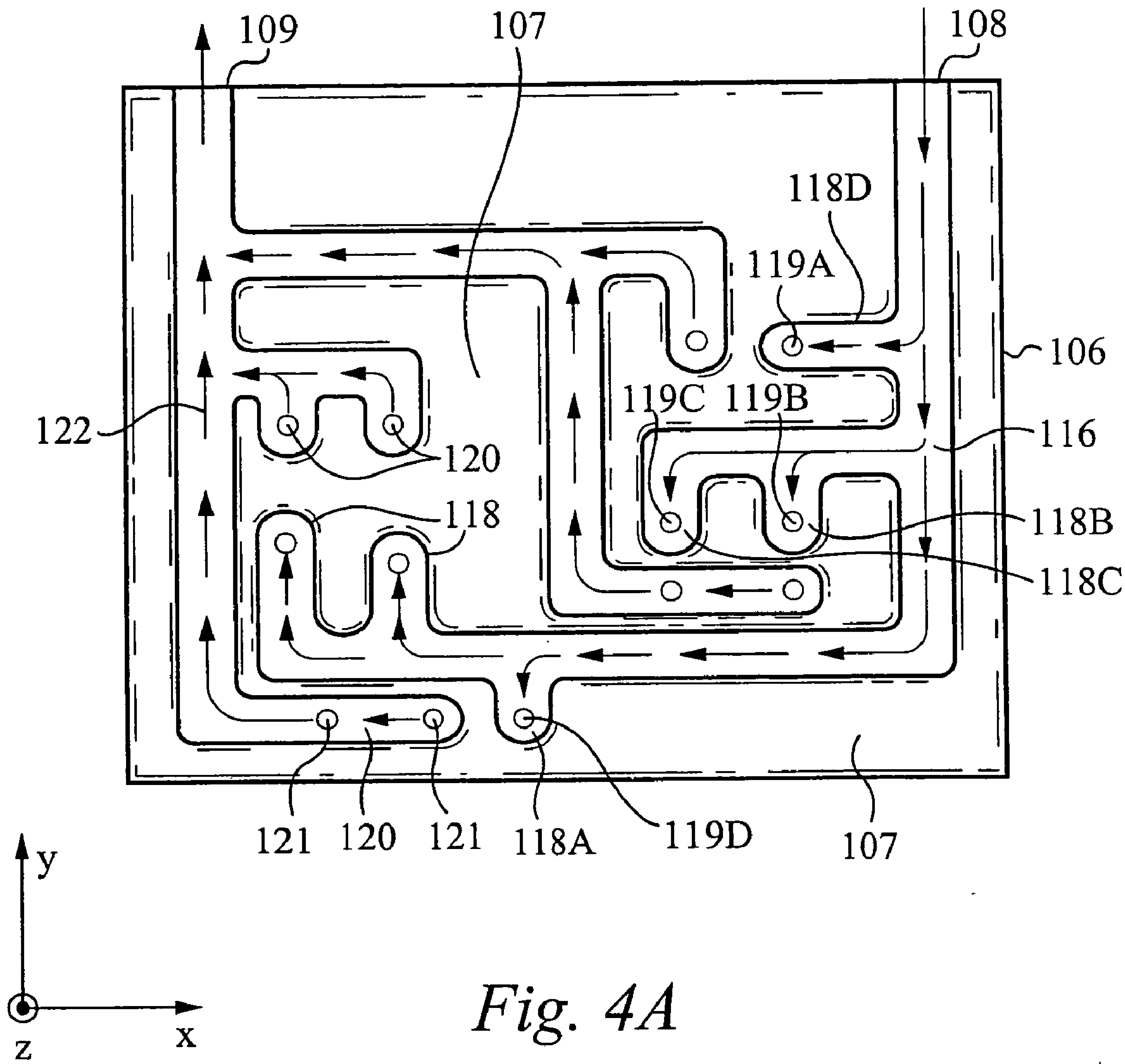

109
107
108
118D
119A
106
119C 119B
122
116
120
118
118B
118C
121
120
121
118A
119D
107
y
z
x

*Fig. 4A*

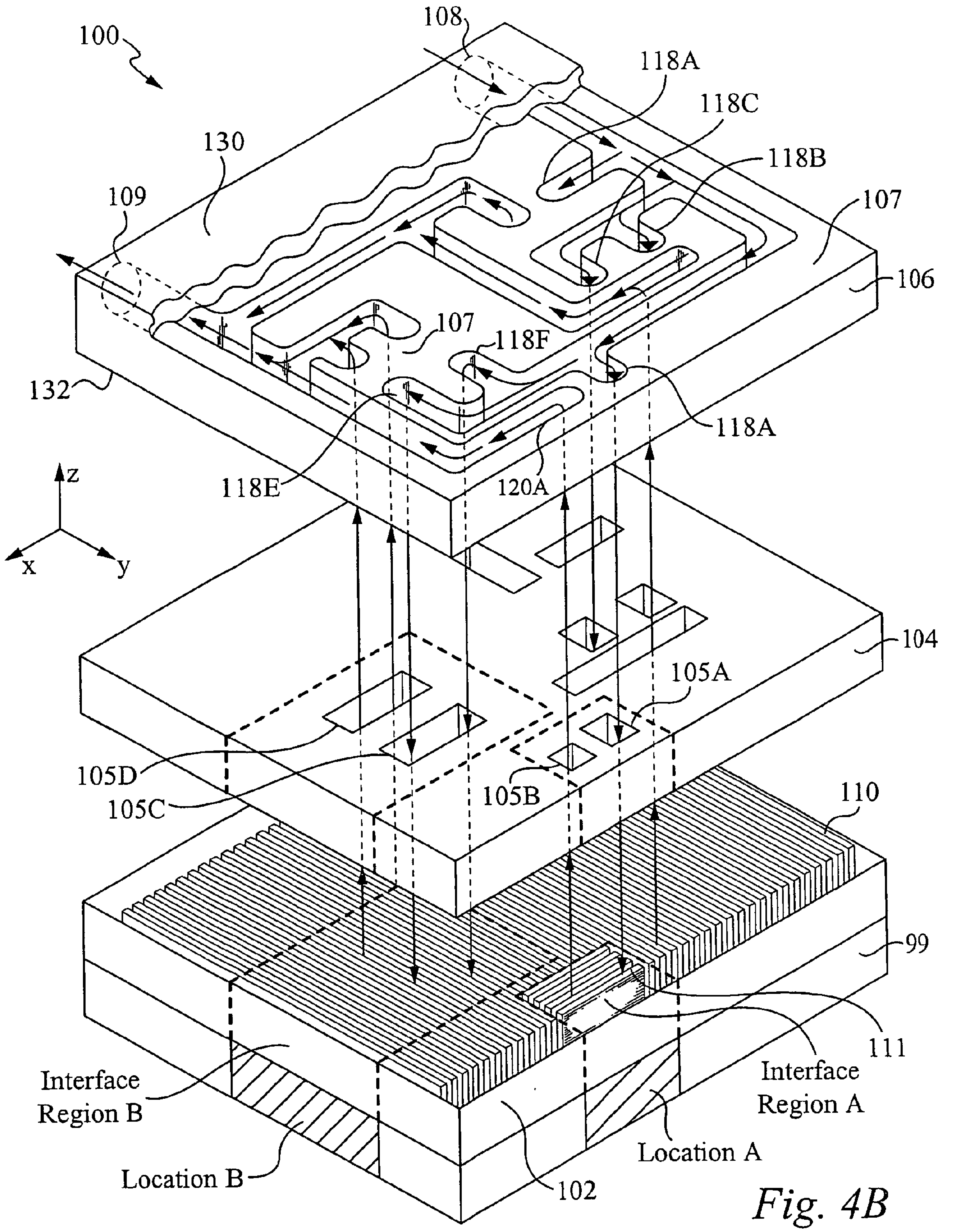

100
108
118A
118C
118B
130
107
109
106
107
118F
132
118A
118E
120A
z
x
y
104
105A
105D
105C
105B
110
99
111
Interface Region B
Interface Region A
Location A
Location B
102

Fig. 4B

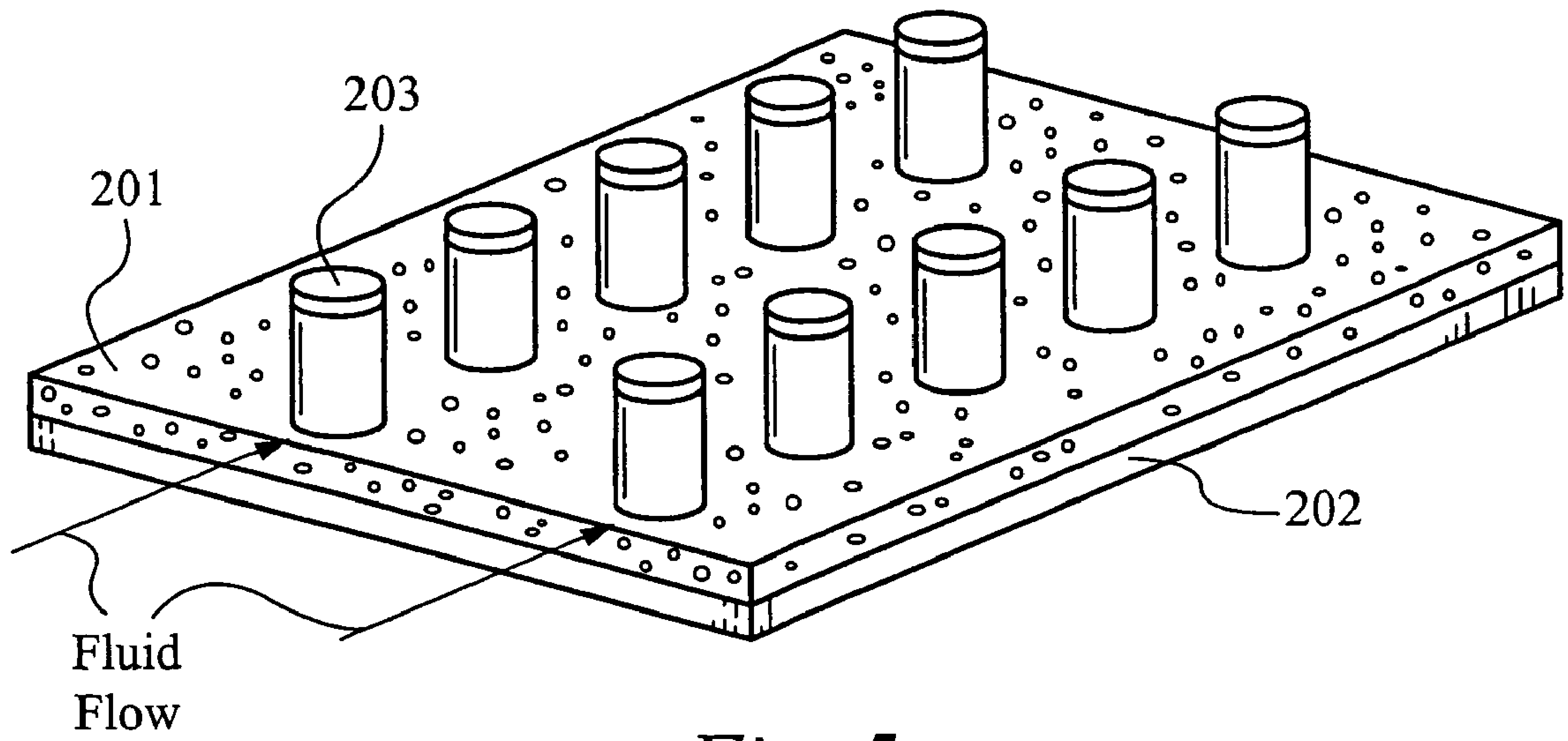



*Fig. 5*

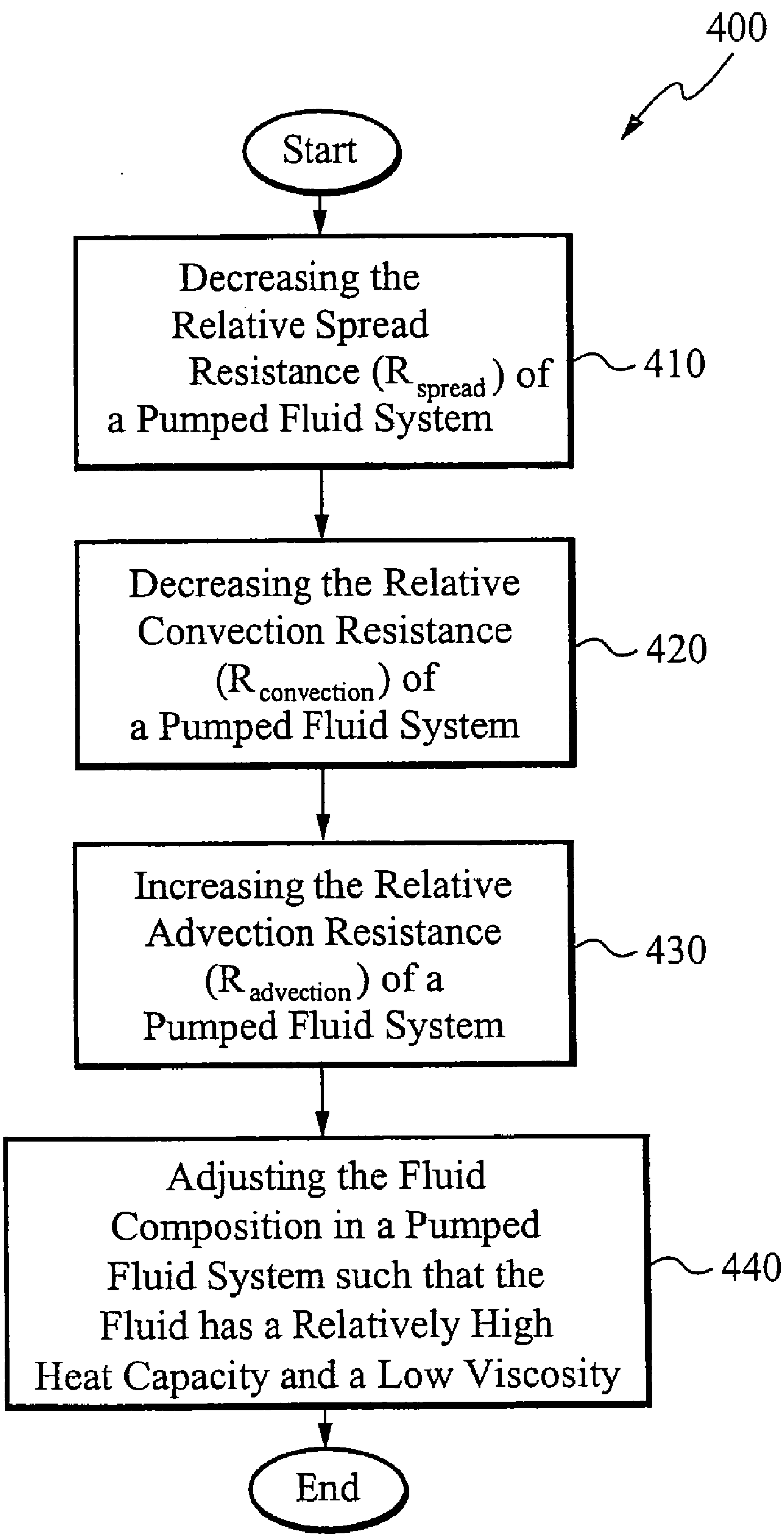

400
Start
Decreasing the Relative Spread Resistance ($R_{spread}$) of a Pumped Fluid System
410
Decreasing the Relative Convection Resistance ($R_{convection}$) of a Pumped Fluid System
420
Increasing the Relative Advection Resistance ($R_{advection}$) of a Pumped Fluid System
430
Adjusting the Fluid Composition in a Pumped Fluid System such that the Fluid has a Relatively High Heat Capacity and a Low Viscosity
440
End

*Fig. 6*

# PUMPED FLUID COOLING SYSTEM AND METHOD

## FIELD OF THE INVENTION

[0001] The present invention relates generally to the field of cooling systems. More specifically, the present invention relates to the field of pumped fluid cooling systems.

## BACKGROUND OF THE INVENTION

[0002] In current pumped fluid cooling systems as depicted in **FIG. 1**, the total "temperature budget," or the difference between the peak device temperature ($T_{Device, peak}$) and the temperature of the cold fluid inlet ($T_{fluid\ inlet}$) is consumed by the total heat power (q(W)) flowing through four separate resistances.

[0003] **FIG. 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 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 ($R_{convention}$) 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 values of the minimum feature size of the hydraulic diameter (d).

[0004] Still referring to **FIG. 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 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 0.3 mm 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.

[0005] Referring now to **FIG. 2**, a resistance model of a current pumped fluid system **20** of 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**.

[0006] 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 flow rates m. Of course, this puts large demands on the pump requirements for a pumped fluid system **20**.

[0007] 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 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.

## SUMMARY OF THE INVENTION

[0008] 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 mass increases.

[0009] 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 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.

[0010] 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 1000 $m^{-1}$. 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 1000 $m^{-1}$.

[0011] 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 1000 $m^{-1}$. 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.

[0012] 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.

**[0013]** 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 1000 $m^{-1}$. 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 1000 $m^{-1}$.

**[0014]** 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 1000 $m^{-1}$. 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.

**[0015]** 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.

**[0016]** 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 1000 $m^{-1}$. 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 1000 $m^{-1}$.

**[0017]** 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 1000 $m^{-1}$. The means for decreasing the spread resistance includes means for 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.

**[0018]** 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 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.

**[0019]** In yet another aspect of the present invention, an apparatus for cooling an integrated 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 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 1000 $m^{-1}$, 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 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.

**[0020]** 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 1000 $m^{-1}$ 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

[0021] **FIG. 1** is a graphical representation illustrating an exemplary temperature budget resistance model.

[0022] **FIG. 2** is a graphical representation illustrating a temperature budget resistance model according to the prior art.

[0023] **FIG. 3** is a graphical representation illustrating a temperature budget resistance model according to an embodiment of the present invention.

[0024] **FIG. 4A** is a graphical representation illustrating a top view of a manifold layer of a heat exchanger in accordance with the present invention.

[0025] **FIG. 4B** is a graphical representation illustrating an exploded view of a heat exchanger with a manifold layer in accordance with the present invention.

[0026] **FIG. 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.

[0027] **FIG. 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

[0028] **FIG. 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 **FIG. 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.

[0029] 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**.

[0030] 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 1000 $m^{-1}$. The structure of the micro hx is explained in greater detail below.

[0031] 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.

[0032] 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.

[0033] **FIG. 4A** illustrates a top view of an exemplary manifold layer **106** of the present invention. In particular, as shown in **FIG. 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 **FIG. 4A** to adequately illustrate and describe the workings of the manifold layer **106**. As shown in **FIG. 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 **FIG. 4B**. Alternatively, the fingers **118** and **120** extend partially through the manifold layer **106** in the Z-direction and have apertures as shown in **FIG. 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**.

[0034] As shown in **FIG. 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 **FIG. 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**.

[0035] 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.

[0036] 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.

[0037] 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. 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.

[0038] 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 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.

[0039] 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 **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.

[0040] Alternatively, as shown in **FIG. 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, as shown in **FIG. 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**.

[0041] 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.

[0042] Although **FIG. 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.

[**0043**] As shown in **FIG. 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**.

[**0044**] 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**.

[**0045**] 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**.

[**0046**] **FIG. 4B** illustrates a perspective view of the interface layer **102** in accordance with the present invention. As shown in **FIG. 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 **FIG. 4B**, whereby fluid flows between the microchannel walls **110** along a fluid path. Alternatively, the microchannel walls **110** have non-parallel configurations.

[**0047**] 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** (**FIG. 5**), roughed surfaces, and a micro-porous structure, such as sintered metal and silicon foam **213** (**FIG. 5**) or a combination. An alternative interface layer **202** incorporating both pillars **203** and foam microporous **213** inserts is depicted in **FIG. 5**. However, for exemplary purposes, the parallel microchannel walls **110** shown in **FIG. 4B** is used to describe the interface layer **102** in the present invention.

[**0048**] Referring back to the assembly in **FIG. 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 **FIG. 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 **FIG. 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.

[**0049**] As shown in **FIGS. 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 **FIGS. 4A and 4B**, fluid flowing along the inlet channel **116** from the inlet port **108** initially branches out to finger **118**D. In addition, the fluid which continues along the rest of the inlet channel **116** flows to individual fingers **118**B and **118**C and so on.

[**0050**] In **FIG. 4B**, fluid is supplied to interface hot spot region A by flowing to the finger **118**A, whereby fluid preferably flows down through finger **118**A to the intermediate layer **104**. The fluid then flows through the inlet conduit **105**A, positioned below the finger **118**A, 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 **FIG. 4B**, although the fluid may travel in any other direction along the interface layer **102**. The heated liquid then travels upward through the conduit **105**B to the outlet finger **120**A. Similarly, fluid flows down in the Z-direction through fingers **118**E and **118**F to the intermediate layer **104**. The fluid then flows through the inlet conduit **105**C 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 **105**D to the outlet fingers **120**E and **120**F. 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**.

[**0051**] 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 drop therebetween. Thus, fluid enters the interface layer **102** via the inlet finger **118**A 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 **120**A. 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. In addition, as shown in **FIGS. 4A and 4B**, the comers in the fingers **118**, **120** are curved to reduce pressure drop of the fluid flowing along the fingers **118**.

[**0052**] It is apparent to one skilled in the art that the configuration of the manifold layer **106** shown in **FIGS. 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, 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.

[**0053**] Referring to **FIG. 4B**, the preferred embodiment of the present invention includes 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 1000 $m^{-1}$. Of course, the present invention contemplates further embodiments contemplating microchannels not entirely within the stated ranges.

[**0054**] Referring now to **FIG. 5**, further embodiments also contemplate utilizing alternatives to the microchannels **110** (**FIG. 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 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.

[**0055**] 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.

[**0056**] 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.

[**0057**] **FIG. 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 1000 $m^{-1}$.

[**0058**] Still referring to **FIG. 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.

[**0059**] 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.

What is claimed is:

**1**. A pumped fluid cooling system for cooling a device, the pumped fluid cooling system comprising:

a. a heat exchanger, the heat exchanger including an interface layer coupled to the device for cooling the device; and

b. 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 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.

**2**. The pumped fluid cooling system as claimed in claim 1 further comprising 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.

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

**4**. The pumped fluid cooling system as claimed in claim 1 further comprising 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.

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

**6**. The pumped fluid cooling system as claimed in claim 1 further comprising 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.

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

**8**. The pumped fluid cooling system as claimed in claim 1 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.

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

**10**. The pumped fluid cooling system as claimed in claim 1 wherein 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.

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

**12**. A method of efficiently cooling a device in a pumped fluid cooling system, the method comprising:

a. decreasing a spread resistance between an interface layer of a heat exchanger and the device;

b. 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;

c. increasing an advection resistance; and

d. 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.

**13**. The method as claimed in claim 12 wherein 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.

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

**15**. The method as claimed in claim 12 wherein 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.

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

**17**. The method as claimed in claim 12 wherein 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.

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

**19**. The method as claimed in claim 12 wherein 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.

**20**. The method as claimed in claim 12 wherein 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.

**21**. The method as claimed in claim 12 wherein 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.

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

**23**. A pumped fluid cooling system for cooling a device, the pumped fluid cooling system comprising:

a. means for decreasing a spread resistance between an interface layer of a heat exchanger and the device;

b. 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,

c. means for increasing an advection resistance; and

d. 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.

**24**. The pumped fluid cooling system as claimed in claim 23 wherein 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.

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

**26**. The pumped fluid cooling system as claimed in claim 23 wherein 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.

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

**28**. The pumped fluid cooling system as claimed in claim 23 wherein 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.

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

**30**. The pumped fluid cooling system as claimed in claim 23 wherein the means for decreasing the spread resistance includes means for 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.

**31**. The pumped fluid cooling system as claimed in claim 23 wherein 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.

**32**. The pumped fluid cooling system as claimed in claim 23 wherein the means for adjusting the composition of the fluid 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.

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

**34**. An apparatus for cooling an integrated circuit, the apparatus comprising:

a. 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;

b. 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 and a surface to volume ration greater than 1000 $m^{-1}$, such that a convection resistance is decreased; and

c. 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.

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

**36**. The apparatus as claimed in claim 34 wherein 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.

**37**. A pumped fluid cooling system for cooling a device, the pumped fluid cooling system comprising:

a. 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;

b. 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 10000 $m^{-1}$; and

c. 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.

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

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